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Research Article

Optimization of geometric design parameters for a parallel plate fins heatsink of broadcast vehicle system

Phan Hoang Cuong*, **Pham Quoc Hoang**

ABSTRACT

This paper researches a design process of a parallel plate-fins heatsink subjected to an impinging airflow in a broadcast vehicle system. First, it is demonstrated theoretically that the thermal conductance between the plate and the air stream can be maximized by optimizing geometric parameters. Next, the research studies the heat transfer theory across parallel plate-fins of the heatsink to optimize the geometric parameters. Those parameters are used to design a three-dimensional heatsink. In the next step, design model is used to simulate temperature under working conditions by using computational fluid dynamics (CFD). The simulation results show that the maximum temperature is smaller than the maximum permitted temperature of electronic equipment given by the manufacturer. Then, the design model is trial manufactured for testing before moving to mass production.

Key words: Heatsink, broadcast vehicle system, optimal geometric parameters, maximum temperature, CFD

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NOMENCLATURE

Table 1

INTRODUCTION

The broadcast vehicle system (Figure 1) consists of 16 heatsinks, in which each heatsink contains 4 electronic equipment that generates extreme heat of more than 0.4 kW. In addition, this system continuously operates day and night in harsh conditions; in summer, the temperature can reach over 40 ^{0}C. Therefore, under the working conditions, the temperature of the electric equipment may exceed the permitted limit of the given manufacturer, which causes failure and damage to the board itself. This reduces the efficiency, functionality, reliability, and uptime of the whole system. Therefore, installing the heatsink to cool the electronic equipment in the broadcast vehicle system is necessary to release the heat into the environment.

Literature review some of the publications of thermal analysis related to this paper as follows. Kuy Hyung Do *et al.*[1] recommended simplified model solutions of velocity and temperature distributions in heatsink subjected to a uniformly impinging jet. This paper also presented a new method for determining the permeability and heat transfer coefficient analytically.[2] showed a method that finds the optimal geometric parameters of fins to maximize the total heat transfer cooled by natural or forced convection.[3] presented a method and practical results for finding the geometries of fixed volume plate-fins heatsink for maximizing dissipated heat flux. Beriache *et al.*[4,5] developed a simple model for expecting thermal and hydraulic performances of a parallel fin heat sink with impinging airflow.[6] focused on fin shapes' effect by numerically examining the flow and temperature distributions of a plate-fins heatsink cooled by jet impingement. The article also considered three distinct shapes are rectangular, round-headed, and elliptic. This found that the fin shapes affect both the heat transfer rate and pressure drop, especially in the short fin case.[7] had another approach, the authors investigated the pressure drop and flow characteristics of a plate-fin heatsink subject to an impinging flow with the elliptic shape at the bottom. Shah, Amit *et al.*[8] evaluated the possibility of increasing the parallel plate fins heatsink performance by improving the airflow characteristics in the vicinity of the heatsink center. This also studied detail the effect of removing fin material near the central region of the heatsink. Zhipeng Duan[9–11] proposed a simple model of impingement flow pressure drop based on developing laminar flow in rectangular channels. Then the model was experimentally measured pressure drop with various dimensions of a heatsink and flow velocities.

Although those publications fully presented many aspects of the thermal analysis, there are no papers that showed a detail for a specific application and stages

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Table 1: Nomenclature

A	Contact area, m^2	R_{ja}	Thermal resistance from junction to ambient, K/W
A_p	Profile area of fin, m^2	R_{jc}	Thermal resistance from junction to case, K/W
A_c	Profile area of fin, m^2	R_{cs}	Thermal resistance from case to heatsink, K/W
a	Base wall thickness, m	R_{sa}	Thermal resistance from heatsink to ambient, K/W
b	Fin height, m	t	Fin thickness, m
c_p	Specific heat at constant pressure, J/kg.K	T_a	The ambient air temperature, K
G	Total amount of required ventilation	T_b	Temperature at baseplate, K
h	Heat transfer coefficient, W/m^2K	T_{case}	The temperature at its case, K
k	Thermal conductivity of Al, W/mK	T_{jmax}	Maximum junction temperature, K
k_{air}	Thermal conductivity of air, W/mK	T_h	Constant wall temperature of heatsink, K
k_f	Thermal interface material	T_∞	Coolant inlet temperature, K
L	Length of heatsink base, m	$\triangle T$	Temperature difference between outlet and inlet, K
m	Fin parameters, $m = \sqrt{hP_f/kA_c}$	u_∞	Stream velocity, m/s
N	Number of fins	W	width of heatsink base, m
n_f	Number of fans for cooling	z	Fin spacing, m
Nu	Nusselt number		
Nu_{dev}	Nusselt number of the fully developed flow	**Greek symbols**	
Nu_{fd}	Nusselt number of the developing flow	α	Thermal diffusivity, m^2/s
P_f	Perimeter of fin, m	μ	Dynamic viscosity of the fluid, Ns/m^2
Pr	Prandtl number, ν/α	ν	Momentum diffusivity, m^2/s
$\triangle P$	Pressure drop, Pa	ρ	Fluid density, kg/m^3
Q	Power of heat source, W	θ_b	Temperature excess, $\theta_b = T_b - T_\infty$, K
q	Heat transfer rate, W	**Subscripts/superscripts**	
q_v	Volumetric flow rate of each fan, m^3/h	a	ambient
Re	Reynolds number	f	fin

