*Journal of Applied Fluid Mechanics*, Vol. 8, No. 3, pp. 515-520, 2015.
Available online at www.jafmonline.net, ISSN 1735-3572, EISSN 1735-3645.
DOI: 10.18869/acadpub.jafm.67.222.22930

# Experimental Study of Air-Cooled Parallel Plate Fin Heat Sinks with and without Circular Pin Fins between the Plate Fins

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(Received April 04, 2014; accepted June 25, 2014)

## ABSTRACT

Experiments were conducted to investigate forced convective cooling performance of an air cooled parallel plate fin heat sink with and without circular pin fins between the plate fins. The original parallel plate heat sink was fabricated consist of 9 parallel plates of length 53 mm with cross-sectional area of 1.4 mm in width by 20 mm height for each plate. The second heat sink has the same geometry of original one but with some circular pins between the plate fins. Thermal and hydrodynamics performances of the heat sinks have been assessed from the results obtained for the pressure drop, thermal resistance and overall performance with the free stream air velocity ranging from 4.7 to 12.5 m/s. Results show that the free stream air velocity has a significant effect on the thermal and hydrodynamics performance of the system. With increasing free stream velocity, the heat transfer coefficient increases and consequently the thermal resistance decreases while pressure drop increases due to higher inertial of fluid at higher velocities. Furthermore, at the same free stream air velocity the thermal resistance for the heat sink with circular pin is about 37.7% lower than that of the original heat sink.

**Keywords**: Parallel plate fin heat sink; Thermal performance; Thermal resistance; Pressure drop.

## 1. INTRODUCTION

Over the past several decades, the heat flux in electronic packages has increased drastically due to the reduction in package size combined with higher power dissipation. Lack of an adequate heat dissipation mechanism cause reduction of the efficiency of the electronic product and possibly damage as the temperature rises. Therefore, the effective thermal management of electronic components is vital. In order to effectively cool small and high-power electronics components, the heat sink module is the most common heat exchanger for CPUs and has been extensively used in cooling of electronic components. The conventional heat sink module utilized the forced convection cooling technique to dissipate heat from heat source to the ambient air. The combination of the fan and heat sink design usually involved in this forced convection cooling technique

Parallel plate heat sinks are one of the widely used heat sinks in electronic cooling because of their many advantages such as low cost, easy manufacturing and simple structure. Nowadays, various forms of parallel plate heat sinks are manufactured and supplied to the market in large quantity because they can be good solution for many thermal issues in electronic cooling industries.

There are many studies about heat transfer and fluid flow characteristic of these types of heat sinks. For example, Shin (1998) investigated rectangular fins and included cooling by natural convection. Iyengar and Bar-Cohen (2003) considered the thermal performance and pumping power requirement of a heat sink with rectangular fins in forced convection flow. Teertstra *et al*. (1999) examined air flow parallel to the base plate, that the heat transfer may be represented by convection in a parallel plate channel. Morega *et al*. (1995) studied the minimization of the thermal resistance between a stack of parallel plates and a free stream. They found that there existed an optimal number of plates that minimized the thermal resistance for a certain dimension of the stack and free stream. The performance of plate fin heat sinks have been studied extensively by Sparrow *et al.* (1981, 1986), Kadle and Sparrow (1986), Lau and Mahajan (1989), Wirtz *et al.* (1994), Iwasaki *et al*. (1994) and Sata *et al.* (1996). Elshafei (2007) experimentally and theoretically evaluated the thermal fluid performance of a plate-fin heat sink under cross flow conditions, by changing fin density, air velocity, and tip-to-shroud clearance. The average heat transfer coefficient increased with increasing Reynolds number and decreased with increasing tip-to-shroud clearance. Increasing the fin density greatly increased the pressure drop and bypasses flow, while the effect of the fin density decreased as the tip-to-shroud clearance increased. Jonsson and Palm (2000) experimentally studied the effected of bypass conditions and fin height on the thermal performance of plate-fin heat sinks and strip-fin heat sinks with staggered and inline arrays. They observed that the fin height significantly affected the thermal resistance, and the heat sink with the highest fins had the lowest pressure drop and thermal resistance. Sata *et al*. (1997) numerically investigated the effects of duct Reynolds number, fin array, fin pitch, width of clearance, and fin length on the fluid flow and heat transfer of a plate-fin array subjected to uniform flow. The heat transfer and friction resistance of the system could be predicted satisfactorily with developing flow between the parallel plates with uniform inlet velocity. El-Sayed et al. (2002) studied the effect of fin width, inter fin space, the distance from the fin tips to the shroud, and fin height, as well as number of fins on performance of a plate-fin heat sink. The results showed that the pressure drop decreased as the inter-fin space and fin width increased but it increased with the duct Reynolds number and fin height. Furthermore, the average duct Nusselt number increased with increasing Reynolds number, fin width and inter-fin space but with decreasing fin height. All these researches focused on optimization of design parameters and the operating condition of a cooling system. However, there exists an intrinsic shortcoming in structures of parallel plate fins which make airflow passing through heat sinks smoother. This is undesirable for enhancing heat transfer performances of heat sinks.

