Automatization of Pin Fin Heat Sink Design with Geometric and Fluid Constraints

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***Abstract*— The sizing of the cooling system is one of the most crucial parts of the design of power electronics since it has a high impact on the overall performance of the packages as well as the lifetime of the chips. The high temperature causes irreversible damage and increases the cost for maintenances and substitutions. However, during the design of the package, some geometric constraints can occur, such as maximum height or width. Moreover, limitations about volume rate and fluid temperature must be taken into account. This work aims to create a tool that can automatize the design of the heat sink cooling system with geometric constraints. The analytical model can consider in-line and staggered configurations. In this study, a genetic algorithm is applied as the optimization algorithm in order to find a solution which respects the boundary and fluid constraints and minimize the heat sink volume. Numerical simulations for the resulted geometries have been performed to validate the tool. The results of the numerical simulations show an error between the numerical and the expected maximum temperature of the plate lower than 2% and 3% for in-line and staggered configuration, respectively. The model can be applied over a wide range of application, and it can be easily adapted to different material and different cooling liquid.**

***Index Terms*—heat sink design, numerical simulations, power electronics, optimisation**

NOMENCLATURE

D	Pin diameter [m]
W	Width of heat sink [m]
L	Length of heat sink [m]
H	Pin height [m]
t	Thickness [m]
S_L	Pin space in the stream-wise direction [m]
S_T	Pin space in the span-wise direction [m]
S_D	Diagonal space [m] $\equiv \sqrt{S_L^2 + 0.25 \cdot S_T^2}$
S_L^*	Normalised stream-wise pitch $\equiv S_L/D$
S_T^*	Normalised span-wise pitch $\equiv S_T/D$
S_D^*	Normalised diagonal pitch $\equiv S_D/D$
N_L	Number of fins in the stream-wise direction
N_T	Number of fins in the span-wise direction
N	Total number of fins
c_P	Specific heat [$J \cdot kg^{-1} \cdot K^{-1}$]
k_f	Thermal conductivity of the fluid [$W \cdot m^{-1} \cdot K^{-1}$]
$\dot{V}$	Volume flow rate [$L \cdot s^{-1}$]
$\dot{m}$	Mass flow rate [$kg \cdot s^{-1}$] $\equiv \dot{V} \cdot \rho \cdot 0.001/60$
μ	Absolute viscosity of fluid [Pa·s]
υ	Kinematic viscosity of fluid [$m^2 \cdot s^{-1}$]
ρ	Density of fluid [$kg \cdot m^{-3}$]
Pr	Prandtl number $\equiv c_p \cdot \mu / k_f$
Q	Dissipated power [W]
Re_D	Reynolds number evaluated with the maximum velocity between fins $\equiv D \cdot U_{max}/\nu$
U	Velocity of fluid [$m \cdot s^{-1}$]
T	Absolute temperature [K]

ASSUMPTIONS

1. Adiabatic lateral wall of the base plate
2. Adiabatic bottom face of fins
3. Isotropic material
4. Uniform velocity
5. Height of the cooling channel = height of the fins
6. Fully developed heat and fluid flow
7. Constant density of the material
8. Constant heat transfer of the material
9. Steady and laminar flow

I. INTRODUCTION

During the last two decades, the reducing size challenge in electronic products has become increasingly popular due to the increase of interest towards the hybrid powertrain development. The increase in the use of power electronics has also increased the attention to reducing costs and space. For this purpose, the design of an efficient cooling system plays a vital role since the maximum allowable temperature of the junctions determines the lifetime of the package. Nowadays, the most used cooling system is the aluminium Pin-Fin heat sink. It provides a good compromise between efficiency and cost due to the vast plate region that reduces the thermal resistance of the package. Two key classification criteria used for heat sinks are the topology and density of the fins, with the topology usually classified as in-line or staggered. (see Fig. 1-a and Fig. 1-b).

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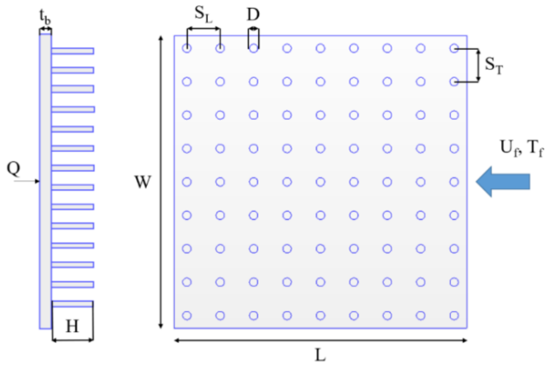

