DESIGN OPTIMIZATION OF A POWER MODULE BOX IN 3D RADAR SYSTEM

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

Abstract: *This paper researches a design process of a heatsink in the power module box of the 3D Radar system. Studying the theory of heat transfer across parallel plate-fins of the heatsink to optimize the geometric parameters. The power module boxes are designed based on those optimum geometric parameters. In the next step, the design model is used to simulate temperature under working conditions by using Solidworks simulation software. The design model is then trial manufactured for testing before moving to the mass production.*

Keywords: Heatsink; Power module box; 3D Radar system; Heat transfer; Optimal geometric parameters; Fans.

1. INTRODUCTION

To meet the modern warfare requirements, the Vietnam Air Defense-Air Force researched, designed, and manufactured the 3D radar system, as shown in figure 1. This system utilizes active electronically scanned phased array antennas with the ability to resist interference, anti-electronic voltage suppression as well as anti-radiation missiles. This system also has a compact size and high mobility on all-terrain ranges. To meet the given requirements, the 3D radar system consists of electronic equipment set up in 16 power module boxes, in which each power module box has a heatsink and 4 electronic equipment. Under the working conditions, the power of the heat source in each module generates more than 0.4 kW. Therefore, the temperature of the electric equipment may exceed the permitted limit of the given manufacturer, which causes failure and damage to the board itself. To release the heat into the environment, installing the cooling equipment into the power module box is necessary.

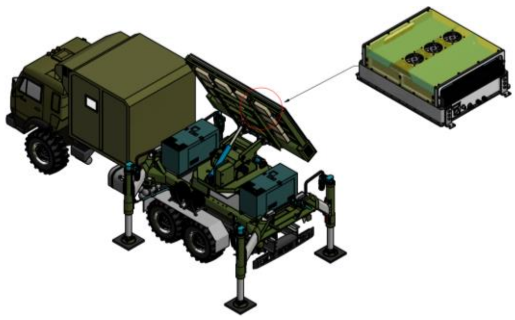


Figure 1. *The three-dimensional model of the 3D Radar.*

Literature review some of the publications of thermal analysis related to this paper as follows. [1-6] comprehensively reviewed the optimization of the thermal design of the heat sinks by examining the pulsating flow and agitation, substrate material of the heat sinks, shapes and orientations of fins, static and rotating heat sinks, single- and double-layer heat sinks, the inlet and outlet of the heat sinks, channel configurations, adding of foams and porous media, using of turbulators, additives to the conventional working fluids, and sizing of the heat sinks. [7-9] summed up the investigational efforts spent for developing the thermal performance of the heat sinks, limitations, and unsolved proposed solutions. The agreement between the temperature-dependent conduction losses of the semiconductor devices and the thermal resistance of the heat sink is considered with the heat sink optimization procedure shown in [10, 11]. [12] theoretically examined whether the cooling system volume can be significantly decreased by applying new advanced composite materials such as isotropic aluminum composites or anisotropic highly orientated pyrolytic graphite. [13] presented an analytical, forced convection model for the average heat transfer rate from a plate-fin heat sink for the full range of Reynolds number, from fully-developed to developing flow. [14] considered the optimal spacing in the heatsink for maximum heat transfer from a package of fin-plates that are cooled by forced convection. Although those publications fully presented many aspects of the thermal analysis, however, there are no papers that showed detail for a specific application as well as stages of the heatsink design process. Thus, the objective of this study is to determine the optimal geometric parameters for maximizing heat transfer rate from electronic packaging of parallel plate-fins heatsink that are cooled by forced convection. These parameters are utilized for designing the heatsink of the power module box. This model is then analyzed by thermal simulation in Solidworks software. If the maximum temperature at the contact area (case to heatsink) is smaller than the value that is given by the manufacturer, then 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 parameters, the number of fans, and simulates the temperature of the heatsink. The results of the thermal analysis of the design process are fully presented and discussed in section 3. Finally, conclusions are made, and future research is proposed in section 4.

