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DESIGN AND THERMAL OPTIMIZATION OF THERMOPLASTIC HEAT SINKS VALIDATED BY OPENFOAM®

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Abstract

With the increase of functionalities present in the electronic devices, namely in the automotive industry, and with the miniaturization of the electronic components, the thermal management is an essential factor to take into account. This factor is raised in importance since the power demand has increased within each device, leading to electronic circuit failure by high temperature. Heat sinks are often used to improve the heat dissipation and avoid the temperature increase on the electronic components. Due to their high thermal conductivity, metals have been used as the more logical solution for the heat sinks. However, from an ecological and economical point of view, introducing thermoplastic composites is an attractive alternative in thermal management. This dissertation has the goal of modeling and simulating various designs of heat sinks with thermoplastic composite materials in OpenFOAM® software to find an optimized solution.

The initial study of how much the fan power and the air inlet position influenced the performance of the heat sink revealed to be highly important, as much as the heat sink design. Regarding the design, a general study was carried out to find the optimal thickness of the fins, revealing that thinner plate fins, of 1 mm, improved the heat sources temperature results by at most 2.6%, when compared to the original plate fins, of 2 mm. An increase of the plate fins height of 2 mm significantly augments the temperature reduction by 4.3%, due to the increase of the heat transfer area. Tests using the bypass airflow effect on the design of the heat sink, as seen in previous works, showed that the disadvantage of lower heat transfer area surpassed the advantage of fresh airflow in the back heat sources. However, an adjustment was made by transferring the ramp to the end of the heat sink, and creating it instead of cutting it, improving the average temperature of the heat sources further apart from the inlet by at most 1.4%. Pin structures were implemented with two different approaches: the first one was implementing pins in the channels between the plates to enhance the turbulence in the flow but the outcome did not work as planned, at least with the considered configurations; the second way was using a heat sink made entirely by pins instead of plate fins and it was investigated two different numbers of pins, as the one with higher density generated better results, but overall the performance was still not as good as the plate fins heat sinks. A heat sink with a thicker base was also considered, to increase even more the heat transfer area, and the optimizations reached a temperature reduction of 3.2%. In the end, resorting to a material with higher thermal conductivity granted the most significant improvement, of 7.1%.

The final heat sink, with plates of 1 mm thickness, 10 mm of height, a ramp of 1 mm of height and a base approximately 1 mm thicker, using the new fan, improved the temperatures at most by 12.0% in comparison to the original heat sink. Using the new material, an improvement of 18.2% in heat sources temperature reduction was achieved.

Keywords:

Thermoplastic; OpenFOAM®; heat sink; optimization; design.

Resumo

Com o aumento das funcionalidades em dispositivos eletrônicos, nomeadamente na indústria automóvel, e com a miniaturização dos componentes eletrônicos, a gestão térmica é um fator essencial a ter em conta. Este fator tem grande importância já que a demanda de energia em cada dispositivo tem vindo a aumentar, provocando falhas de circuitos integrados devido à ocorrência de temperaturas elevadas. Dissipadores de calor são implementados frequentemente para melhorar a dissipação térmica e evitar subida de temperatura nos componentes eletrônicos. Devido às suas elevadas condutividades térmicas, metais são normalmente utilizados em dissipadores de calor. Porém, a nível ecológico e económico, introduzir compósitos termoplásticos é uma alternativa atrativa na gestão térmica. Esta dissertação tem como objetivo modelar e simular várias geometrias de dissipadores de calor de compósitos termoplásticos em OpenFOAM® para encontrar uma solução mais eficiente.

Um estudo preliminar sobre a influência da potência da ventoinha e da posição da entrada de ar no desempenho do dissipador de calor foi realizado e revelou tanta importância quanto a própria geometria do dissipador de calor. Relativamente à geometria, com o objetivo de encontrar a espessura ideal das alhetas, foi feito um estudo geral, cujos resultados revelaram que placas menos espessas, de 1 mm, atingem uma melhoria de 2.6% na redução da temperatura das fontes de calor, comparativamente com as placas originais, de 2 mm. O aumento da altura das alhetas em 2 mm melhorou a redução de temperaturas até 4.3% devido ao aumento da área de transferência de calor. Quando foi considerado o efeito dos escoamentos secundários nas alterações feitas à geometria do dissipador de calor, como observado em trabalhos anteriores, as desvantagens superaram as vantagens. Contudo, ao ajustar esta modificação, transferindo a rampa para o final das alhetas e construí-la sobre as alhetas em vez de sacrificar a altura do dissipador de calor, os resultados melhoraram substancialmente, principalmente na zona onde a rampa foi construída, atingindo uma melhoria de 1.4% na redução da temperatura. Foram utilizados pinos em duas abordagens: a primeira consistiu em implementar pinos nos canais entre as alhetas para aumentar a turbulência no escoamento mas esta adição prejudicou os resultados, pelo menos para as configurações usadas; a segunda forma foi usar o dissipador de calor feito exclusivamente por pinos tendo sido utilizadas duas densidades de pinos diferentes, em que o caso mais denso permitiu melhores resultados, ainda que no global não se mostrassem tão bons quanto os dos dissipadores de calor de alhetas. Um dissipador de calor com a base mais espessa também foi considerado para maximizar a área de transferência de calor e a redução chegou aos 3.2%. Por último, recorreu-se a um material com condutividade térmica mais elevada, que garantiu uma melhoria mais significativa, de 7.1% comparativamente com o mesmo dissipador de calor mas com o material original.

O dissipador de calor final, com placas de 1 mm de espessura, 10 mm de altura, uma rampa de 1 mm de altura, uma base com aproximadamente mais 1 mm de espessura e implementando a ventoinha nova permitiu melhorias até 12.0% em comparação com o dissipador de calor original, enquanto que o uso do novo material melhorou o desempenho até aos 18.2%.

Palavras-Chave: Termoplástico; OpenFOAM®; dissipador de calor; otimização; geometria.

Declaration

I hereby declare, under word of honor, that this work is original and that all non-original contributions are indicated and due reference is given to the author and source

A handwritten signature in black ink, reading "Eduardo José Marques Carneiro". The signature is written in a cursive style with a large initial 'E' and 'C'.

02/07/2021

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Notation and Glossary

A	conduction area	m^2
A_C	area between mediums	m^2
A_R	radiating surface area	m^2
Co	Courant number	
C_P	heat capacity at constant pressure	$\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$
h	heat transfer coefficient	$\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$
k	thermal conductivity	$\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$
Nu	Nusselt number	
q	heat transfer rate	W
Re	Reynolds number	
T	temperature	K
U	velocity	$\text{m}\cdot\text{s}^{-1}$
x	position	m
Δp	pressure drop	Pa
ΔT	temperature difference	K
Δt	time step size	s
Δx	cell characteristic size	m

Greek Letters

ε	emissivity	
ρ	density	$\text{kg}\cdot\text{m}^{-3}$

List of Acronyms

CFD	Computational Fluid Dynamics
IC	Integrated Circuit
LVG	Longitudinal Vortex Generators
MCHS	Microchannel Heat Sink

1 Introduction

1.1 Framing and presentation of the work

Nowadays, technology is part of modern society's life and there is a constant search for evolving the life quality by creating devices that assist every task in a person routine. One of the main goals that has been sought exponentially over the years is the miniaturization of electronic devices, while enhancing their functionalities and consequent power density. However, this means that the heat generated by the power source also raises and with less area for dissipation, this heat will sharply increase the device temperature. This is a problem since for every 1°C raised above the limit temperature of the device, there can be a decreasing in the reliability of 5% [1]. Temperature rise is the main factor of electronic chip failure, corresponding to 55% [2] of the total, and this has been leading to various researchers to investigate how to manage the temperature and heat dissipation. The two principal methods for thermal management is using materials with high heat conduction capacity, so that it can be transferred efficiently through the device, or by improving the design with structures and resources that allows a more efficient heat dissipation.

Now, with the current investigations on exchanging the material from metal to thermoplastic composites, allowing a more lightweight product but sacrificing the thermal properties, it is more important than ever to maximize the thermal dissipation. So, this thesis goal is to maximize the heat dissipation and, consequently, control the device temperature rise by simulating different types of heat sinks in an electronic box with integrated circuits (ICs) as heat sources and air as the convective fluid. From changing the dimensions of the original heat sink to innovating the design completely, various configurations have been considered and simulated in OpenFOAM®, an open source Computational Fluid Dynamics (CFD) software, instead of the more commercial Ansys FLUENT software, allowing deeper adjustments in the simulation.

1.2 Presentation of the company

The theme for this thesis was offered and coordinated by Bosch Car Multimedia Portugal, S.A, located in Braga, Portugal, since 2009 and focuses in the investigation and production of automobile sensors, mobility and multimedia solutions. This corporation is in the top 4 of Portugal exporters and has more than 50 automobile producers that resort to Bosch Car Multimedia resources.

1.3 Contribution of the author to the work

In this dissertation, I first produced the various designs of the heat sinks and the box with the heat sources in Siemens NX™ software with the real dimensions. In this task, along the duration of the dissertation, I had to make decisions in order to simplify the case and the models in a way that kept the problem close to reality while making it viable to solve within the available time. Then, after learning how to use and program with OpenFOAM®, with all the research that it entails, I imported the models to the software, prepared the required simulations with the most adequate and optimized configurations to the case, along with implementing functions that fasten the process, like calculating the average temperature in the solids, and ran them. In the end, I post-processed the results in paraFoam and analyzed them to find the best solution to the problem, regarding temperature reduction, weight and viability.

1.4 Organization of the dissertation

The Context and State of the Art section describes previous research so that a plan of optimizations in the heat sink design could be made. For that, first it was studied the heat transfer methods and their dominance in the case. Then, optimizations in the two main heat sink types regarding dimensions were studied, following possible alternative designs and its applications, as well its limitations. About the convective fluids used, it was taken into account both the effects in the fluid flow in the heat sink and also the type of fluid used and its mechanism in the process of heat transfer.

The Computer Applications section describes the procedure followed using Siemens NX™ and using OpenFOAM®. With OpenFOAM® there was a constant need of keeping the balance between quality of results and limited time of simulation that lead to making various decisions and applying various utilities. This is explained in detail in this section as well as how the results were post-processed.

In the Results and discussion section, the results prepared are compared with each other to understand which design allows the lowest average temperature in the heat sources and lowest weight.

The Conclusions section summarizes the results obtained and the heat sink design chosen, as well as possible consideration for future works in this area of research.

In the Assessment of the work done section, it is deliberated whether the committed objectives were fulfilled, and it is given the feedback of the work behind this dissertation.

2 Context and State of the art

2.1 Heat Transfer

The heat transfer between materials happens everywhere, when temperature gradients occur, and the three mechanisms of this phenomenon are heat conduction, convection and radiation.

2.1.1 Conduction

The thermal conductivity is a key property to be analyzed when choosing a material for electronic cooling because if it is not high enough, the material will not fulfill its job efficiently. It quantifies the ability of a material to conduct heat through the body itself and is especially high in metals, being one of the reasons they have been the most used materials in this area of expertise.

Fourier's Law, or the law of heat conduction, states that one-dimensional heat flow can be described by

$$q = -k A \frac{dT}{dx} \quad (1)$$

where q is the heat transferred by conduction (W), k is the thermal conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$), A is the cross sectional transfer area (m^2) and T is the temperature (K) at a certain position x (m) [3].

Equation (1) represents the internal energy that is transferred by the vibration and collision of particles such as molecules, phonons or free electrons. While in materials like metals, diffusion and collision of electrical particles is the dominant path, materials who tend to be insulators or semiconductors, like polymers, mainly promote heat conduction by phonons [3].

2.1.2 Convection

The heat transfer coefficient is important when choosing the fluid in which the heat is supposed to be dissipated to. It quantifies how well heat is transferred between mediums. Convection is the dominant form of heat transfer in liquids and gases and is usually higher in liquids.

Newton's law of convection describes the heat transferred by convection and can be expressed by

$$q = h A_c \Delta T \quad (2)$$

where h is the convective heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$), A_c is the heat transfer area between mediums (m^2) and ΔT is the temperature difference between the body surface and the surroundings (K).

Convection works with the movement of fluids and can be either natural or forced when the fluid movement is externally driven to increase the heat transfer rate.

2.1.3 Radiation

Even though radiation typically represents less than 5% of all the heat transferred, there are cases where it becomes dominant. Electromagnetic radiation is emitted by every matter with temperature higher than absolute zero and its power can be expressed by the Stefan-Boltzmann law

$$q = \varepsilon \sigma A_R T^4 \quad (3)$$

where ε is the body emissivity, σ is the Stefan-Boltzmann constant ($\sigma = 5.67 \times 10^{-8} \text{ W} \cdot \text{m}^{-2} \cdot \text{K}^{-4}$), A_R is the radiating surface area (m^2) and T is the body's absolute temperature (K).

By analyzing equations (1)-(3), it is evident that the higher the temperature of the power source, the greater is the contribution of the thermal radiation in the heat transfer process.

2.2 Materials

As already mentioned, metals have the right properties to be used for electronic packaging and also in heat sinks for heat dissipation. Aluminum, the most used one, grants a thermal conductivity of $237 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ for pure aluminum, reaching between $90 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ to $150 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ for different alloys, and is interesting due to being a lightweight and low-cost material. Copper, despite not excelling in the latter referred properties, is a good alternative with a higher thermal conductivity, $413 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$ at ambient temperature.

However, a search for more efficient and lucrative options is always on the table, especially when the problem in hands is becoming more and more alarming. Thermoplastics are lightweight, have competitive prices and their high variety of options allow adapting the polymers used with the proper chemical and mechanical properties needed to each specific application [4].

Still, thermoplastics present low thermal conductivities ($\approx 0.1\text{-}0.5 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$) [3] and are usually used more as insulators than as heat conductors. So, a solution for this limitation is implementing fillers with high thermal conductivities in composites with thermoplastic matrix reaching thermal conductivities around $35\text{-}40 \text{ W} \cdot \text{m}^{-1} \cdot \text{K}^{-1}$.

For the matrix, polyamides 6 and polyamides 6/6 are two of the most common materials used since they show good mechanical and thermal stabilities around 90°C and 150°C and have great oil and chemical resistance [5]. Polyethylene (PE) or poly(butylene terephthalate) (PBT) are other thermoplastics usually used which manifest some of the best thermal properties among the polymers [6].

For the fillers, the more appealing are graphite, carbon nanotubes (CNT), carbon fibers and metal particles. When electrical insulation is also important for the application, the best options change to boron nitride (BN) or aluminum nitride (AlN) [3].

Moreover, there are various factors that limit or promote the enhancement of the thermal conductivity of the material, regarding the filler type, the composite morphology and the interaction between the materials.

2.2.1 Filler Shape

The shape of the filler sometimes can be more important to the thermal conductivity than the choice of the filler itself. Using fillers with random shapes, rough areas or edges can result in composites with higher viscosity and hinder the processing and the mixing of the materials [3]. So, smoother fillers help the making of the composites.

In terms of increasing the thermal properties, fillers with high aspect-ratio have the best results, hence, spherical fillers, despite having less edges and improving the fabrication of the composite, do not excel when comparing with fillers with tube-like or platelet-like shapes that easily create paths to transfer the phonons more efficiently. Furthermore, in general, sphere-like fillers are limited to increase the thermal conductivity to, at most, 20 times the matrix thermoplastic pure conductivity whereas fillers with higher aspect-ratio do not have that limitation [4].

2.2.2 Hybrid Fillers

Studies have confirmed that using hybrid fillers with different shapes can improve the thermal properties of the composite. Heat transport is done by phonon vibrations [7]; therefore, it is vital to have an efficient thermal conductive network. The synergetic effect that hybrid fillers produce makes it easier to create those networks by forming bridges between, for example, 2-D fillers such as hBN platelets with 0-D fillers such as AlN particles [8]. Besides that, this type of systems allows a reduction in the filler loading which result in lower viscosities and also increase the packing density of the particles [8].

In anisotropic materials, hybrid systems are very important because the packing and the increased network improve the through-plane thermal conductivity, optimizing the heat transfer.

2.2.3 Filler Size

The size of the fillers has been studied over the years but there are still controversial remarks due to the difficulty in isolating just this topic since particles of different sizes interact with each other differently and sometimes have to be prepared with different methods. Still, bigger fillers are considered a better option than smaller ones due to the smaller interfacial area

between the filler and the matrix. Even though smaller fillers create better thermal conductive networks, the higher matrix-filler interfacial area induce phonon to scatter and the thermal transport to be harder [4,9].

However, nanoparticles can work really well as fillers despite being smaller than the usual microparticles used. This is presumed to be due to its different chemical interaction and to act as bridges between microparticles whose path was not that clear [3].

2.2.4 Filler Quantity

Since the fillers are the contribution to the composite regarding the thermal conductivity, the higher the filler volume fraction the higher is this property. For low filler volume fractions, up to the percolation threshold, a network is not still formed, thus the thermal conduction does not improve significantly. From a point on, higher than the percolation threshold, the network is completely available and the conduction increases sharply.

Having said that, it is important to note that higher volume fractions also implies higher viscosities and harder processing which result in a brittle composite [3].

2.2.5 Dispersion

Just like the size of the fillers, their dispersion is also a factor hard to isolate and there is still some lack of knowledge about this theme. However, from previous studies, it is known that an optimum must be found, that is, despite perfect dispersion bringing out paths with less distance between all the fillers, consecutively, there are more interfaces to cross. The other way around, with lots of agglomerations, the distance between the agglomerates is higher, making it hard to create a clear path to the phonons since the matrix is thermally insulator. Therefore, a system with semi agglomeration may create the most efficient network in the composite [4].

2.2.6 Interactions

One of the reasons why thermal conductivity may be lower in some composites is because of phonon scattering through the composite. This can happen if, for example, the interfacial strength is weak and the interactions between filler-filler or filler-polymer are not strong enough. So, it is beneficial to have covalent or hydrogen bonds between the fillers and matrix instead of Van der Waals bonds and improving these bonds by adding components that functionalize the fillers is a good alternative [10]. There are many procedures and components possible to add, which reinforce these bonds, like using hydroxide compounds to induce hydrogen bonds, as well as using acids or peroxides [9].

2.3 Heat Sinks

Using the design of the device to aid the heat transfer is as important as choosing the right material. In that regard, many studies have been made of what are the best structures or resources to be used to that matter. It happens that heat sinks are the main methodology adopted in electronic cooling since is a simple, low-cost and efficient system that takes advantages of extended surfaces made of high thermal conductive material to direct the heat. The two most used and basic geometries for heat sinks are parallel plate fins or pin fin, but both are suitable for optimizations as well as substitution by other topologies depending on the work case and conditions.

2.3.1 Flat-Plate Fins

Broadly used in heat sinks, flat-plate fins present a basic geometry, as shown in Fig. 1, with many possibilities of optimizing, starting with its dimensions which have more effect in the heat removal when the Reynolds number (Re) number is low, while the number of fins is more relevant when Re is higher [1].

The thickness of the fin t has an optimum value for each case because it should not be too wide so that it limits the fluids to flow efficiently in the channels between the fins, but it also should not be too narrow so that the surface area required to the heat transfer is reduced [1].

Regarding the characteristic dimensions H , D and W (see Fig. 1), the bigger they are, the larger is the heat surface area and, consequently, the heat performance of the heat sink. However, increasing these dimensions also means sharply increasing both the weight and the cost of the heat sink and the thickness influence has been proved to be more influential, in general.

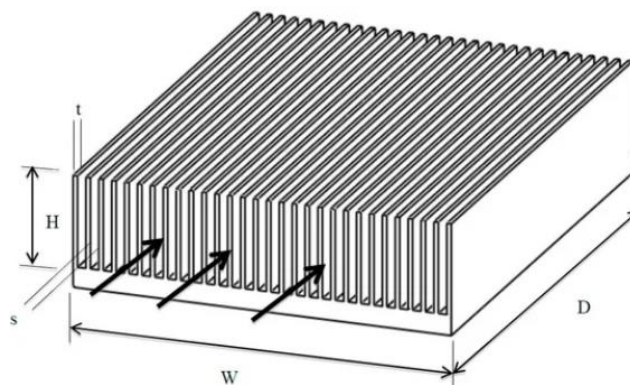


Figure 1- Flat-Plate fin heat sink [11].

Increasing the number of fins results in more heat surface area and can present a solution to reduce the bypass airflow effect but if that means reducing the channel spacing, it can lead to higher pressure drops and inhibit flow turbulence. Another alternative is implementing oblique-plate fins that allow larger surface areas and accelerating flow in the channels. Results show

that there is a reduction of at most 6 °C on the maximum temperature of the device when compared with normal flat-plate fins [1].

Various configurations have already been studied to optimize the heat performance from creating interruptions along the plates and distribute them in-line or scattered to cap them, as represented in Fig. 2. From those studies, it was proved that including the interrupted fins closer to each other did not improve the heat transfer and despite capping the fins revealed good results, this did not compensate the weight increase. L-shaped fins revealed being a good alternative that did not imply the extra weight [12].

