Heat dissipation performance of a heat pipe radiator with rectangular longitudinal fin under natural convection

Pengyang Qu¹, a, Wei Li¹ *, Yu Dong¹, Hanzhong Tao², Jianjie Cheng¹, Yannan Li²
¹College of Urban Construction, Nanjing Tech University, Nanjing, Jiangsu, China
²School of Energy Science and Engineering, Nanjing Tech University, Nanjing, Jiangsu, China
aemail: 201961224058@njtech.edu.cn
*Corresponding author. E-mail: njut001@126.com

Abstract. To solve the cooling problem of electronic equipment with small heat dissipation area under high uneven power, this paper presents a natural convection radiator with L-shaped heat pipe and rectangular longitudinal fin module. The radiator has the advantages of compact structure, flexible layout, and didn’t occupy the side space due to the structural advantages. A numerical study was carried out, and the results are in good agreement with the experimental results. The thermal performance of the rectangular longitudinal fin module is also studied, the radiation heat loss of the fins accounts for about 25% of the total heat dissipation. The temperature and velocity vector distribution near the fin was analysed to research the influence of flow characteristics on the heat transfer of the fin. Additionally, the numerical results also show that the fin heat sink has an optimal fin height, which is 14mm.

1. Introduction
The two main problems of high uneven power consumption but with the small heat dissipation area of electronic equipment and how to fully remove the heat generated by electronic equipment have become the key factors restricting the development of electronic equipment. Therefore, it is very meaningful to develop a kind of radiator with compact structure and flexible arrangement to solve the heat dissipation problem of electronic devices [1-3].

Although various cooling methods have been studied, forced convection cooling, cooled by liquid or thermoelectric cooling etc. [4-6]. Natural convection heat dissipation still shows some advantages in low noise, high reliability and low energy consumption, so the natural convection radiator has been widely used in the heat dissipation of electronic equipment [7,8]. However, in recent years, the traditional natural convection heat transfer method cannot fully satisfy the requirements on heat dissipation requirements of electronic devices, because of its limited ability to take away heat. Therefore, it is often necessary to cooperate with other heat dissipation methods.

As an efficient two-phase heat transfer device, heat pipe has been widely used in the thermal control of high-power density electronic devices for its excellent heat dissipation performance and reliability [9]. Using the isothermal characteristics of heat pipes, multiple heat sources can be arranged in the evaporation section of the heat pipe to meet the heat dissipation requirements while achieving the miniaturization and integration of the heat dissipation device. This will become one of the directions for developing the heat dissipation system of electronic equipment.
Ye et al. [10] and Sharifi et al. [11] research shows that fins play a leading role in convective and radiative heat transfer from heat sink to environment. Some other studies on the application of heat pipe-fin array systems have also indicated that, the limiting system-level thermal resistance are associated with the flow of single-phase fluids external to the heat pipe’s condenser or evaporator sections [12,13]. Therefore, this work will focus on exploring effective methods to reduce the thermal resistance of fin arrays with natural convection to enhance the performance of heat pipe sinks.

The purpose of this study is to development a radiator with the advantages of compact structure, flexible layout, and didn’t occupy the side space. Compared with previous work, although much research has been done on finned radiators, little research has been done on arrays in condensation sections of heat pipes and rectangular longitudinal fins. This paper considered both natural convection and radiation heat transfer in the radiator, and makes relevant numerical simulation and experimental verification. Also, the thermal performance of rectangular longitudinal fin array is numerically studied and analyzed. Finally, the effect of fin height on the heat dissipation performance of fins is analyzed when the ambient temperature is 30°C and the heating power is 30W. The parameters of rectangular longitudinal fins with optimal heat dissipation are obtained.

**Nomenclature**

| Symbol | Description |
|--------|-------------|
| $A$ | Total fin area (m$^2$) |
| $A_{per}$ | Area of each fin (m$^2$) |
| $g$ | Acceleration of gravity (m/s$^2$) |
| $H_s$ | Heat source height (mm) |
| $L_f$ | Fin length (mm) |
| $L_{cond}$ | Length of condensation section of heat pipe (mm) |
| $L_{ins}$ | Length of heat pipe insulation section (mm) |
| $N$ | Number of fins |
| $N_u$ | Nusselt number |
| $Q$ | Heat transfer rate (W) |
| $Q_{rad}$ | Radiative heat transfer rate (W) |
| $R_{th}$ | Heat resistance of convection heat transfer at fin end (°C/W) |
| $T_a$ | Temperature of ambient (°C) |
| $T_{sim}$ | Temperature of simulation (°C) |
| $W_f$ | Fin width (mm) |
| $\alpha$ | Thermal diffusivity (m$^2$/s) |
| $\Delta T$ | Qualitative temperature of air (°C) |
| $\nu$ | Kinematic viscosity (m$^2$/s) |
| $A_b$ | Rib base area (m$^2$) |
| $D_h$ | Diameter of heat pipe (mm) |
| $H_f$ | Fin height (mm) |
| $h$ | Convective heat transfer coefficient (W/(m$^2$.K)) |
| $L_s$ | Heat source height (mm) |
| $L_{evap}$ | Length of evaporation section of heat pipe (mm) |
| $N$ | Number of fins |
| $Pr$ | Prandtl number |
| $Q_{conv}$ | Convective heat transfer rate (W) |
| $Ra$ | Rayleigh number |
| $T$ | Temperature (°C) |
| $T_{exp}$ | Temperature of experimental (°C) |
| $T_{ave}$ | Average temperature of fins (°C) |
| $W_s$ | Heat source width (mm) |
| $\beta$ | Coefficient of thermal expansion of air (1/K) |
| $\eta$ | Fin effectiveness |
2. Mathematical modelling

