International Journal of Heat and Mass Transfer 52 (2009) 2949–2955

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Measurement of performance of plate-fin heat sinks with cross flow cooling

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ARTICLE INFO

Article history:
Received 1 July 2008
Received in revised form 19 February 2009
Accepted 19 February 2009
Available online 28 March 2009

Keywords:
Cross flow
Plate-fin heat sink
Thermal resistance
Infrared thermography

ABSTRACT

This work assesses the performance of plate-fin heat sinks in a cross flow. The effects of the Reynolds number of the cooling air, the fin height and the fin width on the thermal resistance and the pressure drop of heat sinks are considered. Experimental results indicate that increasing the Reynolds number can reduce the thermal resistance of the heat sink. However, the reduction of the thermal resistance tends to become smaller as the Reynolds number increases. Additionally, enhancement of heat transfer by the heat sink is limited when the Reynolds number reaches a particular value. Therefore, a preferred Reynolds number can be chosen to reduce the pumping power. For a given fin width, the thermal performance of the heat sink with the highest fins exceeds that of the others, because the former has the largest heat transfer area. For a given fin height, the optimal fin width in terms of thermal performance increases with Reynolds number. As the fins become wider, the flow passages in the heat sink become constricted. As the fins become narrower, the heat transfer area of the heat sink declines. Both conditions reduce the heat transfer of the heat sink. Furthermore, different fin widths are required at different Reynolds numbers to minimize the thermal resistance.

1. Introduction

Given the increasing miniaturization of electronic products, the transfer of generated heat to the outside becomes more difficult. The efficiency of electronic products suffers from the lack of an adequate heat dissipation mechanism, possibly causing damage as the temperature rises. Therefore, effective thermal management of electronic products is of priority concern. Air cooling is a popular electronic cooling approach. It is adopted widely because of the ease of obtaining the cooling fluid and the simplicity, high reliability and low cost of the required equipment. The heat generated by the electronic element is typically transferred to a heat sink by heat conduction, and then to the air by natural convection or forced convection. Therefore, the air cooling system must be designed properly to promote heat transfer and reduce the temperature of the electronic products.

Morega et al. [1] investigated the minimization of the thermal resistance between a stack of parallel plates and a free stream. The best heat transfer was obtained when the plates were uniformly spaced; there existed an optimal number of plates that minimized the thermal resistance for a specified free stream and overall dimensions of the stack. Sata et al. [2] studied the effects of fin pitch, fin length, number of fins, width of clearance beside the fin array and Reynolds number on the flow and temperature fields of a plate-fin array subjected to uniform flow by numerical simulation. The friction resistance and the heat transfer of the fin array could be predicted satisfactorily with developing flow between the parallel plates having uniform inlet velocity. Jonsson and Palm [3] experimentally examined, using a wind tunnel, the effects of fin height and bypass conditions on the performance of plate-fin heat sinks and strip-fin heat sinks with in-line and staggered arrays. The thermal resistance increased with the height and width of the wind tunnel duct. The fin height significantly affects the thermal resistance, and the heat sink with the highest fins had the lowest thermal resistance and pressure drop. Jonsson and Moshfegh [4] considered the bypass effect of seven types of heat sinks – plate-fin, in-line strip-fin, staggered strip-fin, in-line circular pin-fin, staggered circular pin-fin, in-line square pin-fin and staggered square pin-fin. They examined the effects of duct height, duct width, fin height, fin width, fin-to-fin distance and Reynolds number on the performance of the heat sinks. The Reynolds number and the relative duct height dominated the predicted Nusselt number. El-Sayed et al. [5] varied the fin height, fin width, inter-fin space, number of fins and the distance from the fin tips to the shroud to study the performance of a plate-fin heat sink. They concluded that the pressure drop increased with the Reynolds number and fin height but decreased as the inter-fin space and fin width increased; that increasing the clearance between the fin tips and the shroud reduced the mean Nusselt number, and that the mean Nusselt number increased with increasing Reynolds number, fin width and inter-fin space but with decreasing fin height. Saini and Webb [6] studied the optimization of the plate-fin heat sink with duct