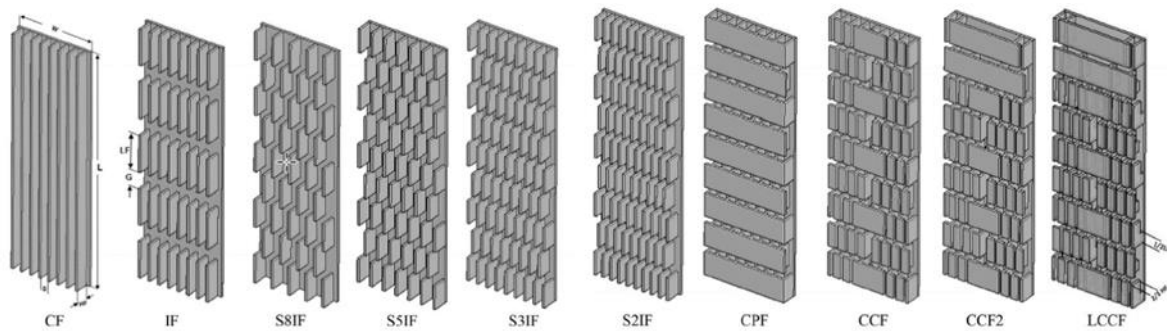


Figure 2- Flat-Plate fin heat sinks with interrupted surfaces, capped and L-shaped fins [12].

The cross-sectional area can also be varied and does not need to be rectangular and uniform along the height. Studies have tested cross-sections of trapezoids or triangles and the latter proved to enhance the performance of the process [12]. Another work showed that using reverse-trapezoidal fins reduces the thermal resistance as much as 15%, for certain thicknesses [13]. Branched-fins heat sinks, especially Y-shaped fins, can reduce the thermal resistance to at most 30%, when compared to original rectangular-shape fins, but this improvement is not so evident when the pumping power is decreased or when the heat sink length increases [14]. Moreover, using fins in which the cross-sections have variable thickness may be a problem if the processing of the heat sink is done using injection molding.

Perforations with different shapes and in different locations can interfere in the performance of the heat sink. For the geometries studied (circle, rectangle and triangle) the rectangular-shaped longitudinal perforation made at a height of 30% of the total height of the heat sink resulted in the best solution [12].

2.3.2 Pin Fins

Pin fins is the second main geometry used for heat sinks and can also be optimized in terms of dimensions and shape. Implementing pin fin heat sinks with height that increases linearly along the direction of the airflow revealed a lower maximum temperature [15]. Optimizations concerning other dimensions, such as width, are as much as important and pins with different widths along the heat sink, as represented in Fig. 3, can reduce the overall thermal resistance

of the heat sink [16]. Pins with variable width along the fin height, similar to how it was already mentioned to the flat-plate fin heat sink, can enhance thermal dissipation of the heat [12].

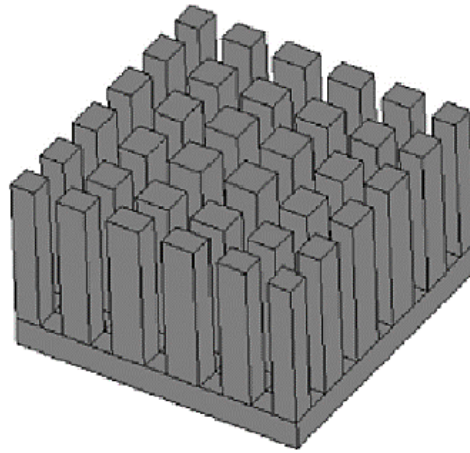


Figure 3- Pin fins with different widths [1].

The shape of the pin base is also very influential in terms of performance of the pins and among all the geometries it is important to analyze the most basic ones first. Rectangular, square, circular and hexagonal shaped fins were studied, as represented in Fig. 4a, and the results show that the circle is the geometry which allows the highest Nusselt number (Nu) for the range of Reynolds numbers experimented. After that, hexagon presents the best results, following the square and in the end the rectangular ones [17]. Circular shaped pin heat sinks were also compared to other geometries like a hydrofoil pin and a cone pin heat sink, as represented in Fig. 4b, and the conical shape showed the highest heat transfer coefficient among the 3 because of the enhancement in the flow mixing process. These pin heat sinks, when compared to flat-plate heat sinks, were 1.6 to 2 times better in terms of thermal performance [18].

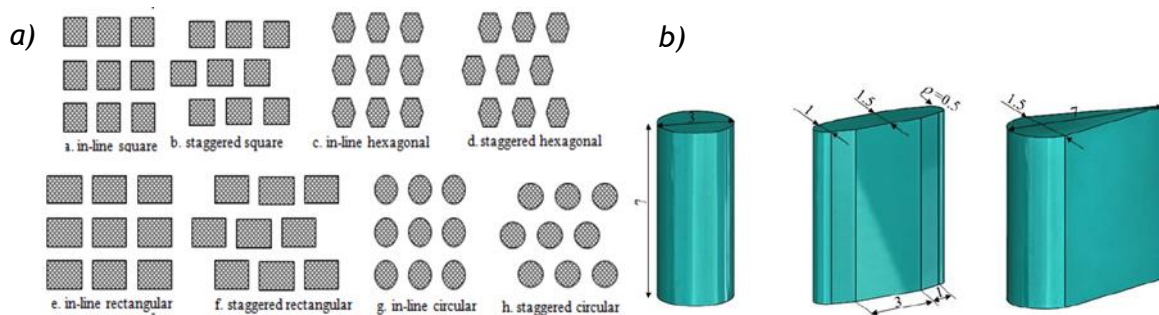


Figure 4- a) Basic geometries and dispositions for pin fin heat sinks [17]; b) Hydrofoil and cone pin fin heat sink [18].

Regarding the disposition of the pins in the heat sink, in-line and staggered pins manifest different thermal performance, with staggered fins being superior than the first due to the induction of more flow turbulence that lead to more efficient mixing and heat transfer [17].

Perforated pins can help reducing the pressure drop and the heat sink weight and they can come with many variations, number of perforations, shape of the perforation and location being some of the relevant degrees of freedom.

In a research where 1 to 5 perforations were used for each pin, as observed in Fig. 5a, the increase in the perforation number reduced the pressure drop. In terms of shapes, a pin fin heat sink with an elliptical perforation can improve the heat transfer rate of at most 40.5%, comparing to a solid pin fin heat sink [12].

Instead of perforations, slots or notches are an alternative that can also reduce the pressure drop and increase the heat transfer area. When comparing slotted pins with two different geometries, cylindrical and kidney, the slotted kidney pin exhibited the best heat transfer rate [12]. In another research, it was concluded that enlarging these perforations, as observed in Fig. 5b, accompanied the increase in the heat transfer rate and the decrease of the pressure drop, hitting a growth of the heat transfer performance of 10% and a reduction of the fan power consumption of 30% [1].

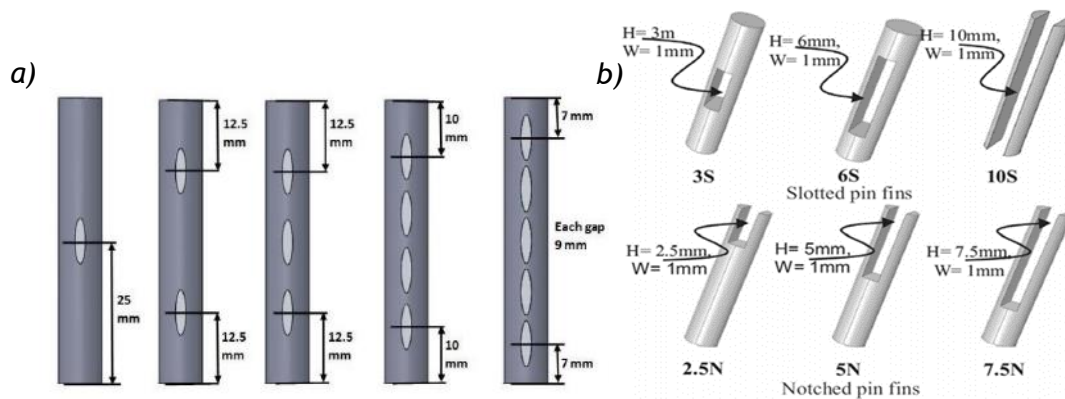


Figure 5- a) Elliptical perforations on pin fin heat sink [12]; b) Slotted and notched pins with different perforation sizes [1].

2.3.3 Microchannel Heat Sink

Using microstructures as heat sinks has been one of the most promising options in recent years, accompanying the tendency of device miniaturization. These microchannel heat sinks (MCHS) are compact and the thermal performance relative to its volume is high since the heat transfer coefficient increases with the decrease of the channel hydraulic diameter, reaching a heat transfer rate 50 times higher than usual heat sinks [1,2]. However, this type of heat sink is characterized by its high pressure drop resulting from the size of the channels. For solving this problem, increasing the MCHS porosity or changing the pumping system may help, but affecting the fabrication cost [1,2].

Implementing ribs on the MCHS can also help to reduce the pressure drop penalty while increasing the Nusselt number and heat transfer rate. From the rib's tests, the triangular shape was the best. Six different configurations for the edges of the heat sink were also investigated, as represented in Fig. 6, and the inverse trapezoidal one revealed the best performance and highest Nusselt number comparing to the other configurations [12].

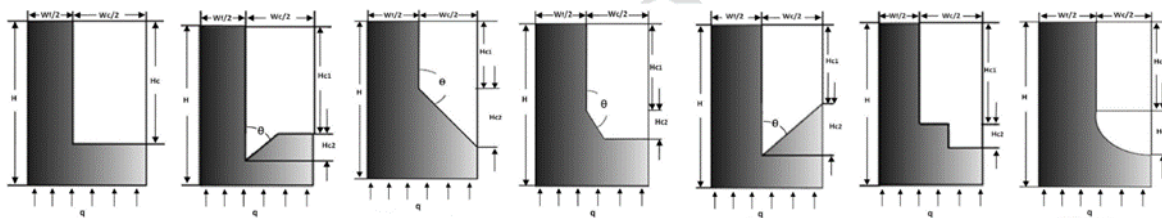


Figure 6- Different configurations for MCHS [12].

2.3.4 Other Designs

Many concepts were investigated over the years and were proved to work under special circumstances, while some disadvantages can be challenging.

Using plain-weave screen laminates with porous structures, such as shown in Fig. 7a, to lower the thermal resistance is one of the concepts tested that ended up being worse than the original heat sink [19]. With a frontal flow velocity below $3.3 \text{ m}\cdot\text{s}^{-1}$, screen laminates heat sink approximates to the plate fin but with higher velocities they manifest poor performance because of the sharp decrease in the pores efficiency at these flow regimes [11].

Converging-diverging heat sinks, like the one represented in Fig. 7b, seek to manipulate the fluid movement, creating disturbances and secondary flows that interfere with the thermal and hydraulic boundary layers. This concept works well and the theory is proved correct but these converging-diverging areas lead to high pressure drop in a way that this disadvantage is more crucial than the advantages already mentioned [11].

Honeycomb heat sinks, as seen in Fig. 7c, are a simple alternative to the original flat-plate fins with an increase in the heat transfer area, with the use of hexagons instead of squares for the cross-sectional shape. The main possible problem of this type of heat sink is the fabrication because if the structures are bound together by thermal adhesives, this can prejudice the performance of the heat sink [20].

coefficient but a large pressure drop penalty too. Three other configurations were investigated and not only the heat transfer coefficient increased but the pressure drop decreased too [22]. Apart from the pressure drop, the difficulty to shape the foam and the overall cost, depending on the fabrication process, can be too high currently to implement in heat sinks commercially [11].

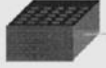
Foam Geometry		Heat transfer coefficient, W/m ² ·K			$\Delta P/L$, kPa/cm			Thermal resistance, °C/W		
		Al-air	Foam-air	Foam-water	Al-air	Foam-air	Foam-water	Al-air	Foam-air	Foam-water
Solid foam		250	2600	23000	<0.14	5.4	5.4	1.3	0.13	0.02
Finned		70	1000	2100	<0.14	<0.14	1.36	1-5	0.38	0.19
Pin-fin		550	1500	2500	<0.14	0.14	1.36	0.7	0.26	0.15
Blind-holes (pin fin negative)		N/A	2000	4600	N/A	2.71	1.36	N/A	0.19	0.09
Blind-holes (parallel to air flow)		N/A	3100	4500	N/A	0.95	1.36	N/A	0.13	0.09
Corrugated		N/A	4100	9500	N/A	0.27	0.09	N/A	N/A	N/A

Figure 8- Comparison of the heat transfer coefficients obtained for graphite foam and aluminum foam heat sinks [22].

2.3.6 Enhancements

The use of interrupted surfaces along the heat sink in order to renew the boundary layer allows a more efficient heat transfer. Louver and slit fins were tested and the results showed that for high frontal velocities and wider channels between fins, these interruptions improve the process. Louver form revealed better results with a fin spacing of 1.65 mm than slit form, but with spacings below that, the pressure drop of louver fins is generally higher than slit fins. Besides this, for smaller fin spacings than that, around 0.8 mm, and at low Reynolds number flows, below $Re = 600$, fully developed regions are created and the performance is poor. The duct flow effect happens in this kind of environment, i.e. with low frontal air velocity, making the flow to not pass through the interruptions but to flow along the channel, as it can be seen in Fig.9 [24].

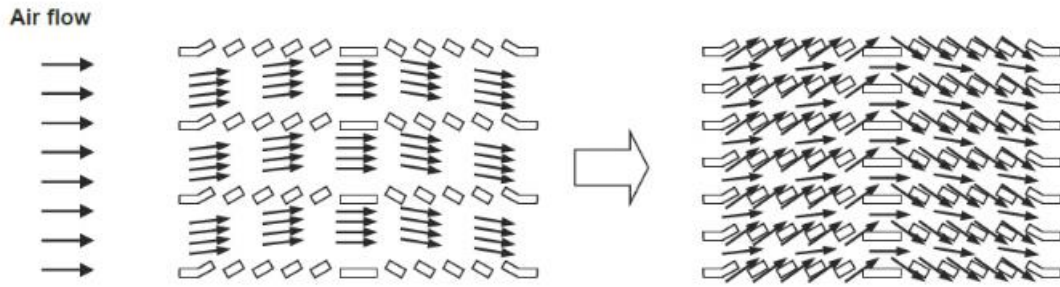


Figure 9- Duct flow effect at low velocities vs. efficient flow trade at high velocities [24].

Manipulating the flow in order to change its directions and intensity or creating secondary flows enhances convection. Longitudinal vortex generators (LVG) takes the most of it by using smaller extended surfaces in the channels to create vortices that form and enhance secondary flows, reducing the thermal resistance.

The advantages of LVG relatively to interrupted surfaces is the reduction of the pressure drop penalty and the efficiency in the developing regions, that is substantially better. However, in the fully developed region or with lower frontal velocities these surfaces do not perform that well, as they do not in tight channels because of the lack of space to create swirls and vortices. Different configurations of LVG were investigated by changing the shape (delta wing, triangular winglet, semi-circular and dimples) and the number of LVGs, as represented in Fig. 10a. First, it was concluded that semi-circular shaped LVGs manifested the highest pressure drop of all the cases. Second, while in large channels, with widths around 1.85 mm, all the LVG improve the original heat sink, in thinner channels, with widths around 1.00 mm, the plain heat sink is usually the best. Also, lower number of LVGs grants less pressure drop penalty [11].

The bypass airflow effect is related to the air that does not flow through the channels of the heat sink but flows either sideways or over it. If the design of the heat sink is optimized to take advantage of this effect, the temperature difference between the fluid and the heat sink remains higher from the front to the rear, increasing the heat transfer rate. For that, some of the frontal area in the heat sink is sacrificed so that more air flows over it and only entered in the heat sink closer to the rear, still cold. That loss of heat transfer area reduces the pressure drop, being important to find the right design in which the lack of performance in the front of the heat sink on the account of the lost area does not overthrow the advantages in the rear of the high temperature difference [11].

Different types of configurations were assessed from the original plate, to steps and then to ramps, some of them represented in Fig. 10b. From the investigation it was possible to conclude that the heat sink with a ramp until 2/3 of the total length, called “trap 1/3”, revealed the lowest pressure drop and highest heat transfer coefficient. In addition, this heat sink has 20.6% less surface [25].

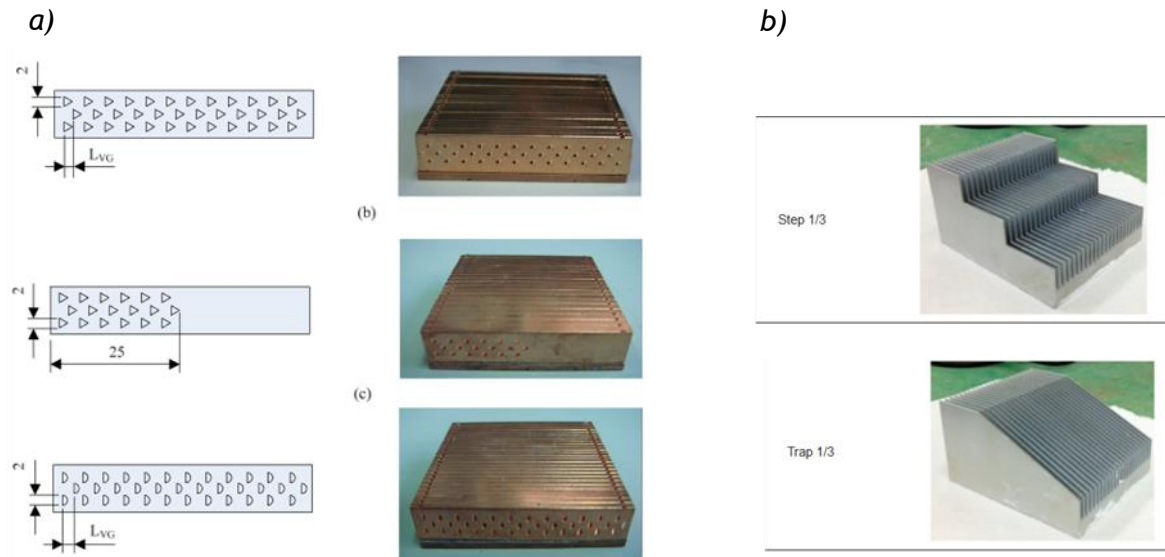


Figure 10- a) Heat sinks with delta wing LVGs (full and partial) and semi-circular LVGs [11];
b) Bypass airflow in heat sinks [25].

2.3.7 Plate-Pin Fin Heat Sink

In order to optimize the already existent flat-plate fin heat sinks, manipulating the flow to enhance turbulence and consequently improve the heat transfer performance, can be made by implementing pins, in-line or staggered, between the fins, as observed in Fig. 11a, acting almost like LVGs [26]. These new plate-pin fin heat sinks can be fabricated by modifying a plate fin heat sink and are able to reduce, according to simulations, the thermal resistance by 30% when comparing to the original plate fin heat sink [26,27].

However, these benefits come with a much higher pressure drop penalty due to the pins acting as obstacles to the airflow, even though the plate-pin fin heat sink ends up needing less pumping power when the goal is obtaining the same heat dissipation power [26,27]. Pins with different shapes were investigated and simulated and from the results it was possible to conclude that even if the plate fin and the pin fin heat sinks suffer much less pressure drop than this new design, the elliptic shaped pin for the plate-pin fin heat sink approximates to those values while granting a high Nu [27].

Another research utilized ribs instead of bigger pins, as represented in Fig. 11b, and concluded that the ideal number of ribs per channel is one since more ribs prevent the creation of vortices and disrupt the disturbance in the flow, combined with extra costs. Despite this, it was also proved that for the same performance, the addition of ribs allows a reduction of the pumping power of 69.65% and if the goal is reducing the fin number, the conclusion was that with new fin number and old fin number ratio of 0.6, and with an increase of ribs, the Nusselt number was 1.55 times better than the original heat sink and the material used reduced by 27.24% [28].

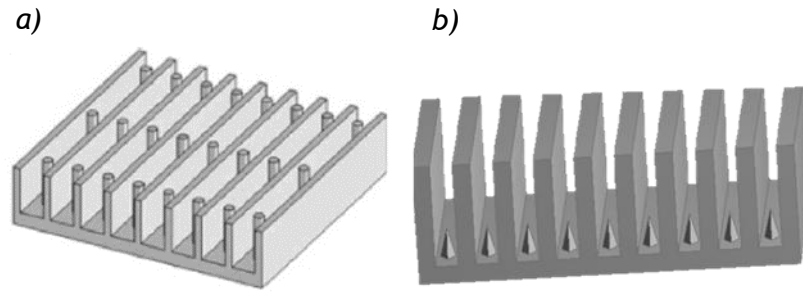


Figure 11- a) Plate-Pin fin heat sink [27]; b) Flat-Plate fin heat sink with ribs [28].