2. MATERIALS AND METHODS

To see clearly the steps of the proposed approach, the paper's research methodology is shown in figure 2. With some constraints and given parameters, this paper determines the optimal geometric parameters of the heatsink and the number of fans. Then those parameters are used to design a three-dimensional model of the heatsink before simulating to determine the maximum temperature. If the maximum temperature does not exceed the T_{case} (the temperature in its case), which is given by the manufacturer, the model is manufactured.

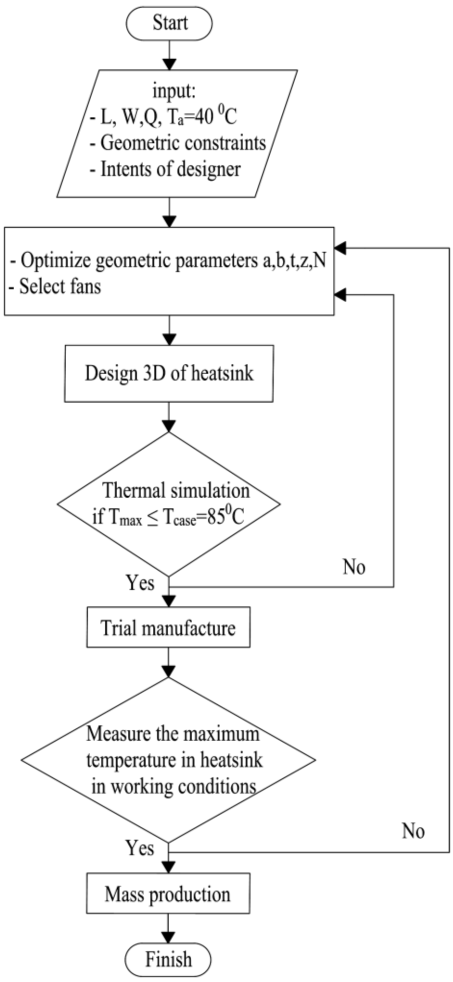
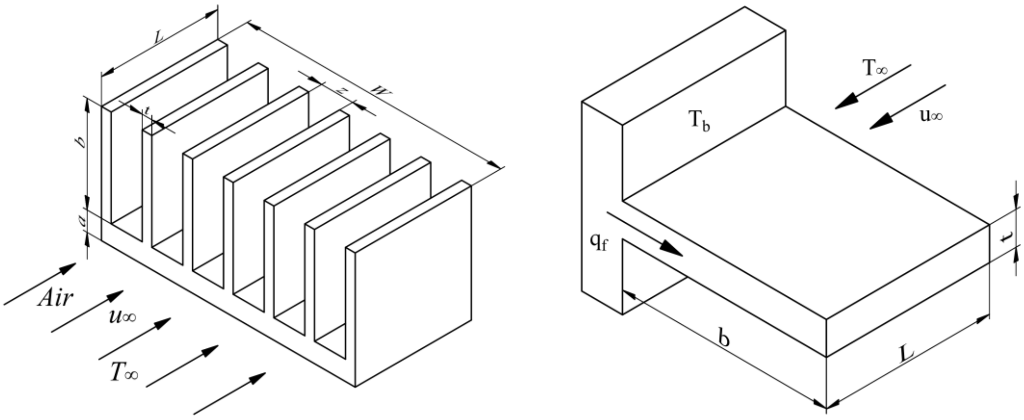


***Figure 2**. Flowchart of the research process of the heatsink.*

2.1. Optimze geometric parameters and the number of fans

In the initial stage of the design process, based on the size of the electronic equipment mounted on the power module box, the sizes of the power module boxes in the overall layout, the designer's intentions, and for other factors, then the width W and length L of the power module box are given. Thus, with a given heat-generating capacity of electronic equipment P, the maximum permitted temperature of electronic equipment T_{case}, this part of the paper is going to figure out the optimal values of 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.



(a) (b)

Figure 3. *(a) Model of the heatsink; (b) Heat transfers through fins.*

a. Optimize spacing of parallel plate-fins

This paper analyses forced convection cooling by fans, so this studies two cases when the cooling air flow passes through the gap between the fins of the heatsink has small and large spacing. The analysis model is shown in figure 3a. Consider the first case, the channel has small spacing, then the total heat transfer rate is given by [14]:

$$q_f = \rho(\frac{z^2 \Delta P}{12\mu L}) b \mathrm{W} c_p (T_h - T_\infty) \tag{1}$$

Consider the second case, the channel has large spacing.