Heat sinks are usually characterized according to their method of operation, i.e., parallel-flow, counter-flow, or cross-flow arrangement. Further considerations include the elements that are used to increase the heat transfer rate in a particular heat sink. Several heat transfer enhancement elements, such as ribs, twisted tapes, wire coils, cross-bar grids and dimples, have been suggested in some researches (e.g., Sahiti *et al.* 2005). However, this kinds of elements are usually not very efficient at heat transfer enhancement since they usually increase the heat transfer coefficient but the heat transfer area basically remains constant. Much higher values of the heat transfer rates can be obtained by applying fins of different geometry, such as strip, corrugated or louver fins, since by these elements one can obtain a larger heat transfer coefficient and larger heat transfer area compared with bare plates and plates with enhancement elements mentioned above. However, these enhancement elements have reached their limit and no further significant improvement in heat exchanger performance seems possible. Hence enhancement of heat transfer continues to be a challenging problem in different industrial fields. Therefore, an attempt to improve the performance of heat exchangers by using pin fins was undertaken in the present study. The main purpose of the present work is to improve the existing design of parallel plate heat sink by adding circular pins between plate fins. Since there are limited numbers of works in this area, there is still a need for new experimental data to validate the theoretical and numerical researches.

## 2. EXPERIMENTAL SETUP

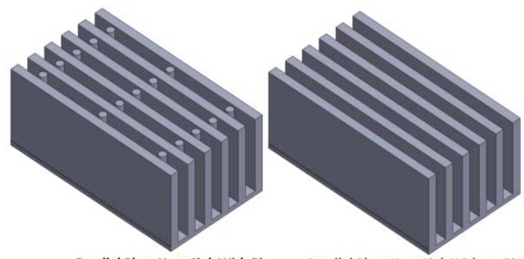



**Fig. 1. schematic diagrams of a parallel plate heat sink with and without pins**

In order to experimentally measure the thermal performance of the heat sink, it is essential that the rate of heat transfer between the heat sink and the flowing air be accurately measured. The problem under consideration is a forced convective flow through a plate-fin heat sink made of aluminum (k=202 W/m-K) with the rectangular base plate as shown in Fig. 1. The construction method was to press fit aluminum plates into a pre-drilled base plate of thickness 0.6 cm. The height of fins is 2 cm with pitch of 0.45 cm. In order to ensure a tight fit between the pin fins and the base plate and to keep thermal losses in the interface to a minimum, the size of rectangular holes incorporating the fins was slightly smaller than the one of the fins. The base plate was heated in an oven to over 600 K while the pin fins were stored in a freezer. The thermal expansion of the base, and of the rectangular hole size, and the thermal contraction of the fins, help to obtain a very tight interference fit between the two parts. For the heat sink with pins between plate fins, the nearest distance between the pin center and fin wall is 2 mm and the center distance of two neighbor pins is set to 20 mm (10 D) in the flow direction. Air is employed as a coolant, passes through the heat sink, thus removing the heat generated by the component attached at the bottom of the heat sink substrate. In analyzing this problem all thermal properties are evaluated at the film temperature of fluid. The experiments have been carried out in a low speed open circuit wind tunnel composed of (i) a contraction cone that accelerate the flow, (ii) inlet section consisting of turbulence control screens and flow straighteners, (iii) a test section with 8×11 cm and length of 100 cm, (iv) a diffuser that reduces the air speed with as little energy as possible, and (v) a variable speed fan to drive the air into test section.