2.3.8 Anisotropy

One of the main features to consider when changing from metals to polymers in electronic cooling is that while metals are thermally isotropic, i.e. the thermal conductivity is the same in all the directions (and the heat transfer rate is also), the polymers are typically anisotropic. Anisotropic materials manifest their properties differently in the in-plane direction, in the through-plane and in the cross-plane directions [29]. With that in mind, in the processing of the device and the heat sink, the direction with the highest thermal conductivity must be aligned with the orientation of the heat flux, as can be seen in Fig. 12 a) which has high temperatures at the base of the heat sink due to low thermal conductivity in the Z-direction, which lead to a higher maximum temperature [6].

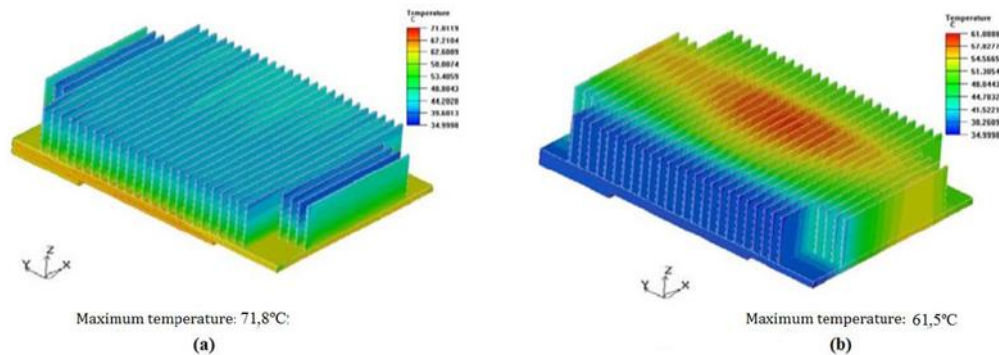


Figure 12- Heat transfer rate profile in anisotropic material heat sinks with different preferential directions [6].

2.4 Fluid Cooling

The second methodology that is widely used are fluids, gases and liquids, to transfer the device heat generated to the outside via convection. There are four types of system level cooling that are going to be mentioned: air cooling, hybrid air-water cooling, liquid cooling and refrigerant cooling. Besides these, heat pipes, Peltier coolers or interface materials are also very important to be mentioned.

2.4.1 Air Cooling Systems

Using air as a cooling system, both in natural or forced convection with a fan, is the most used method in electronic devices because of the simplicity of the implementation, the low cost, the availability of the fluid and the low density that end in a lower pressure drop penalty [27].

Since the air convection typically does not have high heat transfer coefficients, optimizing the system is essential. This could be worked out with a higher fresh flux, i.e. more speed and power in the fan to renew the air or a heat exchanger that cooled it, but it would increase the cost. So, to achieve maximum performance of air-cooling systems, an alternative is to optimize the inlet and outlet positions of the flow to get the best contact with the heat sink. There are diverse articles investigating many flow inlet and outlet positions that reach the lowest maximum temperature possible and great temperature uniformity [1,30].

2.4.2 Hybrid Cooling Systems

Hybrid cooling systems use air as a means to dissipate the heat of the heat sink and the device but then the air is cooled in an air-water heat exchanger, optimizing the process and allowing the use of a serial system, rather than the usual parallel process, since the air is constantly cooled. In this case, the system uses heat exchangers between modules and can absorb near 50% of the heat transferred between the air and the previous heat sink/module [2].

2.4.3 Liquid Cooling Systems

More powerful devices usually produce more heat which requires more dissipation capacities that air might not be able to support with its lower thermal performance. In those situations, using liquids, which have higher heat transfer coefficients, should be considered. This can be fabricated in two ways: indirectly using water cooled plates or other structures that absorb the heat or directly with immersion in dielectric liquids like fluorocarbon fluids that exhibit good electrical and chemical properties, reducing the risks of electrical discharges [1,2]. Adding to the risk of liquid leaking that could cause electronical hazards, liquids are denser and more viscous than air. This leads to a need of more pumping power for the fluid to be moved and distributed through the device and an additional pressure drop penalty not included in air-cooling systems [27].

2.4.4 Refrigerant Cooling Systems

Implementing the Carnot cycle in a electronical device is efficient because it enhances the performance of the electric circuit, but at the same time is hard because it needs to have at least a compressor, a valve, an evaporator, a condenser and a refrigerant in which the process will take place. So, in small packages, it is usually not the best alternative despite the advantages.

3 Computer Applications

3.1 Siemens NX™

Siemens NX™ is an advanced software used mainly for designing, modeling and engineering modules and was used in this dissertation to create the model of the box with all the components inside and the various models of the heat sinks that were studied and simulated.

There was a need of simplifying the model to allow sufficient time to make simulations since both the complexity of the project and of the CFD software used requires a large time to produce results. The produced models resulted in a box with two types of gates for the fluid to flow through: one side of the box has circular perforations with 3mm of diameter; the other has a hexagonal grid corresponding to the fan that generates the flow. Inside the box, the nine ICs that are the heat sources are in their specific places, in contact with the heat sink, that has the function of dissipating the heat produced by them.

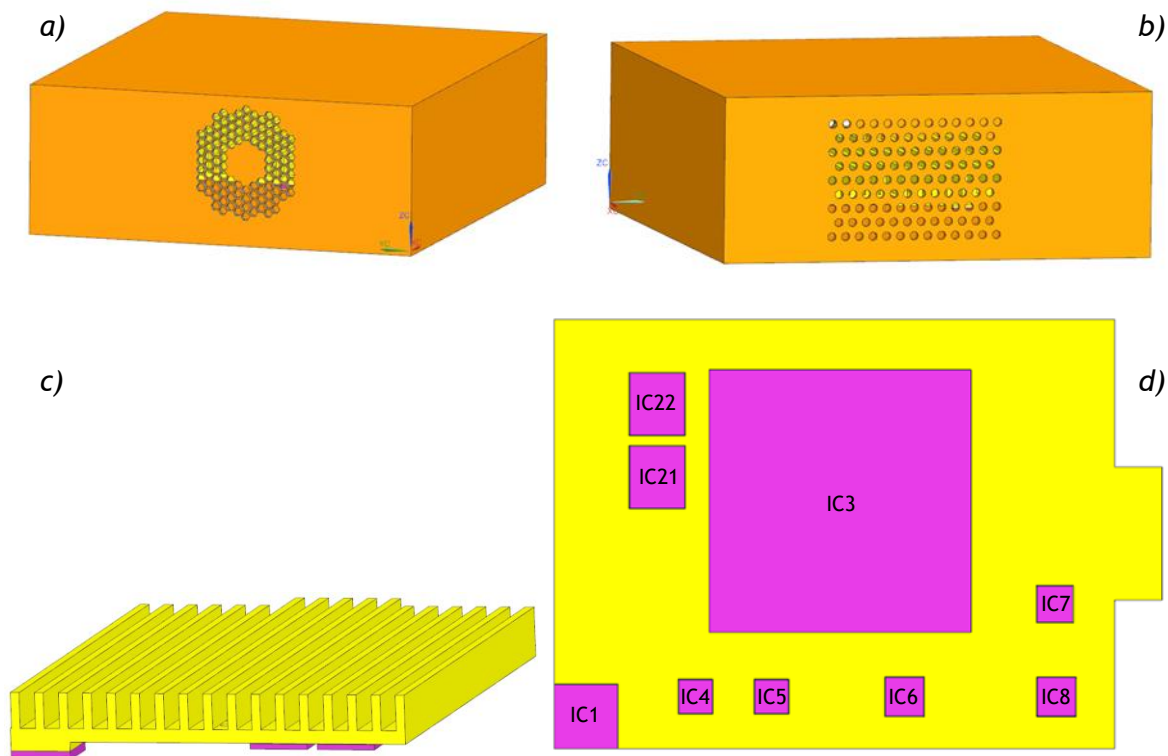


Figure 13- a) Box's face with the grid; b) Box's face with the perforations; c) Original heat sink with ICs; d) ICs numbered under the heat sink.

3.2 OpenFOAM®

3.2.1 snappyHexMesh

The opensource OpenFOAM® software prepares the problem and solves it by reading files from 3 specific folders (0, constant and system) and, with the proper information written in the files, makes the requested calculations and can present the solutions in diverse formats.

The 0 folder is the location of files that indicate the initial conditions for the variables in study, both inside the mesh and in each of the boundaries, as well as how each variable evolves in the boundaries.

The constant folder is the location of files that indicate mesh characteristics and physical properties such as thermal conductivity or models like turbulence models or heat production.

The system folder is the location of files that include information about the solvers used to obtain the solution, numerical schemes, like discretization methods, used to make the calculations, dictionaries used in the problem solving and time control definitions.

To make simulations with simple geometries in OpenFOAM®, it is used the “blockMesh” utility to generate the mesh. The information used by this utility is obtained from a dictionary file, blockMeshDict, located in the system folder, where the geometry points, edges, faces and their main functions are written.

However, to generate complex three-dimensional (3D) meshes in OpenFOAM®, imported from design software such as Siemens NX™, a more complex utility such as “snappyHexMesh” should be used. “snappyHexMesh” works by creating the complex imported mesh inside a simpler mesh, usually a parallelepiped, generated by “blockMesh”. To allow an error-free mesh creation, the complex geometry must fit in this solid, i.e. the dimensions of the parallelepiped must be higher than of the geometry.

After using “blockMesh”, the user must utilize the “surfaceFeatures” utility for OpenFOAM® to read the geometry STL files so that it can be generated. These files have to be in the triSurface folder located inside the constant folder and this utility extracts the geometry points and sharp edges and, therefore, helps in the creation of a better mesh. It is important to notice that there should be a STL file for each region that have specific characteristics and variables that are going to be studied, so that they can be implemented in the future. Again, so that “surfaceFeatures” can work, a dictionary surfaceFeaturesDict must be written with the names of the STL files to be extracted.

“SnappyHexMesh” can be a time-consuming utility depending on the mesh complexity. With that in mind, rather than using a single processor, the case is usually divided in the number of processors available in the computer and execute this utility in parallel. This is available by

writing in the `decomposeParDict` file the number of processors to be used, in this case 6, and then use the utility “`decomposePar`” after using “`surfaceFeatures`” so that a folder for each of these processors is created.

Then, it is time to use the “`snappyHexMesh`” utility to generate the mesh inside the solid. The `snappyHexMeshDict` is the biggest dictionary used in this project and has many configurations and definitions that must be adjusted in order to create both a refined and viable mesh according to every step through “`snappyHexMesh`”. The mesh generation begins by refining specific edges and features of the regions written in the dictionary according to certain specifications like the features refinement level per region (which usually goes from 0 if no refinement is needed to 5 if there is a need to a high refinement in a specific region), maximum number of cells and number of cells between levels of refinement. After these features are refined, all the surfaces associated with them also suffer the same treatment, according to the minimum and maximum refinement level needed for each of the regions. In this step, the user must also write an `InsidePoint`, i.e. a point inside each of the regions so that region cell zones are created. After all cell zones are generated and the regions are completely closed the utility removes all the cells outside the mesh that are no longer needed.

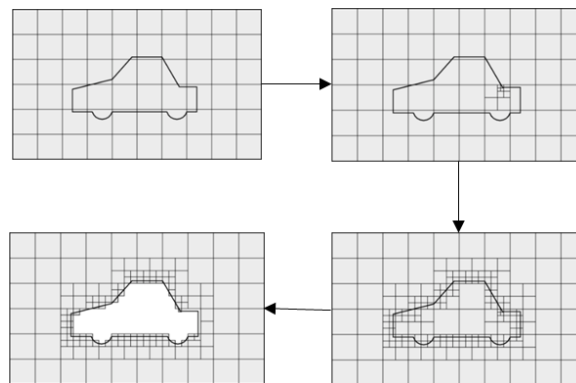


Figure 14- Illustration of the `snappyHexMesh` process [31].

The process to get “`snappyHexMesh`” to run without any errors was time-consuming because there were various issues that had to be understood and corrected adequately.

The main issue was that when trying to generate the box, the heat sink and all the nine ICs, the internal mesh ended up being only the parallelepiped from the “`blockMesh`” utility. First, there was a necessity to make various corrections in the geometry in Siemens NX™ in order to obtain a decent mesh which closes perfectly without leaks when generated with “`snappyHexMesh`”. There was also a need to understand the concept on how to make the fluid in OpenFOAM® since the inlet of the fluid should be the grid in the box and the outlet the perforations on the other side of the box. After many attempts, the solution was to create a box outside the actual box, as close as possible, with the walls with just a 0.1 mm of distance,

to act as boundaries for the case. The inlet was considered an octagon with the dimensions of the grid and the outlet was a rectangle in the face of the perforations, represented in Fig. 15. Since Siemens NX™ just exports objects, without the option to export individual walls, the text file of the outside box had to be divided in three different files, one for the inlet, one for the outlet and another for the rest of the walls so that they could be differentiated in OpenFOAM®. With this solution, when the “snappyHexMesh” was used, the two boxes were so close that they fused together, making the grid the inlet, the perforations the outlet and the walls of the inside box the boundaries.

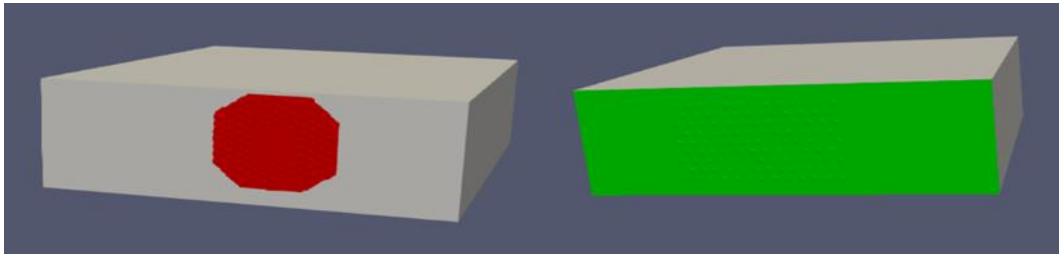


Figure 15- Illustration of the inlet face (left) and the outlet face (right).

After the mesh is generated, the simulation can be done. Since the geometry was very complex, each “snappyHexMesh” process took approximately 5 hours which resulted in simulations of $\Delta t = 1$ s taking 2 hours and 30 minutes. This is impracticable since each case studied needed around 3 simulations only to calculate the operation point and then one more simulation in the operation point.

To circumvent this problem, a Printed Circuit Board (PCB), where the ICs were put, was left out of the final mesh and some quality of the mesh was sacrificed, without compromising the results, bringing the refinement levels from 3-5 to 1-3. However, as some regions are especially complex, like the grid, with this refinement levels they kept creating bad cells that leaked. A refinement region that consists in a region limited by the user in which the mesh inside is more refined than the rest of the mesh, as represented in Figure 16, was used around the grid to focus the refinement there. After the mesh is generated, the process ends with smoothing the geometry, according to the snappyHexMeshDict. After this, the utility “reconstructParMesh - constant” is used to combine the information of the generated mesh located in the processors folders back to the constant folder. The final mesh is shown in Figure 17.

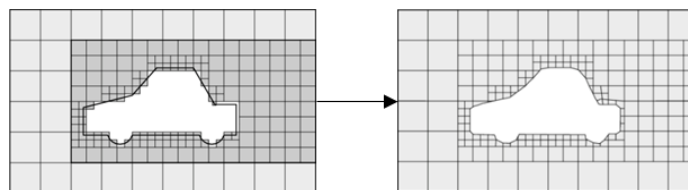


Figure 16- Process of the refinement box creation [31].

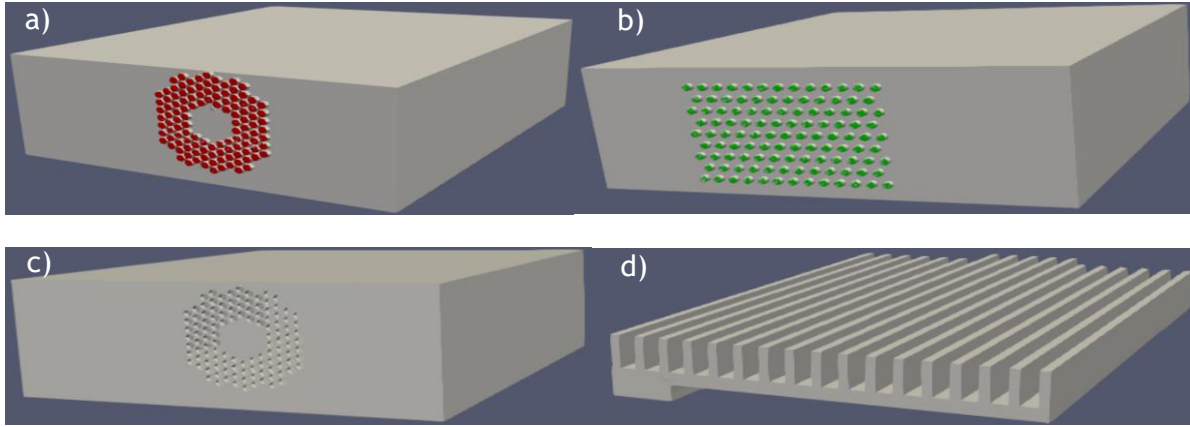


Figure 17- a) Inlet face; b) Outlet face; c) Box; d) Heat sink with ICs.

3.2.2 chtMultiRegionFoam

With the mesh generated, the processor folders must be deleted. Now, it is time to use the “splitMeshRegions -overwrite -cellZones” utility to divide the mesh into the regions that are supposed to be studied, so that it is possible to control how each region and the boundaries between them act. With this utility, a folder for each region is created in the 0, constant and system folders so that the respective files for that region have a specific location. From here on is the preparation of the problem so that it can be simulated afterwards. First, in the constant folder it is necessary to include a regionProperties file in which the user indicates whether each region is a solid or a fluid. Also in the constant folder, in each solid region folder it is necessary to have a thermophysicalProperties file with the principal region properties, from the material density, ρ , its heat capacity at constant pressure, C_p , and the thermal conductivity, k . Since this project is based on thermoplastic composites, the thermal conductivity is anisotropic, so, in this file there must be a coordinateSystem written with the type of coordinate used and its directions. This behavior must be specified, as well as every other consideration like constant density or type of mixture. In this case, for the box it was considered $\rho = 1360 \text{ kg}\cdot\text{m}^{-3}$, $C_p = 1500 \text{ J}\cdot\text{kg}^{-1}\cdot\text{°C}^{-1}$ and $k = 2.5 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ in-plane (x and y) and $k = 0.5 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ through plane (z). The heat sink and the ICs, in other hand, have a $\rho = 1850 \text{ kg}\cdot\text{m}^{-3}$, $C_p = 1250 \text{ J}\cdot\text{kg}^{-1}\cdot\text{°C}^{-1}$ and $k = 20 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ in-plane and $k = 6 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ through plane.

To enable the heat generation in each of the ICs, in their folders there must be a fvOptions file where the duration of the heat production and its power per volume of region is provided. In Appendix A presents the power of the ICs used.

For the fluid region folder, there must be a thermophysicalProperties file with the fluid properties, which is air in this case, a radiationProperties file indicating whether or not a radiation model is considered (it was not included), a momentumTransport file where it was written the model considered for the fluid (the k - ϵ turbulence model was used), and a fvOptions

file where the temperature of the fluid for the first iterations was limited between 65 °C and 127 °C, to prevent initial destabilizations.

In the 0 folder, each region folder must have a file for each variable to be used in the simulation and their initial values. The solids and the fluid started these simulations at the same temperature, $T = 65$ °C. Regarding the velocity and pressure of the fluid, the relative pressure in the outlet was considered 0 Pa and at the inlet various entrance velocities were given so that after the simulation was done, the average pressure at the inlet was calculated (using “postprocess -func ‘patchAverage(name=inlet,p)’ -region fluid”). With the pressure loss across the device for each inlet velocity, a curve of the fluid behavior was created and intersected with the fan curve, obtaining this way an estimate of the operation point for each case. The parameters for the k- ϵ turbulence model used were the default found in OpenFOAM tutorials, $k = 0.375 \text{ m}^2 \cdot \text{s}^{-2}$ and $\epsilon = 0.54 \text{ m}^2 \cdot \text{s}^{-3}$.

With all the regions prepared, it is time to use the simulation utility but, as it is also time consuming, it is better to use “decomposePar -allRegions” to simulate the problem in parallel (using the command “mpirun -np 6 chtMultiRegionFoam -parallel”).

During the simulation, the user can accompany the stability by monitoring the Courant number (Co), which is described by

$$Co = \frac{\Delta t U}{\Delta x} \quad (4)$$

where Δt is the time step size, U is the velocity ($\text{m} \cdot \text{s}^{-1}$) and Δx is a mesh cell characteristic dimension (m). This parameter should be low, preferentially below 1 and it is useful to understand if the time step size chosen is too big for mesh size.

At the end of the simulation, the command “reconstructPar -allRegions” can be used to fuse the parallel results. Afterwards, the command “paraFoam -touchAll” is used to create paraFoam files for each region and the results are ready to be analyzed in paraFoam.