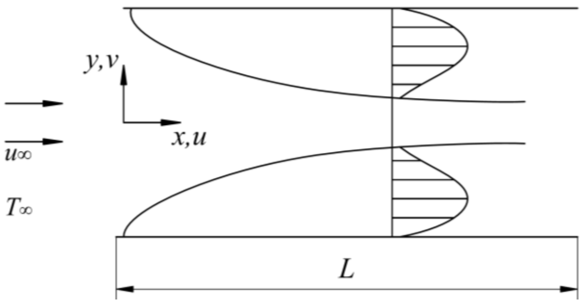


Figure 4. *Model of the large channel.*

The total heat transfer rate released [14]:

$$q_f = 1{,}208 \mathrm{W} bk (\frac{\mathrm{Pr}\,\Delta P L}{\rho \nu^2})^{\frac{1}{3}} z^{-\frac{2}{3}} (T_h - T_\infty) \tag{2}$$

To optimize the spacing between the fins, the curves of the total heat dissipation in equations (1) and (2) must intersect at a point. This is expressed by:

$$z = 3{,}24 L\,\mathrm{Re}^{-\frac{1}{2}}\,\mathrm{Pr}^{-\frac{1}{4}} \tag{3}$$

Where: T_∞ and u_∞ are temperature and velocity of air immediately around device to be cooled, respectively; T_h is the constant wall temperature of heatsink; ΔP is pressure drop; c_p is specific heat at constant pressure; k is fluid thermal conductivity; Pr is Prandtl number; ρ is fluid density; ν is kinematic viscosity; Re is Reynolds number.

b. Optimize the thickness and height of parallel fins

Heat transfer from a fin (figure 3b) with the adiabatic tip is described in [3, 13]:

$$q_f = \sqrt{hP_f kA_c}\,\theta_b \tanh mb \quad (4)$$

Assuming that $L \gg t$, $P_f \approx 2L$, $A_c = Lt$, $A_p = bt$ and $mb = \sqrt{hP_f / kA_c}\, b$, then (4) becomes $q_f = \sqrt{2hLkLt}\,\theta_b \tanh \sqrt{2hL / kLt}\,b$. This is rewritten:

$$q_f = (2hk)^{\frac{1}{2}} L\theta_b t^{\frac{1}{2}} \tanh \alpha \quad (5)$$

Considering heat transfer rate at the base wall between two fins:

$$q_w = hLz\theta_b \quad (6)$$

The total heat transfer rate is given by:

$$\text{q}=\frac{W-z}{t+z} L\theta_b [(2hk)^{\frac{1}{2}} t^{\frac{1}{2}} \tanh \alpha + hz] \quad (7)$$

Where: m is fin parameters, $m = \sqrt{hP_f / kA_c}$; h is heat transfer coefficient; A_p, A_c are the profile area of fin; P_f is perimeter of fin; T_b is the temperature at baseplate; θ_b is temperature excess ($\theta_b = T_b - T_\infty$). Differentiate (7) with respect to t:

$$\frac{dq}{dt} = 0$$

Thus, the optimal thickness of fins is:

$$t = \frac{z}{2(\text{W}-z)L\theta_b - 1} \quad (8)$$

There is a relationship that is:

$$\alpha = mb_o = (\frac{2h}{kt})^{\frac{1}{2}} b_o \quad (9)$$

Therefore, the optimal height of fin is calculated by:

$$b_o = \alpha(\frac{kt}{2h})^{\frac{1}{2}} \quad (10)$$

c. The number of fins

Based on figure 3a the number of fins is determined as follows:

$$N = \frac{W-z}{t+z} \quad (11)$$

d. Determine the number of fans

The total amount of required ventilation for cooling is determined by:

$$Q = \frac{60.P}{\rho.c_p (T_{out} - T_{in})} \quad (12)$$

The number of fans that needs to be installed:

$$n = \frac{Q}{q_v} \quad (13)$$

Where: Q is the total amount of required ventilation for cooling; q_v is the volumetric flow rate of each fan; P is the heat source dissipating; $\Delta T = T_{out} - T_{in}$ is the temperature difference between outlet and inlet of the heatsink.