2.1. Numerical model

Fig. 1 Schematic view of the model for simulation (a) Model and calculation area (b) Top view of fin (c) Front view of fin.

Fig. 1 shows the L-shape heat pipe heatsink investigated in the present study. The length of evaporator section, adiabatic section and condenser section of heat pipe are 500mm, 75mm and 150mm respectively. And the diameter of heat pipe is 8mm. The length of per fin is 150mm and each fin is arranged in a ring at an angular interval of 15 degrees. The height of fin in this study is varied from 12mm to 20mm. In addition, the geometric dimension of the base plate is 500mm×50mm×25mm.

To ensure that the effect of the size of the calculation region on the numerical results is negligible, the calculation area is set to two times the model size in the X and Z directions and three times the model size in the Y direction. The top of the calculation area is set as the pressure outlet and the bottom is set as the pressure inlet, and all other sides are set to open freely.

2.2. Solution algorithm

The ambient temperature of the numerical simulation is set to 303.15K, the total heating power of the three heat sources can be varied within the 25W-40W. The gravity is taken into account in the Y-axis direction in Cartesian coordinates, and the acceleration of gravity is 9.8 m/s². Besides, in order to satisfy the convergence rate and the accuracy of the calculation results at the same time, the following assumptions are made:

(1) three-dimensional steady-state heat transfer.
(2) the airflow is steady, turbulent, and incompressible. The Boussinesq approximation was adopted for the density of air.
(3) the solid surface is diffuse and gray.

The establishment of the numerical model is based on the finite volume method, and the SIMPLE algorithm is used to was chosen for coupling the momentum pressure equations. The second-order upwind scheme was applied to the convective terms of the governing equations to improve the accuracy of the analysis.

2.3. Data reductions

The Ra number at the fin end is defined as,

\[ Ra = \frac{g \beta \Delta T l^3}{\nu a} \]  

(1)
Nusselt number can be obtained from the relationship between Churchill and Chu [33],

\[ Nu = 0.68 + \frac{1}{9} \frac{0.67 Ra^{\frac{1}{9}}}{1+(0.492/Pr)^{\frac{9}{16}}} \]  

The convective heat transfer rate,

\[ Q_{\text{conv}} = hA(T_w - T_x) \]  

The convective heat transfer resistance,

\[ R_{\text{th}} = \frac{T_{\text{max}} - T_x}{Q_{\text{conv}}} \]  

The total fin area,

\[ A = A_{\text{per}} \cdot N + A_p \]  

The fin effectiveness,

\[ \eta = \frac{T_{\text{ave}} - T_x}{T_{\text{max}} - T_x} \]  

3. Experiment and validation

To validate the accuracy of the numerical model, a series of related experiments were carried out. Fig.2 shows the device schematic diagram for the research and test of heat transfer performance of L-type heat pipe fin radiator. The device consists of a radiator test piece, an experimental chamber environment regulation system, a measurement system, and a data acquisition system.

In order to facilitate manufacturing, aluminium alloy T6063 is selected as the material of fin, and its thermal conductivity is 193W/m·K, while aluminium alloy T6061 is used in aluminium base plate, and the corresponding thermal conductivity is 170W/m·K. These aluminium surfaces are sandblasted with an emissivity \( \varepsilon = 0.82 \).
To reduce the heat loss to the environment and keep the internal temperature of the chamber relatively constant, each wall of the section is filled with thermal insulation materials with low thermal conductivity, which can be approximately considered as adiabatic. And the temperature automatic control system is used to adjust and maintain the ambient temperature. For the sake of simulating the heat generated by electronic chips or integrated circuits, three heating films with a maximum power of 16W are evenly placed at the bottom of the aluminium plate, as shown in Fig.2, and a DC power supply used to adjust the input power of the heat source.