To calculate the operation point, 3 simulations with different inlet velocities are done for each case and for the 2 points closer to intersect the fan curve a fit of the following form is used,

$$\Delta p = a U^2 \quad (5)$$

where Δp is the pressure drop in the device and the operation point is estimated. The provided fan curve was given as Δp vs. air flow rate (Q) but in the plots it was decided to convert to Δp vs. U where U is the inlet average velocity. The temperature results presented in the following tables are displayed with two decimal places to facilitate the comparison between the cases, but the accuracy of the computed temperature results is not that high.

4 Results and discussion

4.1 Solver Verification

Since there is no information about the fluid flow behavior, a transient simulation using the first order implicit Euler method was initially done. However, when the transient problem was simulated it took approximately 8 entire days to finish and achieve a stabilized temperature field, which is impracticable due to the limit time available. The alternative found was to test the same problem with the same conditions with a steady state solver and verify if the solution is the same and the approximation to steady state is suitable. Using this approach, the simulation took 2 hours and 30 minutes to stabilize and the results revealed that while using the Euler solver the final average temperature of the heat sink was 127.12 °C, using the steady state solver it was 127.57 °C, as represented in Fig. 18 and 19.

The approximation between the two models has, then, a difference of 0.11% and, with these results, it was validated the use of the steady state solver in the following simulations.

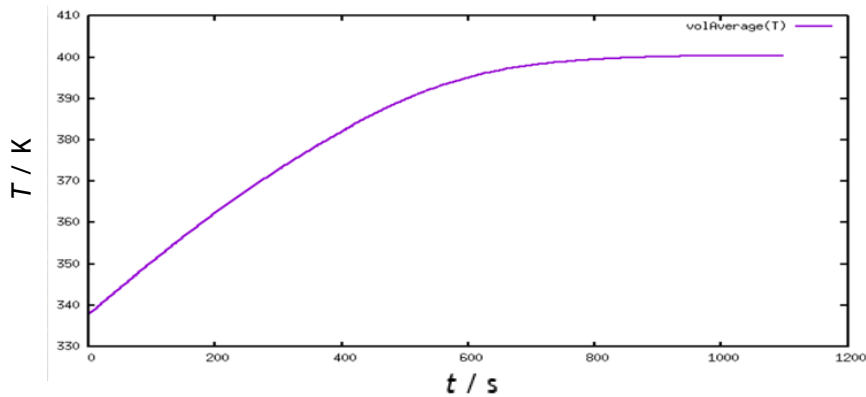


Figure 18- Heat sink average temperature for Euler solver.

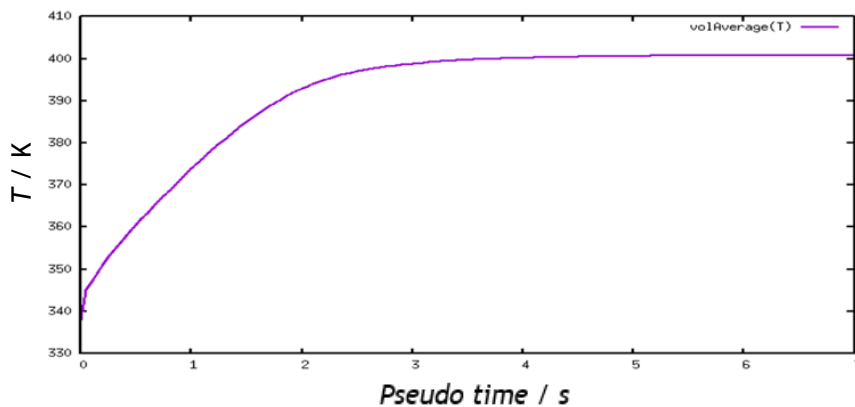


Figure 19- Heat sink average temperature for SteadyState solver.

4.2 Original Geometry

The first case analyzed, as a matter of comparison to the future optimizations, is the original heat sink which can be described by 17 parallel plate fins with 2 mm of thickness and 8 mm of height each. Both the fan curve and the fluid behavior are represented in Fig. 20, where U represents the average velocity at the inlet boundary.

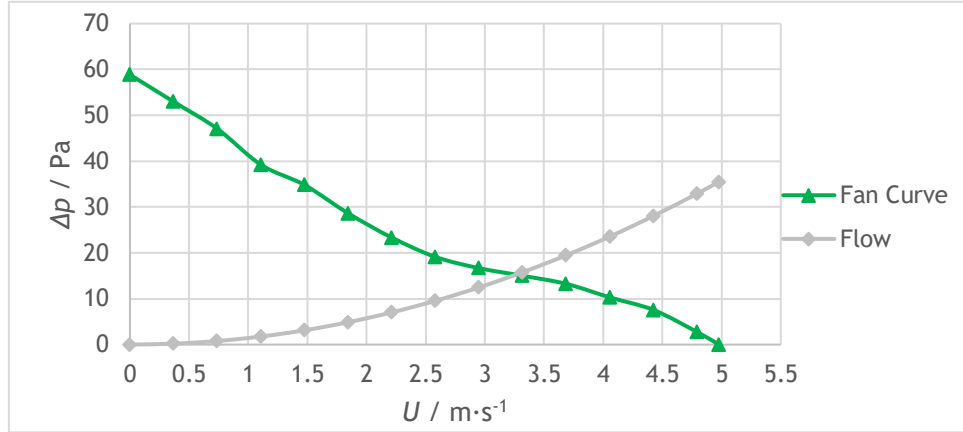


Figure 20- Flow behavior and fan curve for the original case.

The operation point obtained was $U = 3.3 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 15.6 \text{ Pa}$. As this dissertation's main goal is optimizing the heat sink efficiency in reducing the ICs temperatures, those are the most important results to be analyzed. In Table 1 are represented the temperatures for the original heat sink case. The computed pressure drops in the device are not highly accurate, due to entrance and exit effects (we are not including the fan which will have some effect on the flow field), but the operation point is not very sensitive to the Δp accuracy. Considering, as example, an increase of 50% in the pressure drop across the device, leads to a decrease of only 14% in the operation point inlet velocity. On the other hand, all simulations are subjected to the same inaccuracies, thus the comparison between different cases is not affected significantly.

Table 1- Temperatures for each piece of the original case

Model Piece	Original		
	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	82.44	121.89	139.53
IC1	123.19	125.35	126.56
IC21	99.37	103.80	107.67
IC22	97.56	101.80	105.72
IC3	115.06	134.20	139.99
IC4	127.50	128.35	128.84
IC5	131.45	131.94	132.21
IC6	131.82	132.21	132.45
IC7	134.71	135.59	135.97
IC8	134.45	134.99	135.25

4.3 Old Fan vs. New Fan

The air flow rate through the box is important because if the heat sink is highly optimized but the air flow rate is low, then the heat cannot be transferred efficiently between the solid and the fluid and the results are suboptimal. So, it is essential to improve the air flow right at the beginning of these optimizations to prevent limitations in the future as well as understanding the influence of the fluid flow rate in the heat management. Implementing a more powerful fan was the first step in this dissertation, which brings a new fan curve and consequently a new operation point, as can be seen in Fig. 21.

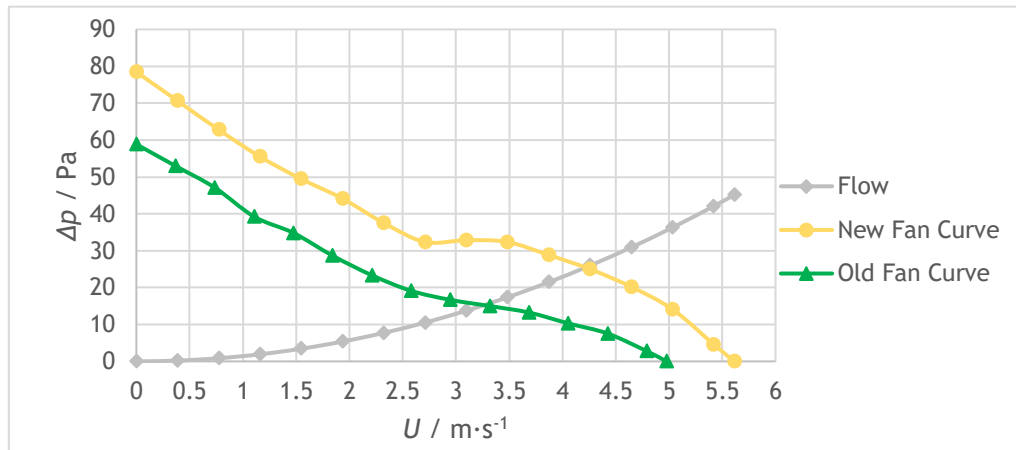


Figure 21- Comparison between the old and the new fan in the flow behavior.

With the new fan, the operation point increases to $U = 4.2 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.3 \text{ Pa}$. As shown in Table 2, increasing the air inlet velocity also means an improvement in the heat transfer process by convection and, consequently, reduced temperatures, as expected.

Table 2- Temperatures for each piece of the original case with the new fan

Model	New Fan		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	78.79	117.90	136.47
IC1	117.21	119.36	120.55
IC21	93.96	98.02	101.57
IC22	92.19	96.08	99.67
IC3	108.40	129.86	136.92
IC4	121.64	122.54	123.08
IC5	126.84	127.48	127.87
IC6	129.80	130.32	130.59
IC7	133.08	133.99	134.39
IC8	133.27	133.83	134.10

These results revealed that most of the ICs average temperature decreased around 4-5 °C, especially the ones closer to the inlet. In terms of temperature reduction, it was normalized using the same air inlet temperature, 65 °C, but in general it could also be normalized based

on temperature differences. The maximum reduction in this case is 5.6%, for the IC22, and the minimum reduction is 0.9%, for the IC8, the furthest heat source. Moreover, the IC with the largest range of temperatures is IC3 since it is the biggest and more powerful heat source and, consequently, is influenced by the distance to the inlet, as observed in Appendix B. The new fan is, then, going to be implemented in the project.

4.4 Normal Flow vs. Reverse Flow

As referred before, the inlet and outlet positions are also of interest and can completely change the performance of the heat sink and of the heat transfer rate. Up until now, the grid fan was used as the inlet and the perforation matrix in the opposite wall as the outlet but in this section their roles were reversed to understand how the efficiency is influenced. It is already evident by observing Fig. 22 that the flow behaves differently.

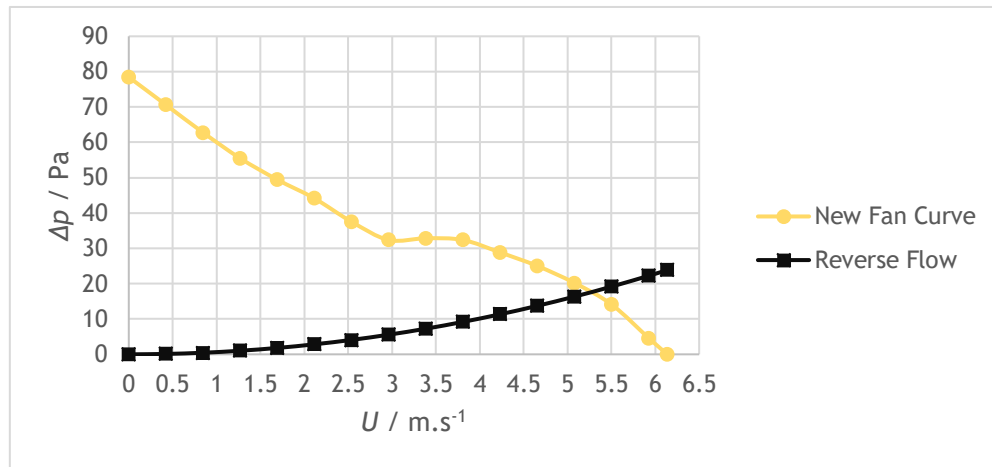


Figure 22- Flow behavior and fan curve for the reverse flow case.

For reverse flow, the operation point increases to $U = 5.28 \text{ m}\cdot\text{s}^{-1}$, while the pressure drop decreases to $\Delta p = 17.6 \text{ Pa}$. The temperatures for the reverse flow case are shown in Table 3.

Table 3- Temperatures for each piece of the original reverse flow case

Model	Original Reverse Flow		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	73.44	115.83	136.06
IC1	133.47	135.43	136.31
IC21	126.43	128.04	129.09
IC22	123.23	125.74	127.40
IC3	102.55	127.74	136.49
IC4	133.61	133.93	134.11
IC5	131.65	132.31	132.64
IC6	119.22	120.52	121.51
IC7	99.56	101.71	103.08
IC8	108.05	109.22	109.84

Changing the inlet position and flow direction completely inverses the behavior of the ICs temperature since the ones that are further away from the inlet in one option are the ones closer to the inlet in the other. In Appendix C it is possible to visualize that while both have a high temperatures zone, mainly because of the high power of IC3, the zones with lower temperatures are obviously the ones closer to the inlet. To decide which alternative will be the best adaption in this particular case, it was resorted to the reference limit temperatures that each IC can go up to without shutting down. In Table 4 it is possible to compare these limit temperatures with the average temperatures from the two alternatives in study.

Table 4- Comparison between the limit temperatures with the Normal and Reverse flow case

Piece	Limit T / °C	Normal Flow T / °C	Reverse Flow T / °C
IC1	125	119.36	135.43
IC21	105	98.02	128.04
IC22	105	96.08	125.74
IC3	105	129.86	127.74
IC4	125	122.54	133.93
IC5	125	127.48	132.31
IC6	125	130.32	120.52
IC7	150	133.99	101.71
IC8	125	133.83	109.22

At first stance, it is possible to see that while the normal flow alternative allows 5 ICs to work in good condition, below the shut-down temperature, the reverse flow alternative only has 3 ICs in such condition. With deeper analysis to the ICs above the limit, it is also understandable that the difference between the real temperature and the shut-down temperature is generally a lot lesser in the normal flow alternative, not taking into account IC3 temperature, which is high in both situations. The gap usually goes approximately from 2.5 °C to 9 °C in the normal flow option and from 7 °C to 23 °C in the reverse flow option. From these results, normal flow will be used in forward simulations and will be considered the base case from now on since the modifications on the design of the heat sink will now be explored.

4.5 Plate Fins Thickness

As described in the state of the art, the thickness of the plate fins is probably the heat sink dimension that mostly influences its performance. At the same time, there is an optimum point to be found in order to maximize the heat transfer process since bigger thickness reduce the amount of fluid flowing in the channels between the plates and smaller thickness reduce the heat transfer area, both in heat conduction and convection. So, it is important to make a broad analysis in this parameter to understand if the heat sink is already working in its optimal thickness. To this end, two heat sinks with different thicknesses were simulated, one with

narrower plate fins, of 1 mm, and one with wider plate fins, of 3 mm, as represented in Fig. 23.

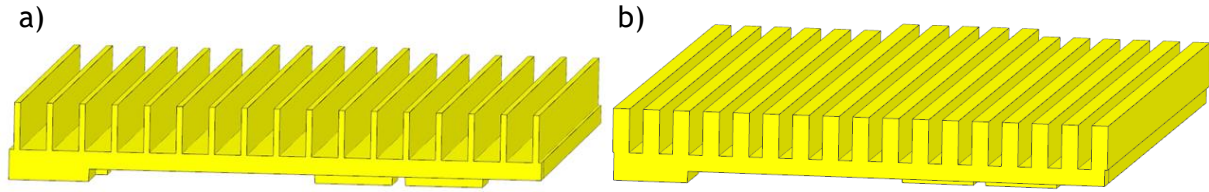


Figure 23- a) Heat sink with plate fins with 1 mm of thickness; b) Heat sink with plate fins with 3 mm of thickness.

From here on, both the calculated operation points and the U vs. Δp plots are similar with each other, so, they will instead be transferred to Appendix D.

Table 5 presents the results obtained for the temperatures of the three heat sinks with different thicknesses in the corresponding operation points and the column charts comparing the temperature reduction for each case are located in Appendix E.

The heat sink with 1 mm of thickness has revealed the best performance, improving the temperature reduction of almost all the ICs with a maximum reduction of 8.0%, in the IC21, when compared to the first case, and of 2.6% when compared to the base case. The IC7 and IC8, which did not benefit from wider channels as much as the other ICs, have their results improved by at most 1.9%, comparatively to the base case, when using the plate fins heat sink with 3 mm of thickness. This is possibly justifiable due to a vortex created in that region, illustrated in Appendix F, that has more contact area in this heat sink and influences the heat transfer rate around those ICs. However, the other ICs temperatures of this heat sink are the worst of the 3 cases, with a maximum reduction of only 3.7% when comparing to the first case.

Table 5- Temperatures for each piece of the base, 1 mm and 3 mm fin thickness case

Model	1 mm			Base Case			3 mm		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C	Minimum T / °C	Average T / °C	Maximum T / °C	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	75.76	115.73	134.55	78.79	117.90	136.47	82.52	118.96	137.02
IC1	115.03	117.43	118.76	117.21	119.36	120.55	118.88	120.78	121.89
IC21	91.64	95.51	98.88	93.96	98.02	101.57	97.95	102.29	106.08
IC22	89.98	93.70	97.09	92.19	96.08	99.67	96.19	100.35	104.24
IC3	105.96	127.50	134.99	108.40	129.86	136.92	113.01	131.59	137.45
IC4	119.26	120.23	120.84	121.64	122.54	123.08	123.40	124.40	124.96
IC5	125.30	126.00	126.43	126.84	127.48	127.87	128.09	128.68	129.03
IC6	129.29	129.84	130.14	129.80	130.32	130.59	129.54	130.00	130.23
IC7	132.50	133.44	133.90	133.08	133.99	134.39	130.43	131.50	131.97
IC8	133.49	134.13	134.42	133.27	133.83	134.10	131.78	132.31	132.58

Also benefiting the heat sink with 1 mm of fin thickness, it is important to note that this alternative also interests in the accomplishment of this project secondary goal: obtaining a

lighter product. In this subject matter, this heat sink has 24.3% lesser mass than the base case heat sink. Also, there are two effects that are important to visualize. The first is a factor that contributes to the lesser efficiency of the thicker-plates heat sink in the more central ICs, which is possible to see in Fig. 24. Unlike with the plate fins heat sink with 1 mm of thickness, in which the air flows through all the channel, the plate fins heat sink with 3 mm of thickness has so narrow channels that the airflow that enters there heats up more and faster. The consequence is a higher reduction of the air density and a faster ascent of the air, that ends up leaving the channels. Some of it might return to the channels as soon as it cools down, but a part of the heat sink loses efficiency.

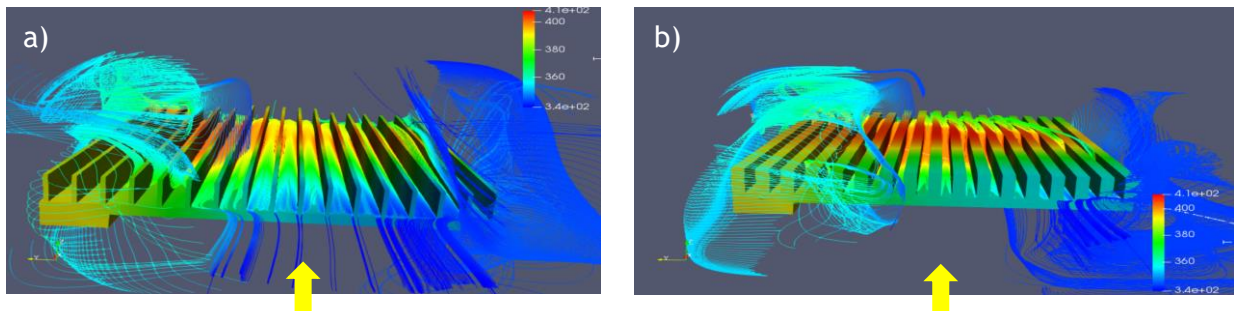


Figure 24- a) Air flowing through the channel of the plate fins heat sink with 1 mm of thickness; b) Air leaving the channels of the plate fins heat sink with 3 mm of thickness. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

The other effect can be observed in the two heat sinks, as represented in Fig. 25a, and is about the fact that the two heat sink channels in the left do not receive direct airflow which makes this region vulnerable. Figure 25b shows, in the right of the heat sink, the effect of this airflow flow that prejudice the heat transfer process and has the consequence of high temperatures profiles in some ICs. Taking all into account, the heat sink with fin thickness of 1 mm is the best choice of the three and this change will be adapted in the remaining tests.