Figure 1-a. Nomenclature for In-Line configuration

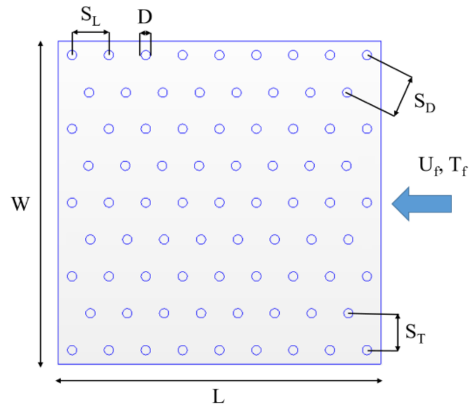

Figure 1-b. Nomenclature for Staggered configuration

Over the years there have been many studies performed into the optimal geometric parameter combinations of Pin-Fin heat sinks. A particular study about the effect of the separation between two fins shows that the optimum separation in span-wise direction is 1.0 ± 0.2 mm $<S_T<3.0\pm0.2$ mm for both in-line or staggered configuration as well as 7.6 ± 0.2 mm and 7.8 ± 0.2 mm in stream-wise direction for in-line and staggered, respectively [1].

The same authors also studied the correlation between the fins' distance and the total dimension. The study shows that the best solution can be achieved inside the range between $0.004\leq S_T/W\leq0.332$ and $0.033\leq S_L/L\leq0.152$ [2]. The technical investigation about the shape of the fins was resolved in 1996 in Sweden. In the KTH laboratories, the best shape for the Pin-Fin turned out to be elliptical or circular for high and mid-range velocity, respectively [3]. The fin space ratio had also been studied in 1998. During this experimental study, the researcher found $2.5D$ both in line and staggered. They also studied the effects of missing fins [4]. In 1996, a study conducted with experimental data discovered that the optimal ratio between the length and the diameter of the cylinder Pin-Fin falls within the range of $6.2\leq H/D\leq20$ [5]. In 2004, a Canadian paper presented the results of an optimisation algorithm that took into account the entropy of the system with respect to the ratio between the plate and fins dimensions [6].

The effect of Pin-Fin density was investigated in 2010. The presented results show that the thermal resistance is influenced by the flow pattern and that the optimal fin spacing ratio S_T/D is 0.5 and 0.3 for 5 mm and 11 mm fin length respectively [7]. Regarding the thermal characteristics, in 1986, studies found a relation between the fin position and the thermal resistance. During the experiments, the bottom tubes immersed in a hot fluid flow presented a different average heat transfer coefficient concerning the higher tubes [8]. One year later, a similar study in Switzerland found the relation between heat transfer and space as well as between pressure drop and space [9]. On the same topic, an experimental study was conducted with two arrays of tubes and the Reynolds number in a range starting from 7500 to 32000. The result shows that the pressure drop and the heat transfer are strongly influenced by the pitch distance [10]. Same results were achieved in Japan, where the variation of heat transfer and pressure drop are compared with the change of the Nusselt number, which represents the ratio between convective and conductive heat transfer [11]. According to the literature, it is thus clear that all the studies optimised a specific geometric parameter concerning either the entropy or the fluid characteristics. In the literature, there are no constraints to space or fluid characteristics. In this study, all the geometric parameter are isolated and constrained to deal with engineering applications.

II. Model Development

The thermal resistance of the heat sink is defined as:

$$R_{hs}=\frac{\Delta T}{Q} \tag{1}$$

where $\Delta T=T_{IGBT}-T_f$ is the difference between the maximum IGBT temperature and T_f which is the temperature of incoming fluid. The dissipated power Q can be divided into three groups:

$$Q=Q_c+Q_{sw}+Q_b\approx Q_c+Q_{sw} \tag{2}$$

where Q_c and Q_{sw} are the conduction and switching losses, respectively. The blocking losses Q_b are normally neglected. Further details are presented in the literature [12, 13].

The total resistance of the heat sink can be written as the sum of three different resistances as:

$$R_{hs}=R_m+R_e+R_t \tag{3}$$

where R_e is the additional resistance between the IGBT and the heat sink is, R_m is the resistance of the base and R_t is the resistance of the fins and the plate (see Fig. 2).