of the heatsink design process. Especially, there have been few studies on impinging flow for cooling the heatsink.[2,3,12–15] mainly focused on the optimization of the spacing between parallel plate-fins of heatsink without considering other geometric parameters such as the height, thickness, and base wall thickness of the heatsink. Thus, the objective of this study is to determine the optimal geometric parameters for maximizing heat transfer from a package of parallel plate-fins that are cooled by forced convection. Those parameters are utilized for designing the heatsink. This model is then analyzed by thermal simulation in CFD. If the maximum temperature at the contact area (case to heatsink) is smaller than the value that is given by the manufacturer, the model is trial manufactured for testing before moving to the mass production period. The rest of this paper is organized as follows: Section 2 determines the optimal geometric design parameters and the number of fans. The results of the three-dimensional design and the thermal analysis are fully presented and discussed in Section 3. Finally, conclusions are made, and future researches are proposed in Section 4.

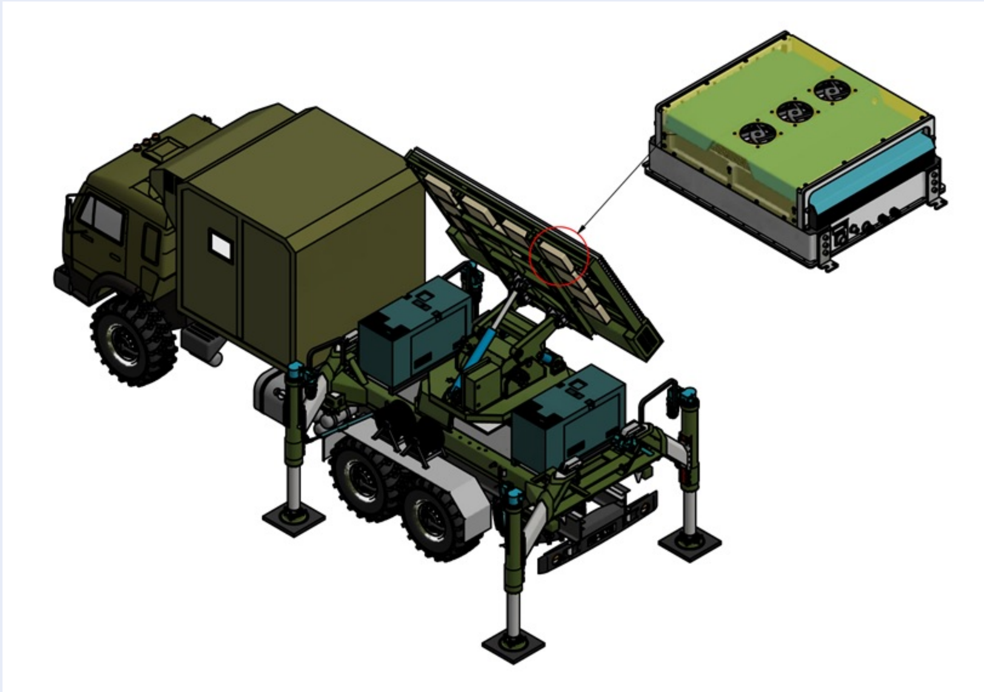

Figure 1: The three-dimensional model of the broadcast vehicle system

MATERIALS AND METHODS

To see clearly the steps of the proposed approach, the paper's research methodology is shown in Figure 2. With some the restricting geometrical limits and given parameters, determining the optimal geometric parameters of the heatsink and the number of fans. Then those parameters are used to design the heatsink before simulating the temperature. If the maximum temperature does not exceed the T_{case}, which the manufacturer gives, the model is manufactured.

In the initial stage of the design process, based on the size of the electronic equipment mounted on the base of the heatsink, the sizes and locations of 16 heatsinks in the overall layout of the broadcast vehicle system, the designer's intentions, manufacturing constraints, and for other factors, then the width W and length L of the heatsink are preselected 0.400x0.477 m, respectively. Thus, with known power dissipation of electronic equipment Q, the maximum permitted temperature of electronic equipment T_{case}, this part of the paper is going to figure out the optimal geometric design parameters as the base wall thickness a, the fin height b, the fin thickness t, the fin spacing z, the number of plates N, and the number of fans for cooling.