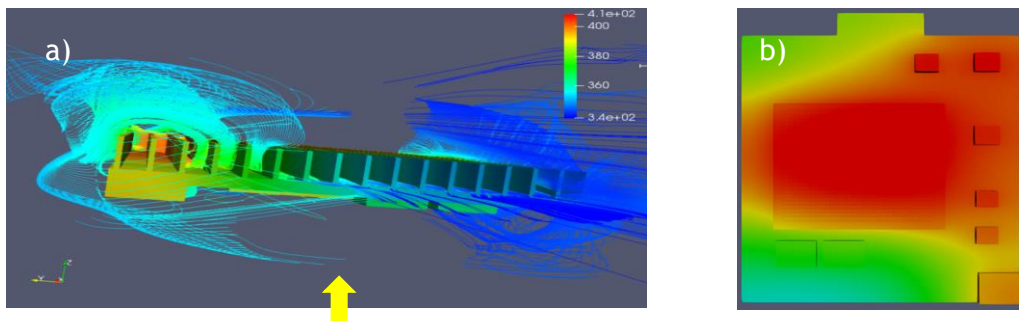


Figure 25- a) Two channels in the left of the heat sink with no direct flow; b) Effect of less strong flow through specific channels in the temperature profiles of the heat sink. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

4.6 Fins Height

It is already known that increasing the height of the heat sink fins increases the heat transfer area by conduction and convection and with the reduction of mass obtained by the last optimization it is acceptable to increase the heat sink fins height without increasing the overall mass. Utilizing 25% of the mass removed in the last section is enough to increase the fins height by 2 mm, reaching 10 mm, as represented in Fig. 26.

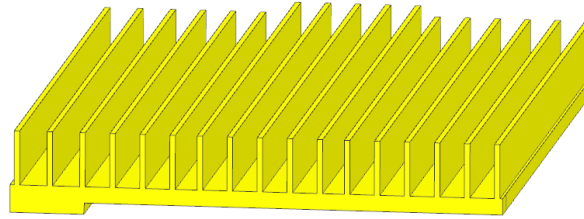


Figure 26- Heat sink with plate fins with 10 mm of height.

The temperature results obtained by this heat sink are shown in Table 6. As expected, the reduction occurs in all the ICs and the temperature reductions can go up to approximately 5 °C and reach a maximum temperature reduction of 4.3% when compared to the plate fins heat sink with 1 mm of thickness and a maximum reduction of 6.3% when compared to base case. It is also evident that, with this optimization, the IC5 average temperature is now under its shut-down temperature, 125 °C. This heat sink is, then, the best option until now.

Table 6- Temperatures for each piece of the high plate fins case

Model	High Fins		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	73.64	111.08	131.22
IC1	111.71	114.15	115.50
IC21	88.38	91.89	94.95
IC22	86.81	90.18	93.24
IC3	101.05	122.08	130.27
IC4	115.24	116.16	116.71
IC5	120.70	121.36	121.76
IC6	125.07	125.68	126.02
IC7	128.83	129.79	130.28
IC8	130.82	131.41	131.68

4.7 Bypass Airflow

Since the ICs closer to the inlet are below their limit temperatures and have already been well optimized, the use of the bypass airflow effect is an alternative to consider in an attempt to reduce further the temperatures of the IC3 and the heat sources on the back of the heat sink.

This adaption may or may not work depending if the temperature reduction of those ICs compensate the sacrifice made with the ICs in the front and the length of the cut made in the heat sink is highly influential in the end results. In this case, a ramp will be cut through 27.7% of the length of the heat sink so that the ramp is finished when the IC3 is encountered, as represented in Fig. 27. This hypothesis was tested since IC3 is the first IC with high temperatures and above the shut-down.

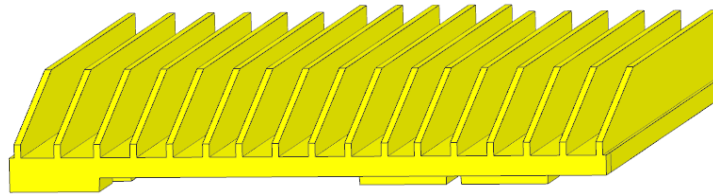


Figure 27- Heat sink optimized to use the bypass airflow effect.

The temperature results obtained in this case are presented in Table 7. Analyzing this table it is possible to see that while the last 3 ICs have a temperature reduction, reaching a maximum of 0.3%, it is basically residual when compared to the increase in temperature of all the other ICs, that reached a maximum of 4.4%, when compared with the high plate fin heat sink case. Even IC3 increased its average temperature by 0.3%, comparing to the same case, since the loss of heat transfer area near it surpassed the benefits of the bypass airflow effect. To get a more significant improvement in the back ICs, the length of the ramp would probably need to increase but that also would mean a sacrifice even higher in the others heat sources. Considering this, even though the concept works, as it is right now is impracticable and does not compensate the disadvantages it brings.

Table 7- Temperatures for each piece of the bypass airflow case

Model	Bypass Airflow		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	80.31	113.06	130.89
IC1	115.65	118.49	120.12
IC21	92.78	95.85	98.38
IC22	91.21	94.17	96.78
IC3	103.12	122.48	130.19
IC4	117.26	118.02	118.43
IC5	121.42	122.01	122.34
IC6	124.98	125.55	125.88
IC7	128.80	129.77	130.27
IC8	130.39	131.05	131.32

4.8 Plate-Pin Fin

4.8.1 Pins Size

With an attempt to generate vortices that would improve the heat transfer process, the plate-pin fin geometry was considered as an alternative. Just as referred in the State of the Art, more than one pin per channel would reduce the turbulence created in the flow, so one pin in the beginning of each channel was implemented in this case.

The first parameter investigated was the influence of the height of the pins in the performance of the heat sink. Three different pins height were considered, 1 mm, 3 mm and 5 mm, all with 1.5 mm of base diameter, as represented in Fig. 28.

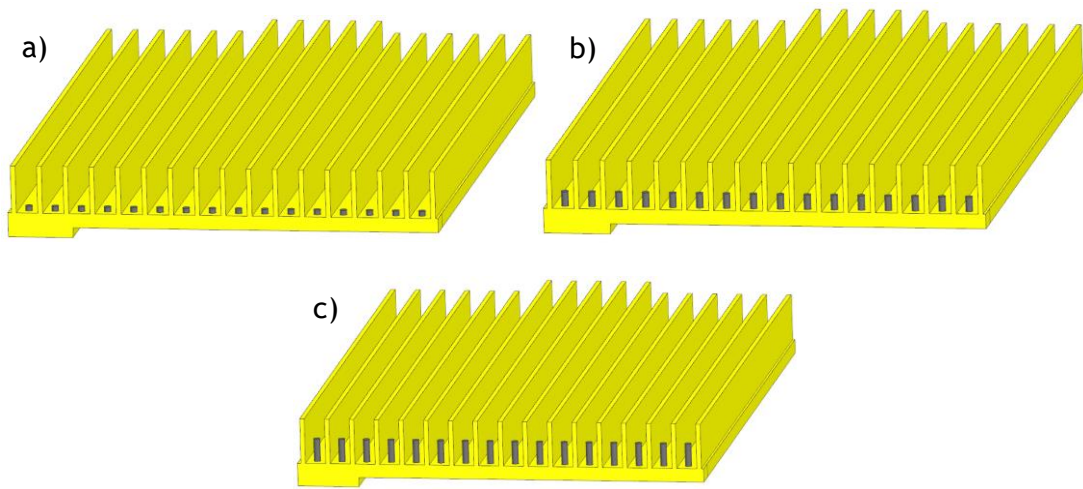


Figure 28- Plate-Pin fin heat sinks with pins with height of a) 1 mm; b) 3 mm; c) 5 mm.

As Table 8 reveals, the introduction of pins in the heat sinks ended up worsening the performance and the pin height increase deteriorates even more the heat transfer process.

Table 8- Temperatures for each piece of the small, medium and high pin plate-pin fins case

Model	Small Pins			Medium Pins			High Pins		
Piece	Minimum $T / ^\circ\text{C}$	Average $T / ^\circ\text{C}$	Maximum $T / ^\circ\text{C}$	Minimum $T / ^\circ\text{C}$	Average $T / ^\circ\text{C}$	Maximum $T / ^\circ\text{C}$	Minimum $T / ^\circ\text{C}$	Average $T / ^\circ\text{C}$	Maximum $T / ^\circ\text{C}$
Heat Sink	73.86	111.65	131.63	74.31	114.26	133.34	74.34	115.54	135.18
IC1	112.12	114.52	115.86	113.48	115.81	117.11	113.92	116.22	117.50
IC21	89.02	92.57	95.64	90.91	94.77	98.15	91.41	95.45	99.02
IC22	87.35	90.78	93.88	89.12	92.89	96.34	89.61	93.55	97.17
IC3	101.73	122.73	130.80	104.94	126.39	133.81	106.48	128.38	135.62
IC4	115.77	116.70	117.25	117.76	118.72	119.31	118.64	119.62	120.23
IC5	121.25	121.90	122.30	123.57	124.26	124.68	124.69	125.40	125.84
IC6	125.61	126.22	126.56	127.74	128.34	128.66	128.67	129.24	129.55
IC7	129.37	130.33	130.81	131.43	132.37	132.82	132.39	133.32	133.75
IC8	131.22	131.82	132.09	132.55	133.15	133.43	132.94	133.55	133.83

When compared to the high plate fin heat sink case, this case heat sinks all lead to an increase in temperature, as the plate-pin fin with small pins leads to a maximum temperature increase of 0.7%, with medium pins increases at most 3.5% and with high pins leads to 5.2% of maximum increase.

Regarding the turbulence of the airflow, when analyzing the results and Fig. 29, a stagnation zone can be visualized after the pins that influences and limits the heat transfer by convection in this area. Also, the higher the pins, the wider and longer is this stagnation zone and that affects the results of Table 8, despite higher pins meaning more area of heat transfer.

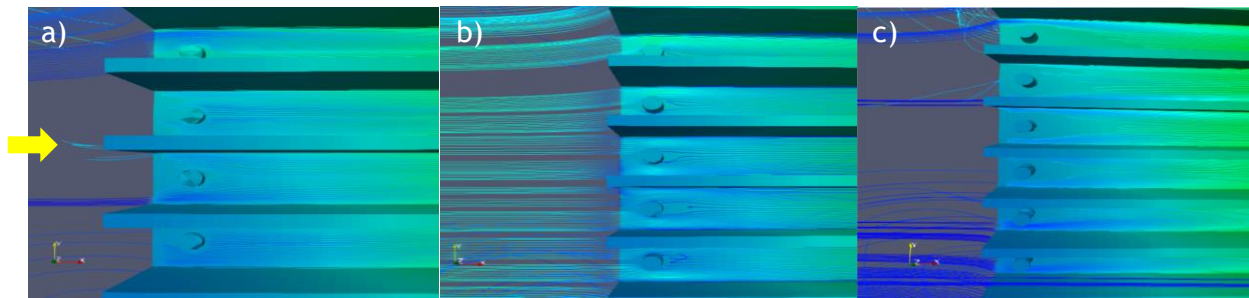


Figure 29- Stagnation zone created by a) small pins in plate-pin fin heat sink; b) medium pins in plate-pin fin heat sink; c) high pins in plate-pin fin heat sink. The yellow arrow indicates the main flow direction in the 3 figures.

4.8.2 Pins base geometry

In order to try to avoid the stagnation zone created with the pins, another base geometry was tested, a drop geometry, which enables better flow mixing capacities [18]. Since the pins that created less stagnation flow and that lead to less temperature increase were the smaller ones, this test will also be done with pins with 1 mm of height, as represented in Fig. 30.

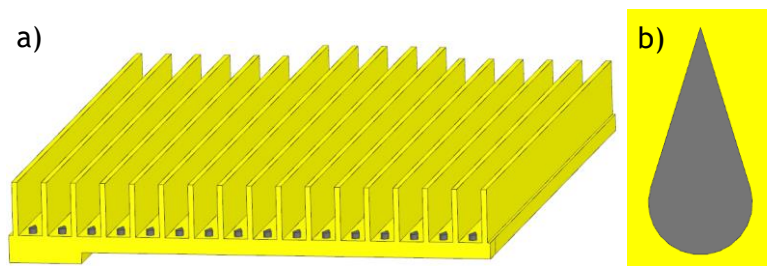


Figure 30- a) Plate-Pin fin heat sink with drop geometry pins; b) Pins base geometry.

The temperature results, as presented in Appendix G, ended up being similar to the plate-pin fin heat sink with the circular base geometry pins, with a maximum temperature increase of 0.8%, when compared with the high plate fin heat sink case. With this information, this adaptation of the pins was not considered a good choice for this case and should not be used, at least in these conditions.

4.8.3 Staggered Pins

Even though the literature specifies that more than one pin in a plate-pin fin heat sink deteriorates the turbulence of the flow created by the first pin, an example in which multiple pins were introduced in each channel was simulated, in a staggered disposition, as represented in Fig. 31, in an attempt to create the turbulence that was not observed in the sections 4.8.1 and 4.8.2 of the plate-pin fin heat sink. Pins with circular base geometry, 1.5 mm of diameter and 1 mm of height were used in this heat sink.

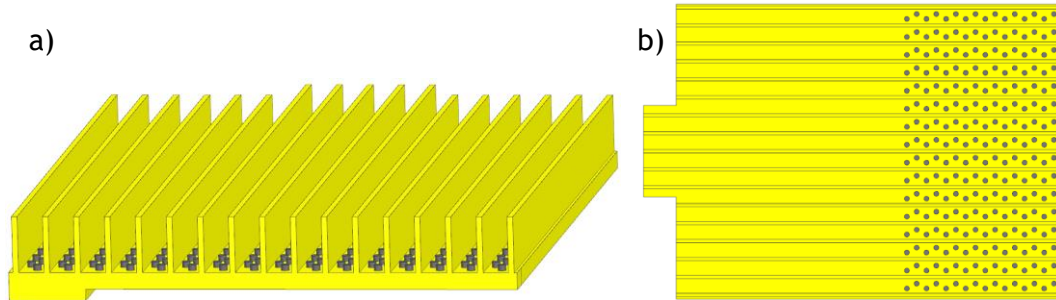


Figure 31- Plate-Pin fin heat sink with staggered pins: a) frontal view; b) top view.

From the temperature results shown in Table 9, it is possible to conclude that, in comparison with only a pin in the front of the channel, this case is worse in terms of thermal performance. The maximum temperature increase observed was 1.6% relative to the plate-pin fin heat sink with only one pin per channel. On the other hand, in terms of turbulence, it is possible to see in Fig. 32 that it increased compared to the previous results and there are more secondary flows created but ultimately it looks there are still some preferential flow paths, just as the middle of the pins and between the pins and the walls of the plates. A study with more pins, arrangements, shapes and dimensions of this type of heat sink is in need for futures applications. Meanwhile, it is still a modification not worthy of implementing.

Table 9- Temperatures for each piece of the staggered pins plate-pin fins case

Model	Staggered Pins		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	74.34	113.04	132.39
IC1	113.07	115.42	116.74
IC21	90.26	94.00	97.26
IC22	88.58	92.21	95.50
IC3	103.79	124.71	132.33
IC4	117.08	118.02	118.59
IC5	122.67	123.34	123.74
IC6	126.83	127.42	127.75
IC7	130.38	131.34	131.81
IC8	131.99	132.59	132.86

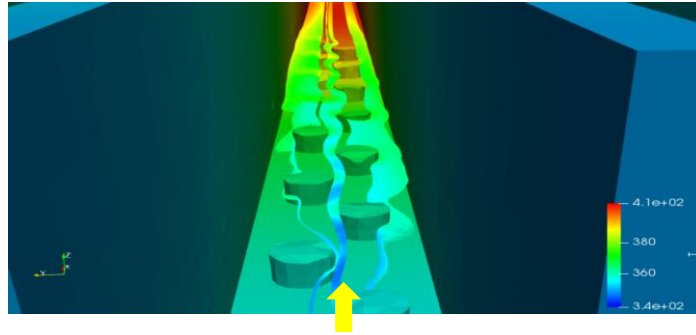


Figure 32- Creation of secondary flows with staggered pins (detailed view). The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

4.9 Pin-Fin

After analyzing the viability of the plate-pin heat sink, it is also adequate to make a quick study using other mainstream heat sink besides plate-fin heat sink: pin-fin heat sink.

Two heat sinks using solely pins were simulated, differentiating them the number of pins used, one with 373 pins and another with 899 pins, as represented in Fig. 33. All the pins have 2 mm of diameter and 10 mm of height.

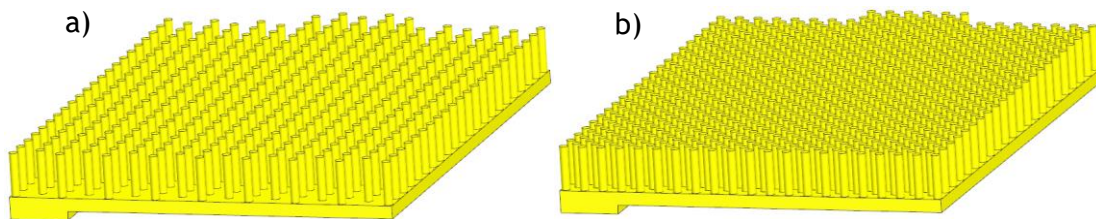


Figure 33- a) Heat sink with 373 pins; b) Heat sink with 899 pins.

From Table 10, it can be concluded that the heat sink with more pins revealed generally better heat transfer performance than the one with less pins.

Table 10 - Temperatures for each piece of the 373 and 899 pin-fin case

Model Piece	373 Pins			899 Pins		
	Minimum T / °C	Average T / °C	Maximum T / °C	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	77.42	121.17	144.85	75.68	118.44	139.68
IC1	116.18	118.50	119.76	114.17	115.82	116.83
IC21	97.83	102.50	106.74	101.27	107.14	112.45
IC22	95.94	100.74	105.18	99.18	105.06	110.49
IC3	117.82	138.53	145.21	121.55	135.56	140.13
IC4	124.71	126.05	127.06	123.89	125.17	126.15
IC5	132.98	133.69	134.09	130.61	131.19	131.50
IC6	132.55	133.45	133.91	128.30	129.10	129.49
IC7	133.43	134.55	135.07	127.08	128.31	128.89
IC8	131.80	132.74	133.16	126.06	127.07	127.50

The 899 pins heat sink granted an IC1 average temperature close to the IC1's high heat sink temperature and both the IC7 and IC8 were 1.1% and 3.3% better, respectively, than the ones from the high heat sink. However, the remaining ICs did not reveal good results, so this option was not more studied.

Besides the increase in heat transfer area, the 899 pin-fin heat sink results are also consequence of the turbulence of the airflow and creation of secondary flows and interaction between different paths. This behavior does not happen in the other heat sink due to the existence of easier flow paths that reduce collisions and high space between pins, as observed in Fig. 34. Applying closer pins and with a higher number of pins may be the solution to improve the performance of the plate-pin heat sinks too.

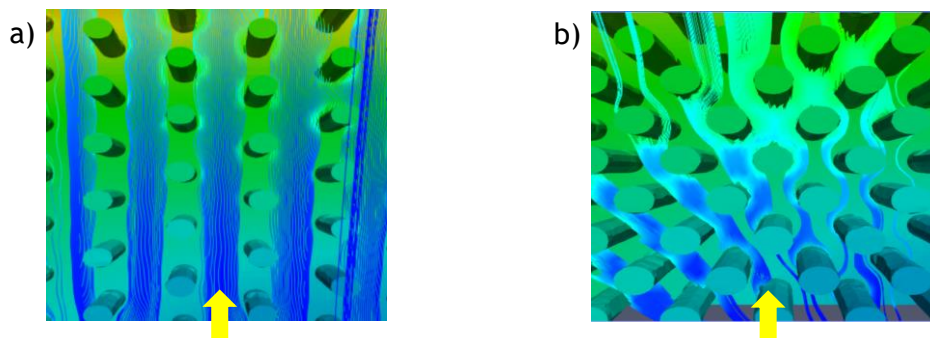


Figure 34- a) Easy paths in the 373 pin-fins heat sink; b) Secondary paths created in the 899 pin-fins heat sink. The yellow arrow indicates the main flow direction, in both cases.

4.10 Inverse Bypass Airflow

With the information obtained so far, and considering that the ICs with higher temperatures are the ones more in the back of the heat sink, in this section, instead of cutting a ramp in the beginning of the heat sink, a ramp was created in the end, elevated until 1 mm of height and persisting with the 1 mm through near 25% of the plate fins length, as represented in Fig. 35.

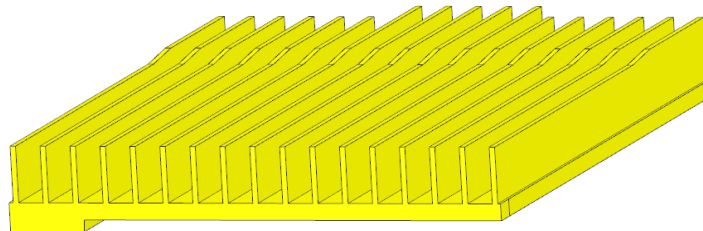


Figure 35- Heat sink with a ramp created in the back.

In addition to the reasons already expressed, the use of this ramp also accompanies the rise of the flow with the reduced density, consequence of the increasing air temperature, observed in Fig. 36. So, this heat sink is the result of the information obtained in sections 4.5, 4.6 and 4.7. The temperature results, represented in Table 11, expressed a reduction of almost 2 °C

corresponding to an improvement of 1.4% to the IC7 and IC8 average temperatures, when compared to the high plate fins heat sink, and a maximum temperature reduction of 6.4%, when compared to the base case. It is also notable that with this optimization, the IC6 average temperature goes under the shut-down temperature, 125 °C.

