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A novel heat dissipation fin design applicable for natural convection augmentation☆

Roody Charles, Chi-Chuan Wang *

Department of Mechanical Engineering, National Chiao Tung University, Hsinchu 300, Taiwan

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ABSTRACT

In this study, a comparative study of heat sink having various fin assembly under natural convection is investigated. The fin pattern includes a rectangular, a trapezoidal and an inverted trapezoidal configuration. Tests were performed in a well controlled environmental chamber having a heat load ranging from 3 to 20 W. From the test results, the heat transfer coefficient of the conventional rectangular fins is higher than that of the trapezoidal fins while the heat transfer coefficient of the inverted trapezoidal fins is higher than the trapezoidal one by approximately 25%, and it exceeds that of convectional rectangular fin by about 10%. The heat transfer improvements of the inverted trapezoidal fin are mainly associated with a larger temperature difference and inducing more air flow into the heat sink.

1. Introduction

Air-cooling is still the most widely used methods for heat dissipation in electronic applications. This is because air cooling is reliable and easy to implement. However, because the considerably low thermal conductivity of air heat sink normally incorporated with substantial fin surfaces to reimburse the poor heat transfer performance of air for effective heat removal. Yet the added surfaces may occupy a lot of space and weight especially when heat sinks were operated under natural convection. Hence it is imperative to reduce the size/volume as much as possible provided the junction temperature is below the threshold requirement.

Many researchers had devoted efforts to fin design for natural convection condition and Table 1 tabulated some related studies for the past decade. Among them, Bar-Cohen et al. [1] demonstrated that there is a least-material optimization for the vertical rectangular longitudinal plate fin arrays in natural convective heat transfer. Mokheimer [2] considered the effect of the variation of local heat transfer coefficient alongside the fin surface, and reported a considerable deviation of the conventional constant heat transfer coefficient assumption that may result in a significant underestimation of the fin efficiency. Khan et al. [3] examined some selected fin geometries subject to influences of axis ratio, aspect ratio, and Reynolds number. Their results clearly indicated that the preferred profile is very dependent on these parameters. Huang et al. [4] concluded that the optimal porosity of a plate heat sink is around 83% for the upward arrangement and is around 91% for the sideward arrangement. Goshayeshi and Ampofo [5] found that vertical plate with vertical fins gives the best performance in natural convection. Suryawanshi and Sane [6] investigated the performance between plate and inverted notched fin array. They reported that the average heat transfer coefficient for the inverted notched fin arrays is nearly 30–40% higher than that of conventional plate fin array.

Zhang and Liu [7] investigated the optimal spacing between isothermal laminar natural convection analytically and numerically. They showed that the optimal plate spacing depends on the outlet velocity of the heat sink. By inserting multi-scale plates in the boundary layer, the heat transfer enhancement could be achieved effectively. Fahiminia et al. [8] stated that a maximum heat transfer rate is attainable for an optimum fin spacing. Similar results were reported by Goshayeshi et al. [9] who concluded that at a given fin height and a temperature difference, the heat transfer rate may first increase with fin spacing and it reaches a maximum, followed by a noted decline. Torabi et al. [10] highlighted that the concave parabolic fin yields the optimum utilization of material, giving the highest heat transfer rate, fin efficiency, and fin effectiveness. For all three geometries, the effect of the temperature-dependent heat transfer coefficient is to decrease the fin heat transfer rate as compared to those with a constant heat transfer coefficient. However, the fin efficiency is higher when the convective heat transfer coefficient is temperature-dependent than when it is a constant. Tari and Mehrtash [11] reported effectiveness of the selected fin design with various fin thickness by visualizing temperature distributions. The effect of tilt angle is thoroughly investigated and their results suggested that within some small inclinations from the vertical in both directions, the inclination does not reduce the convection heat transfer rate and the heat transfer rate can even slightly increase at a very small downward inclination due to the thinner boundary layer. Mehrtash and Tari [12] investigated the optimum fin spacing for all inclinations varying from the downward facing horizontal to the upward horizontal arrangement, the optimum tilt angle is vertical arrangement. They concluded that

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* Corresponding author.
E-mail address: ccwang@mail.nctu.edu.tw (C.-C. Wang).

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Nomenclature

A	cross sectional area of the fins, m^2
A_b	cross sectional area of the Bakelite, m^2
h	heat transfer coefficient, W m^{-2} K^{-1}
k_b	thermal conductivity of the Bakelite, W m^{-1} K^{-1}
k	thermal conductivity of fluid, W m^{-1} K^{-1}
Nu	Nusselt number, dimensionless
Pr	Prandtl number, dimensionless
Q	Rate of heat transfer, W
Ra	Rayleigh number, dimensionless
T_f	fin temperature, °C
T_a	ambient temperature, °C
T_b & T_w	base plate temperature, °C
T_s	surface temperature of the heat sink, °C
ΔT	effective temperature difference, °C
x	thickness of the Bakelite, m
θ	inclined angle, degree

Subscripts

f	plate-fin
b	base
c	convection
i	input
t	total
l	loss

optimum fin spacing for the vertical orientation is most favorable. Tari and Mehrtash [13] later investigated the previously uncovered inclination angle for the inclined cases and developed a set of correlations. These correlations were shown to be very accurate in predicting heat transfer rates using the available horizontal data from the literature. Kim et al. [14] showed that the thermal resistance revealed an optimal value at a specific fin number. However, the thermal resistance decreases continuously without reaching an optimal value as the fin height is increased. Naserian et al. [15] concluded that by increasing the fin number and the fin spacing, the ratio of natural convection heat transfer coefficient of various fin configurations to the corresponding vertical plate is increased.