2.2. Design three-dimensional of the heatsink

Each power module box consists of 4 heat sources, in which sizes and power are shown in table 1.

Table 1. *The sizes and power of electric equipment.*

No.	Sizes (m)	Power (W)
1st	0.137 x 0.058	30
2nd	0.385 x 0.050	5
3rd	0.385 x 0.112	328
4th	0.385 x 0.090	40

Based on the requirements for cooling, geometric constraints, and the intentions of the designer, the 8314HU axial fan is selected. The maximum volumetric flow rate of this fan is 47.1 cfm ($\approx$ 79.85 m^3/h). The number of fans is determined from equations (12) and (13): $n \approx 2{,}98$. Thus, the number of fans is 3. Based on the size of the electronic devices, the installation size of the power module boxes in the overall 3D radar system, the technological capabilities of the workshop. Therefore, the length and width dimensions are preselected 0.477 x 0.4 m, respectively. Some of the initial values for determining the optimal geometric parameters are listed in table 2.

Table 2. *The initial values of some parameters.*

No.	Denotation	Value	Unit
1	L	0.477	m
2	W	0.400	m
3	u	4.6	m/s
4	ν	$1.702\text{x}10^{-5}$	m^2/s
5	$\mathrm{Re} = u.L/\nu$	128920	
6	Pr (40 ^{0}C)	0.7255	
7	k_{air}	0.02662	W/m.K
8	$k_{aluminium}$	205	W/m.K

The electronic equipment is BLS9G2731L-400 LDMOS S-band radar power transistor, which has $\mathrm{T_{jmax}} = 225\ ^o\mathrm{C}$ [15]. The thermal resistance from the case to ambient is given by [10]:

$$R_{ja} = R_{jc} + R_{cs} + R_{sa} \tag{14}$$

There is also a relationship as follows:

$$R_{cs} = \frac{a}{k_f A} \tag{15}$$

From equations (14) and (15), one gets:

$$a = k_f A(R_{ja} - R_{jc} - R_{sa}) \tag{16}$$

Where: $R_{ja} = (T_{j\max} - T_{\infty})/P$ is thermal resistance from junction to ambient; $T_{j\max}$ is the maximum junction temperature, the maximum allowable temperature that device will see at its silicon junction; $\mathrm{R_{sa}}$ is the thermal resistance from heatsink to ambient; R_{jc} is the thermal resistance from junction to case, this is provided by the device manufacturer; $\mathrm{k_f}$ is thermal interface material; A is the contact area of this material.

From equations (8), (10), (11), and (16) it is easy to determine the optimal geometric parameters, which are listed in table 3.

Table 3. *The optimal geometric parameters of the heatsink.*

No.	Denotation	Value	Unit
1	a	10	mm
2	b	60	mm
3	t	2.1	mm
4	z	4.5	mm
5	N	60	
6	Fans	3	

The three-dimensional model of the power module box is shown in figure 5.

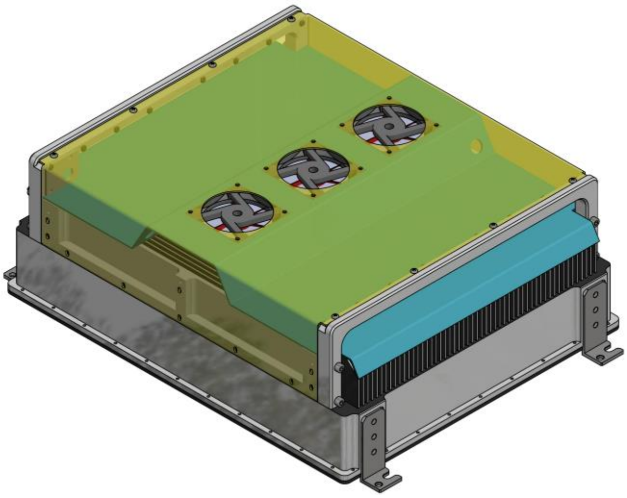


Figure 5. *The three-dimensional design of the power module box.*

2.3. Thermal analysis

The thermal analysis aims to allow the designer to quickly evaluate whether the theoretical calculation results 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 quite accurately the critical positions, thus help them make some important decisions in the initial design process. The boundary conditions for the thermal analysis are shown in figure 6.