Table 11- Temperatures for each piece of inverse bypass airflow case

Model	Inverse Bypass Airflow		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	73.72	110.45	129.36
IC1	111.59	114.01	115.36
IC21	88.53	92.04	95.08
IC22	86.93	90.30	93.36
IC3	101.12	121.58	129.38
IC4	115.14	116.06	116.60
IC5	120.49	121.13	121.52
IC6	124.17	124.70	125.01
IC7	127.07	128.01	128.49
IC8	128.93	129.54	129.81

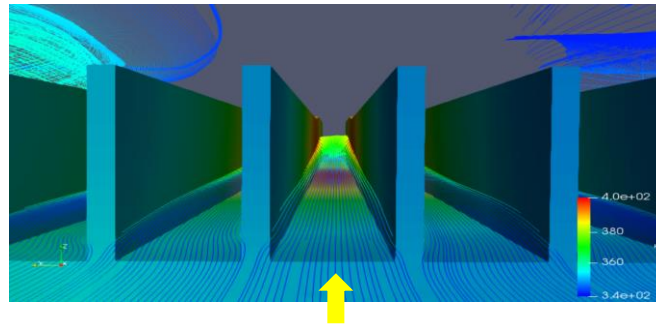


Figure 36- Rise of the air flow near the ramp area. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

4.11 Base Thickness

In order to further increase the heat transfer area, thickening the base of the heat sink is an efficient method that enhances the process of heat conduction globally and decreases the temperature in every IC. However, it is important to note that to maintain the goal of lightness of the product, the amount of thickness increased must be restrained to a certain level. In this dissertation, it was stipulated that the final mass of the heat sink should be equal or lower than the original one. Accordingly, the mass used in this process summed to the mass used to increase the height of the plates and to create the back ramp must be equal to the mass removed from the thickness reduction of the plates. Therefore, the base of the heat sink must increase approximately 1.07 mm of thickness relatively to the original, totalizing 4.07 mm, as represented in Fig. 37.

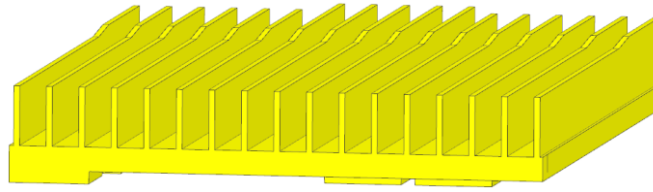


Figure 37- Heat sink with a thicker base.

The temperature results are presented in Table 12 and, as expected, there was a significant improvement in most of the ICs, with a maximum reduction of 3.2% relatively to the inverse bypass airflow heat sink, corresponding to 3.7 °C. The maximum reduction was of 7.6%, when comparing to the base case, and this reduction reached the 12.0%, relative to the first case.

Table 12- Temperatures for each piece of the thick base heat sink case

Model	Thick Base		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	74.83	109.84	126.94
IC1	108.40	110.31	111.43
IC21	89.61	93.02	95.96
IC22	87.99	91.27	94.29
IC3	101.42	120.44	127.41
IC4	113.20	114.09	114.70
IC5	118.95	119.55	119.97
IC6	122.90	123.27	123.50
IC7	124.58	125.39	125.79
IC8	126.08	126.53	126.75

4.12 New Material

Up until now, optimizations were done at ventilation and heat sink design level but, according to the Context and State of the art section, the heat sink material can also be changed to improve the heat transfer rate by conduction.

In consideration of that, to understand the influence of the material, a test using the design of the last section and replacing the material with $\rho = 1850 \text{ kg}\cdot\text{m}^{-3}$, $k = 20 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ in-plane and $k = 6 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ through plane to a material with $\rho = 1610 \text{ kg}\cdot\text{m}^{-3}$, $k = 35.02 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ in-plane and $k = 6 \text{ W}\cdot\text{m}^{-1}\cdot\text{°C}^{-1}$ through plane was done. The results are presented in Table 13 and, when compared to the old material heat sink, it can reach a maximum reduction of 7.1%, corresponding to 7.78 °C. Relatively to the first case, the temperature reduction reached 18.2%. Moreover, the IC8 average temperature is, with this optimization, below the shut-down temperature, leaving only the IC3 above it. Despite the results validating this material and revealing a good improvement, its application must overcome the cost vs. advantages balance.

Table 13- Temperatures for each piece of the new material heat sink case

Model	New Material		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	76.27	108.23	123.30
IC1	101.31	102.53	103.25
IC21	88.98	92.25	95.26
IC22	87.72	90.99	94.02
IC3	101.03	117.65	123.58
IC4	107.10	108.28	109.20
IC5	114.91	115.70	116.30
IC6	120.57	120.81	120.98
IC7	119.12	119.87	120.24
IC8	120.60	121.06	121.29

Fig. 38 shows a comparison between all the heat sink design cases relative to the base case. It is evident that the modifications that influence more the heat sink performance was the plate fin height increase and the thicker base in the heat sink. It can be also observed that the main improvement in the back ICs was especially achieved with the inverse bypass airflow case.

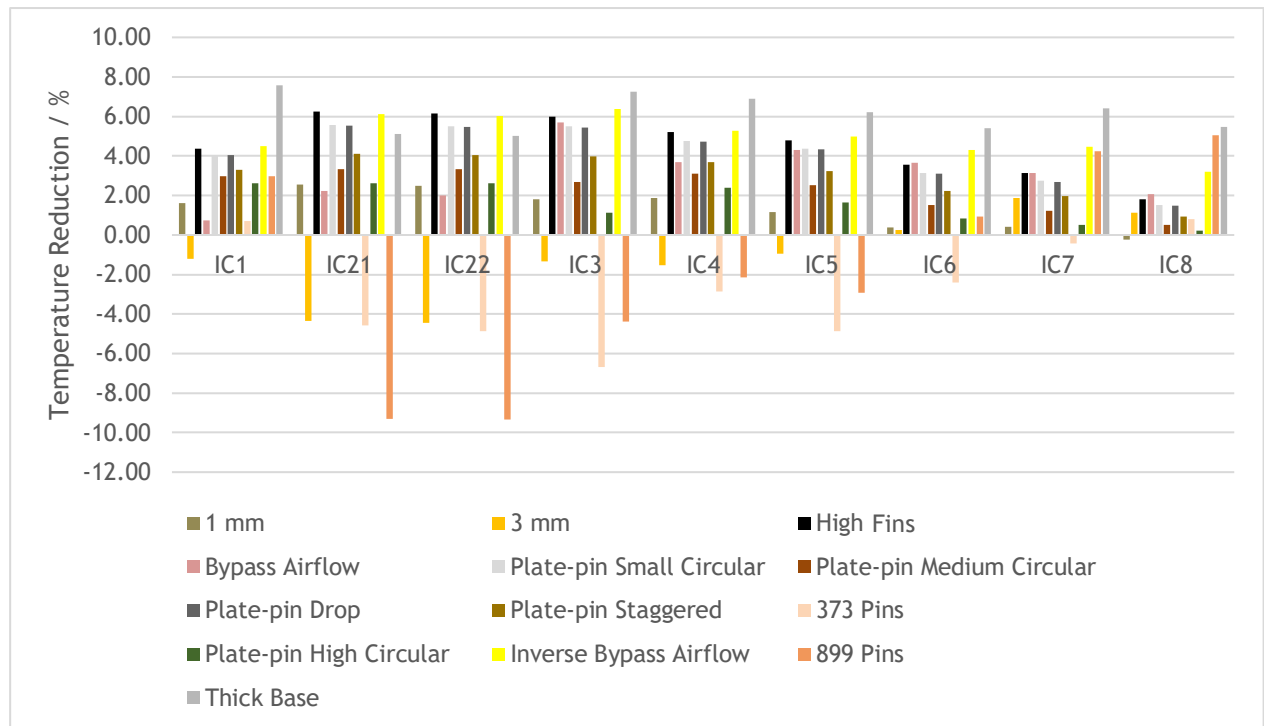


Figure 38- Temperature reduction of all the cases relative to the base case for each IC.

At the end of the optimizations, it was reached a maximum temperature reduction of 5.6% by improving the ventilation, a maximum temperature reduction of 7.6% by changing the heat sink design and a maximum temperature reduction of 7.1% by replacing the heat sink material.

5 Conclusions

In this dissertation, the goal was to optimize the heat management supplied by 9 ICs spread under a heat sink inside an electronic housing, with variable sizes and powers. For that purpose, the heat sink used had flexibility of changing its design and constitution in order to maximize the heat transfer with the convective fluid, air. Seventeen cases were simulated in the OpenFOAM® software in which various types of optimization were tested at the level of ventilation, design and materials. In the end, both the main goal and the secondary goal of keeping a light product were reached.

First, in terms of ventilation, using a fan with more power improved the ICs temperature results by 5.6%. Inverting the inlet and outlet positions changed completely the results, being the normal flow the most beneficial in this particular case.

Regarding the heat sink design optimization, while an optimal point was not yet found for the plate fins thickness, the tests revealed that this optimum is around 1 mm, since the ICs temperatures reduce, at most, 1.9% when comparing to the 2 mm heat sink. As for the fin's height, the higher the plates, the most efficient will be the heat sink but it is important to secure space between the other components of the housing and to maintain the heat sink weight. Using the bypass airflow effect on the design of the heat sink worked but the length of the ramp used conducted to more disadvantages than advantages. However, when this effect was used together with a localized increase of height, the improvements reached 1.4% in the back ICs when compared to the already high plate fins heat sink, which proved a good result since these ICs were the least optimized until then. The implementation of pins, both between the plates and exclusively on the heat sink, revealed problems in finding a good number of pins and a good distribution in order to enhance turbulence in the airflow and to reduce the stagnation zones. In the end, none of the pin cases worked out good enough, introducing more complexity in the fabrication. Finally, increasing the heat sink base thickness by approximately 1.07 mm improved the results of the inverse bypass airflow design by at most 3.2%. The total design optimization was 7.6% of temperature reduction, when compared to the base case.

Optimizing the heat sink material was also tested, changing from a material with $k = 20 \text{ W}\cdot\text{m}^{-1}\cdot^\circ\text{C}^{-1}$ in-plane to a material with $k = 35.02 \text{ W}\cdot\text{m}^{-1}\cdot^\circ\text{C}^{-1}$ in-plane, and had a significant improvement on the results with a reduction on temperature of 7.1%. However, for implementing this material, other factors like the cost must be considered.

A heat sink with 17 plates with 1 mm of thickness, 10 mm of height, a ramp with a total 1 mm of height, a heat sink base with a thickness increased by 1.07 mm and a new material optimized the ICs average temperatures by 18.2%, when compared to the first case with the original fan.

6 Assessment of the work done

6.1 Objectives Achieved

The objectives of this dissertation were to create new models and designs of heat sinks that allowed a higher thermal dissipation to the air, simulate those designs in a new software to the student and to the company and assess which of the designs granted better optimizations and influenced more in the temperature reduction. This would lead to the viability of the use of thermoplastic composites in thermal management and to maintain a more lightweight product.

Regarding these goals, various designs were created from the research done in the beginning of the dissertation with regular and innovative alterations. A significant amount of time was invested in learning how to use OpenFOAM® and in the end all the designs were able to be simulated in time with the conditions close to the requested. The optimizations were analyzed and the improvements can be compared by the temperature reduction they generate on each IC. Almost all the ICs were below the shut-down temperature by the end of the optimizations, and the weight of the final heat sink is the same as the original. It is important to refer that since many simplifications needed to be done, and a lot of not so much relevant pieces of the product were discarded in the simplified model, these results will eventually differ from the reality. However, this work can, despite those approximations, make a valid comparative study between the optimizations done and the influence of each one of them.

6.2 Final Assessment

This work was a learning experience in which I enjoyed both the design part in Siemens NX™, in which I was allowed total freedom to try different models, and the OpenFOAM® part in which the learning curve was slow and challenging but in the end and with a lot of effort the feeling of being able to put it to work paid off. Having said that, it is my opinion that this theme needed more time to learn and consolidate all the OpenFOAM® functionalities and settings and to make more profound studies in each of the optimizations implemented, as well as implementing others, more alternative, designs.

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Appendix A - ICs Power

Table A.1- ICs Power

Heat Source	Power / mW
IC1	1 580
IC21	750
IC22	750
IC3	20 510
IC4	687
IC5	550
IC6	390
IC7	1 175
IC8	975

Appendix B - IC3 Temperature Range

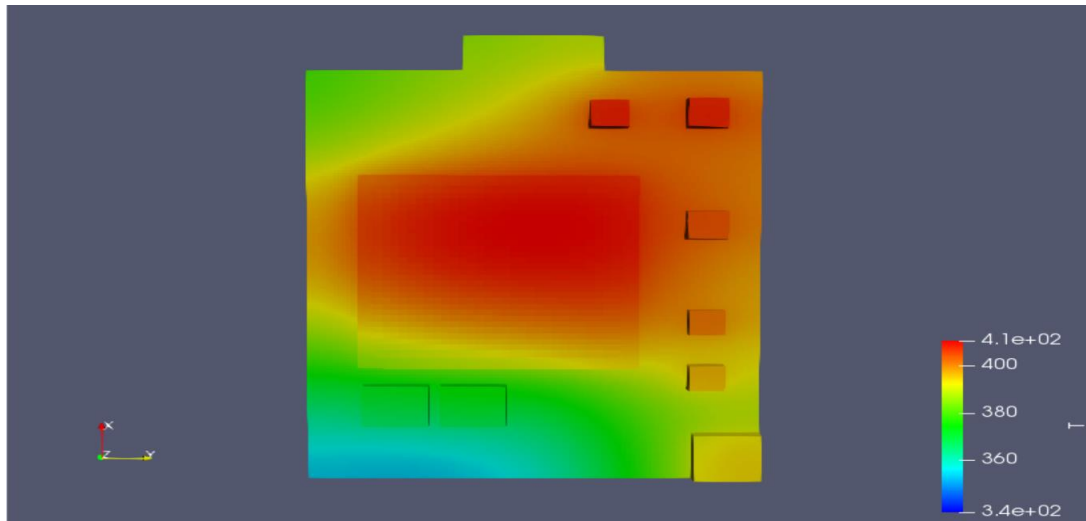


Figure B.1- ICs temperature (in K) map.

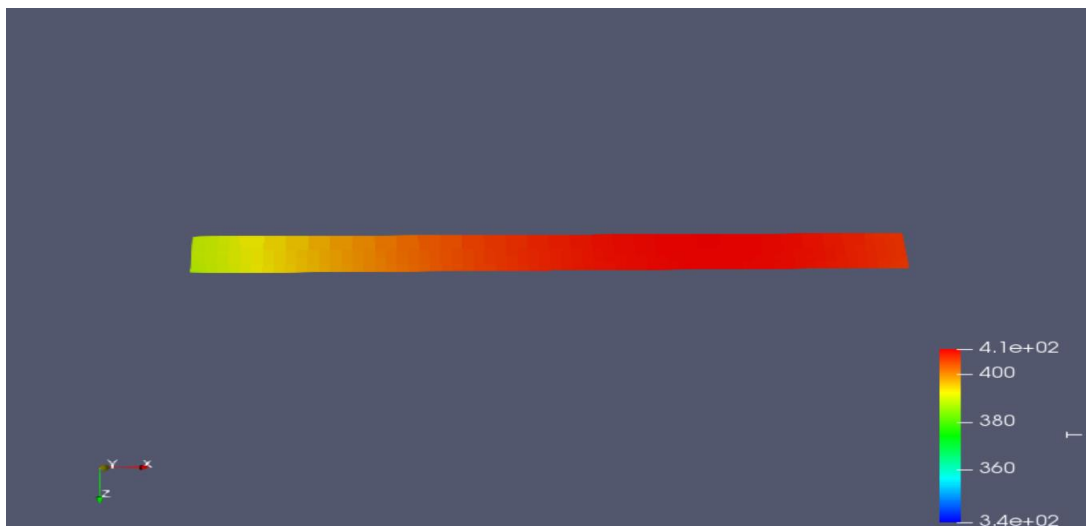


Figure B.2- IC3 vertically cut to visualize its temperature (in K) range.

Appendix C - Normal Flow vs. Reverse Flow

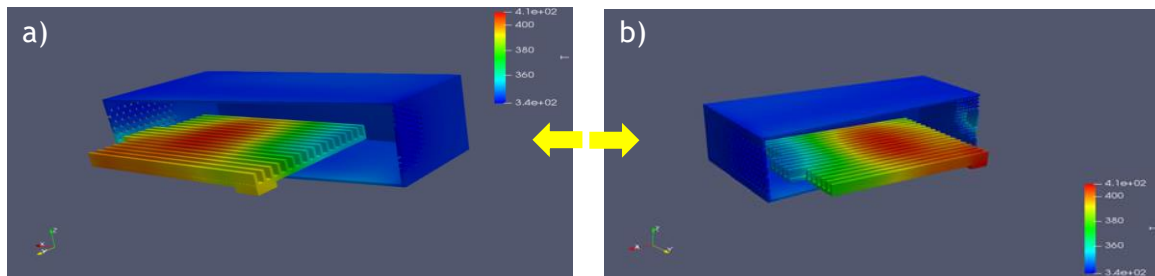


Figure C.1- a) Heat sink temperature profile in Normal flow case; b) Reverse flow case. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

Appendix D - Plots of U vs. Δp

D.1 Plate Fins Thickness

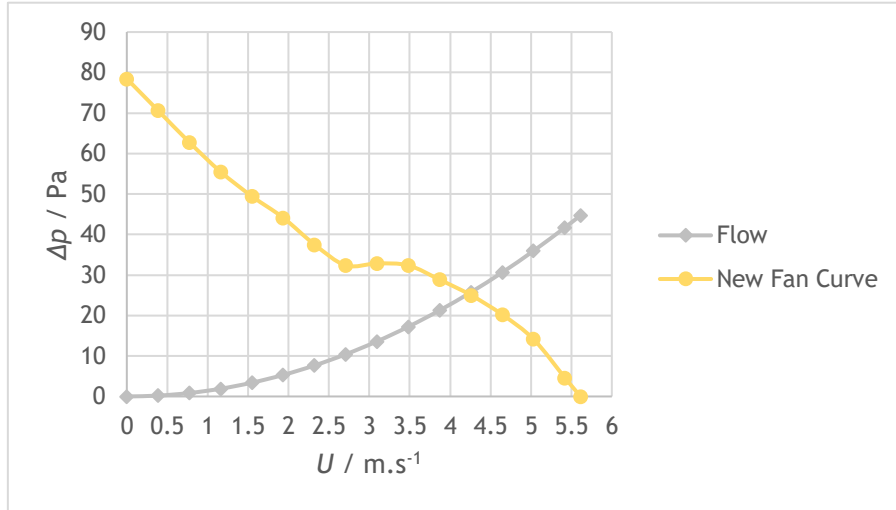


Figure D.1- Flow behavior and fan curve for the 1 mm thickness plate fin heat sink case.

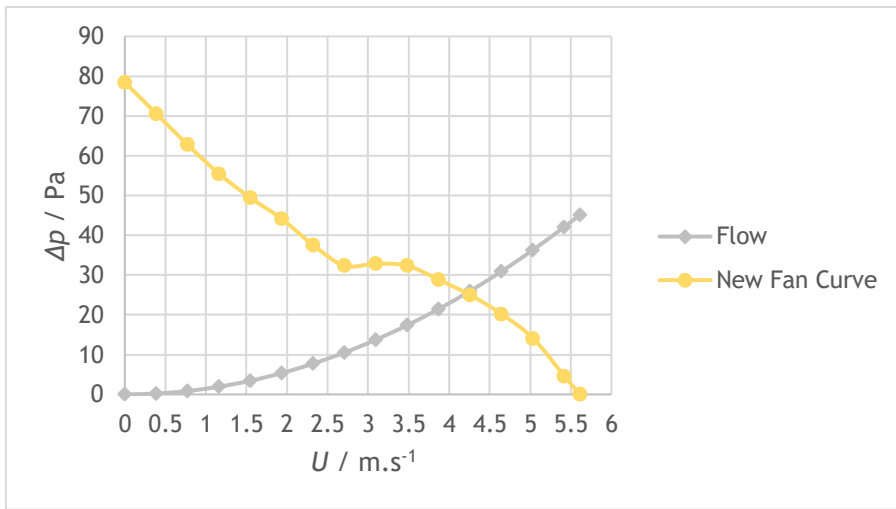


Figure D.2- Flow behavior and fan curve for the 3 mm thickness plate fin heat sink case.

For the 1 mm plate fins heat sink, the operation point is $U = 4.22 \text{ m}\cdot\text{s}^{-1}$ and $\Delta p = 25.4 \text{ Pa}$ and for the 3 mm plate fins heat sink is $U = 4.21 \text{ m}\cdot\text{s}^{-1}$ and $\Delta p = 25.5 \text{ Pa}$.