Each heatsink consists of 4 heat sources attached at its base, sizes and power are shown in Table 2.

Table 2: The sizes and power of electric equipment

No.	Length (m) x Width (m)	Power (W)
1^{st}	0.137x0.058	30
2^{nd}	0.385x0.050	5
3^{rd}	0.385x0.112	328
4^{th}	0.385x0.090	40

The geometry of the heatsink subject to an impingement flow is shown schematically in Figure 3. The flow enters at the middle top and exits out the sides. Fluid uniformly impinges on the heatsink along the z-axis, then flows parallels to the x-axis. The following assumptions are made to the model of the heatsink and fluid flow:

- Steady-state;
- The fluid is incompressible;
- Laminar flow;
- Constant fluid and solid properties;
- Negligible viscous dissipation and radiation heat transfer.

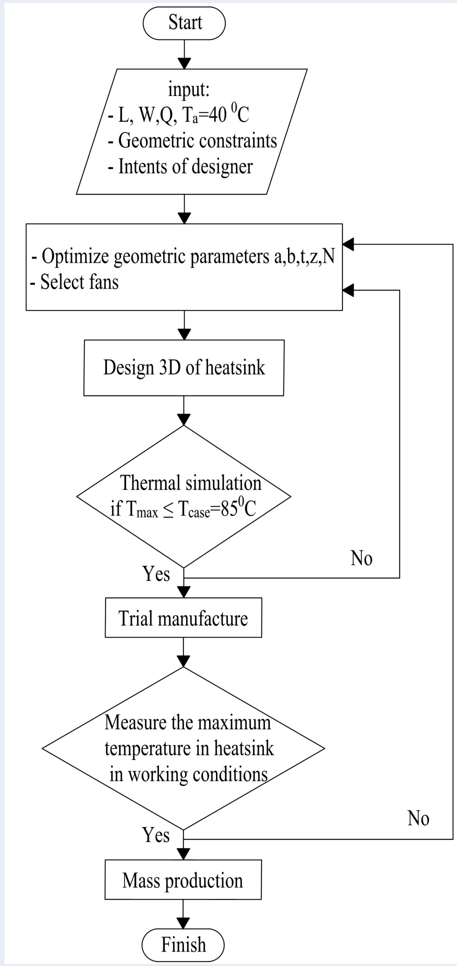

Figure 2: Flowchart of the research process of the heatsink

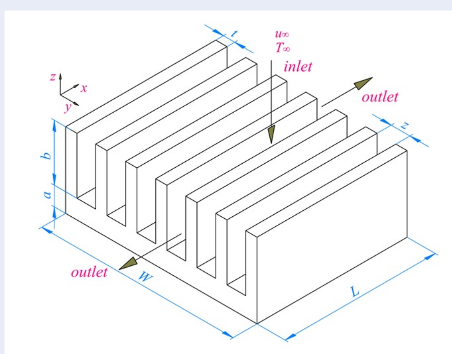

Figure 3: Model of the heatsink

The number of fins

Based on Figure 3 the number of parallel plate-fins is determined as follows:

$$N = \frac{W+z}{t+z} \quad (1)$$

Optimize spacing of parallel plate-fins

The research analyses forced convection cooling by fans that set up above the heatsink, the physical basis of the existence of an optimal spacing between parallel plate-fins z for maximum heat transfer as follows. On the one hand, the heat transfer decreases when $z \to 0$ because the fluid ceases to penetrate through the fins. On the other hand, when $z \to \infty$ the heat transfer also declines because the fins surfaces disappear. Therefore, there exists an optimal z, that is z_{opt} for which the heat transfer is maximum. The optimal parallel plate-fins z_{opt} can be determined by the interesting the total heat transfer rate of the heatsink's small and large channel spacing.

Consider the first case of the channel has small spacing; when the spacing between parallel plate-fins small enough, the flow is fully developed all along with L. The total heat transfer rate removed is given by[12]:

$$q = \rho\left(\frac{z^2 \triangle P}{12\mu L}\right) bWc_p\left(T_h - T_\infty\right) \quad (2)$$

Eq. (2) presents that the total cooling rate reduces as z^2. This tendency is showed as curve (red line) in Figure 4.