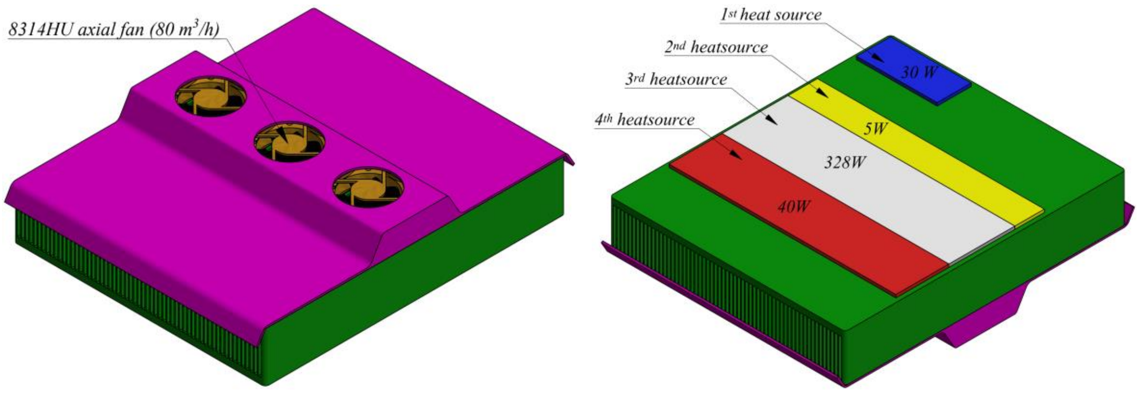


Figure 6. *The boundary conditions for thermal analysis.*

The three-dimensional power module box in figure 4 is simply modified for simulating in the Solidworks simulation software. The boundary conditions for the thermal analysis as follows: The power module box is made from 6061 aluminium. The input temperature of the simulation process is the ambient temperature $T_a = 40\ ^oC$. There are three 8314HU axial fans set up above the power module box so that the heatsink subjected to an impingement flow. The electronic equipment is BLS9G2731L-400 LDMOS S-band radar power transistor, 4 heat sources located at different positions as illustrated in figure 5. The electronic equipment integrates at the bottom of the heatsink in which dimensions and power dissipation are listed in table 1.

