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NATURAL CONVECTION HEAT TRANSFER FROM A HEAT SINK WITH FINS OF DIFFERENT CONFIGURATION

Murtadha Ahmed and Abdul Jabbar N. Khalifa

Mechanical engineering department, Alnahrain University, P.O. Box 64040, Baghdad, Iraq

ABSTRACT: Experiments were carried out on natural convection heat transfer from square pin fin heat sinks subject to the influence of its geometry and heat flux. A total of 50 fins were bolted into the upper surface of the base plate. The area of the base plate is 250mm by 250mm. The base plate and the fins were made of aluminum .Over the tested range of Rayleigh number, $12.45\times10^6 \leq Ra \leq 58.59\times10^6$, it was found that the solid pin fin heat sink performance for upward orientations depended on the Rayleigh number and generally shows higher heat transfer coefficients than those of the perforated/hollow pin fin ones. For all tested hollow/perforated pin fin heat sinks, however, the heat transfer performance for heat sinks with hollow/perforated pin fins was better than that of solid pins. The temperature difference between the base plate and surrounding air of these heat sinks was less than that of solid pin one.

KEYWORDS: Natural convection; Square solid/perforated pin fin; Heat sinks

1 INTRODUCTION

Natural convection heat transfer from a heat sink has been the subject of a large number of experimental and theoretical investigations. Heat sinks with extended surfaces have been widely used in various engineering applications especially in air conditioning as well as in electronic devices, electrical and internal combustion, solar energy applications, cooling of nuclear reactor fuel elements, improving heat transfer in radiators for air conditioning and in air cooled heat exchangers. In most of the electronics cooling applications, the final heat transfer medium is air.

Heat transfer from a finned heat sink depends on many parameters; temperature difference between the heat sink and the fluid, shape and size of the fins, number of fins, the type of flow and the fluid, thermal conductivity and spacing between the fins.

Natural convection heat transfer from solid and perforated pin fin heat sinks can be found from experimental and theoretical works. An experimental study was conducted on natural convection heat transfer from square pin fin and plate fin heat sinks subject to the influence of orientation by Huang et al. [1]. Over a tested range of $1.8\times10^6 < Ra < 4.8\times10^6$, it was reported that the downward facing orientation yielded the lowest heat transfer coefficient for both pin fin and plate fin geometry. The finning factor Φ (which represents the total surface area, A_t divided by the base plate area, A_{bp}) for the perforated pin fin was greater than from the solid pin fin .The heat transfer coefficient increases with decreasing the finning factor. In addition, the advantage of the pin fins escalates with the increase of the Rayleigh number due to the more open ends for air ventilation. Elshafei [2] studied the natural convection heat transfer from circular pin fin heat sinks subject to the influence of its geometry, heat flux and orientation. Their results showed that the solid pin fin heat sink performance for upward and sideward orientations was a competitive nature, depending on Rayleigh number ($3.8 \times 10^6 \leq Ra \leq 1.65 \times 10^7$) and generally shows higher heat transfer coefficients than those of the perforated/hollow pin fin. The heat transfer performance for heat sinks with hollow/perforated pin fins was better than that of solid pins. The temperature difference between the base plate and surrounding air of these heat sinks was less than that of solid pin. Zografos and Sunderland [3] investigated the heat transfer performance of inline and staggered pin fin arrays in natural convection with different inclination angle, and concluded that the inline arrays generally yielded higher heat transfer rates than the staggered ones. In addition, their investigation showed little influence of inclination when the inclination angle was less than 30^o from the vertical. Sparrow and Vemuri [4] studied the fin orientation on natural convection/radiation heat transfer from pin fin arrays. Their results revealed that the upward facing orientation yielded the highest heat transfer rates, followed by the sideward facing and the downward facing ones. Dialameh, et al. [5] made a numerical study to predict natural convection from an array of aluminum horizontal rectangular thick fins of (3 mm< t < 7 mm) with short lengths (L< 50 mm) attached on a horizontal base plate. The results showed that natural convection heat transfer coefficient increases with increasing temperature differences and fin spacing and decreases with fin length, and the fin thickness and fin height does not affect the value of average heat transfer coefficient considerably. Kobus and Oshio [6] carried out a theoretical and experimental study on the performance of pin fin heat sinks. A theoretical model was proposed to predict the influence of various geometrical, thermal and flow parameters on the effective thermal resistance of heat sinks. Subsequently, Kobus and Oshio [7] investigated the effect of thermal radiation on the heat transfer of pin fin heat sinks and presented an overall heat transfer coefficient that was the sum of an effective radiation and a convective heat transfer coefficient. Fisher and Torrance [8] presented the analytical solutions relevant to the limits of free convection for pin fin cooling. They suggested that the design of pin fin heat sink could be optimized by properly choosing the pin fin diameter and the heat sink porosity. For conventional heat sinks, the minimum thermal resistance was about two times greater than that in an ideal limit according to the model of in viscid flow with idealized local heat transfer.