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doi:10.1016/j.ijheatmasstransfer.2009.02.025

Nomenclature

A	cross-sectional area of the heating element (mm^2)
A_t	heat transfer area (mm^2)
b	thickness of the base of the heat sink (mm)
D	hydraulic diameter (mm)
d	distance between the two thermocouples in the heating element (mm)
G	inter-fin spacing (mm)
H	fin height (mm)
k_{al}	thermal conductivity of aluminum alloy (W/mK)
L_x	length of base of the heat sink (mm)
L_y	width of base of the heat sink (mm)
n	fin number
ΔP	pressure drop (Pa)
Q	heating power (W)
Re	Reynolds number
R_{th}	thermal resistance (°C/W)
T_{ave}	average temperature of the base of the heat sink (°C)
T_l	temperature of the lower thermocouple in the heating element (°C)
T_u	temperature of the upper thermocouple in the heating element (°C)
T_∞	temperature of the cooling air (°C)
U	overall heat transfer coefficient (W/°C mm^2)
V	inlet velocity of the cooling air (m/s)
V_t	volume of the heat sink (mm^3)
W	fin width (mm)

Greek symbol

ν	kinematic viscosity (m^2/s)

flow and impinging flow. According to that study, increasing the fan speed and base area only slightly affected the optimal geometry; increasing the fin height reduced the convective thermal resistance of the heat sink, and the plate-fin heat sink outperformed the offset-strip-fin heat sink in duct flow. El-Sayed et al. [7] investigated the heat transfer and fluid flow of a plate-fin heat sink for parallel flow, impinging flow and reverse impinging flow; a parallel flow yielded the greatest rate of heat transfer and the lowest pressure drop. Kim and Webb [8] showed that for a particular airflow frontal area, flow depth and fan curve, the optimized offset strip-fin heat sink provided a slightly lower convective thermal resistance than the optimized plate-fin heat sink. However, the convective thermal resistance increased drastically as the length of the offset stripe increased. Additionally, the in-line circular pin-fin and staggered circular pin-fin heat sinks had markedly higher convective thermal resistance. Elshafei [9] evaluated the thermal fluid performance of a plate-fin heat sink under cross flow conditions, both experimentally and theoretically, by varying air velocity, fin density and tip-to-shroud clearance. Increasing the fin density greatly increased the bypass flow and the pressure drop, while the effect of the fin density decreased as the tip-to-shroud clearance increased. The average heat transfer coefficient increased with increasing Reynolds number and decreased with increasing tip-to-shroud clearance.

A review of pertinent literature shows that the performance of the heat sink in a cross flow is influenced markedly by the flow velocity and the geometries of the heat sink. The heat sink is oriented such that the inlet flow direction of the cooling air is parallel to the heat sink base. In this cross flow cooling configuration, the cooling air enters and exits from the sides of the heat sink. The infrared thermography technique uses the radiant exitance in the infrared spectral band from measured objects to measure temperature. It is non-intrusive, applicable remotely and suitable for measuring a large area. It has been adopted successfully in various applications [10,11]. This work investigates the effects of the Reynolds number, fin width and fin height on the performance of the plate-fin heat sinks using infrared thermography.

2. Experimental apparatus and method

The experimental apparatus, schematically depicted in Fig. 1, comprises an infrared thermography system, a wind tunnel system, heat sinks, a heating system and devices for measuring flow velocity, temperature and pressure drop. The infrared thermography system, which includes a ThermaCAM SC500 camera from FLIR systems and a PC with AGEMA Researcher software, can measure temperatures from −20 °C to 1500 °C with an accuracy of ±2%. The infrared camera uses an uncooled focal plane array detector with 320 × 240 pixels which operates over a wavelength range of 7.5–13 μm. The field of view is 24° × 18°; the instantaneous field of view is 1.3 mrad, and the thermal sensitivity is 0.1 °C at 30 °C.