![Fig.3 Grid independence verification](image1.png)

![Fig.4 Verification of numerical simulation results](image2.png)

As shown in Fig.3, when the number of grids increases from 1326543 to 1869796, the transformation of the average Nusselt number at the fin end is less than 0.5%, so in order to meet both calculation accuracy and running speed, the grid parameter values set when the grid is 1326543 elements are used for all future calculations.

In order to verify the accuracy of the selected model for numerical simulation, the temperature values measured by numerical simulation and experiment are given in Fig.4. When the ambient temperature is 303K and the total heating power is 25W and 30W respectively. Under the condition of the heating power of 25W, the maximum error of experiment and simulation is 4.8%, and about 3.9% at 30W.

4. Results and discussion

4.1. Thermal performance of the rectangular longitudinal fin array

![Fig.5 The relationship between input power and heat dissipation of fins](image3.png)

![Fig.6 Input power and fin convection heat transfer coefficient and thermal resistance](image4.png)
The relationship between convective and radiative heat dissipation and total heat dissipation under different input power is given in Fig.5. It can be seen that with the increase of input power, $Q_{\text{rad}}$ scalar variability is about 0.1 (3.32W-5.89W), while $Q_{\text{conv}}$ expanded from 11.80W to 16.65W. With the continuous growth of heating power, the temperature of the fin end will gradually increase, and the radiation heat transfer is proportional to the fourth power of temperature. In contrast, the convective heat transfer presents a linear change with temperature. Therefore, the proportion of radiation heat transfer in the total heat transfer process has increased, but the counter-flow heat transfer is still dominant. Among them, the radiative heat loss accounts for 21.95%, 22.76%, 23.98%, 24.92% and 26.01% of the total heat dissipation of fins, respectively, when the heating power increases from 25W to 40W. This is basically consistent with the results of reference [14]. For the longitudinal rectangular fins, the radiation heat loss accounts for about one-fourth of the total heat dissipation.

As shown in Fig.6, for convective heat transfer, the thermal resistance of convective heat transfer at the fin end decreases obviously at 20W-35W, and then when the heating power increases to 40W, the decreasing trend of thermal resistance slows down due to the decrease of the temperature difference between the fin and the surrounding environment. Therefore, the corresponding convective heat transfer coefficient increases from 4.427 W/m$^2$∙K to 4.512 W/m$^2$∙K, but the slope of the curve decreases obviously (from 0.03 to 0.02), that is, the enhanced heat transfer will be weakened. The convective heat transfer coefficient is mainly used to characterize the physical quantity of convective heat transfer, that is, when the ambient temperature is constant, the heat dissipation capacity of the heat sink will reach a limit.

4.2. Temperature and velocity vector distribution

As shown in Fig.7, in order to analyze the influence of flow characteristics on the heat transfer of the fin, the velocity vector diagram near the fin is drawn. In general, since air flows vertically from the end of each fin, a cluster of convergent three-dimensional vertical streamlines will be formed in the upper part of the center of the fin array. This flow is called a "chimney or plume", observed by Harahap and MacManus [15] through experiments, but they did not measure the specific velocity of the air flow. Due to the existence of the geometric characteristics of the rectangular longitudinal fin, when the hot air moves upward from the fin array, the flow in the gap between two continuous fins will bring larger heat transfer area, thus improving the heat transfer rate.

Fig.7 Temperature distribution and velocity vector diagram near the fin

As shown in Fig.7, in order to analyze the influence of flow characteristics on the heat transfer of the fin, the velocity vector diagram near the fin is drawn. In general, since air flows vertically from the end of each fin, a cluster of convergent three-dimensional vertical streamlines will be formed in the upper part of the center of the fin array. This flow is called a "chimney or plume", observed by Harahap and MacManus [15] through experiments, but they did not measure the specific velocity of the air flow. Due to the existence of the geometric characteristics of the rectangular longitudinal fin, when the hot air moves upward from the fin array, the flow in the gap between two continuous fins will bring larger heat transfer area, thus improving the heat transfer rate.
4.3. Thermal performance of the rectangular longitudinal fin array

Fig. 8 shows the relationship between fin height and temperature of fin and heat source. It can be seen that when other geometric parameters (except fin height) are fixed, with the gradual rise in fin height, the average temperature of fin end and heat source presents a downward trend, which is because the increase of fin height leads to the increase of total heat exchange area of radiator, which is more conducive to heat transfer. Simultaneously, the average Nusselt number at the fin end gradually decreases with the fin height, and the downward trend slows down gradually, which is basically consistent with the results of E.M Sparrow [16]. There is a negative correlation between Nusselt and fin height. This is because at a more considerable fin height, the boundary layers of the two fins influence each other, so it is difficult for the air to go deep into the fin root region, and then the airflow stagnates, which worsens the heat transfer in this area.

