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Investigating the Impact of Heating Parameters on Base Temperature Profiles and Thermal Resistance in Macro Scale Plate-Fin Heat Sinks

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Abstract

In this study, plate-fin heat sinks base 100×60 with constant heating (10,20,30W) under forced convection were experimentally investigated. The base temperature profiles of the plate-fin heat sinks were measured for various heating lengths, heating positions, flow rates, and channel widths. From the experimental data, the effects of heating length, heating position, and flow rate on the base temperature profile and the thermal performance were investigated. Finally, the characteristics of the optimal heating position were investigated. As a result, it was shown that the optimal heating position was on the upstream side in the case of the heat sinks under laminar developing flow, as opposed to the heat sinks under turbulent flow. It was also shown that the optimal heating position could change significantly due to heat losses through the front and back of the heat sink, while the effects of the heat loss through the sides of the heat sink on the optimal heating position were negligible. In addition, it was shown that the one-dimensional numerical model with empirical coefficients could predict the important trends in the measured temperature profiles, thermal resistances, and optimal heating lengths.

INTRODUCTION

Background

Electronics have become an indispensable component of today's lifestyles. The tremendous rise of consumer electronics and industries need good heat management of electronic components, which can only be accomplished through system downsizing. As a result, scientists have been trying to make the cooling system more efficient and efficient. Electronic heat management is now a critical factor in their design and development. Previous research has shown temperature as one of the primary causes of field failures in electronic equipment. To maintain reliable functioning, it is critical to maintain the working temperature of electronic components within a regulated range. However, numerous classic cooling solutions (for example, natural convection and forced convection) are no longer useful in addressing the evaporation needs of today's high-performance electronic constituents in order to achieve this. The demand for high-performance and compact products has resulted in a continuous increase in heat dissipation from electronic devices in various systems, such as power converters, supercomputers, and electric vehicles. The high heat dissipation from electronic products results in a high junction temperature, which negatively affects the overall performance and durability of the product. Therefore, efficient cooling of these devices is essential and various cooling methods have been developed.

In this study, plate-fin heat sinks under partial heating with air as the working fluid were experimentally investigated. The base temperature profiles of plate-fin heat sinks under laminar developing flows were measured for various heating lengths, heating positions, flow rates, and channel widths. From the experimental data, the effects of various engineering parameters on the base temperature profile and the thermal performance were investigated. Finally, the characteristics of the optimal heating position were investigated. It is shown that the conclusions of the previous studies on optimal heating position of plate-fin heat sink under turbulent flow did not apply to the heat sink under the laminar developing flow examined in this study. In is also shown that the optimal heating position could change significantly due to heat losses through the front and back of the heat sink. In addition, a simple one-dimensional numerical model for predicting the thermal performances is presented.

LITERATURE REVIEW

Dong-Kwon Kim, et al, 2009, compared plate and pin-fin heat sinks subjected to an impinging flow for different flow rates and channel widths. They found that optimized pin-fin heat sinks shows lower thermal resistances when dimensionless pumping power is small and the dimensionless length of heat sinks is large. (Abdullah H, et al, 2009) compared solid and perforated horizontal rectangular fin in natural convection by using finite element technique. They observed that, perforated horizontal rectangular fin has more heat transfer and less weight compared to equivalent solid fins.

E.A.M.Elshafei, 2010, compared solid circular pin fins with perforated circular pin fins in staggered arrangement in natural convection. They also tested the performance of the heat sink for two different orientations and observed that, the perforated pin fins gives better heat transfer in sideward arrangement than upward arrangement.

Monoj Baruah, et al, 2011, investigated the performance of elliptical pin fins which were arranged in a staggered manner. Three perforations were provided on elliptical pin fins and its performance compared with corresponding solid elliptical pin fin. They have observed that, the three perforations on elliptical pin fins give more heat transfer.

Anupam Dewan, et al, 2011, also studied heat transfer from an array of circular pin fins and observed that, the perforation on pin fin reduces the weight and the cost of pin fin. (Ji-Jinn Foo, et al, 2012) studied heat transfer with perforated pin fin by varying the number of horizontal perforation and the diameters horizontal and vertical perforation on pin fin. They found that the perforated pin fin array gives more heat transfer than the corresponding solid pins.

G. Ganesh Kumar, 2013, observed that the variation of local temperature difference is more for hollow perforated pin fin compared to that the solid perforated pin fin.

Ali Shakir Baqir, et al, 2014, studied staggered perforated pin fin array in a rectangular channel. They have compared HLV (horizontal, vertical, lateral) perforated pin fin with HV perforated and solid pin fin. They observed that the HLV perforated pin fin shows higher heat transfer than HV perforated and solid pin fin heat sink.