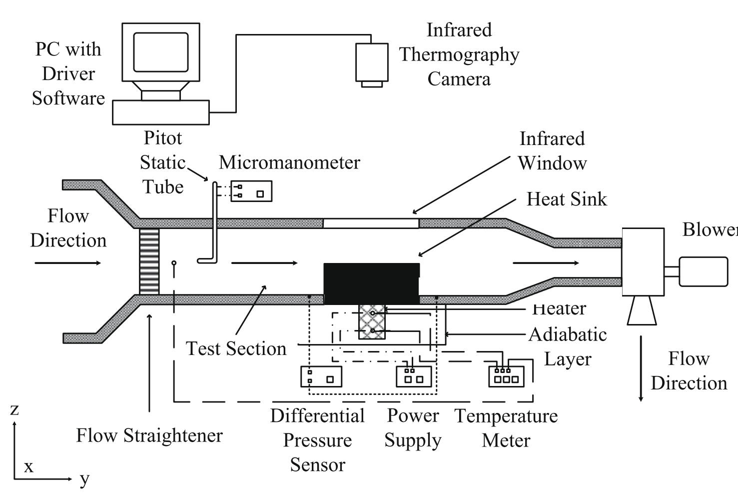

Fig. 1. Schematic diagram of experimental apparatus.

The images captured by the infrared camera are displayed and recorded using a computer for further analysis.

The wind tunnel system has a flow straightener, a test section and a blower. The top wall of the test section has a cut-out region where a zinc-selenide window of transmissivity 0.98 is installed to enable the infrared camera to measure the temperature distribution of the heat sink in the test section. The blower, which has a maximum flow rate of 14 m^3/min, sucks the required cooling air though the test section via a centrifugal fan regulated with an inverter. The air flows steadily across the heated heat sink which removes heat continuously. The effect of the air velocity on the performance of the heat sink is studied.

The plate-fin heat sinks are made of aluminum alloy 6061 which has high thermal conductivity and is low cost, and easily processed. The models are produced by CNC milling or electrical discharge machining. The surfaces of the heat sinks are coated with flat black paint with a radiation emissivity of 0.96 to increase the accuracy of temperature measurement. The length (L_x), the width (L_y) and the thickness (b) of the base of the heat sinks are 70, 77 and 8 mm, respectively. The height and width of the fins are varied as experimental parameters. This study involves 25 heat sink models with five fin widths and five fin heights, presented in Table 1 and Fig. 2. The heat sinks are mounted at the bottom of the wind tunnel such that the heat sink base is flush with the wind tunnel wall.

The heating system has a DC power supply and a heating element that is covered with insulated material, except on the top, where it is in contact with the heat sink. The DC power supply provides a heating power of

$$Q = \frac{k_{al}A(T_l - T_u)}{d} \tag{1}$$

where k_{al} is the thermal conductivity of aluminum alloy 6061. T_u and T_l are the temperatures of the upper and lower T-type thermocouples, respectively, in the heating element. A denotes the cross-sectional area of the heating element, and d is the distance between the upper and lower thermocouples.

The velocity of the flow that enters the test section of the wind tunnel is measured using a Furness Controls FCO510 micromanometer, which has an accuracy of 0.25%, and a Pitot static tube. A T-type thermocouple is inserted to measure the temperature of the cooling air. A Druck LPM5480 low differential pressure sensor is adopted to measure the pressure drop as the cooling air flows across the heat sink where the two pressure taps are located – 112 mm upstream and 112 mm downstream of the heat sink, respectively. The operating differential pressure range of the sensor is 0–1 mbar and its accuracy is 0.25%. A high pressure drop corresponds to a high power consumption of the air blower. The measured data are obtained using a Yokogawa MV220 series data acquisition system for further analysis.

