Analysis of Fin Interruptions on the Natural Convection of Heat Sinks

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Abstract. For heat sinks with high fins used in outdoor high-power electronic equipment, this paper proposes a natural convection heat transfer calculation method based on the chimney effect and uses this method and numerical simulation to analyze the heat transfer performance of the heat sinks with interrupted fins. Results show that the interrupted fins effectively improve the convective heat transfer coefficient of the fins by breaking the thermal boundary layer of the fins, and have a significant effect on the airflow between the fins. When the fin interruption length is 1mm and the number of interrupts is 15, the natural convection heat transfer performance of the heat sink is the best. Compared to heat sinks of the same size without interrupted fins, the thermal performance has been improved by 80% while reducing weight by 7%.

1 Introduction
Outdoor electronic equipment is a whole composed of numerous electronic components, and during its normal operation, each electronic component becomes a huge heat source under the action of current. If in a low-temperature environment, these electronic components can transfer most of the heat to the environment through radiation to reduce their own temperature. However, in high-temperature environments, the thermal radiation ability decreases, and the heat generated during the operation of electronic components cannot be effectively transmitted to the surrounding environment in a timely manner, resulting in a sharp increase in the temperature of electronic equipment, causing changes in the properties of related materials, and even causing electronic components to be damaged at high temperatures, directly affecting the normal operation of electronic devices. Therefore, the issue of heat dissipation in electronic components has always received widespread attention[1-4]. Heat sinks are often used for heat dissipation in electronic components, with common fin forms including plate fin, interrupted fin, and pin fin. Among them, the structures of straight fin and interrupted fin are shown in Fig. 1.

![Fig. 1. Schematic of the heat sink with (a) continuous fins, (b) interrupted fins.](image-url)

When optimizing the structure of finned heat sinks, it is common to change the geometrical parameters of the fins. For example, Welling et al. [5] studied the effect of different geometric parameters of the fins on natural convective heat transfer, and the results showed that given the temperature, the optimal value of the ratio of fin height to fin length can be found. Cohen et al. [6] discussed the fin spacing for vertical arrangement of plate fins and fitted the experimental correlation for the heat transfer coefficient of natural convection. Harahpa and Setio[7] conducted experimental studies on the heat transfer performance of several horizontally placed vertically-finned arrays and explored the effects of different fin schemes on their heat transfer performance. The controlled variables included fin spacing, fin height, fin thickness, fin numbers, fin length, and base...
temperature. Yazıcıoğlu and Yüncü ([8] studied the influence of various parameters of fins on natural convection heat transfer, and found that adding fins on a vertical flat plate can effectively enhance heat transfer. Hong ([9]) conducted numerical simulation and experimental analysis of natural convection heat transfer in plate fin radiators, revealing that with the change of fin spacing, the heat transfer rate reaches its maximum value, and the optimal fin spacing increases with the decrease of Prandtl number.

Taking the natural convection heat transfer problem of finned heat sinks for outdoor high-power electronic equipment as an example, due to the large error in using traditional heat transfer calculation models for finned heat sinks with high fin height, this paper has improved their natural convection theoretical model and analyzed the thermal performance of heat sinks with interrupted fins. In this paper, the main focus is on the heat transfer performance of interrupted fins under natural convection. The material parameters of the heat sink are shown in Table 1.

### Table 1. Material of heat sink

<table>
<thead>
<tr>
<th>Material name</th>
<th>Thermal Conductivity (W·m⁻¹·K⁻¹)</th>
<th>Density (kg·m⁻³)</th>
<th>Surface Emissivity</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum</td>
<td>236</td>
<td>2710</td>
<td>0.92</td>
</tr>
</tbody>
</table>

(1) The surface is coated with ZS 411)

#### 2 Theoretical Model of Heat Transfer Performance of Heat Sink

The conventional calculation of natural convection of heat sink is to calculate the Nusselt number based on the geometric parameters of fins and the physical property parameters of the air, and then obtain the convective heat transfer coefficient of the fins, and ultimately obtain the heat transfer between the fin and the environment. The calculation method cannot check the balance between heat transfer rate of fins and heat absorption rate of air in the channel. Moreover, the method of taking ambient temperature as the air temperature in the channel between adjacent fins is only suitable for low fin height, and when used for high fin height, the calculation error is relatively large. Therefore, this paper has improved the theoretical calculation method by the air flow process in the channel between adjacent fins.