The above literature indicates that many studies have been carried out for different types of fin arrays, but still there is lack of knowledge of the natural convection heat transfer from a surface with hollow /perforated pin fins. The main objectives of the present experimental work are to determine the effects of geometric parameters and base to ambient temperature difference on the natural convection heat transfer from square solid and perforated pin fins heat sink. Another objective is to correlate the experimental data of overall Nusselt number with the average modified Rayleigh number for such configurations.

2 EXPERIMENTAL SETUP AND PROCEDURE

2.1 PIN FIN ASSEMBLY

A total of 50 fins are used in this experimental investigation. Twenty fins have two holes, one of the holes is horizontal and the other is vertical. The fins are bolted into the upper surface of the base plate, the dimensions of the base plate is 250mm × 250mm. Thermal paste was used to minimize the thermal contact resistance between the pin fins and the base plate; the pin fins were fixed at equal spacing between the span wise directions of 18.125 mm. The base plate was made of the same material as the fins because of the considerations of conductivity machinability, and cost. The base plate and the fins were made of aluminum because high thermal conductivity, low radiative emissivity and good contact between them were assured. The experimental rig is shown in Fig. 1.

2.2 HEATING SYSTEM

During experiments, the heat sink base plate was heated by an attached electric heater, with an identical size as the base plate, which could supply a specific heat flux. The voltage and current of the electric input to the 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 a digital clamp meter (UT200A). The presence of thin layer of high thermal conductivity mica ensured that good thermal contact existed between the heater and the heat sink base. The bottom surfaces of the base plate as well as the heater block were insulated by two layers of insulation, firebrick and glass wool blanket. The whole assembly, base, heater with associated thermal insulation, was located in a well-fitted open-topped wooden box of 20mm thick as shown in Fig. 1. The upper edges of the wooden box and the top surface of the laterally-placed thermal insulation were flushed with the upper surface of the heat sink base in which the pin fins protruded perpendicularly. The power supplied to the heater was controlled by a variac transformer to obtain constant heat flux along the base plate, and was measured by in line multi-meter (an ammeter and voltmeter).

A total of 12 K-type thermocouples were appropriately distributed among the base plate to measure its average temperature, T_{bp}.These thermocouples were pressed into fine holes and glued in position with epoxy so as to ensure good thermal contact. All thermocouples were pre-calibrated with an accuracy of ±0.1 ºC. The temperatures at various locations on the surface of the base plate were measured by calibrated K- type thermocouples connected to a 12-channel temperature recorder (Model BTM-4208SD). The whole assembly was located in an environmental chamber inside the laboratory. Heat sinks of solid and perforated pin fins as shown in Fig. 2 were tested in the enclosed room at nearly fixed temperature with the power inputs ranging from 150 W to 1400 W. Steady state conditions were indicated by noticing the repeatable readings of each thermocouple at different locations of the heat sink. Each test run took nearly 2-hours to reach equilibrium when the power is turned on.