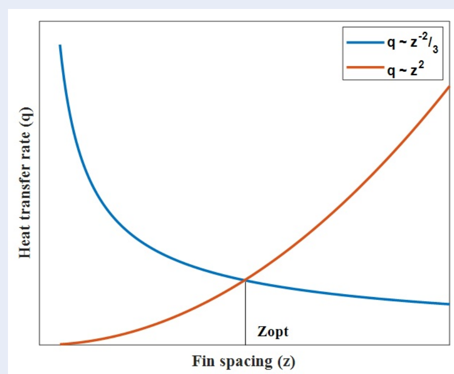

Figure 4: The optimal z_{opt} value

Consider the second case of the channel has large spacing. In this case, each boundary layer is isolated, the flow is developing. Hence, the total heat transfer rate released[12]:

$$q = 1,208Wbk_{air}\left(\frac{Pr\triangle PL}{\rho v^2}\right)^{1/3} z^{-2/3}\left(T_h - T_\infty\right) \quad (3)$$

Eq. (3) presents that the total cooling rate reduces as $z^{-\frac{2}{3}}$. This tendency is shown as a curve (blue line) in Figure 4. To optimize the spacing between the fins, the red and blue line curves intersect z_{opt}. This parameter is expressed by:

$$z_{opt} = 3,24LRe^{-\frac{1}{2}}Pr^{-\frac{1}{4}} \quad (4)$$

Optimize the thickness and height of parallel plate-fins

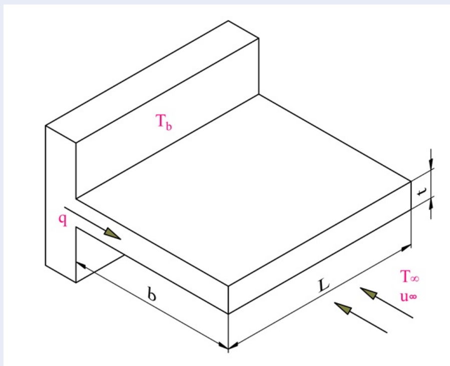

Figure 5: Heat transfers through fins

Heat transfer from a fin with the adiabatic tip is described in [16,17]:

$$q_f = \sqrt{hP_f kA_c}\ \theta_b tanh\, mb \quad (5)$$

Assuming that $L\square\ t$, $P_f \approx 2L$, $A_c = Lt$, $A_p = bt$ and $mb = \sqrt{hP_f/kA_c b}$, then (5) becomes $q_f = \sqrt{h2LkLt}\theta_b tanh\sqrt{h2L/kLtb}$. This is rewritten:

$$q_f = (2hk)^{1/2} L\theta_b t^{1/2} tanh\alpha \quad (6)$$

herein, $\alpha = mb = \sqrt{2h/kt}b$

Considering heat transfer rate at the base wall between two fins

$$q_w = hLz\theta_b \quad (7)$$

The total heat transfer rate is given by:

$$\begin{aligned} q &= q_f + q_w \\ &= NL\theta_b\left[(2hk)^{1/2} t^{1/2} tanh\alpha + hz\right] \end{aligned} \quad (8)$$

To find out the maximum total heat transfer with respect to t, one sets the derivative equal to zero:

$$\frac{dq}{dt} = 0 \quad (9)$$

Thus, the optimal thickness of fins is received:

$$t_{opt} = \frac{z}{2(W-z)L\theta_b - 1} \quad (10)$$

The convection coefficient is taken from the model proposed by P. Teerstra [17]. It is given by:

$$h = \frac{Nu.k_{air}}{z} \quad (11)$$

In parallel plate-fins the flux is found developing conditions, fully developed conditions or combination of both conditions. [18] proposed a composite solution:

$$Nu = \left[(Nu_{dev})^{-n} + \left(Nu_{fd}\right)^{-n}\right]^{-\frac{1}{n}} \quad (12)$$

The Nusselt number of the fully developed flow asymptote can be written:

$$Nu_{fd} = \frac{1}{2}\frac{z}{L}RePr \quad (13)$$

The Nusselt number of the developing flow asymptote is defined by:

$$Nu_{dev} = 0.664\left(\frac{z}{L}Re\right)^{0.5} Pr^{1/3}\left(1 + \frac{3.65}{\left(\frac{z}{L}Re\right)^{0.5}}\right)^{0.5} \quad (14)$$

where, n is combination parameter that depends on the model, in the model that proposed by [17], n has the value of 3.

Therefore, the optimal height of fin is calculated by:

$$b_{opt} = \alpha\left(\frac{kt}{2h}\right)^{\frac{1}{2}} \quad (15)$$

Determine the base thickness

The thermal resistance from the case to ambient is determined by [19]:

$$R_{ja} = R_{jc} + R_{cs} + R_{sa} \quad (16)$$

The total thermal resistance of the heatsink from junction to ambient is given:

$$R_{jc} = \left(T_{j_{max}} - T_a\right)/Q \quad (17)$$

The thermal resistance from the case to the heatsink:

$$R_{cs} = a/k_f A \quad (18)$$

The thermal resistance from the heatsink to ambient:

$$R_{sa} = (T_s - T_a)/Q \quad (19)$$

From Eqs. (16) - (19), one gets:

$$a = k_f A\left(R_{ja} - \left(R_{jc} + R_{sa}\right)\right) \quad (20)$$

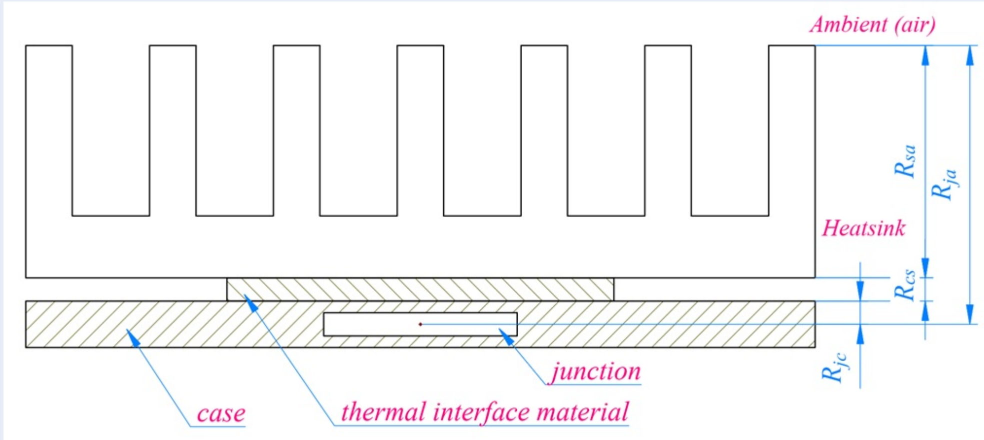

Figure 6: Thermal resistance of the heatsink

Determine the number of fans

Fan selection plays an important role in cooling the system. The required volume flow rate G depends on heat dissipation Q, the temperature difference between inlet and outlet of the heatsink $\triangle T = T_{out} - T_{in}$, fluid density ρ, and specific heat c_p. Therefore, the total amount of required ventilation for cooling is determined by[20]:

$$G = Q/\left(\rho.c_p\left(T_{out} - T_{in}\right)\right) \quad (21)$$

The number of fans that needs to be installed:

$$n_f = G/q_v \quad (22)$$

Based on the cooling requirements and the designer's intentions, the 8314HU axial fan is selected. The maximum volumetric flow rate of this fan is $q_v = 47.1\ (CFM) \approx 79.85\ \left(m^3/h\right)$. The number of fans is determined from Eqs. (21) and (22) is $n_f \approx 2.98$. Thus, the number of fans is 3.

RESULTS AND DISCUSSION

Design three-dimensional of the heatsink

Initial values for determining the optimal geometric design parameters are listed in Table 3.

From Eqs. (4), (10), (15), and (20) it is easy to determine the optimal geometric parameters, which are listed in Table 3.

The optimal geometric parameters from Table 4 are utilized to design the three-dimensional model of the heatsink as shown in Figure 7.

Table 3: The initial values of some design parameters

No.	Denotation	Value	Unit
1	L	0.477	m
2	W	0.400	m
3	u	4.6	m/s
4	v	1.702x10-5	m^2/s
5	$Re = uL/v$	128920	
6	Pr (40 ^{0}C)	0.7255	
7	k_{air}	0.02662	W/m.K
8	k	205	W/m.K
9	T_{jmax}	225[21]	^{0}C

Table 4: The optimal geometric parameters of the heatsink

No.	Denotation	Value	Unit
1	a	$10x10^{-3}$	m
2	b	$60x10^{-3}$	m
3	t	$2.1x10^{-3}$	m
4	z	$4.5x10^{-3}$	m
5	N	60	
6	n_f	3	

Results of thermal analysis

In order to evaluate the thermal response of the three-dimensional heatsink under working conditions. The thermal analysis aims to allow the designer to quickly evaluate whether the theoretical calculation results

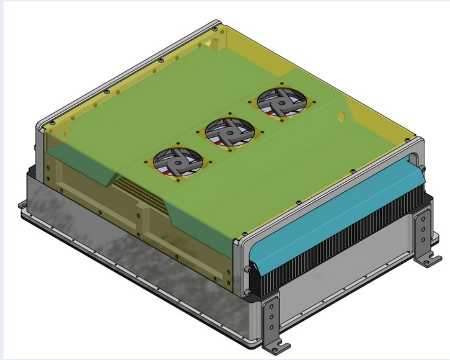

Figure 7: Three-dimensional of the heatsink

are reasonable or not. In addition, the thermal simulation also shows the distribution and maximum temperature under working conditions of the heatsink so that the designer can quickly adjust the design parameters to meet the given requirements. Besides, the designer can predict the critical positions quite accurately, thus helping them make some important decisions in the initial design process.