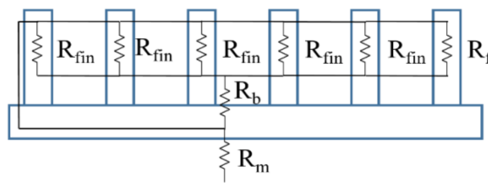

Figure 2. Thermal conductivity model

R_m can be defined as:

$$R_m=\frac{t}{k\cdot L\cdot W} \tag{4}$$

where k is the thermal conductivity of heat sink material. R_t can be written as follows:

$$R_t = \frac{1}{\frac{N}{R_{fin}} + \frac{1}{R_b}} = \frac{1}{\frac{N}{h_{fin} \cdot A_{fin} \cdot \eta_{fin}} + \frac{1}{h_b \cdot A_b}} \quad (5)$$

where R_{fin} and R_b are the thermal resistances of the fins and the plate, respectively, which are in contact with the fluid, A_{fin} and A_b are the contact area fluid-solid of the fin and plate, respectively. The mean heat transfer coefficients h_{fin} and h_b can be written as [14]:

$$h_b = \frac{0.75 \cdot k_f}{D} \cdot \sqrt[2]{\frac{S_T^* - 1}{N_L \cdot S_L^* \cdot S_T^*}} \cdot \sqrt[2]{\mathrm{Re}_D} \cdot \sqrt[3]{\mathrm{Pr}}$$
$$h_{fin} = \frac{C_1 \cdot k_f}{D} \cdot \sqrt[2]{\mathrm{Re}_D} \cdot \sqrt[3]{\mathrm{Pr}} \quad (6)$$

C_1 is a dimensionless constant which depends on the arrangement of the pins, as it can be seen in Equation (7) [14].

$$C_1 = \begin{cases} 0.2 + \exp\left(-0.55 \cdot S_L^*\right) \cdot S_T^{*0.285} \cdot S_L^{*0.212}, in-line \\ \frac{0.61 \cdot S_T^{*0.091} \cdot S_L^{*0.052}}{1 - \exp(-1.09 \cdot S_L^*)}, staggered \end{cases} \quad (7)$$

The contact area can be calculated as:

$$A_{fin} = \pi \cdot D \cdot H$$
$$A_b = L \cdot W - N \cdot \frac{\pi \cdot D^2}{4} \quad (8)$$

The fin resistance includes η_{fin}, which represents the efficiency of the fin and which can be calculated as:

$$\eta_{fin} = \frac{\tanh(m \cdot H)}{m \cdot H}$$
$$m = \sqrt[2]{\frac{4 \cdot h_{fin}}{k \cdot D}} \quad (9)$$

By combining all the equations and isolating the length of the fins H, we obtain the following system:

$$\tau^* \cdot H^{\frac{1}{4}} \cdot \tanh\left(\varepsilon^* \cdot H^{\frac{3}{4}}\right) + z^* - H^{\frac{1}{2}} = 0$$

$$\tau^* = \tau \cdot N \cdot \pi \cdot \frac{C_1^{\frac{3}{2}}}{D} \cdot \sqrt[2]{\mathrm{Pr}} \cdot 2 \cdot \sqrt[2]{\frac{1}{k}} \cdot k_f^{\frac{3}{2}} \cdot \alpha^{\frac{3}{4}}$$

$$z^* = \tau \cdot \frac{0.75}{D} \cdot k_f \cdot \sqrt[2]{\frac{S_T^* - 1}{N_L \cdot S_L^* \cdot S_T^*}} \cdot \sqrt[3]{\mathrm{Pr}} \cdot \sqrt[2]{\alpha} \cdot \left(L \cdot W - \frac{N \cdot \pi \cdot D^2}{4}\right) \quad (10)$$

$$\alpha = \frac{D \cdot c \cdot \dot{V} \cdot 0.001}{L \cdot 60}$$

$$\tau = \left(\frac{\Delta T}{Q} - R_m - R_e\right)$$

$$\varepsilon^* = \sqrt[2]{\frac{4 \cdot C_1 \cdot k_f \cdot \mathrm{Pr}^{\frac{1}{3}} \cdot \alpha^{\frac{1}{2}}}{k \cdot D^2}}$$

$$c = \max\left\{\frac{S_T^*}{S_T^* - 1}, \frac{S_T^*}{S_D^* - 1}\right\}$$

The system presented in Equation (10) contains the geometry parameters and the pump characteristics.
These characteristics can be constrained as follows:

$$\begin{gathered} L_{IGBT} \le L \le L_{MAX} \\ W_{IGBT} \le W \le W_{MAX} \\ 0.004 \cdot W \le S_T \le 0.332 \cdot W \\ 0.033 \cdot L \le S_L \le 0.152 \cdot L \\ \dot{V}_{\min} \le \dot{V} \le \dot{V}_{\max} \\ T_{\min} \le T_f \le T_{\max} \\ H \le H_{\max} \\ t_{IGBT} \le t \le 2 \cdot t_{IGBT} \end{gathered} \quad (11)$$

In the literature, several methods could be used to solve non-linear problems. Most of these methods however, have additional requirements such as convexity or continuity, or they can be applied only to a specific type of problem such as Quadratic Programming.

