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Natural convection heat transfer enhancement in new designs of plate-fin based heat sinks

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Experimental investigation was conducted to measure the convective heat transfer coefficient and thermal performance of plate fins and plate cubic pin-fins heat sinks, under natural convection regime. The investigation was conducted for Rayleigh number from 8×10^6 to 9.5×10^6 and input heat of 10 W to 120 W. The fin spacing and fin numbers are varied between 5–12 mm and 5–9, respectively. The results demonstrated that plate cubic pin-fins heat sinks have lower thermal resistance and higher heat transfer, compared to plate fins heat sinks. Heat transfer enhancement of new-designed heat sinks is about 10–41.6% higher, compared to normal pin-fins. Increasing fin spaces in all types of studied heat sinks cause lower thermal resistance. But, increasing fin numbers does not cause better heat transfer. The best heat sink design was a plate cubic pin–fin heat sink with 7 fins and 8.5 mm fin spacing. Finally, empirical equations have been developed to correlate the average Nusselt number as a function of number of fin plates, fin spacing to height ratio as well as Rayleigh (and consequently Grashof) number.

1. Introduction

Heat transfer has wide applications around the world, including electronic systems, MEMS and robotics [1,2]. Fast development pace in technologies related to circuits and their components has exacerbated the heating problem. As a result, limiting the maximum temperature is essential in modern electronic devices. Also, the amount of generated heat has been augmented in recent electronic products, while their heat dissipation surface area becomes smaller. Nowadays, complex packaging configurations have become prevalent in electronic systems with packing densities as high as 10^6 chips/m^3 [3]. As a result, in order to extract heat from components and keeping the devices in low temperatures (T < K), efficient cooling systems are needed [3]. For this purpose, there are two main solutions to manage the heat generation: active and passive cooling. Active cooling works by forced motion of cooling fluid through and over the electronic components to absorb their exhaust heat [4,5]. On the other hand, passive cooling benefits from radiation and free convection heat transfer in order to transfer generated heat to the environment [6,7]. Obviously, active cooling has the higher value of heat transfer rate, but it requires greater energy consumption because fans or pumps are used to carry the cooling fluid, and it generates noise as well [8,9]. On the other hand, heat removal from many electronic and telecommunication devices are done by passive cooling, because of its good traits such as being silent, reliable, and cost effective. One of the important techniques for thermal design of such devices is air cooling, with the advantages of being low cost and reliable and also exhibiting safe operation when used in antagonistic environments [10,11].

Fins, which are a common name for extended surfaces, are generally used in air-cooled systems. Across all of the applications, utilized fins in electronics application are used in arrays, called heat sinks. Usually, the transfer of the generated heat to the heat sinks occurs through heat conduction, and dissipation of such heat to the environment is done through convection [12]. To achieve a better heat sink design, different parameters as shape [13], size [14], material [15], inclination angle [16], flow regime (laminar, transitional, turbulence) [17–20], type of heat transfer (phase change, natural, forced or mixed convection) [21–23], and heat transfer rate [24] should be considered.

Many techniques have been introduced in the past years to improve the efficiency of heat sink systems through natural convection heat transfer. Over the past recent years, a wide variety of designs have evolved to meet the rising heat dissipation demand. A number of studies concerning rectangular fins have

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Nomenclature

h	convection heat transfer coefficient [W/m^2 K]
I	current [A]
L	fin length [m]
d	gap between pieces [m]
g	gravitational acceleration [m^2/s]
Gr	Grashof number
q″	heat flux [W/m^2]
q	heat transfer rate [W]
H	height of the plate [m]
S/H	fin spacing to height ratio
n	number of rows
N	number of fin plates
Nu	Nusselt number
P	power [W]
Pr	Prandtl number
PCPF	plate cubic pin-fin heat sink
PF	plate-fin heat sink
PPF	plate-pin fin heat sink
Ra	Rayleigh number
S	fin spacing [m]
T	temperature [K]
k	thermal conductivity [W/m K]
R	thermal resistance [K/W]
U	total uncertainty
V	voltage [V]

Greek symbols

β	thermal expansion coefficient [1/K]
μ	fluid viscosity [Pa s]
ρ	density [kg/m^3]
Δ	difference