Saurabh D. Bahadure, and G. D. Gosavi, 2014, carried out the thermal performance of a pin fin heat sink. In experiment, they varied the material of pin fin such as Aluminum, Copper and mild steel and number of perforation on pin fin from 1 to 3 respectively. They found that the average heat transfer coefficient is higher for three perforations compared to solid, single, and two perforations respectively.

Murtadha Ahmed and Abdul Jabbar N. Khalifa, 2014, compared solid and perforated square pin fin heat sinks in natural convection. They found that the temperature drop across perforated pin fin was less than that of solid pin fins. Vishvas S. Choure, 2015, studied heat transfer enhancement from perforated pin fin in staggered arrangement. The number of Perforation varied from one to five and diameter of perforation from 3 to 5 respectively. They have observed that the Nusselt number increases with increase in number of perforation and diameter of perforation.

In present study, experiment is carried out to investigate heat transfer rate in natural convection of parallel plate fin heat sink with circular pin fins placed between the plate fins. The firstly parallel plate heat sink consists of without perforated circular pin fin and its performance is compared with perforated circular pin fin. The variation of height of circular pin fin and effect of number perforation on heat transfer coefficient is also examined.

Heat Sinks Dimensions

The dimensions of the setup are given as below; Base plate = 100 mm x 60 mm. Height of parallel plate fin = 30 mm. Thickness of parallel plate fin = 2 mm. Height of pin fin = 30 mm. Diameter of circular pin fin = 5 mm. Diameter of perforation = 2 mm. No. of perforation =1.

EXPERIMENTAL SET UP AND PROCEDURE

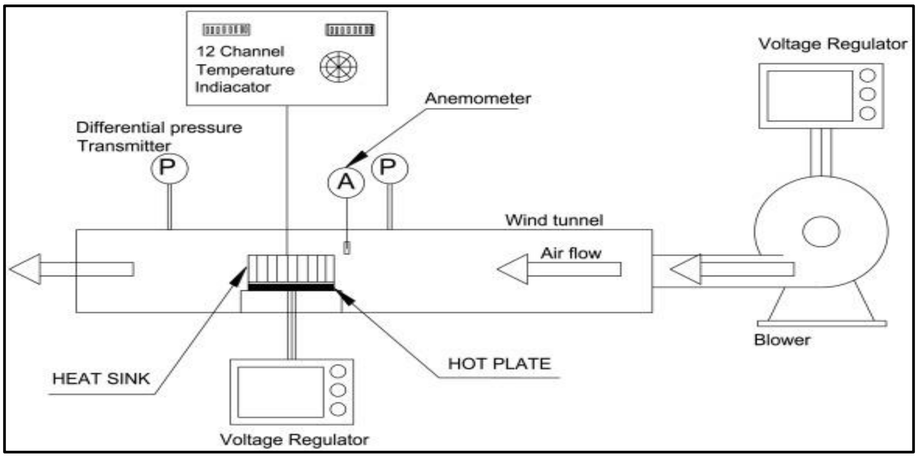


Fig 1 Experimental Set up

A schematic diagram of a heat sink is shown in Figure 1. The length L, width W, height H, base thickness tb, and fin thickness tf of the heat sink were 100 mm,60 mm, 30 mm, 5mm, and 2 mm, respectively. The experiment was conducted for five different velocities 1 to 5 m/s. The heat sink was made of Al (k = 210 W/m K). As shown in Figure 2, 22 T-type thermocouples were attached to the base of the heat sink. More specifically, in order to measure the profile of the base temperature Tb along the flow direction, thermocouples were installed at 5 points in the diagonal directions. The thermocouples were attached at two points in the y direction to check that the measured base temperature profile along the flow direction was reliable. Silver epoxy was used to minimize contact thermal resistance between the thermocouple and the base. As shown in Figure 1, the signals from the thermocouples were converted to temperature data using a data acquisition system (34970A DAQ; Agilent Technology) and were stored in a computer. As shown in Figure 2, film heaters of the same size were attached to the bottom surface of the heat sink base to locally apply heat to the heat sink. The film heater was made of mica (100W) and was attached using a Kapton tape. The heaters were powered and controlled by a DC power supply to be turned on and off, independently.

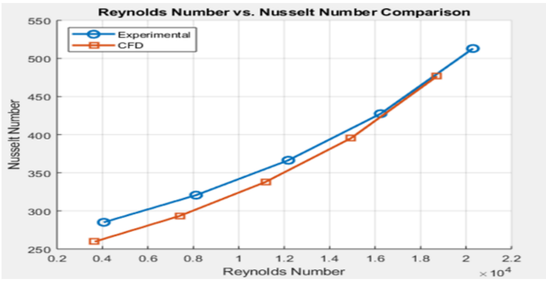
The total power input to the heater was fixed at 10,20,30W. As shown in Figure 1, the heat sink with the attached thermocouples and heaters was installed inside a rectangular duct in acrylic. The duct was connected to a wind tunnel system with a nozzle flow meter and a blower (TB-150; InfoTech) to measure the flow rate of air while generating air flow in the duct. The nozzle flow meter consisted of the nozzle and the differential pressure meter and it measured the volume flow rate by measuring pressure difference between the inlet and the outlet of the nozzle caused by flow restriction by the nozzle. Each measurement was repeated three times to ensure the reliability of data. The temperature was measured until the temperature change dropped to less.