By considering these limitations, further methods have been implemented, such as Genetic Algorithm, Ant Colony Optimization and Particle Swarm Optimization [15, 16, 17]. The problem is given by nonlinear objective function f, which is to be minimised with respect to the design variable $\overline{x}$={L, W, S_T, S_L, $\dot{V}$, T, D, t} and to the inequality and non-linear equality constraints presented in Equation (11).

The function to minimise is the volume of the IGBT; however since the H is linearly dependent on the design variable $\overline{x}$, it was included into the fitness function as a penalty value, as presented in Equation (12).

$$f_{fitness} = L \cdot W \cdot t + N \cdot \frac{\pi \cdot D^2}{4} \cdot H + (bool)(H > H_{\max}) \cdot 10^6 \quad (12)$$

To solve this problem, a genetic algorithm has been used with 800 generations and 350 as population size. Moreover, the crossover fraction and the constraint tolerance were 0.8 and 0.001, respectively.

III. Refrigerant

TABLE I. Fluid Properties

T [K]	μ [Pa·s]	ν [$m^2 \cdot s^{-1}$]	ρ [$kg \cdot m^{-3}$]	c_p [$J \cdot kg^{-1} \cdot K^{-1}$]	k_f [$W \cdot m^{-1} \cdot K^{-1}$]
278.15	0.00748	6.91400E-06	1081.6		0.382650918
283.15	0.00610	5.64800E-06	1079.0	3250.00	0.385676006
288.15	0.00505	4.69200E-06	1076.3		0.388693026
293.15	0.00424	3.94600E-06	1073.5	3310.00	0.391742873
298.15	0.00360	3.35900E-06	1070.7		0.394868824
303.15	0.00309	2.89000E-06	1067.8	3370.00	0.398095512
308.15	0.00268	2.51300E-06	1064.8		0.401402361
313.15	0.00234	2.20400E-06	1061.8	3420.00	0.404617044
318.15	0.00206	1.94800E-06	1058.6		0.407607237
323.15	0.00184	1.73900E-06	1055.4	3470.00	0.410498206
328.15	0.00164	1.55800E-06	1052.1		0.413453304
333.15	0.00147	1.40600E-06	1048.8	3520.00	0.416569059
338.15	0.00134	1.27800E-06	1045.4		0.419858628
343.15	0.00121	1.16400E-06	1041.9	3560.00	0.423091604
348.15	0.00111	1.06800E-06	1038.3		0.426080508
353.15	0.00102	9.84000E-07	1034.7	3590.00	0.428971478
358.15	0.00094	9.09000E-07	1031.0		0.431944832
363.15	0.00087	8.43000E-07	1027.2	3620.00	0.435041165

The fluid considered for this application is a mix 1:1 of ethylene glycol and water. Table I presents the data used for the calculations. Additionally, all the fluid properties are interpolated over the temperature in order to obtain a continuous domain for the fluid characteristics.
Equation (13) shows the considered interpolations:

$$
\begin{aligned}
a &= 6.16786E-04 \\
b &= 2.11186E-01 \\
c &= -1.449690E-03 \\
d &= 2.896245E-01 \\
e &= 1.113209E+03 \\
f &= 3.439972E-11 \\
g &= 6.348257E-08 \\
h &= -4.875081E-05 \\
i &= 1.994455E-02 \\
l &= -4.608157E+00 \\
m &= 5.811936E+02 \\
n &= -2.907926E+04 \\
o &= -6.5766563467E-05 \\
p &= 2.3757437465E-02 \\
q &= -6E-08 \\
r &= 2E-05
\end{aligned}
\qquad
\begin{aligned}
k_f &= a \cdot T + b \\
\rho &= c \cdot T^2 + d \cdot T + e \\
c_p &= f \cdot T^6 + g \cdot T^5 + h \cdot T^4 + i \cdot T^3 + l \cdot T^2 + m \cdot T + n \\
\mu &= o \cdot T + p \\
\upsilon &= q \cdot T + r
\end{aligned}
\tag{13}
$$

IV. Results

The purpose of this section is to validate the analytical findings with the use of CFD Software STAR-CCM+. The examples are computed assuming aluminium as the material of the heat-sink with a value of 237 $W \cdot m^{-1} \cdot K^{-1}$ for the thermal conductivity. For the optimisation, using the inequalities presented in Equation (11), the following constraints are applied:

$$
\begin{gathered}
32mm \le L \le 60mm \\
18mm \le W \le 45mm \\
0.004 \cdot W \le S_T \le 0.332 \cdot W \\
0.033 \cdot L \le S_L \le 0.152 \cdot L \\
1L/\min \le \dot{V} \le 8L/\min \\
30^\circ C \le T_f \le 80^\circ C \\
3mm \le t \le 6mm
\end{gathered}
\tag{14}
$$

Table II shows the obtained values for the geometry. In order to validate the results, the derived geometry has been replicated into the CFD software. By applying the same boundary conditions, the error between the maximum temperature of the plate recorded by the program and that entered as input by the genetic algorithm is calculated.