D.2 Fins Height

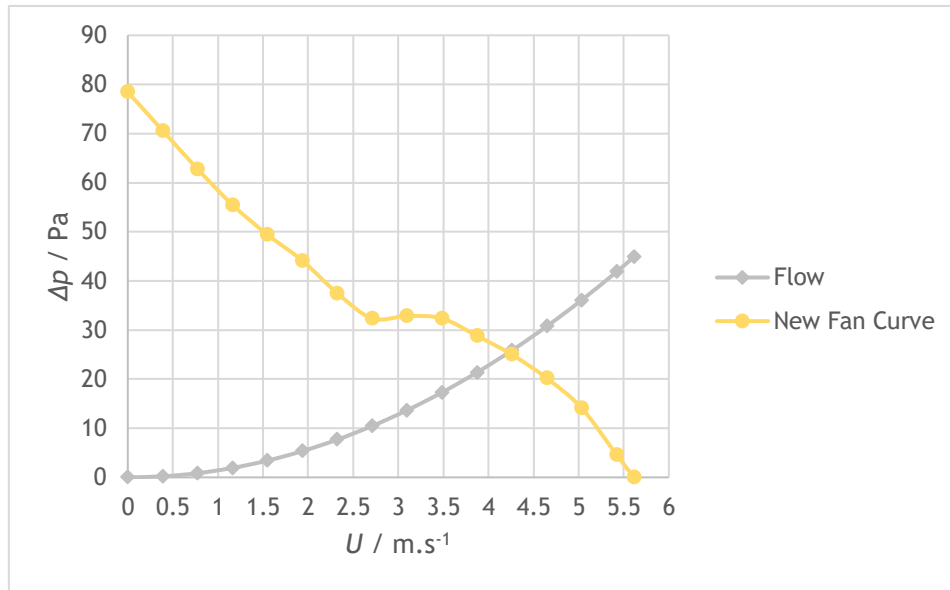


Figure D.3- Flow behavior and fan curve for the high plate fin heat sink case.

For the high plate fins heat sink, the operation point is $U = 4.23 \text{ m}\cdot\text{s}^{-1}$, with $\Delta p = 25.6 \text{ Pa}$.

D.3 Bypass Airflow

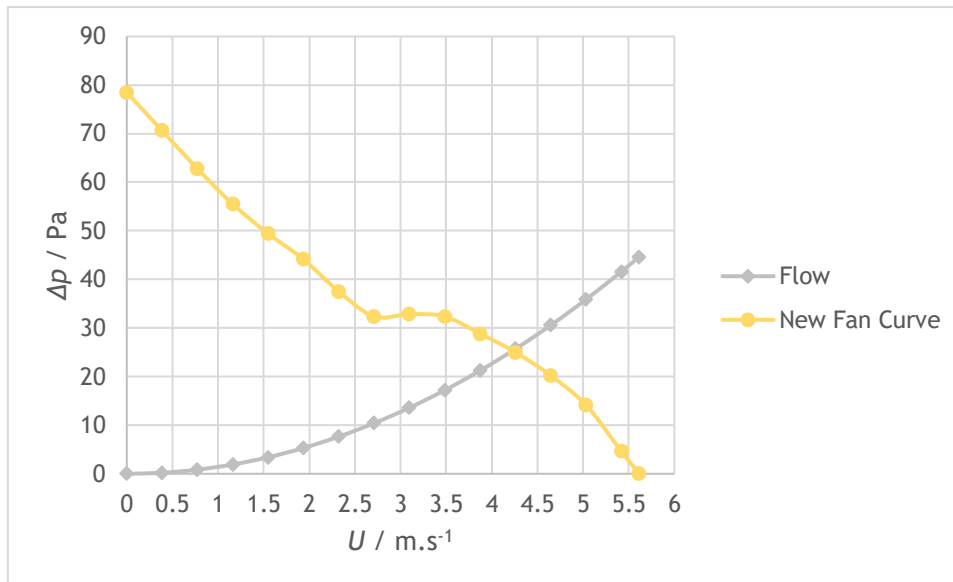


Figure D.4- Flow behavior and fan curve for the bypass airflow case.

For the heat sink optimized to use the bypass airflow effect the operation point is $U = 4.23 \text{ m}\cdot\text{s}^{-1}$ and $\Delta p = 25.6 \text{ Pa}$.

D.4 Plate-Pin

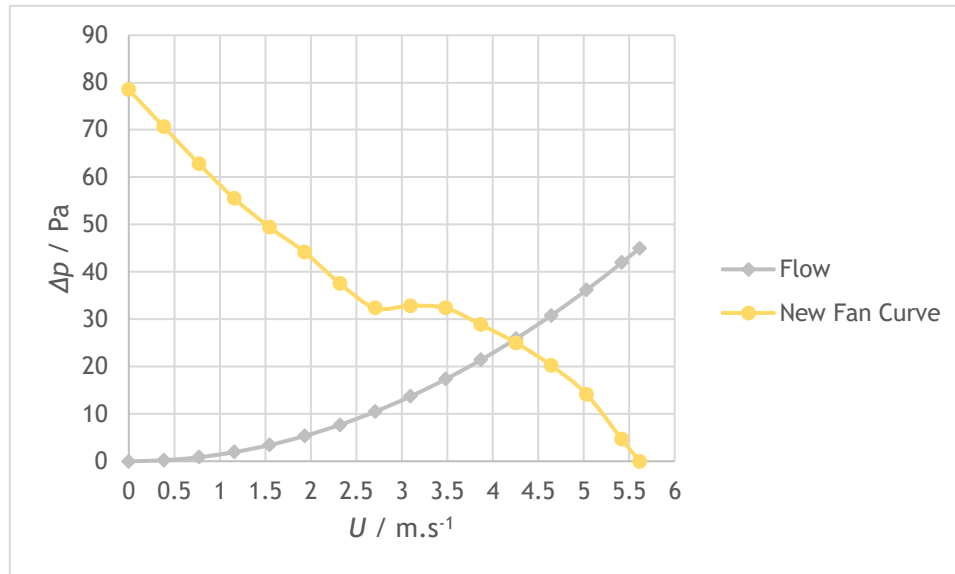


Figure D.5- Flow behavior and fan curve for the small pin plate-pin fin heat sink case.

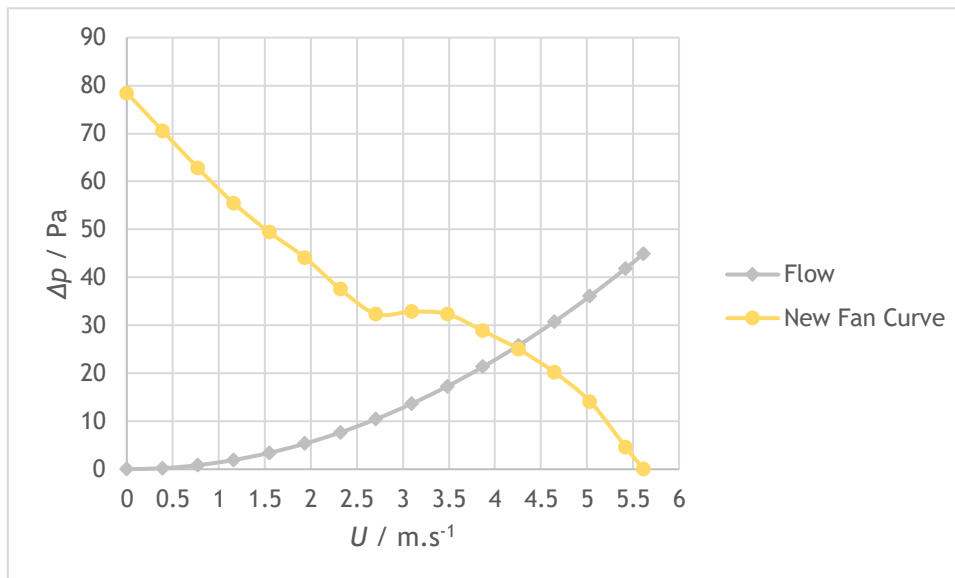


Figure D.6- Flow behavior and fan curve for the medium pin plate-pin fin heat sink case.

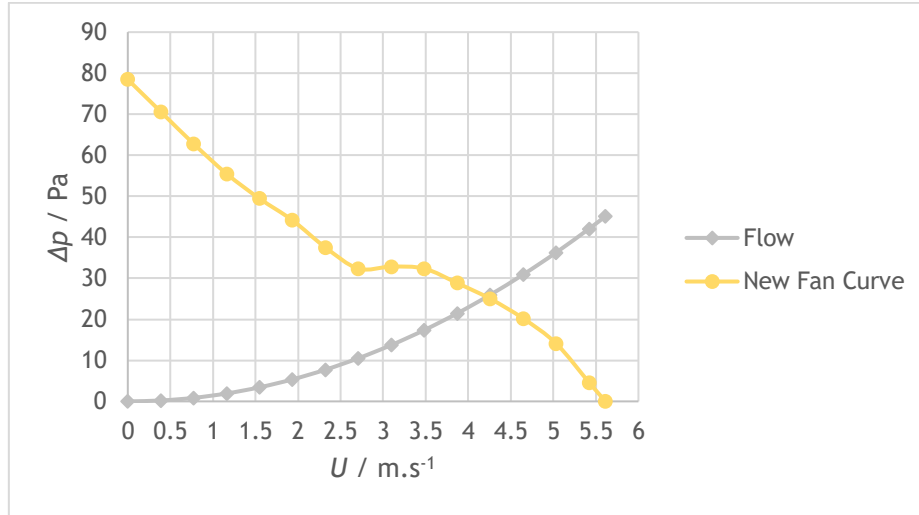


Figure D.7- Flow behavior and fan curve for the high pin plate-pin fin heat sink case.

The operation points are $U = 4.2 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.6 \text{ Pa}$, $U = 4.2 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.5 \text{ Pa}$ and $U = 4.2 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.4 \text{ Pa}$ for small, medium and high pin plate-pin fin heat sinks, respectively.

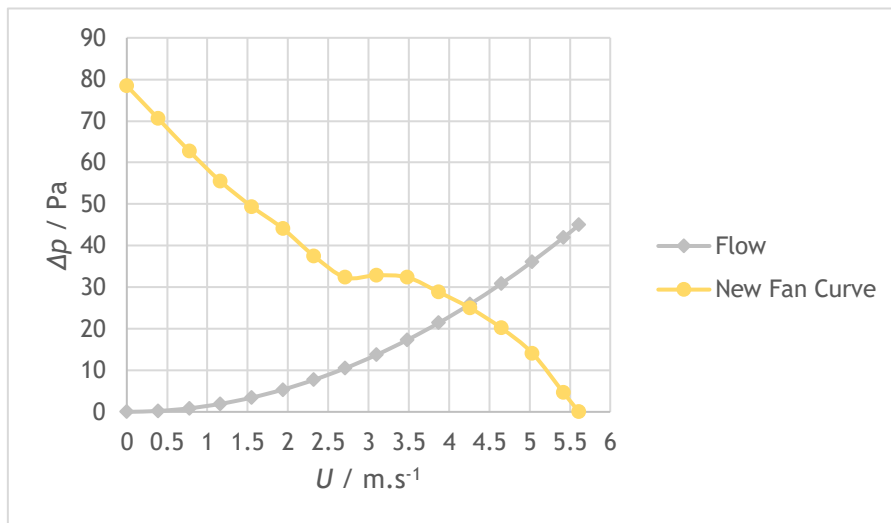


Figure D.8- Flow behavior and fan curve for the drop pin plate-pin fin heat sink case.

The operation point for the drop pin plate-pin fin heat sink is $U = 4.21 \text{ m}\cdot\text{s}^{-1}$, with $\Delta p = 25.4 \text{ Pa}$.

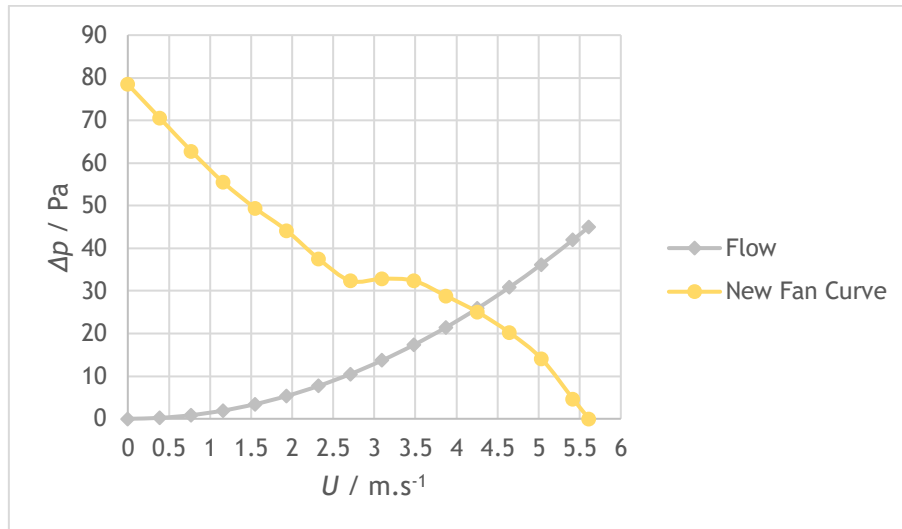


Figure D.9- Flow behavior and fan curve for the staggered pins plate-pin fin heat sink case.

The operation point for the plate-pin fin heat sink with staggered pins is $U = 4.22 \text{ m}\cdot\text{s}^{-1}$ and $\Delta p = 25.6 \text{ Pa}$.

D.5 Pin-Fin

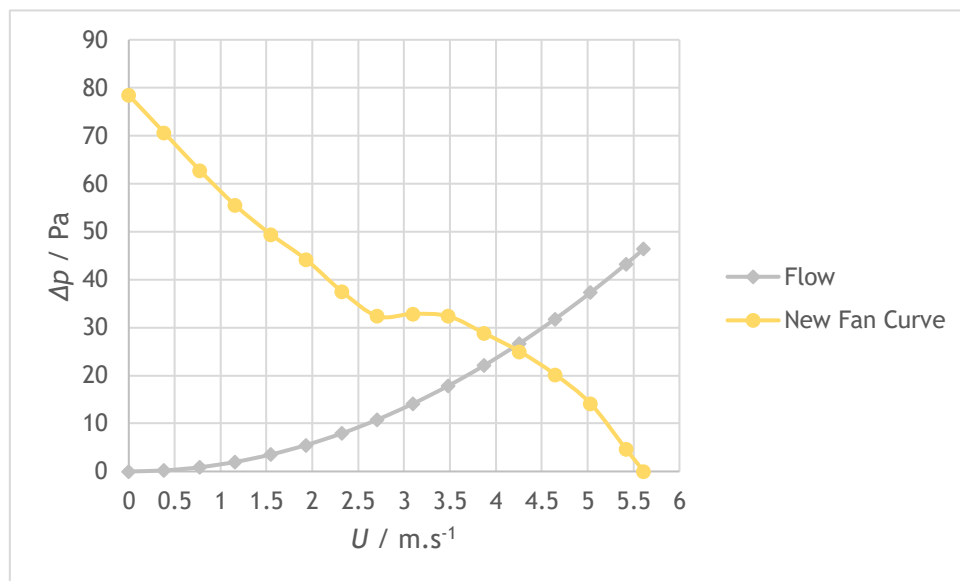


Figure D.10- Flow behavior and fan curve for the 373 pin-fin case.

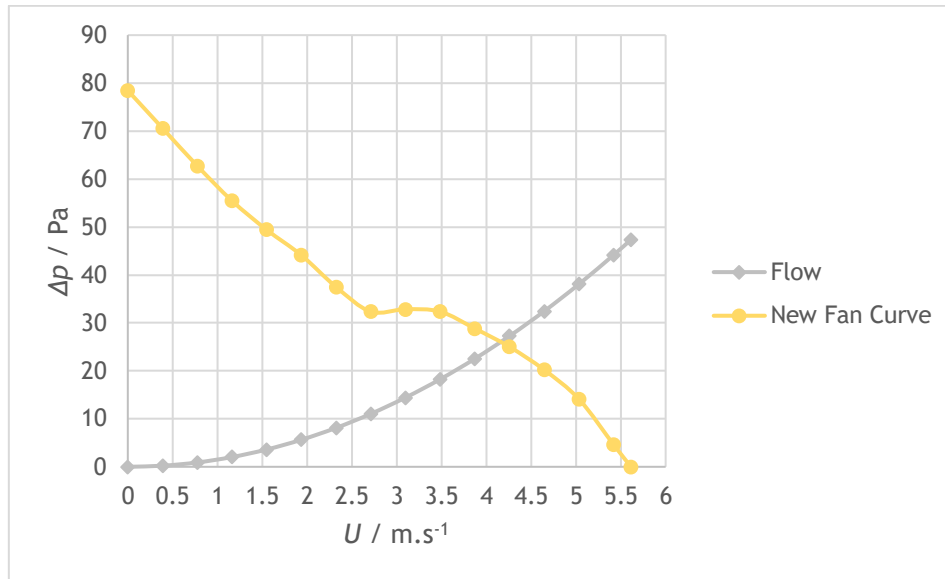


Figure D.11- Flow behavior and fan curve for the 899 pin-fin case.

The 373 pins heat sink operation point is $U = 4.18 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.8 \text{ Pa}$ and with 899 pins is $U = 4.16 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 26.1 \text{ Pa}$.

D.6 Inverse Bypass Airflow

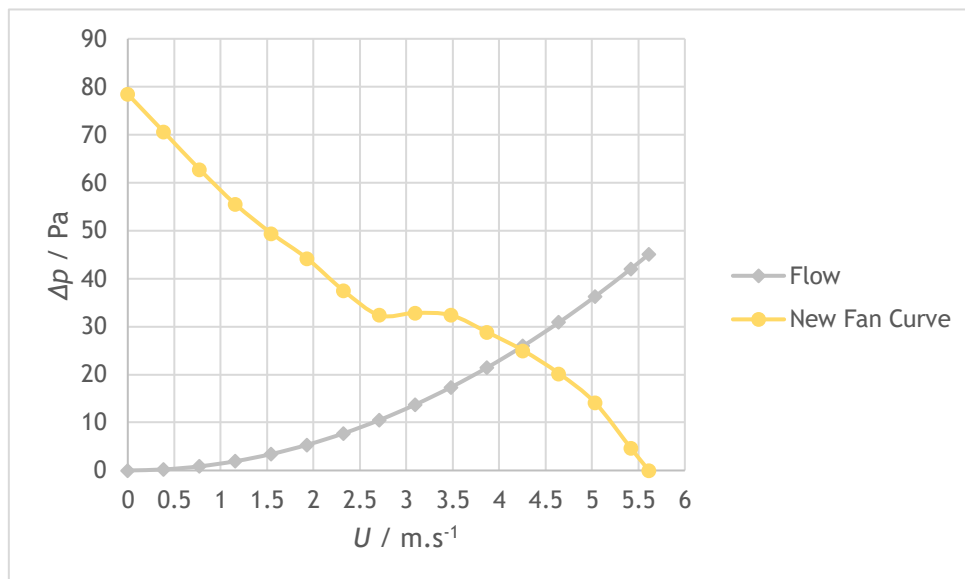


Figure D.12- Flow behavior and fan curve for the inverse bypass airflow case.

The operation point in this case is $U = 4.21 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.5 \text{ Pa}$.

D.7 Base Thickness

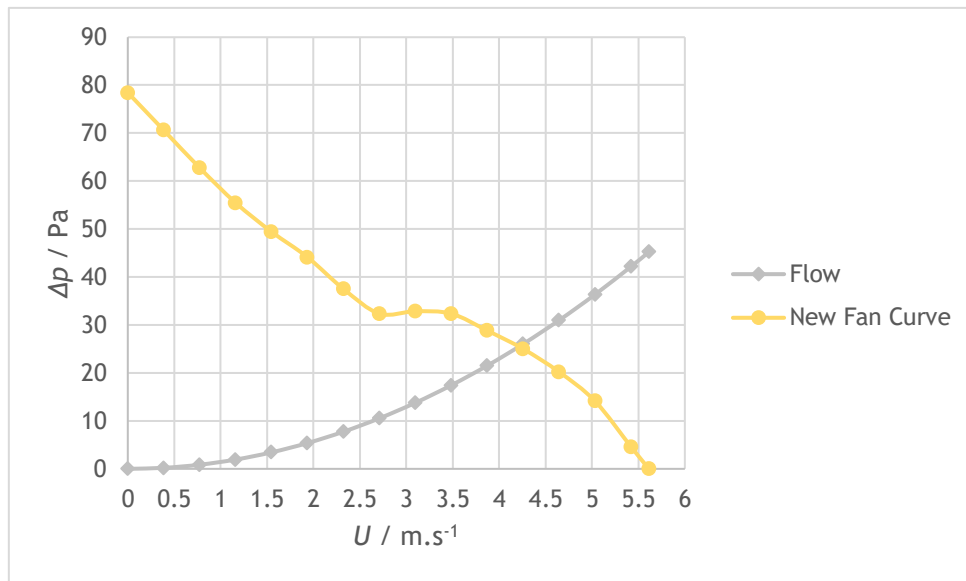


Figure D.13- Flow behavior and fan curve for the thicker base case.