The three-dimensional heatsink as shown in Figure 7 is simply modified for thermal simulation. The boundary conditions for the thermal analysis are as follows: the heatsink is made from 6061 aluminum. The input temperature of the simulation process is equal to the ambient temperature at $T_\alpha = 40\ ^0C$. There are three 8314HU axial fans set up above the heatsink. The electronic equipment is BLS9G2731L-400 LDMOS S-band radar power transistor, 4 heat sources located at different positions as illustrated in Figure 8. The electronic equipment integrates at the bottom of the heatsink, in which dimensions and power dissipation are listed in Table 2.

Thermal analysis was conducted by Solidworks simulation tool. The figures below show the temperature distribution results in the heatsink and velocity trajectory of the airflow.

Discussions

Table 4 and Figure 7 show the optimal geometric parameters and the three-dimensional design of the heatsink, respectively. The values of the design parameters in Table 4 are rounded off to satisfy manufacturing constraints. These values are used to design the heatsink as shown in Figure 7 that consists of 4 heat sources, 3 fans set up above the heatsink, a plate to cover the heatsink and mount the fans, 4 support brackets to mount the heatsink into the frame of the broadcast vehicle system.

In Figure 9 and Figure 10, numerical simulations for the temperature and velocity distribution are obtained by using a tool in Solidworks simulation. Overall, the high-temperature field distributes in the area where the electronic equipment is attached to the heatsink. Figure 9a and Figure 9b are the side view and bottom view of the heatsink; these figures clearly show the maximum temperature appears at the bottom where the third electronic components directly integrate with the heatsink; the highest temperature is approximately 55.7 ^{0}C. On the contrary, the lowest temperature is equal to ambient occurs near the input flow of fans, which is roughly 43 ^{0}C. Figure 9c shows the temperature field of the cross section view, the simulation in fins illustrates that the temperature gradually decreases from the bottom to the top. Also, the temperature and the trajectory velocity of airflow are nearly symmetric through the plane of symmetry of the heatsink. As can be seen from Figure 10, the velocity of air flow where sets up the fans is maximum, but the flow stagnates near the center of the heatsink and the under surface of the cover plate. This results in slowing down the heat transfer rate at the center of the heatsink. The results of the simulation process show that the maximum temperature in working conditions is smaller than the given maximum temperature of the manufacturer $T_{case} = 85\ ^0C$[21]. Therefore, the model design is manufactured. Some of the images of the trial manufacture process are as follows:

CONCLUSIONS

The main conclusions from the research results of the current work can be drawn as follows. Thermal analysis was studied to figure out the optimal geometric design parameters of the heatsink. The heatsink was designed based on these parameters, by using the Solidworks simulation tool to simulate the maximum temperature under working conditions. The simulation results show that the maximum temperature of the model design is smaller than the upper limit of the electronic equipment. The design model is then produced for testing the maximum temperature in the laboratory before moving on to the mass production process. The products were installed on the broadcast vehicle system to meet the given requirements of the whole system.

The height, spacing, and thickness of parallel plate-fins are roughly 60 mm, 4.5 mm, and 2.1 mm, respectively. Thus, it is necessary to fully study the effect of process parameters in the high-speed machining of thin walls in the future. Moreover, it is necessary to research other main factors that affect productivity

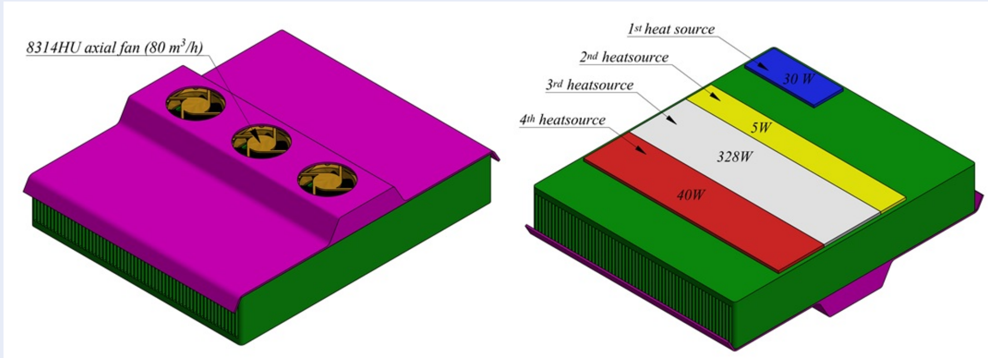

Figure 8: The boundary conditions for thermal simulation

and quality of machining processes (surface roughness) such as tools, machines, tool wears, force cutting, and tool life.

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CONFLICT OF INTEREST

All of the authors have no conflict of interest in the publishing of the paper

AUTHOR CONTRIBUTION

Pham Quoc Hoang proposed ideas, research methods, quality monitoring, and correcting the manuscript. Phan Hoang Cuong wrote the manuscript.