3. RESULTS AND DISCUSSIONS

3.1. Results

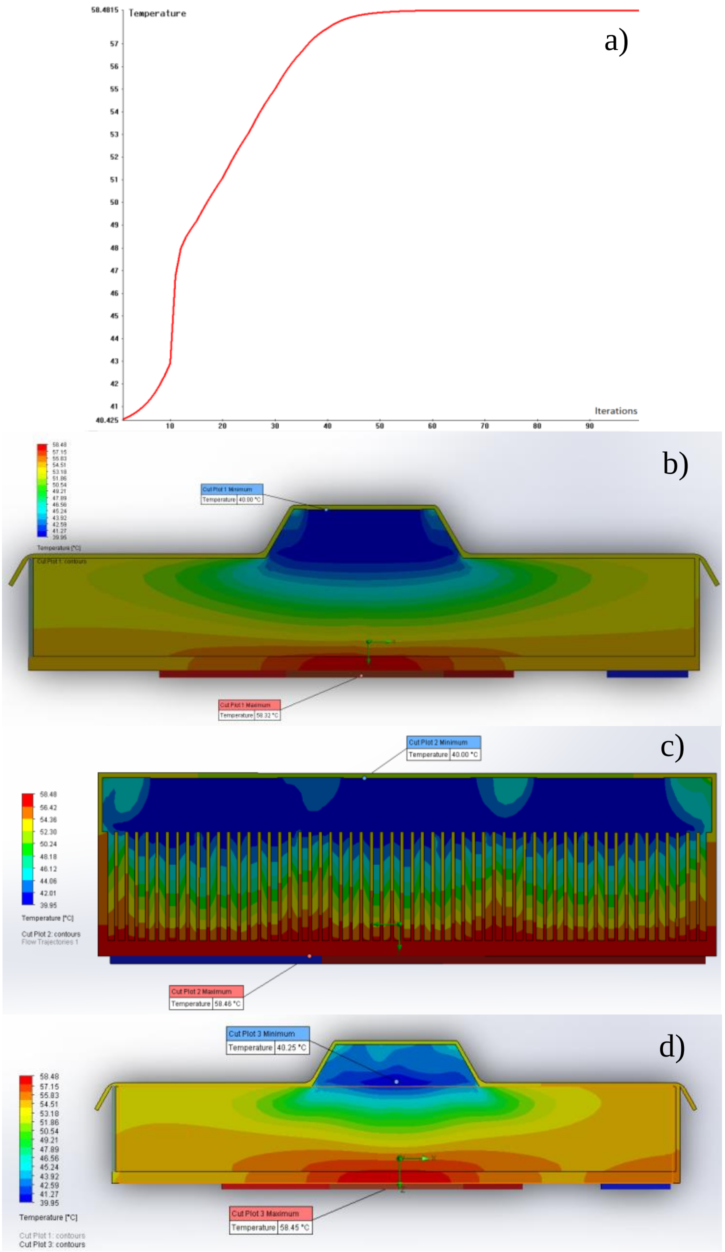


Figure 7. *The distribution of temperature in the heatsink.*

3.2. Discussions

The simulation results of the heatsink is shown in figure 6. Overall, the high-temperature field is distributed in the area where the electronic equipment is attached to the base of the heatsink. As shown in figure 6a, the iterations of the simulation process converge at approximately 50. Figures 6b and 6d are the side views of the heatsink, these figures clearly show that the maximum temperature appears at the bottom where the electronic components directly integrate with the heatsink. To be more precise, the highest temperature is 58.5 ^{0}C in the power module box shown at the contact area of the third electronic component with the heatsink. On the contrary, the lowest temperature in the power module box is equal to ambient and occurs near the input flow of fans, that is roughly 40 ^{0}C. Figure 6b shows the temperature gradually reduces from the center to both two sides of the heatsink. Moreover, figure 6c illustrates the distribution of temperature in fins, the temperature gradually decreases from the bottom to the top. Also, the temperature is symmetric through the plane of symmetry of the power module box. 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 ^{0}C [15]. Therefore, the model design is manufactured.

4. 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 parameters of the heatsink in the power module box. The heatsink is made from 6061 aluminium with the height, spacing, and thickness of fins are 60, 4.5, and 2.1 mm, respectively. Using Solidworks simulation software to simulate the maximum temperature under working conditions. The simulation results show that the model design satisfies the given requirements. The design model is then produced for testing the maximum temperature in the laboratory before moving on to the mass production process. Trial products with the optimal geometric parameters were installed on a 3D radar system. The temperature test results under working conditions are always lower than the permissible temperature and the device works well.

REFERENCES

[1]. A. Bar-Cohen and W. W. Rohsenow, *"Thermally optimum spacing of vertical, natural convection cooled, parallel plates," J. Heat Transfer*, vol. 106, no. 1, pp. 116–123, 1984, doi: 10.1115/1.3246622.

[2]. P. Bhambare, D. Kaithari, and G. Kharote, *"Experimental Investigation on Fin Spacing Optimization in Natural Convection Heat Transfer for Isothermal Rectangular Aluminum Fins on a Vertical Base,"* vol. 7, no. 3, pp. 887–893, 2016.

[3]. J. R. Culham and Y. S. Muzychka, *"Optimization of plate fin heat sinks using entropy generation minimization,"* IEEE Trans. Components Packag. Technol., vol. 24, no. 2, pp. 159–165, 2001, doi: 10.1109/6144.926378.

[4]. C. Gammeter, F. Krismer, and J. W. Kolar, "*Weight optimization of a cooling system composed of fan and extruded-fin heat sink,"* IEEE Trans. Ind. Appl., vol. 51, no. 1, pp. 509–520, 2015, doi: 10.1109/TIA.2014.2336977.