The fan induces the outside air through the test section via the bell-mouthed entrance section, with the motor and fan assembly on the exhaust side of the system which prevents the airstream being heated by motor prior to its passage through the heat exchanger assembly. The duct is insulated with 2-cm thick Aero flex standard sheet in order to minimize the heat losses to the environment. An air velocity transducer of cylindrical shape with accuracy of 0.2% is inserted from the side walls of the test section to obtain the flow rate and the velocity profile. To allow vertical movement of the transducer, a gutter of the same width as the transducer diameter is milled into the side wall of the channel. In order to evaluate the overall pressure difference between the inlet and the outlet of the heat sink, two static pressure tap are located on the wind tunnel wall and is connected to a manometer (FCO510, Furness). The pumping power is calculated by multiplying the measured flow rate and pressure drop. A schematic diagram of the experimental apparatus is shown in Fig. 2.

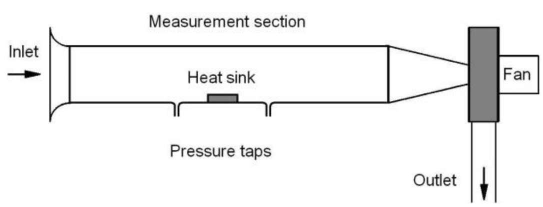



**Fig. 2. Experimental setup**

One of the most important parts of experiment is the heat source that plays an important role in the design of the experimental setup. Three cartridge heaters rated 250 W each were inserted into a high thermal conductivity copper block (about 401 W/K- m) with thickness of 6 mm and the same area as the heat sink so the spreading resistance effects can be neglected. The heat supplied from the power supply is calculated by measuring the voltage drop and the current through the heater. It was found that all heat dissipated by the heaters was nearly uniformly distributed through the heater block. This has been verified by confirming the uniformity of the heater surface temperature during its operation in natural convection environment. To reduce the heat loss, the bottom of the heater is insulated with Teflon. The heat losses are estimated from measuring the temperature across the insulation substrate. It was found that the maximum heat loss from the sides and the bottom of the heater plate was about 4.2% of the total power. To provide the maximum output power of 750 W, the cartridges were wired in parallel. The lower horizontal part and sides of the main heater copper block, such as sides of the heat sink were insulated with a 10 cm layer of fiber glass blanket, sandwiched with mica sheet. The entire assembly was firmly bolted together. The temperature at the base of the fins is measured by mounting seven 30 gauge J-type thermocouple (omega) with accuracy of 0.1% through a 3 mm deep holes of a base plate. An Agilent 34970A DAQ is used to transport data measured by the thermocouples. The inlet and outlet temperatures of the air are measured by three and five thermocouples, respectively, with 1 mm diameter probes extending into the duct in which the air flows. The air temperature distributions while flowing through the heat sink are measured by three and five thermocouples, respectively, with 1 mm diameter probes extending into the duct in which the air flows. The air temperature distributions while flowing through the heat sink are measured in 20 positions. All the thermocouple probes are pre-calibrated by a dry box temperature calibrator with 0.02 ˚C precision.

Experiments were conducted with various heat fluxes and flow rates of air entering the test section. The flow rate is set by regulating the metering value. The heater is then powered up to a heat load of 30W and allowed to stabilize. It is worth to mention that the voltage and current of the electric input to the cartridge heaters were controlled by an AC power supply unit. The supplied power was calculated using the measured voltage and current supplied to the heaters. The supplied voltage and current into the heaters were measured by the digital clamp meter (3PK-2002A). The fan was set to reach the maximum velocity achievable. At steady state conditions, pressure drop and temperature were recorded. For the same input power, four different velocities were tested. The temperature distribution at the base of the heat sink is measured until the change in the temperature is smaller than ±0.1 ˚C in a 2 min period. Each experiment was conducted ten times. To discern the accuracy and reliability of experimental measurement, an experimental uncertainty analysis was performed.