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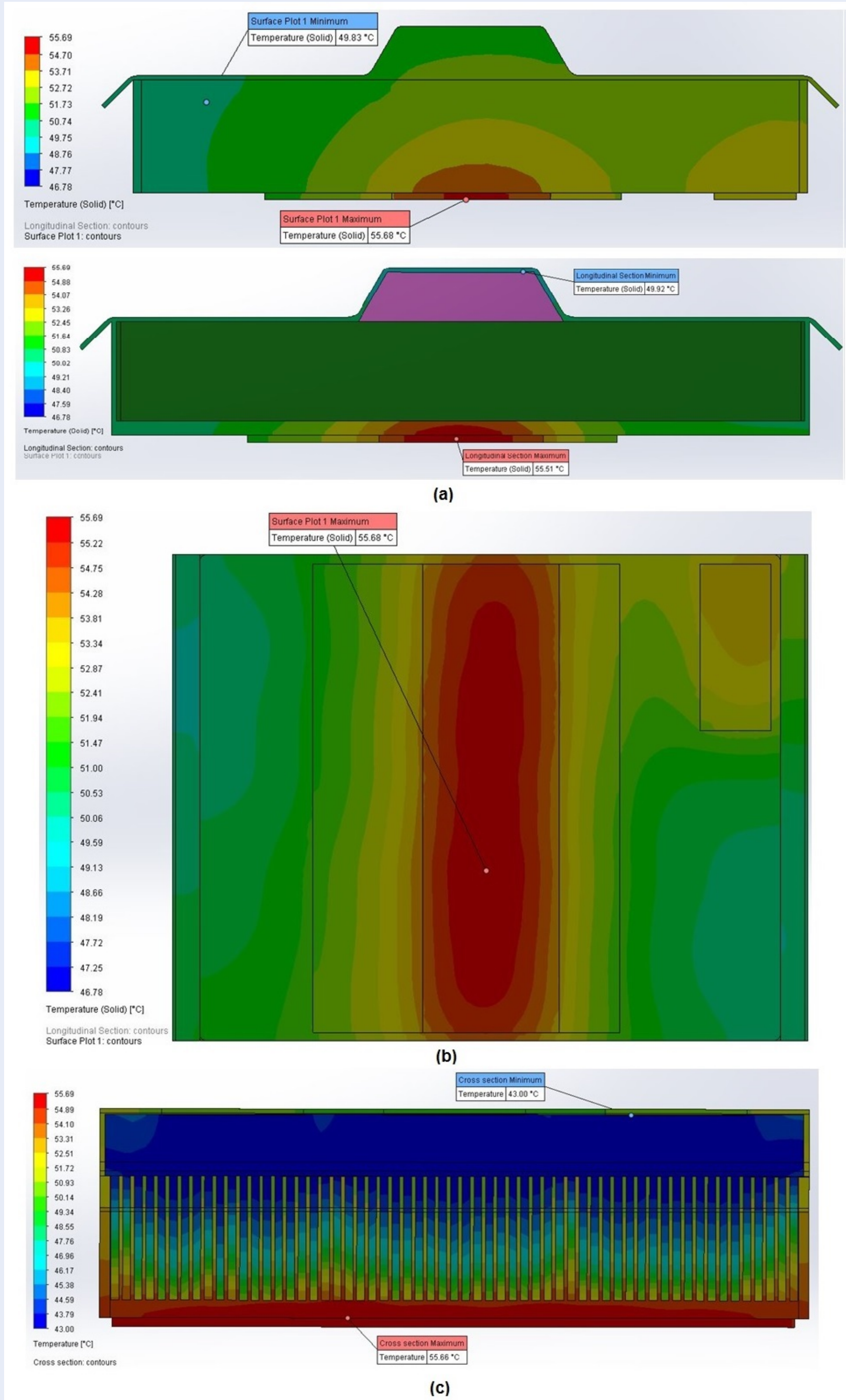