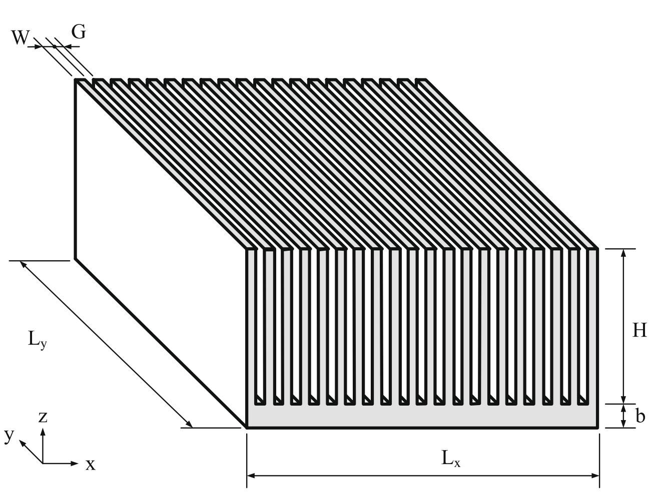

Fig. 2. Diagram of plate-fin heat sink.

In evaluating the thermal performance of the heat sink, the thermal resistance is defined as

$$R_{th} = \frac{T_{ave} - T_\infty}{Q} = \frac{1}{UA_t} \tag{2}$$

where T_{ave} is the average temperature of the base of the heat sink; T_∞ is the temperature of the cooling air, and U is the overall heat transfer coefficient. The heat transfer area A_t is given by

$$A_t = L_xL_y + 2nH(W + L_y) \tag{3}$$

where L_x and L_y are the length and width of base of the heat sink, respectively; n is the fin number; W is the fin width, and H is the fin height. The thermal resistance is the reciprocal of the product of the overall heat transfer coefficient and the heat transfer area. Accordingly, a larger heat transfer area or a higher overall heat transfer coefficient yields a lower thermal resistance. If the thermal resistance is small, then the resistance to the heat transfer is low, and the thermal performance is favorable. The Reynolds number is given by

$$\mathrm{Re} = \frac{VD}{\nu} \tag{4}$$

where V is the inlet velocity of the cooling air; D is the hydraulic diameter of the test section, and ν is the kinematic viscosity of air.

The relative uncertainty in the thermal resistance is given by [12]

$$\frac{\delta R_{th}}{R_{th}} = \left\{ \left[\frac{\delta(T_{ave} - T_\infty)}{T_{ave} - T_\infty}\right]^2 + \left(\frac{\delta Q}{Q}\right)^2 \right\}^{1/2} \tag{5}$$

The relative uncertainties in the heating power Q, the Reynolds number Re, and the pressure drop ΔP can be obtained similarly. The maximum relative uncertainties in the thermal resistance, the heating power, the Reynolds number and the pressure drop in the experiments are estimated to be 12.32%, 3.43%, 0.13%, and 0.25%, respectively.

3. Results and discussion

This study investigates the effects of the Reynolds number (Re) of the cooling air, the fin height (H) and the fin width (W) on the performance of plate-fin heat sinks under cross flow conditions. Six Reynolds numbers (Re = 10,000, 20,000, 30,000, 40,000, 50,000 and 60,000) and 25 plate-fin heat sinks with various fin widths (W = 0.5, 1.0, 1.5, 2.0 and 2.5 mm) and fin heights (H = 15, 30, 45, 60 and 75 mm), as presented in Table 1 are used in the experiments. The surface temperature of the base of the heat sink, measured using the infrared thermography system, is analyzed to examine the effects of the fin dimensions and the Reynolds number on the thermal resistance.