To analyze the heat transfer relationship between air flow and fins in the channel and establish its relationship, the heat sink has been simplified as follows: (i) the side air intake of heat sink is ignored, which means that air only flows in from the bottom of fins, (ii) the various channels of heat sink are equivalent, without considering the influence of heat dissipation boundaries, (iii) the air in the channel flows in a one-dimensional steady state and is considered an ideal gas. Based on the above assumptions, the single channel between adjacent fins is simplified as shown in Fig. 2.

In the channel, air flow has driving force and flow resistance. Its driving force comes from the density difference generated by the heated air, while the resistance is composed of laminar flow friction resistance and the local resistance of channel inlet and outlet. The driving force calculation method adopts the self-generated wind force calculation method in the chimney effect. Therefore, based on the chimney effect, the specific calculation process is as follows:

![Fig. 2. Schematic of the channel between fins.](image-url)

The self-generated ventilation force of air in the channel is also as follows:

\[ P_v = \frac{\rho_k g L}{T_\infty} \]  \( (1) \)

where \( P_v \) is the self-generated ventilation force, \( \rho_k \) is the density under standard air conditions, \( L \) is the height of the channel, \( T_\infty \) is the ambient temperature, \( T_b \) is the average temperature of the air in the channel, respectively.

Flow resistance in the channel:

\[ P_w = c_f \frac{L}{D_e} \left( \frac{\rho_1 u_1^2}{2} + \xi \rho_1 \frac{u_1^2}{2} + \xi \rho_2 \frac{u_2^2}{2} \right) \]  \( (2) \)

where \( P_w \) is the flow resistance, \( c_f \) is the friction factor, \( D_e \) is the equivalent diameter of the channel, \( \xi \) is the coefficient of local resistance, \( u \) is the flow velocity of air, respectively.

The self-generated ventilation force is equal to the flow resistance:

\[ P_v = P_w \]  \( (3) \)

Furthermore, air adopts the ideal gas law:

\[ P = \rho RT \]  \( (4) \)

The equation for the air mass flow rate in the channel is as follows:

\[ M = \rho_1 u_1 A \]  \( (5) \)

Channel inlet velocity:

\[ u_1 = \frac{(1/2) \Delta T g \Delta \rho A T^2}{12 \nu} \]  \( (6) \)

where \( \Delta T \) is the difference between base temperature and ambient temperature.

The natural convection heat transfer rate in the channel:

\[ Q = \frac{M c_p}{T_2 - T_1} = \eta h A (T_b - T_k) \]  \( (7) \)

According to the control equation system, all unknown parameters for stable flow can be calculated.
by determining the outlet temperature $T_2$ of the channel. $P_N$ and $P_a$ are both functions of $T_2$, so $T_2$ can be iteratively calculated.

Fig. 3 shows a typical U-shaped groove surrounded by any two adjacent fins of a plate-fin heat sink. If the surface of the groove is a diffuse gray surface and its surrounding medium is very large or blackbody, the radiative heat transfer rate of the groove towards the surrounding environment is [10],

$$Q_{\text{r,ch}} = \frac{e(s+2h)L(T_b^4 - T_{\text{r}}^4)}{\frac{1}{\varepsilon} + \frac{1}{\varepsilon_{\text{b-surf}}}}$$  

where $F_{\text{b-surf}}$ is the total surface view factor of the groove, as:

$$F_{b-\text{surf}} = 1 - \frac{2h}{2h + (1 + \varepsilon^2)^{1/2} - 1}$$  

where $\bar{H}$ and $\bar{L}$ is the normalized length, as:

$$\bar{H} = \frac{H}{S}$$  

$$\bar{L} = \frac{L}{S}$$

Fig. 3. The groove surrounded by adjacent fins of heat sink.

The total radiative heat transfer rate:

$$Q_{\text{r,total}} = \left[ 2 \delta_k (L + W) + 2HL \right] \sigma \left( T_b^4 - T_{\text{r}}^4 \right) + N_f \delta (L + 2H) \sigma \left( T_b^4 - T_{\text{r}}^4 \right) + (N_f - 1) Q_{\text{r,ch}}$$

where $N_f$ is the number of fins, $\varepsilon$ is the surface emissivity coefficient, $\sigma$ is the Stefan-Boltzmann constant equals to $5.67 \times 10^{-8} \text{ W/(m}^2 \text{ K}^4)$, respectively.