TABLE II: CONSTRAINED OPTIMISATION CASES

Parameters	Case A	Case B
L [MM]	50	37
W [MM]	40	26
T [MM]	5	5
N_L	10	18
N_T	8	13
TYPE	IN-LINE	STAGGERED
D [MM]	2	1
S_L [MM]	5	2
S_T [MM]	5	2
H [MM]	4.3	4.2
T_f [°C]	55	55
POWER [W]	1154	2000
T_{MAX} [°C]	140	175
T_{CFD} [°C]	142.63	178.82
ERROR	1.88%	2.18%

Fig. 3 and Fig. 4 present the results obtained with STAR-CCM+ of Case A and Case B, respectively.

The sub-figure .a and .b show the 3D overview and the temperature graph, respectively.

The two sub-figures .b prove that the simulation reaches the steady-state situation where the temperature remains constant.

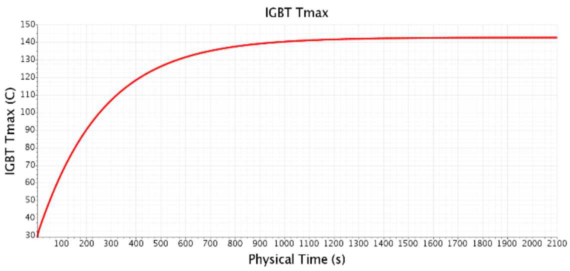

Figure 3-a. IGBT Temperature graph of Case A

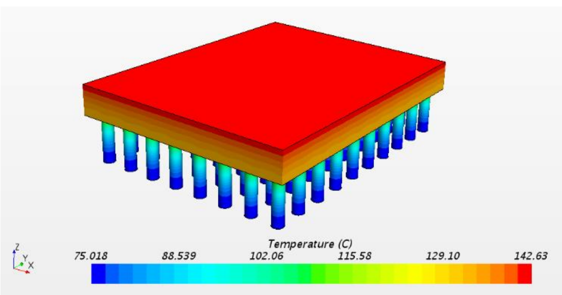

Figure 3-b. 3D View of Case A

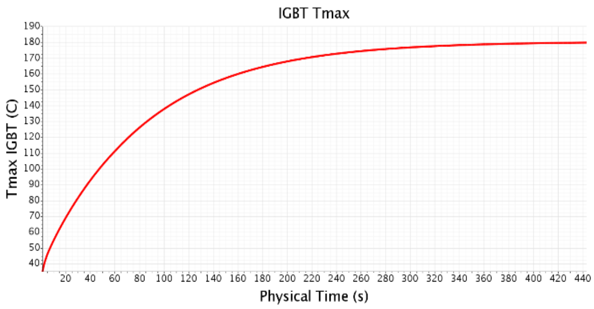

Figure 4-a. IGBT Temperature graph of Case B

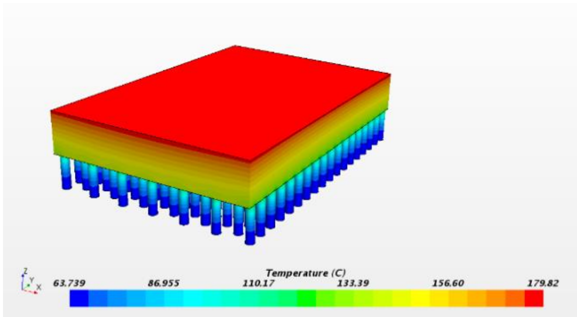

Figure 4-b. 3D View of Case B

V. CONCLUSION

An analytic approach is presented in order to find the optimal heat sink design with geometric and pump constraints. The effects of velocity, pin density, temperature and thermal conductivity are examined with respect to their influence on the design. The designs respect the technical constraints, and the plate temperature exceeds the maximum allowable temperature with an error less of 2% and 3% for in-line and staggered configuration. The difference in temperature between the two setups was found in all the test cases analysed. The higher pressure drop, as well as the change in velocity direction, can be the causes of this deviation.

CONFLICT OF INTEREST

The authors declare no conflict of interest.

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