![Fig. 8 The effect of fin height on the temperature of fin end](image1)

![Fig. 9 Effect of fin height on fin effectiveness and fin Nusselt number](image2)

Although the temperature of the fin shows a downward trend as a whole, it can be seen from Fig. 9 that the fin efficiency does not improve, but decreases at first, and then the change of fin efficiency becomes less evident after the height reaches 14mm. And when the fin height reaches a particular value, the increase of the thermal resistance of the fin is enough to offset the benefit brought by the increase of the heat transfer area, and the further growth of the fin height will not bring the apparent improvement of the heat transfer performance, so there is an optimal value of the fin height. In this study, considering various factors, including heat dissipation performance and manufacturing cost, we think that for the finned radiator described in this paper, the best fin height can be set as 14mm.

5. Conclusion

In this paper, a numerical simulation study and experimental verification of L-shape heat pipe heatsink with a rectangular longitudinal fin module was carried out. The following conclusion were obtained:

1) The thermal performance of the rectangular fin is studied and analyzed, and it is found that the radiation heat dissipation of the rectangular longitudinal fin accounts for about 25% of the total heat dissipation, and as the heating power increases, the proportion of heat emitted by fins through thermal radiation will gradually increase.

2) The geometric characteristics of rectangular longitudinal fins make that when the hot air flows upward from the fin array, the flow in the gap between two continuous fins will bring larger heat transfer area, which can improve the heat transfer rate.

3) When other geometric parameters (except the fin height) are fixed, considering all factors including heat dissipation performance and manufacturing cost, the fin heat sink has an optimal fin height, which is 14mm.
Acknowledgments
The present study was supported by Postgraduate Research & Practice Innovation Program of Jiangsu Province (NO. SJCX20_0328, NO. SJCX20_0330).

References
[1] Sohel Murshed, S. M., Nieto, D. C. C. A. (2017) A critical review of traditional and emerging techniques and fluids for electronics cooling. Renewable and Sustainable Energy Reviews, 78 :821-833.
[2] Marcinichen J B, Olivier J A, Oliveira V D, et al. (2012) A review of on-chip micro-evaporation: Experimental evaluation of liquid pumping and vapor compression driven cooling systems and control. Applied Energy, 92: 147-161.
[3] Tavakkoli F, Ebrahimi S, Wang S, et al. (2016) Analysis of critical thermal issues in 3D integrated circuits. International Journal of Heat and Mass Transfer, 97: 337-352.
[4] Maalej S, Zayoud A, Abdelaziz I, et al. (2020) Thermal performance of finned heat pipe system for Central Processing Unit cooling. Energy Conversion and Management, 218:112977.
[5] Roger S. (2004) Challenges in Electronic Cooling—Opportunities for enhanced thermal management techniques—microprocessor liquid cooled minichannel heat sink. Heat Transfer Engineering, 25: 3-12.
[6] Santosh K S, Mihir K D, Prasenjit R. (2016) Application of TCE-PCM based heat sinks for cooling of electronic components: A review. Renewable and Sustainable Energy Reviews, 59: 550-582.
[7] Bergles, Arthur E. (1997) Heat transfer enhancement—the maturing of second-generation Heat Transfer Technology[J]. Heat Transfer Engineering, 18: 47-55.
[8] Ahmed H E, Salman B H, Kherbeet A S, et al. (2018) Optimization of thermal design of heat sinks: A review. International Journal of Heat and Mass Transfer, 111: 129-153.
[9] Tang H, Tang Y, Wan Z, et al. (2018) Review of applications and developments of ultra-thin micro heat pipes for electronic cooling. Applied Energy, 223: 383-400.
[10] Ye H, Li B, Tang H, et al. (2014) Design of vertical fin arrays with heat pipes used for high-power light-emitting diodes. Microelectronics Reliability, 54:2448-2455.
[11] Sharifi, Nouroddin, Stark, et al. (2016) The influence of thermal contact resistance on the relative performance of heat pipe-fin array systems. Applied Thermal Engineering,105: 46–55.
[12] Zhao Y. (2015) Study of the heat transfer characteristics of semiconductor cooling box based on the natural convection heat pipe radiator, South China University of Technology, 2014: 1–48.
[13] Faghi A. (2012) Review and advances in heat pipe science and technology, Journal of Heat Transfer,134:1–18.
[14] Maji A, Bhanga D, Patowari, PK. (2017) Numerical investigation on heat transfer enhancement of heat sink using perforated pin fins with inline and staggered arrangement. Applied Thermal Engineering, 125: 596-616.
[15] Harahap F, Mc M H. (1967) Natural convection heat transfer from horizontal rectangular fin arrays. Heat Transfer, 89: 32–38.
[16] Sparrow E M, Kadle D S. (1986) Effect of tip-to-shroud clearance on turbulent heat transfer from a shrouded, longitudinal fin array. Journal of Heat Transfer, 108: 519-524.