3. Performance Analysis of the Heat Sink with Interrupted Fins

The geometric parameters of the heat sink analyzed in this section are shown in Table 2. Among them, $W$, $L$, $\ell_b$ remain unchanged in the analysis, mainly focusing on five factors: $N_i$, $L_i$, $H$, $\delta$, and $N_f$.

3.1. Effects of Fin Interruption on the Heat Transfer Performance

The effects of fin interruption on the heat transfer performance of heat sink are shown in Fig. 4 and Fig. 5. By interrupting the fins, the thermal boundary layer on the fins can be disrupted, which can effectively improve the natural heat transfer coefficient. At the same time, fin interruption also have an impact on the heat transfer area. Due to the geometrical parameters of the heat sink in this paper, however, the lost heat transfer area is exactly equal to the increased heat transfer area when the $L_i = 1$mm. So the heat transfer area remains unchanged as shown in Fig 4 (a).

As $N_i$ increases, in addition to a decrease in fin efficiency, the natural convection heat transfer coefficient and heat transfer temperature difference both increase. In addition, the increase in $N_i$ also increases the mass air flow rate in the channel. The natural convection heat transfer rate shows a trend of increasing with $N_i$ and tends to flatten out when the $N_i = 14$, as shown in Fig 4 (d).

Increasing $L_i$ significantly reduces the heat transfer area compared to increasing $N_i$ (based on Fig 4 and 5), it also reduces the mass air flow rate in the channel. Although the increase in $L_i$ leads to an increase in natural convective heat transfer coefficient, it ultimately leads to a downward trend in the natural convective heat transfer rate of the heat sink.

Radiation heat transfer rate accounts for approximately 13-20% of the total heat transfer rate. Therefore, the influence of radiation cannot be ignored in the heat transfer of heat sinks. The radiation heat transfer is mainly influenced by the emissivity of the material, and is less affected by the fin interruption.

3.2. Effects of Fins on the Heat Transfer Performance

Under the same geometric parameters of the base, increasing the $N_i$ can greatly improve the overall heat transfer area of the heat sink, as shown in Fig. 6 (a). As the $N_i$ increases from 26 to 32, the heat transfer area increases by 21.62%. As the number of fins increases from 26 to 32, the heat transfer area of the fins increases by 21.62%.

But when increasing the $N_i$, as the base width remains unchanged, the fin spacing will decrease. And the distance between the thermal boundary layers generated by natural convection on different fins will also shrink. When the distance between the two is too close, they will interfere with each other, leading to a significant decrease in the heat transfer of the heat sink.

For the heat sink in this paper, the optimal heat transfer performance is achieved when the number of fins is 30.
Table 2. Geometric parameters of heat sink

<table>
<thead>
<tr>
<th>Geometry of Base</th>
<th>Geometric Parameter</th>
<th>Geometry of Fins</th>
<th>Standard Parameter</th>
<th>Variation Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Base width W (mm)</td>
<td>264</td>
<td>Number of interruptions (N_i)</td>
<td>15</td>
<td>1-15</td>
</tr>
<tr>
<td>Base length (L) (mm)</td>
<td>210</td>
<td>Fin interruption length (L_i) (mm)</td>
<td>1</td>
<td>1-5</td>
</tr>
<tr>
<td>Base thickness (\delta) (mm)</td>
<td>2</td>
<td>Fin height (H) (mm)</td>
<td>75</td>
<td>69-75</td>
</tr>
<tr>
<td>Base temperature (T_s) (°C)</td>
<td>105</td>
<td>Fin thickness (\delta) (mm)</td>
<td>1</td>
<td>1.0-4.0</td>
</tr>
<tr>
<td>Ambient temperature (T_\infty) (°C)</td>
<td>45</td>
<td>Number of fins (N_f)</td>
<td>30</td>
<td>26-32</td>
</tr>
</tbody>
</table>
The effect of fin thickness on the performance of the heat sink is shown in Fig. 7. When the fin thickness $\delta$ is between 1.0 mm and 2.0 mm, the mass air flow rate in the channel is higher. When $\delta$ is greater than 2 mm, the mass air flow rate begins to sharply decrease. For $\delta = 2.0$ mm, the heat transfer performance is strongest. While $\delta$ exceeds 3 mm, the heat transfer coefficient begins to significantly decrease, and the convective heat transfer also begins to decrease.