(a) Pif fins, (b) Base plate, (c)Heater, (d)Firebrick, (e)Glass wall blanket, (f)Wooden box

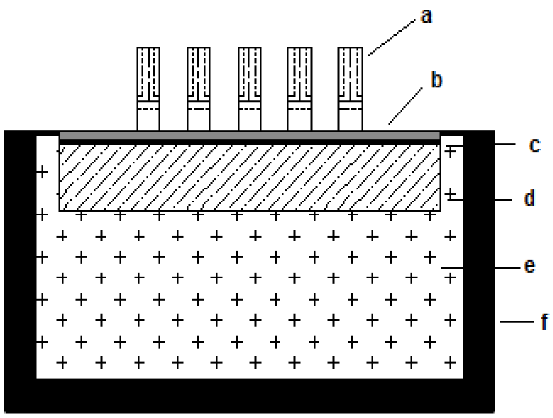

Fig. 1. Heat sink assembly

3 RESULTS AND DISCUSSION

For the sake of comparison, the thermal performance of heat sink is measured by detecting the variation of the temperature difference between the base plate and the surrounding air (ΔT) with the heat input rate (Q). Figure 2 indicates the variation of ΔT with Q for two tested modules (solid and perforated heat sink) arrangements. Generally, the results indicate that ΔT increases with increasing heat input rate (Q). In addition, the temperature difference (ΔT) for the perforated heat sink is less than for the solid heat sink for each heat input.

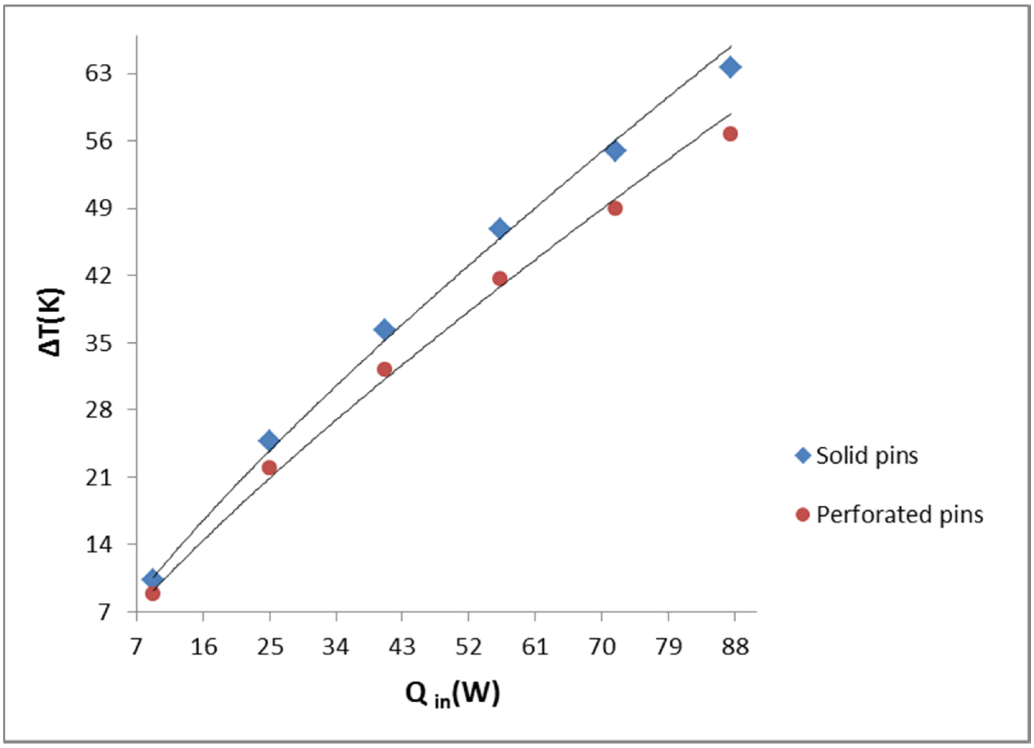

Fig. 2. Temperature difference variations ΔT with heat input rate, Q_{in} for solid and perforated heat sink