The operation point obtained for the thicker base heat sink is $U = 4.21 \text{ m}\cdot\text{s}^{-1}$ with $\Delta p = 25.6 \text{ Pa}$.

Appendix E - Temperature Reduction associated with the Heat Sink Design

E.1 Plate Fins Thickness

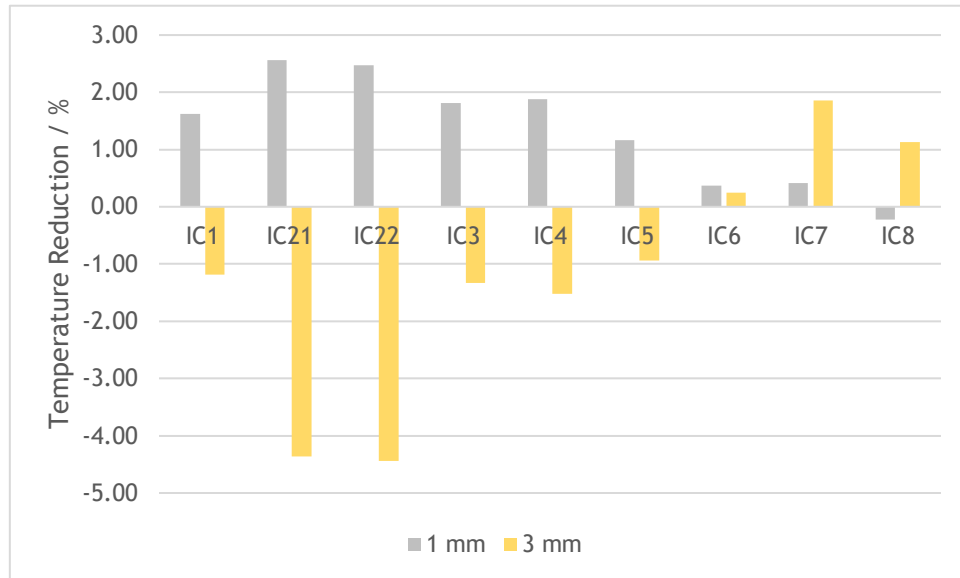


Figure E.1- Temperature reduction of heat sinks with different plate fins thickness when compared with the base case, for each IC.

E.2 Fins Height

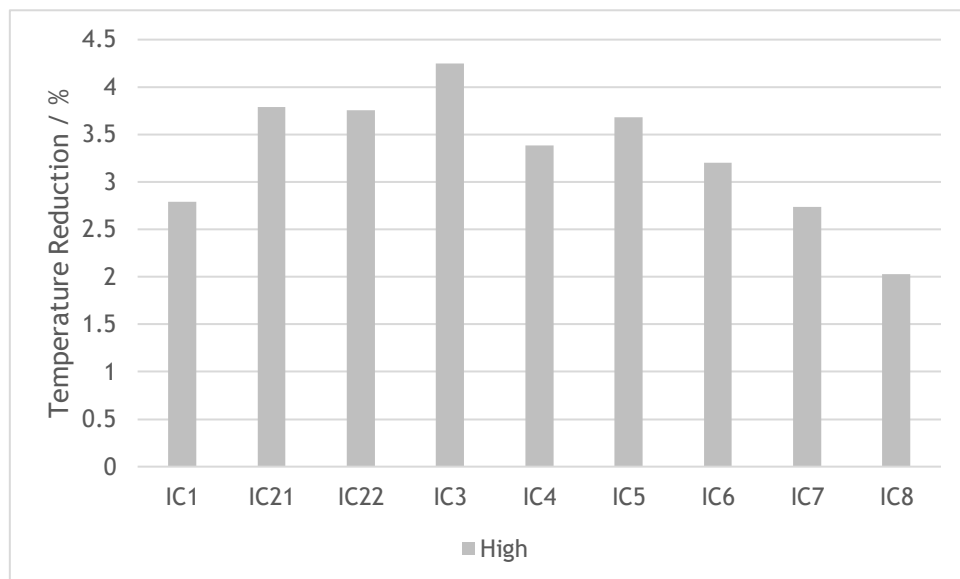


Figure E.2- Temperature reduction of the high plate fins heat sink when compared to the 1 mm plate fins heat sink, for each IC.

E.3 Bypass Airflow

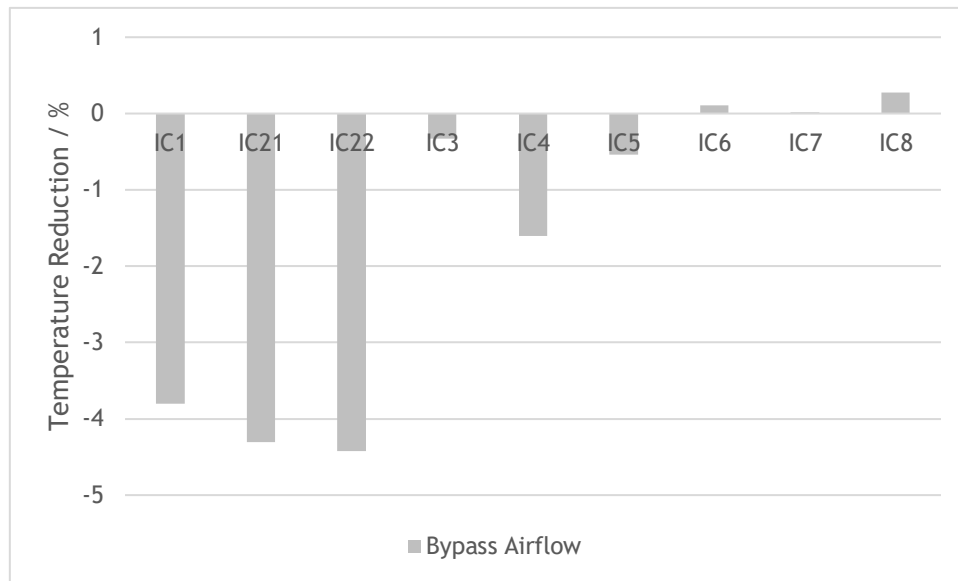


Figure E.3- Temperature reduction of the bypass airflow design heat sink when compared to the high plate fins heat sink, for each IC.

E.4 Plate-Pin

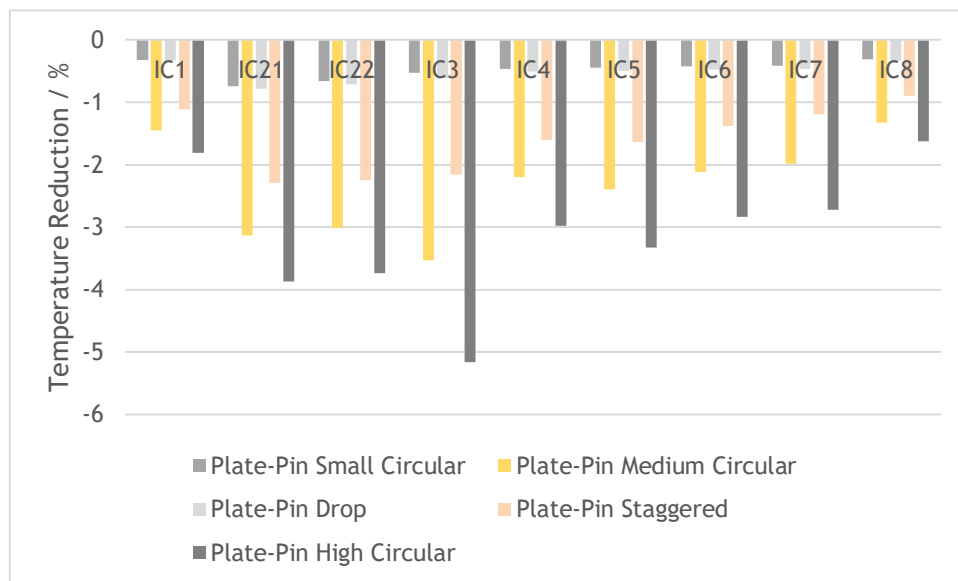


Figure E.4- Temperature reduction of the plate-pin heat sink cases when compared to the high plate fins heat sink, for each IC.

E.5 Pin-Fin

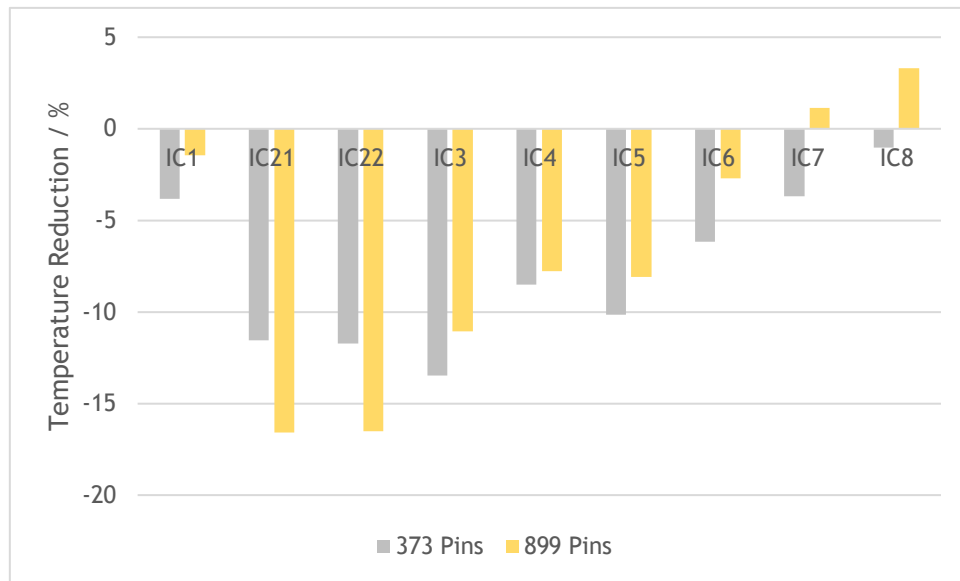


Figure E.5- Temperature reduction of the pin-fin heat sinks when compared to the high plate fins heat sink, for each IC.

E.6 Inverse Bypass Airflow

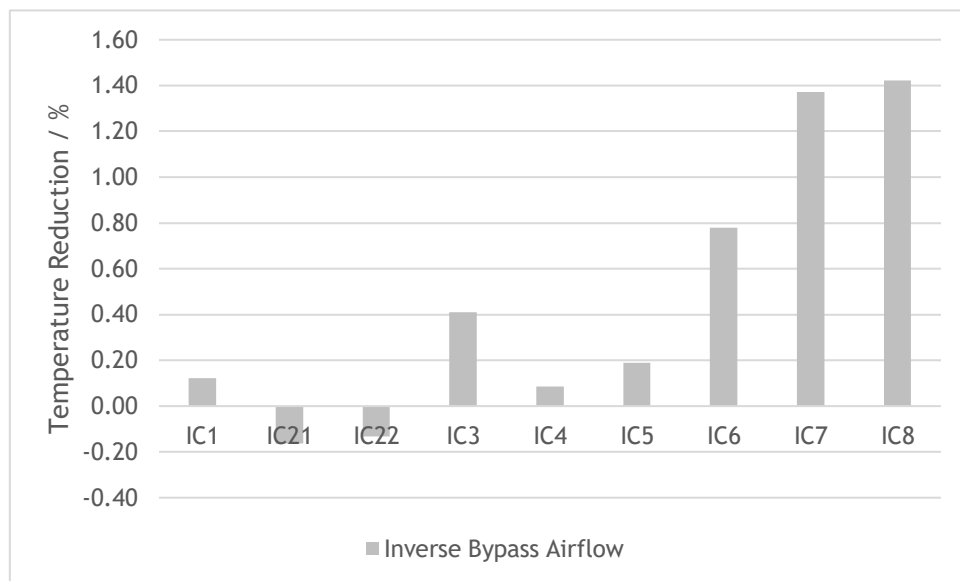


Figure E.6- Temperature reduction of the inverse bypass airflow design heat sink when compared to the high plate fins heat sink, for each IC.

E.7 Base Thickness

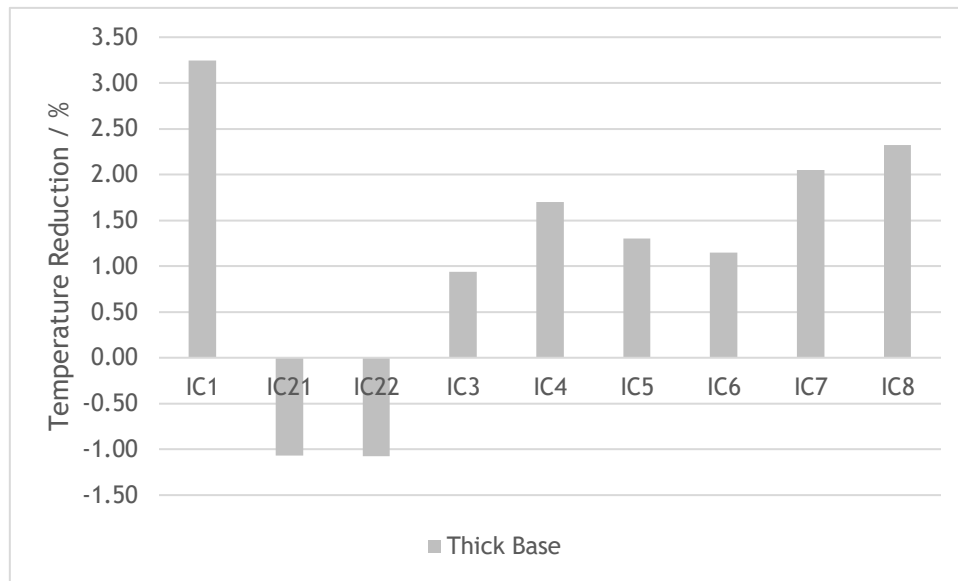


Figure E.7- Temperature reduction of the thick base heat sink when compared to the inverse bypass airflow design heat sink, for each IC.

Appendix F - Main Vortices

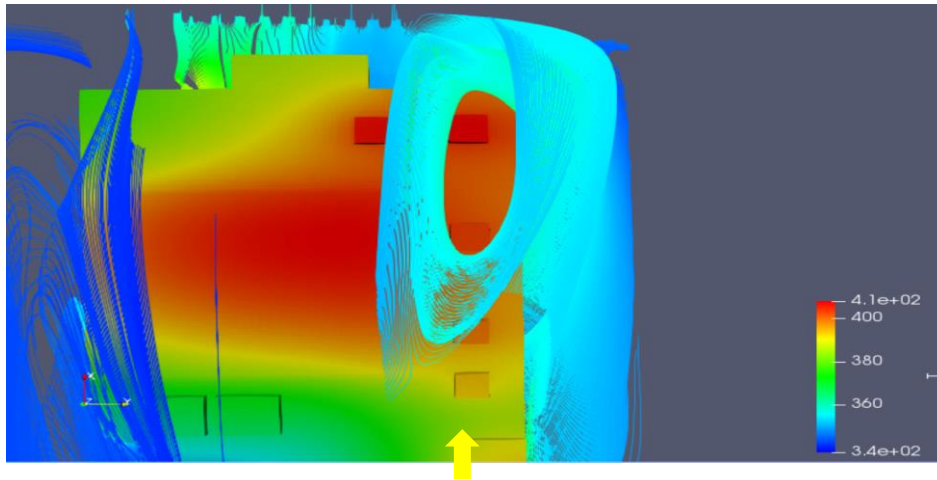


Figure F.1- Vortex in the back of the plate fin heat sink with 1 mm of thickness. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

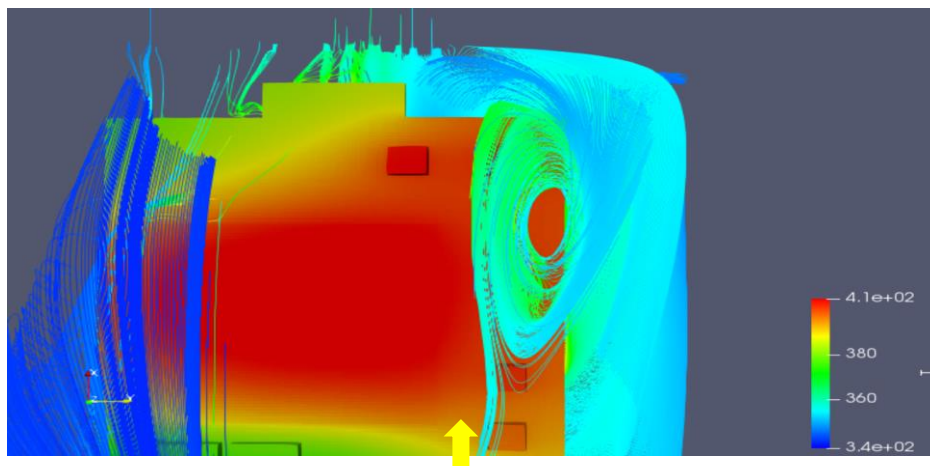


Figure F.2- Vortex in the back of the plate fin heat sink with 3 mm of thickness. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

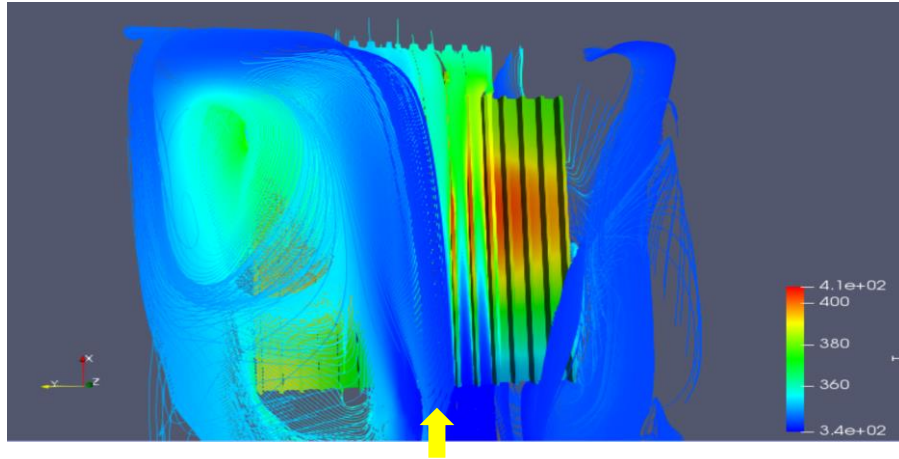


Figure F.3- Vortex in the top of the plate fin heat sink with 1 mm of thickness. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

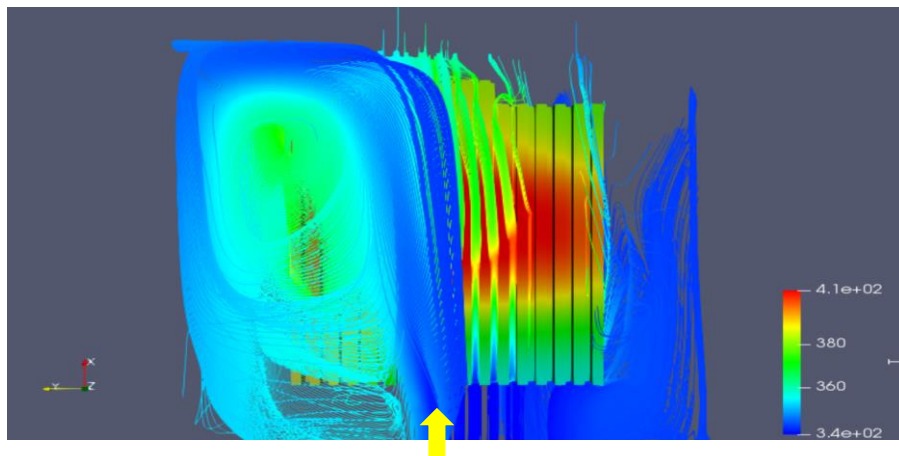


Figure F.4- Vortex in the top of the plate fin heat sink with 3 mm of thickness. The yellow arrow indicates the main flow direction. The color map refers to the absolute temperature (in K).

Appendix G - Pins Base Geometry Results

Table G.1- Temperatures for each piece of the drop pins plate-pin fin heat sink case

Model	Drop Pins		
Piece	Minimum T / °C	Average T / °C	Maximum T / °C
Heat Sink	73.78	111.70	131.66
IC1	112.13	114.53	115.87
IC21	89.05	92.61	95.70
IC22	87.38	90.82	93.94
IC3	101.80	122.82	130.87
IC4	115.82	116.75	117.30
IC5	121.31	121.96	122.36
IC6	125.67	126.28	126.62
IC7	129.44	130.39	130.87
IC8	131.25	131.85	132.12