Based on the foregoing discussion, apparently the fin design casts significant influences on the natural convection. Normally, the easiest way is just to impose more surface area for effective reduction of thermal resistance. However, as aforementioned from previous studies, the induced air flow under natural convection is normally quite restricted and can be significantly impaired by dense fin design, jeopardizing the heat transfer accordingly. In this regard, the principal objective of this study is to propose a novel fin design applicable for natural convection.

2. Experimental setup

The designated concept of the fin pattern is simple. Instead of increasing the surface area, the basic idea is to increase the effective temperature difference, especially at the rear part of fin surface. Fig. 1 illustrates the proposed concept in association with the conventional design. Fig. 1(a) is the convectional rectangular fin with identical cross-sectional surface area at the fin base and fin tip, Fig. 1(b) is the trapezoidal fin whose cross-sectional area at the fin base is larger than that in the fin tip, and Fig. 1(c) is the proposed inverted trapezoid fin in which the cross-sectional area at the fin tip is larger than that at the fin base. Note that the effective surface areas amid these three designs are the same, and their detailed dimensions are shown in Fig. 1(d). Experiments are performed in an environmental chamber whose volume is 900 mm × 900 mm × 1240 mm. The environmental chamber can provide a temperature condition in the range of 20–50 °C with a controlled resolution of 0.2 °C. In this study, the ambient temperature is fixed at 25 °C. To simulate the natural flow condition, the air ventilator is turned off inside the test chamber when the ambient temperature reaches 25 °C. In particular, the air conditioner outside the test chamber continues to operate to maintain the room temperature at 25 °C. The test facility is inside the test chamber which consists of a heat sink, a heater, and insulation box, and a tilting mechanism as illustrated in Fig. 2(a).

The heat sinks are made of aluminum alloy 5083 with a thermal conductivity 121 W m^{-1} K^{-1}. Five pin fin heat sinks are made via CNC machining with a manufacturing precision of 0.03 mm. The heat sinks are operated with the power inputs from 3 to 20 W. Detailed dimensions of the test samples are also shown in Fig. 1. A Kapton heater with identical size as the base plate of the heat sink is used to eliminate the spreading resistance. An insulation box made of bakelite with a low thermal conductivity of 0.233 W m^{-1} K^{-1} is placed beneath the heater to reduce the heat loss. In addition, a high thermal conductivity grease ($k = 2.1$ W m^{-1} K^{-1}) is used to connect the heat sink and the heater for further minimization of the contact resistance. The heater is powered by a DC power supply.

As seen in Fig. 2, a total of five T-type thermocouples which are equally divided and located at beneath the base plate are used to obtain the mean temperature of the base plate (T_b) of the heat sink. In addition, a total of 10 T-type thermocouples are installed inside the insulation box at two cross positions to calculate the heat loss from the bottom of the Kapton heater. Each cross-section is equally instrumented with five T-type thermocouples to obtain the mean temperature of that cross section. The average temperature is then used to estimate the heat loss via Fourier's law of conduction. The thermocouples were pre-calibrated with an accuracy of 0.1 °C. The exact total heat transfer rate by natural convection supply (Q_t) is then obtained by subtracting the estimated loss from the power input. The signals from thermocouples are then transmitted to a data acquisition system for further data reductions. Usually, each test run needs approximate 2.5 h to reach equilibrium when the power is turned on.

3. Data reduction

In the present study, the ambient air temperature is always controlled at 25 °C and the thermophysical properties are evaluated at the film temperature, i.e.

$$T_f = 1/2(T_a + T_b). \quad (1)$$

The actual heat transfer rate, Q_t, is determined by subtraction the heat loss Q_l from the measured heat input Q_i of the Kapton heater:

$$Q_t = Q_i - Q_l \quad (2)$$

$$Q_l = \frac{k_b A_b (T_b - T_a)}{x}. \quad (3)$$

The average heat coefficient can be calculated from the following:

$$h = \frac{Q_t}{A(T_b - T_a)}. \quad (4)$$

The experimental uncertainty is estimated using the uncertainty propagation equation proposed by Kline and McClintock [16]. The maximum measured uncertainties of the heat transfer coefficient are about 11%, occurring at the lowest input power of 3 W. In particular, this uncertainty drastically decreases to less than 3% when the power input is larger than 5 W.