Uncertainties in temperature differences were within ±0.4 ˚C, volumetric air flow rates within ±4% of full scale readings and power measurements within ±1.2% of readings.

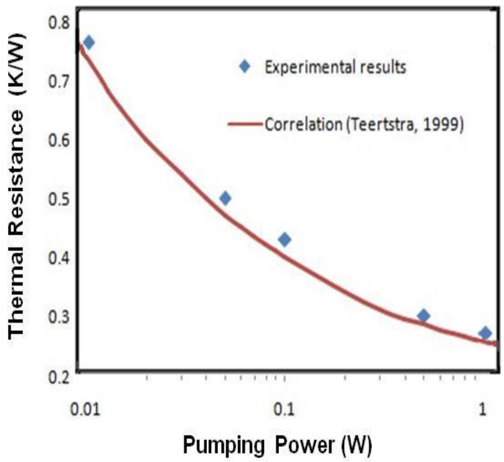
## 3. RESULTS AND DISCUSSIONS

The thermal resistance of the heat sink is defined by:

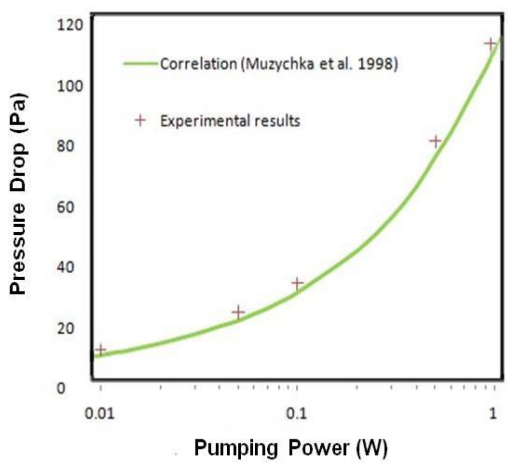
$$R = \frac{\Delta T}{Q} \tag{1}$$

where Q is the heat dissipation power applied on the fin base and $\Delta T$ is the temperature difference between the highest temperature on the fin base and the ambient air temperature. In order to check the validity of the present study, the experimental results of the thermal resistance and pressure drop of the parallel plate heat sink without pin were compared to those obtained from existing correlations. As shown in Fig. 3, the experimental results of the thermal resistance are in good agreement with the correlation suggested by Teertstra *et al*. (1999). Furthermore, the pressure drop through the heat sink is in good agreement with correlation proposed by Muzychka and Yovanovich (1998) with less than 5% error.

The test results of heat sink thermal resistance and pressure drop vs. inlet air velocity for both parallel plate heat sinks with and without pins between the fins are plotted in Figs. 4 and 5, respectively. It can be seen that the pressure drop of heat sink with pin is much higher than that of the heat sink without pin between the fins but the thermal resistance of heat sink with pin is about 37.7% lower. The reason for the lower thermal resistance of heat sink with circular pins is due to higher surface area and higher streamlined flow patterns in assisting flow mixed convection between fins. The existence of pins between the fins cause the deflection of flow and consequently vortex structure behind pins and better mixing and heat transfer performance.




**(a)**




**(b)**

**Fig. 3. Comparison of present experimental data with available correlation in the literature, (a) thermal resistance; (b) pressure drop through the heat sink vs. pumping power**