Figure 9: The temperature distribution in the heatsink

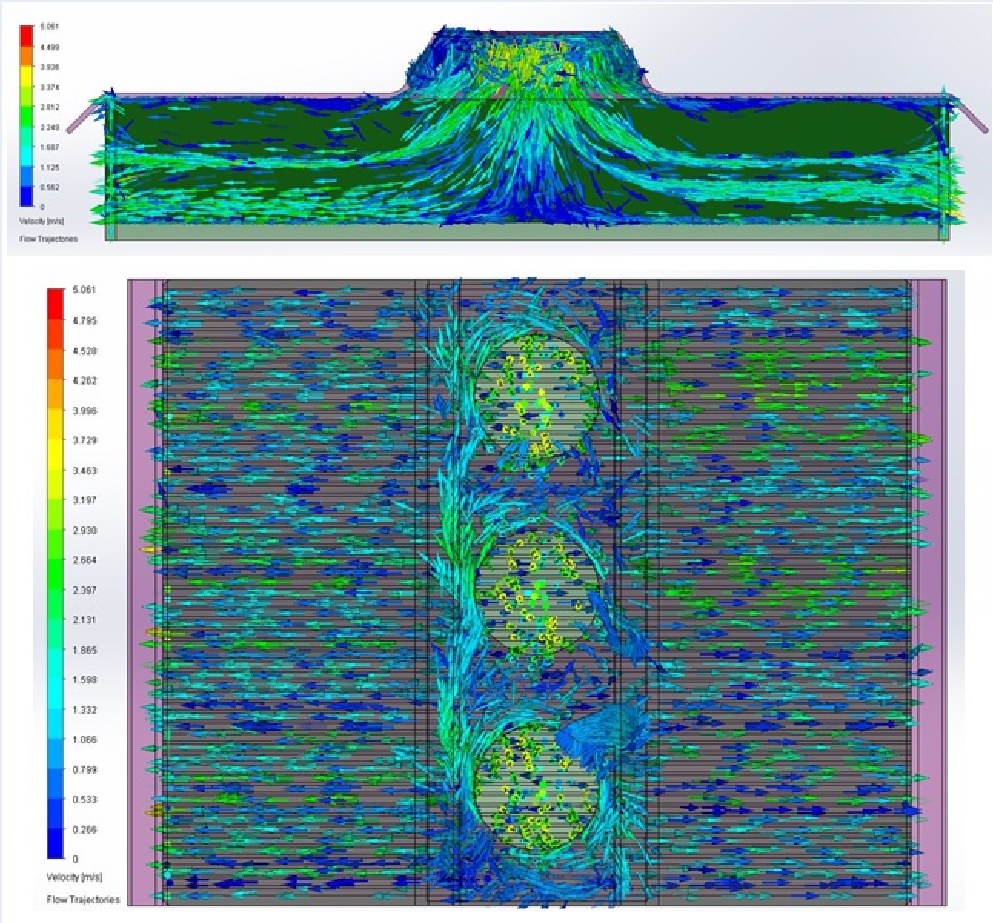

Figure 10: The velocity trajectory of the heatsink

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Figure 11: Trial manufacture of the heatsink

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Hình thức truy cập: Truy cập mở (Open Access)
Ngôn ngữ bài báo: Tiếng Anh
Tỉ lệ chấp nhận đăng 2021: 72%
Phí xuất bản: Miễn phí
Thời gian phản biện: 43 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Khoa học Tự nhiên
ISSN: 2588-106X
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt
Tỉ lệ chấp nhận đăng 2021: 75%
Phí xuất bản: Miễn phí
Thời gian phản biện: 30-45 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Kĩ thuật và Công nghệ
ISSN: 2615-9872
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt
Tỉ lệ chấp nhận đăng 2021: 61%
Phí xuất bản: Miễn phí
Thời gian phản biện: 50 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Kinh tế-Luật và Quản lý
ISSN: 2588-1051
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt
Tỉ lệ chấp nhận đăng 2021: 65%
Phí xuất bản: Miễn phí
Thời gian phản biện: 45 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Khoa học Xã hội và Nhân văn
ISSN: 2588-1043
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt
Tỉ lệ chấp nhận đăng 2021: 62%
Phí xuất bản: Miễn thu phí đối với tác giả là CBVC của ĐHKHXHNV, ĐHQG-HCM; Tác giả khác: 500.000 VNĐ/bài
Thời gian phản biện: 75 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Khoa học Trái đất và Môi trường
ISSN: **2588-1078**
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt và tiếng Anh
Tỉ lệ chấp nhận đăng 2021: 87%
Phí xuất bản: liên hệ tòa soạn
Thời gian phản biện: 45 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ - Khoa học Sức khỏe
ISSN: 2734-9446
Hình thức xuất bản: In & trực tuyến
Hình thức truy cập: Truy cập mở
Ngôn ngữ bài báo: Tiếng Việt
Tỉ lệ chấp nhận đăng 2021: 70%
Phí xuất bản: Miễn phí
Thời gian phản biện: 30 ngày
Lập chỉ mục (Indexed): Google Scholar, Scilit
SCAN ME

Tạp chí Phát triển Khoa học và Công nghệ, Đại học Quốc gia Tp.HCM
25 năm xuất bản học thuật (1997-2022)
Tòa soạn: Nhà điều hành Đại học Quốc gia Tp.HCM, P. Linh Trung, TP. Thủ Đức, TP. HCM
Email: stdj@vnuhcm.edu.vn; tcptkhcn@vnuhcm.edu.vn; Website: *http://www.scienceandtechnology.com.vn*