Plate Fin Heat Sink (PFHS)

Table 1. Design parameters for the experiment

Heat Supplied Q (W)	**Velocity (m/s)**	**Reynolds No (Re)**	**Average Base Temperature (°C)**	**h (Heat Transfer Coefficient) W/m²K**	**Nusselt Number (Nu)**	**R_{th} (°C/W)**
10	5	20303.8	30	134.40	513	0.5
20	5	22923.07	41	84.00	320.63	0.8
30	5	25880.14	56	65.036	230	1.03

Data Reduction: Sample Calculations,
Heat input supplied to fins is given by,
$Q_{in} = VI$
The most of the heat lost by radiation to surrounding air which is calculated as,
$Q_{rad} = \varepsilon\sigma As (T_s^4 - T_a^4)$
Heat lost by convection $Q_C = Q_{in} - Q_{rad}$
This rate of heat transfer by convection can be given by,
$Q_C = h A_s (T_s - T_a)$
Calculation of base and exposed area of fins Area of base plate (A_{bp}) =L x W
Area of one plate fin= (2xHxL) + (2xHx t)+ (L x t)
Area of all vertical plate fin,
(A_f)= Area of one plate fin x No. of fin
Area of all vertical pin fins with perforation is,
(A_{pf})= N_f + (πDH) – (n + nπdD)}
Total surface area As=Abp+ Af + Apf
The mean temperature is calculated as,
$T_{bavg} = (T_{in} + T_{out})/2$



Graph 1 Nusselt Number Vs Reynolds Number for Q = 10W

This dataset compares experimental and CFD (Computational Fluid Dynamics) results for Reynolds number and Nusselt number at various flow rates for a heat transfer system where Q=10W for Case 1. The Reynolds number (Re) shows the flow regime, with experimental values consistently higher than CFD values. Both sets increase with flow rate, indicating intensified convective heat transfer. Similarly, the Nusselt number (Nu), representing convective heat transfer efficiency, also rises with flow rate. Experimental Nu values are consistently higher than CFD predictions, highlighting potential modeling limitations in CFD simulations.

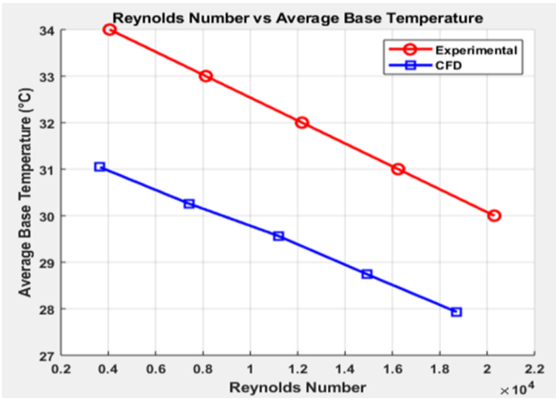
For instance, at Re=4060.77 (experimental), Nu=285.01 experimentally but only 259.64 computationally. At the highest flow rate (Re=20303.87), Nu reaches 513.01 (experimental) versus 476.59 (CFD). These differences suggest refinement opportunities in CFD models for better accuracy in predicting heat transfer.

The average convective heat transfer coefficient is given by,
$Nu = hL / K_{air}$
Experimentation: A schematic of the experimental test setup is as shown in fig.5.1. The experimental apparatus mainly consisted of Wind Tunnel Air Flow Bench, Heat sink, Heater (100W, 230V), mica sheets and other accessories. The air was supplied by the Wind tunnel air flow bench through its Centrifugal Blower. A Honeycomb structure is used inside the plenum chamber in order to provide streamlined flow prior to entering the jet plate and a butterfly valve used in order to regulate the discharge.
Data Acquisition System: The main aim of this system is to measure steady state temperatures in the jet impingement cooling experiment. For this purpose, high reliable accuracy is required which is possible by using right sensors, signal conditioners, noise reduction and high resolution A/D converter