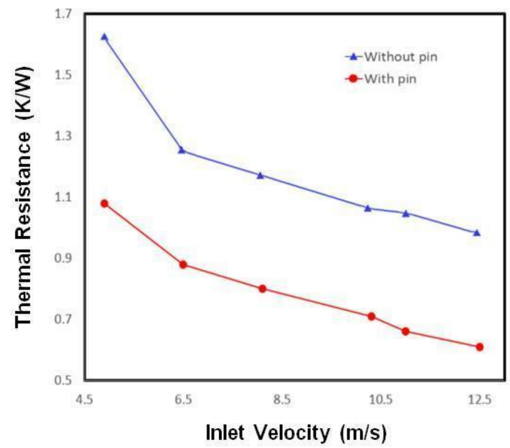



**Fig. 4. Effect of inlet velocity on thermal resistance of the heat sinks**

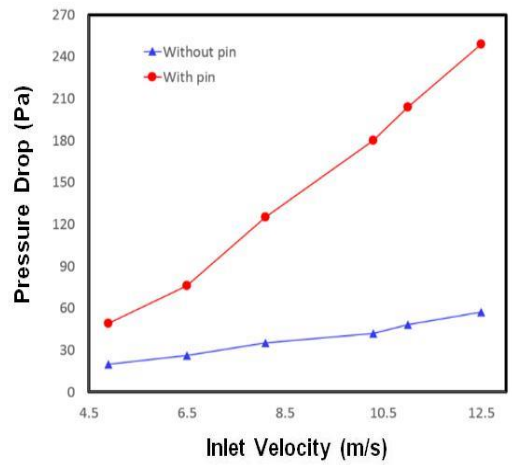



**Fig. 5. Effect of inlet velocity on pressure drop through the heat sinks**

For overall performance assessment of investigated heat sinks the overall heat transfer efficiency is represented by the ratio of power to the pumping power. This definition relates the hydrodynamic and thermal performance to obtain an indication about the overall heat sink performance.

$$\gamma = {}^{Q}\!/_{Q_p} \tag{2}$$

where $Q_p$ is the pumping power required to propel a mass flow ($\dot{m}$) through a pressure drop ($\Delta$p).

$$Q_p = \left({}^{\dot{m}}\!/_{\rho}\right)\Delta p = wA_p\Delta p \tag{3}$$

$$\mathrm{A_p = H\,.\,a.\,(N-1)} \tag{4}$$

where N and H are the number and height of the plate fins respectively.

Figure 6 illustrates the effect of pumping power on the overall performance of both heat sinks. It can be seen that the performance decreases as the pumping power increases. Moreover, the performance of heat sink with pin between plate fins is 24.4% higher than that of the heat sink without pin with the same pumping power. Therefore, one can conclude that the heat sink with pin needs less pumping power than the heat sink without pin when heat dissipation power is the same for both of them. This is an interesting result because it shows that adopting heat sink with pin can make the volume of air-cooling system smaller with very small increase in the cost of manufacturing. The process of manufacturing of heat sink with pin is not complex. It can be constructed by drilling holes in flow passages of a parallel plate heat sink and inserting columnar pins into these holes.

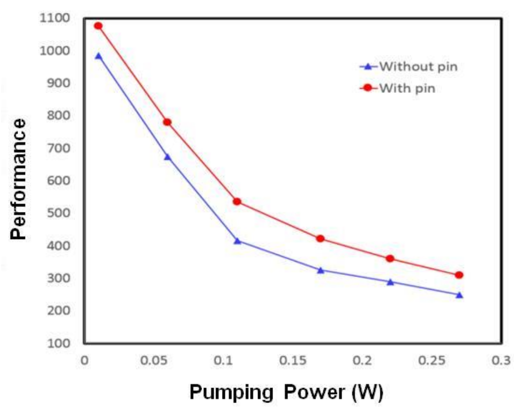



**Fig. 6. Effect of pumping power on overall performance of the heat sinks**

## 4. CONCLUSION

An experimental evaluation of a parallel plate heat sink with and without circular pins between fins is carried out. The thermal resistance and pressure drop, as well as overall performance of the heat sinks, are determined for a range of inlet velocities, i.e., 4.7-12.5 m/s. The thermal resistance of the system is the main design criteria for a real application, while the pressure drop is another important parameter which is related to pumping power. Therefore, heat sink should be designed by considering not only heat dissipation conditions, but also a reasonable trade-off between the heat rejection performance and pressure drop. In this study, the heat sink with circular pins between fins is shown to provide favorable cooling performance. However, the pressure drop in the heat sink with pins is higher than that in without pins between the fins. The overall performance of the system which is the ratio of dissipated heat to pumping power shows the heat sink with circular pins has higher overall performance. Furthermore, with increasing the inlet velocity, the pressure drop increases but the overall performance and thermal resistance decrease. Hence, in order to obtain higher thermal and overall performance the heat sink should operate at lower operating inlet velocities.

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