[5]. S. Lee, *"Optimum Design and Selection of Heat Sinks,"* IEEE Trans. Components Packag. Manuf. Technol. Part A, vol. 18, no. 4, pp. 812–817, 1995, doi: 10.1109/95.477468.

[6]. M. B. De Stadler, *"Optimization of the Geometry of a Heat Sink,"* Virginia Sp. Grant Consort. Student Res. Conf., pp. 1–10, 2007.

[7]. H. E. Ahmed, B. H. Salman, A. S. Kherbeet, and M. I. Ahmed, *"Optimization of thermal design of heat sinks: A review,"* Int. J. Heat Mass Transf., vol. 118, no. January, pp. 129–153, 2018, doi: 10.1016/j.ijheatmasstransfer.2017.10.099.

[8]. H. T. Dhaiban and M. A. Hussein, *"The optimal design of heat sinks: A review,"* J. Appl. Comput. Mech., vol. 6, no. 4, pp. 1030–1043, 2020, doi: 10.22055/jacm.2019.14852.

[9]. K. Londhe and V. R. Kaushik, *"Heat Sink Design for Optimal Performance of Compact Electronic Appliances - a Review,"* J. Adv. Res. Appl. Sci., vol. 4, no. 5, pp. 13–21, 2017.

[10]. D. Christen, M. Stojadinovic, and J. Biela, *"Energy Efficient Heat Sink Design: Natural Versus Forced Convection Cooling,"* IEEE Trans. Power Electron., vol. 32, no. 11, pp. 8693–8704, 2017, doi: 10.1109/TPEL.2016.2640454.

[11]. S. W. Churchill and R. Usagi, *"A general expression for the correlation of rates of transfer and other phenomena,"* AIChE J., vol. 18, no. 6, pp. 1121–1128, 1972, doi: 10.1002/aic.690180606.

[12]. U. Drofenik, A. Stupar, and J. W. Kolar, *"Analysis of theoretical limits of forced-air cooling using advanced composite materials with high thermal conductivities,"* IEEE Trans. Components, Packag. Manuf. Technol., vol. 1, no. 4, pp. 528–535, 2011, doi: 10.1109/TCPMT.2010.2100730.

[13]. P. Teertstra, M. M. Yovanovich, and J. R. Culham, *"Analytical forced convection modeling of plate fin heat sinks,"* J. Electron. Manuf., vol. 10, no. 4, pp. 253–261, 2000, doi: 10.1142/S0960313100000320.

[14]. A. Bejan and E. Sciubba, *"The optimal spacing of parallel plates cooled by forced convection,"* International Journal of Heat and Mass Transfer, vol. 35, no. 12. pp. 3259–3264, 1992, doi: 10.1016/0017-9310(92)90213-C.

[15]. L. S-band, *"Bls9g2731l-400; bls9g2731ls-400,"* no. April, pp. 1–13, 2017.

TÓM TẮT

TỐI ƯU HÓA THIẾT KẾ HỘP MÔ-ĐUN CÔNG SUẤT TRÊN ĐÀI RA-ĐA 3D

Bài báo nghiên cứu thiết kế thiết bị tản nhiệt trong hộp mô-đun công suất cho đài ra-đa 3D. Nghiên cứu lý thuyết truyền nhiệt qua cánh để tối ưu hóa các tham số hình học. Hộp mô-đun công suất được thiết kế dựa trên các tham số hình học tối ưu. Sau đó, mô hình thiết kế được sử dụng để mô phỏng nhiệt độ trong điều kiện làm việc bằng phần mềm Solidworks. Mô hình này được sản xuất chế thử để thử nghiệm trước khi đưa vào sản xuất hàng loạt.

Từ khóa: Thiết bị tản nhiệt; Hộp mô-đun công suất; Hệ thống đài ra-đa 3D; Truyền nhiệt; Các tham số hình học tối ưu; Quạt.

Received 26th March 2021
Revised 20th April 2021
Accepted 10th June 2021

Author affiliations:

Le Quy Don Technical University.

*Email: phanhoangcuongatc@gmail.com.