Therefore, for heat sink with interrupted fins, by appropriately increasing the fin thickness, the heat transfer area and fin efficiency have been improved, and the heat transfer coefficient has not significantly decreased, which can effectively improve their natural convective heat transfer.

However, the weight of heat sink increases significantly with the increase of fin thickness. For $\delta = 3.0$ mm, it increases by 162% compared to 1.0 mm, resulting in an increase in the cost of heat sinks. Based on cost factor analysis, the optimal fin thickness is still 1mm.

For the fin height, its increase has no effect on the fin spacing, therefore it has no significant effect on the heat transfer coefficient of heat sink. As fin height increases, the heat transfer area and mass air flow rate slightly increase. Although the fin efficiency has decreased, there is still a positive correlation between fin height and natural convection heat transfer.

For the heat sink in this paper, the convective heat transfer increases by approximately 0.8% for every 1mm increase in fin height, as shown in Fig. 8. Due to the impact of high fin height on the stability of the heat sink which is vertical, increasing fin height should be considered after the optimization design of other geometrical parameters is completed.

Similar to fin interruptions, the effect of fin geometric parameters on radiative heat transfer is also not significant. Although the changes in geometric parameters of fins have a more significant impact on the heat transfer area, the proportion of radiation heat transfer rate is still between 13% - 20% of the total heat transfer rate.
4 Numerical simulation study on natural convection of fins interruption

For the heat sink with interrupted fins, the effect of geometrical parameters is diversified. In this section, the numerical simulation of the effects of $N_i$ and $L_i$ on the convective heat transfer of interrupted fins is emphasized. The geometrical parameters of heat sink used in the numerical simulation research is shown in Table 2. Numerical simulation studies were conducted on three cases: (i) $N_i = 10$, $L_i = 1$mm, (ii) $N_i = 15$, $L_i =$...
1 mm, (iii) $N_i = 15$, $L_i = 3$ mm. The temperature contours are all selected from the cross-section at fin height $H = 37.5$ mm and the central section of the channel between the adjacent fins.

Fig. 9. Temperature contour of the heat sink with $N_i = 10$, $L_i = 1$ mm.

Fig. 10. Temperature contour of the heat sink with $N_i = 15$, $L_i = 1$ mm.

Fig. 11. Temperature contour of the heat sink with $N_i = 15$, $L_i = 3$ mm.

As shown in Fig. 12 and Fig. 13, the error between the numerical simulation and the theoretical model is within 4%, indicating that the theoretical model for heat transfer of heat sink with high fin height is relatively accurate. And from the temperature contours in Fig. 9- Fig. 11, it can be seen that as the number of interrupts increases from 10 to 15, the overall air temperature slightly increases, but the natural convective heat transfer coefficient significantly increases, resulting in better overall heat transfer performance. As $L_i$ increases from 1 mm to 3 mm, the heat transfer dead zone of heat sink can be significantly reduced, but at the same time, it reduces the heat transfer area of the fins. Therefore, in terms of heat transfer performance, $L_i = 3$ mm is not as good as 1 mm, which is consistent with the theoretical model conclusion.

5 Conclusion

This paper takes the heat sink of high-power outdoor electronic equipment as the research object, establishes and improves its natural convection heat transfer theoretical model. And by analyzing its natural convection and comparing the effects of different fin heights, fin thicknesses, fin numbers, and different interruption numbers and fin interruption lengths on heat transfer rate, and draws the following conclusions:

1) The interrupted fin greatly improves the convective heat transfer coefficient of the heat sink, and by breaking the heat transfer boundary layer on the fins, the area of the heat transfer dead zone on the upper part of the fins is reduced, effectively improving the heat transfer performance of the heat sink. After theoretical analysis and numerical simulation, the optimal number of interruption for the heat sink proposed in this paper is 15, and the interruption is 1 mm.

2) Under the same available space and operating conditions, the interrupted fin can not only improve heat transfer performance but also reduce the weight of heat sink compared to continuous fin. The interrupted fin heat sink, which has 15 interruptions and 1 mm interruption length, reduces weight by more than 7%, improving economic benefits.
References