Subscripts

b	base plate
H	hydraulic
in	input
opt	optimum

been done from both applied and theoretical point of view, recently. Feng et al. [9] designed a new heat sink that is consisted of a number of long fins and a number of short fins arranged perpendicularly, which aimed at maximizing free convection. Their outcomes showed 15% convective heat transfer enhancement, in comparison with a plate-fin heat sink, without any overall volume/material usage augmentation, or extra expenses. Jeon and Byon [25] used ANSYS Icepack commercial software to investigate thermal efficiency of plate fin heat sinks with dual-height fin segment under free convection regime. Their results illustrated that the dual-height configuration shows its superiority for fins with a height greater than a threshold value. Lee et al. [26] experimentally studied the influence of fin numbers on natural convection heat transfer from vertical cylinders with triangular fins. They found that the Nusselt number is not influenced by spacing between fins when the fin spacing is large. However, when small fin spacing is employed, by decreasing the fin spacing, Nusselt number is reduced monotonically. Micheli et al. [27] studied the heat transfer performance of plate and pin-micro fin arrays under free convection regime, experimentally. Their study indicated that the efficiency of pin micro-fins is higher than plate-type, while manufacturing process needs less material. Joo and Kim [28] investigated the impacts of width and length spacing on heat transfer and pressure reduction of fin-pins with multiple arrangements. They discovered that by decreasing length and width of pin-fins, heat transfer is enhanced; whilst this enhancement depends more to length distance, compared to the width of fin. A modern design of plate-pin fin heat sink (PPF) was studied experimentally and numerically by Yu et al. [29]. Their design was based on adding several column-shaped pins with staggered arrangements between plate-fins to a simple plate fin heat sink. Their outcomes indicated that in similar conditions, PPF has 30% less thermal resistance compared to traditional studied plate-fin heat sink.

Reviewing the literature makes it clear that although many papers have considered different types of heat sinks and investigated various aspects of it, still plate fin heat sinks are the more common and mainstream choice because of their simple geometry and well thermal performance. As a result, having a superior performance is not the only consideration in manufacture of practical heat sinks, but a simple geometry is also considered a merit along with low cost of manufacturing. The goal of present study is to compare heat transfer coefficient of a new type of plate cubic pin fin (PCPF) and plate fin (PF) heat sinks under free convection heat transfer. Based on plate pin-fin, a new type of plate cubic pin-fin heat sink is designed and constructed which is consisted of a plate fin heat sink and some cubical pins with linear arrangements between plate fins. Then, some experimental tests were carried out to compare the thermal performances of these two kinds of fin arrangements. The outcomes of this research can be beneficial for engineers whom work on electronics cooling systems.

2. Experimental apparatus and uncertainties

In the experimental setup, various instruments were used to measure temperature at different points including ambient temperature, input power of heaters, front side of guard heater, back side of the main heater and fins.

Thermo-foil laminar heater is mounted on the front side of the base plate in the considered fin to provide Dirichlet boundary condition. Insulation of the front side of main heater is done perfectly by means of a guard heater to guarantee complete heat transfer to the fins. Also, the gap between the front side of the guard heater and the main heater is filled by hardener and epoxy. The epoxy plate is in contact with heaters and its temperature is controlled at the same temperature of heaters; in order to omit the heat loss from main heater's back side. The front side of the guard heater and lateral sides of both heaters are also insulated via rock wool.

Experimental apparatus is showed in Fig. 1. The experiments are performed at constant air temperature and inside a laboratory, without any HVAC system. Precise values of input power are feed to the main heater; using a Variac (variable transformer). Input heat was varied from 10 W to 120 W. A voltmeter-ammeter is used to monitor the current flow and voltage drop of the system in order to calculate the supplied power (supplied power = heater current × voltage drop). Temperature difference is considered to be negligible and steady state condition is supposed to be reached, when the recorded temperature difference of each thermocouple becomes less than 0.5 °C/h. All the observations are recorded after confirmation of steady state condition.

Configurations of examined heat sinks as well as studied geometrical parameters are demonstrated in Fig. 2. Aluminum alloy 6061 (thermal conductivity = 170 W/m K) is selected as the heat sink material. For the studied configurations, the dimensions