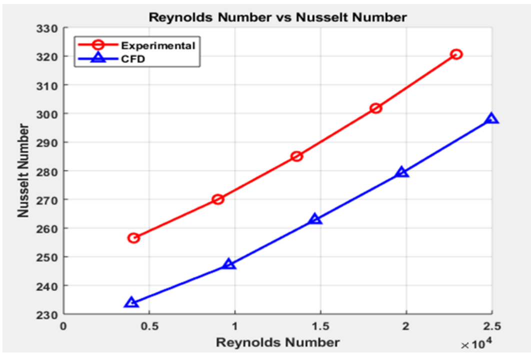
RESULTS AND DISCUSSION

The diagrams illustrate the temperature differential profiles between the base (Tb) and ambient (T_a) temperatures along the flow direction. The following trends were observed:

GRAPHS

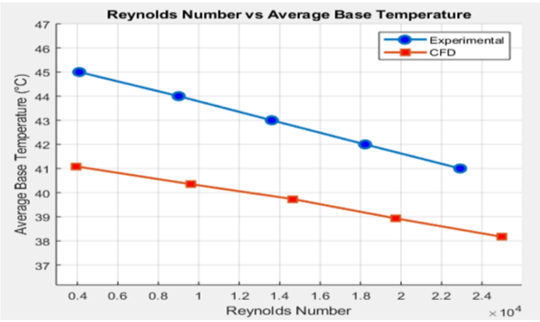


Graph 2 Reynolds Number Vs Average Base Temperature for Q = 10W



Graph 3 Reynolds Number Vs Nusselt Number For Q= 20W

The data compares experimental and CFD (Computational Fluid Dynamics) results for Reynolds number (Re) and Nusselt number (Nu) at Q=20W in different cases. It shows a consistent trend of increasing Reynolds and Nusselt numbers as the flow rate rises. While both methods align well, the CFD results tend to slightly underestimate the Nusselt number numbers is observed at lower Reynolds numbers, gradually decreasing with increasing flow.



Graph no 04 Reynolds Number Vs Average Base Temperature for Q = 20W

The thermal resistance exhibited a consistent decrease as the flow rate increased, regardless of heating position or channel width. This occurred because the temperature of the cooling fluid within the heat sink decreased with higher flow rates. As the heating length decreased, the thermal resistance increased, irrespective of channel width and flow rate. This was attributed to the more concentrated heating of the heat sink base, resulting from an increased heat flux (heat per unit area) applied to the heated portion as the heating length diminished. In the case of uniform heating of the heat sink base, heat dissipation to the fluid occurred evenly across the entire heat sink surface. Conversely, when the heat sink was locally heated, only a specific portion of the surface was primarily responsible for heat dissipation.

The following thermal resistance characteristics were observed in the graphs. The thermal resistance decreased steadily with increasing flow rate, regardless of heating position and channel width. This occurred because higher flow rates led to lower temperatures of the cooling fluid within the heat sink. This was due to more localized heating of the heat sink base, resulting from increased heat flux (heat per unit area) applied to the heated portion as the heating length diminished. In uniformly heated heat sinks, heat dissipation to the fluid occurred evenly across the entire surface. Conversely, locally heated heat sinks primarily utilized only a portion of the surface for heat dissipation. compared to the experimental values. For Reynolds numbers ranging from 4089.199 to 22923.07, the experimental Nusselt numbers rise from 256.50 to 320.63, while the CFD values increase from 233.68 to 297.87. The maximum percentage deviation between experimental and CFD Nusselt

CONCLUSION

In this study, plate-fin heat sinks with a base dimension of 100×60 mm were experimentally investigated under forced convection with constant heating powers of 10W, 20W, and 30W. The base temperature profiles of the heat sinks were analyzed for various heating lengths, heating positions, flow rates, and channel widths to understand their impact on thermal performance. The experimental results demonstrated that heating length, heating position, and flow rate significantly influenced the base temperature distribution and thermal resistance. It was observed that the optimal heating position for heat sinks under laminar developing flow was located on the upstream side, whereas heat sinks subjected to turbulent flow exhibited different optimal heating characteristics. Furthermore, heat losses from the front and back surfaces of the heat sink significantly affected the optimal heating position, while sidewall heat losses had a negligible impact. A one-dimensional numerical model with empirical coefficients was used to predict temperature profiles, thermal resistances, and optimal heating lengths. The numerical model successfully captured key trends observed in the experimental data. The thermal resistance consistently decreased with increasing flow rate, regardless of heating position or channel width. This was attributed to the cooling fluid's lower temperature at higher flow rates, which enhanced heat dissipation. Additionally, as the heating length decreased, the thermal resistance increased, irrespective of channel width or flow rate. This was due to localized heating, where heat flux (heat per unit area) was more concentrated over a smaller heated portion of the base. In cases of uniform heating, heat dissipation occurred evenly across the heat sink surface, whereas in localized heating, only a specific region was responsible for heat dissipation. Overall, this study provides valuable insights into the thermal optimization of plate-fin heat sinks, highlighting the importance of heating position, flow rate, and heat losses in determining their thermal performance. The findings can be applied to enhance the design of heat sinks for various thermal management applications.

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