The variation between the temperature difference (ΔT) and the heat transfer coefficient (h) is shown in Fig. 3. Generally, the results indicate that the heat transfer coefficients for two test samples increase with the temperature difference between the base plate and the surrounding, this increase in both temperature difference and heat transfer coefficient is due to the increase in the heat input gradually. A comparison between the solid and perforated heat sink shows that the heat transfer coefficient for solid pin fin is greater than from perforated pin fin.

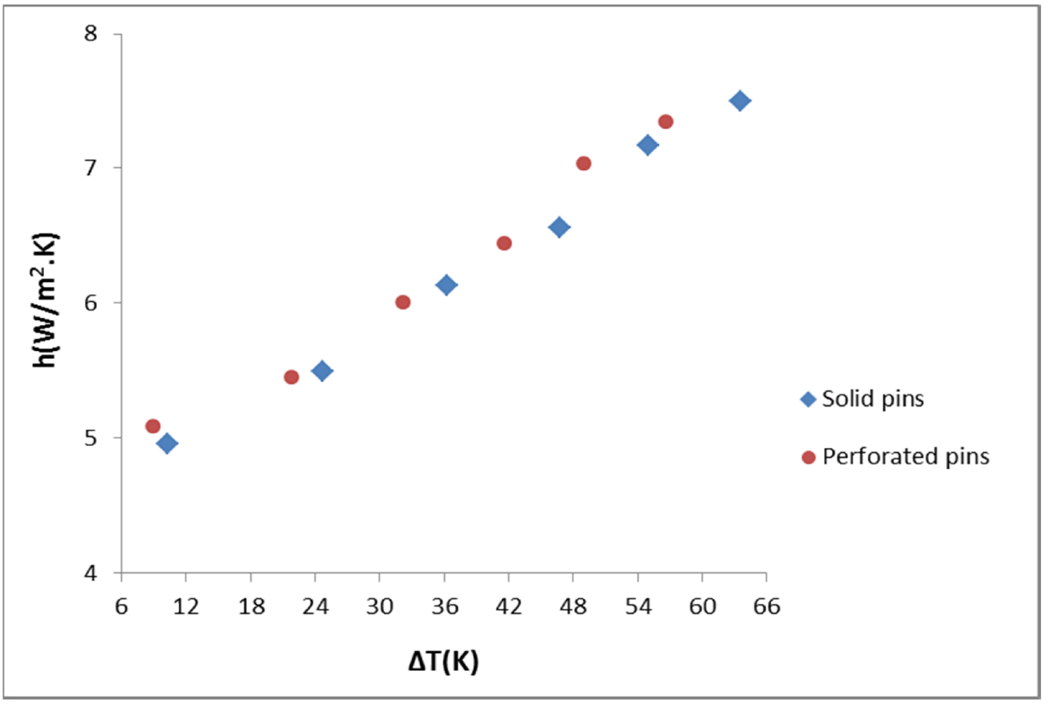

Fig. 3. Temperature difference variations ΔT with average heat transfer coefficient h for solid and perforated heat sink

4 CONCLUSIONS

In this study, natural convection heat transfer from square solid and perforated pin fins heat sink at different heat flux values was investigated experimentally. The effect of geometric parameters and base to ambient temperature difference on the heat transfer performance of fin arrays was discussed. Based on the preceding discussions, the following conclusions may be drawn:

- At the same heat input rate, the temperature difference between the base plate and surrounding air was found to be less for hollow/perforated pin fin heat sink than that for solid pin one.
- The heat transfer coefficient for solid pin fin is greater than from perforated pin fin.
- Temperature difference between the fin base and fin tip becomes larger, by adding perforation
- Perforated fins have higher fin effectiveness than solid fin and it rises remarkably by adding more perforations.

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