3.1. Surface temperature distribution of heat sink

Fig. 3 presents the surface temperature distribution of the heat sink measured by infrared thermography. The temperature decreases from the bottom to the top and from the middle to the exterior. It is lower in the front than in the rear of the heat sink. It is because heat generated by the heating element, which is in the center of the heat sink, is transferred to the heat sink by conduction and then to the air by convection.

Table 1
Specifications of plate-fin heat sinks.

No.	H (mm)	W (mm)	G (mm)	V_t (mm^3)	A_t (mm^2)
1	15	0.5	3.16	51,890	54,670
2		1.0	2.63	52,190	66,220
3		1.5	2.11	52,490	77,770
4		2.0	1.58	52,790	89,320
5		2.5	1.05	53,090	100,870
6	30	0.5	3.16	98,390	66,220
7		1.0	2.63	98,990	89,320
8		1.5	2.11	99,590	112,420
9		2.0	1.58	100,190	135,520
10		2.5	1.05	100,790	158,620
11	45	0.5	3.16	144,890	77,770
12		1.0	2.63	145,790	112,420
13		1.5	2.11	146,690	147,070
14		2.0	1.58	147,590	181,720
15		2.5	1.05	148,490	216,370
16	60	0.5	3.16	191,390	89,320
17		1.0	2.63	192,590	135,520
18		1.5	2.11	193,790	181,720
19		2.0	1.58	194,990	227,920
20		2.5	1.05	196,190	274,120
21	75	0.5	3.16	237,890	100,870
22		1.0	2.63	239,390	158,620
23		1.5	2.11	240,890	216,370
24		2.0	1.58	242,390	274,120
25		2.5	1.05	243,890	331,870

b = 8 mm, n = 20.

3.2. Effects of fin dimensions and Reynolds number on thermal resistance

The total size of a heat sink is determined by the available volume in the system; the interior dimensions are adjusted to improve heat transfer. Therefore, this work demonstrates the effects of fin dimensions on the performance of heat sinks for various Reynolds numbers.

Fig. 4 plots the thermal resistances of the plate-fin heat sinks under a cross flow of Re = 10,000. The thermal resistance increases with the fin width for all fin heights. Therefore, when the Reynolds number is low, the inter-fin channels become narrow as the fin width increases, inhibiting the cooling air from flowing through to remove the heat, and thus worsening the thermal performance. The thermal resistance clearly declines as the fin height increases from 15 to 75 mm for all five fin widths, indicating that higher fins have larger heat transfer areas and thus greater heat transfer for the same fin width. In summary, the thermal performance is improved as the fin height increases or the fin width decreases for cooling flow with a low Reynolds number.

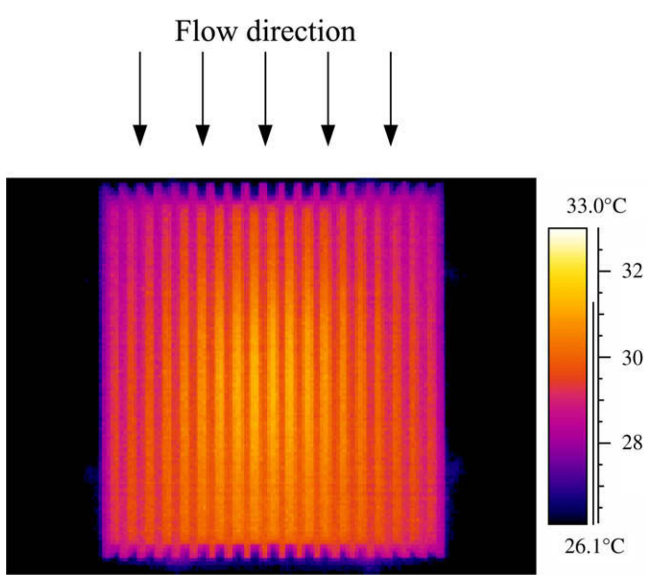

Fig. 3. Surface temperature distribution of heat sink for W = 1.5 mm, H = 45 mm, Re = 30000 and Q = 27.19 W.

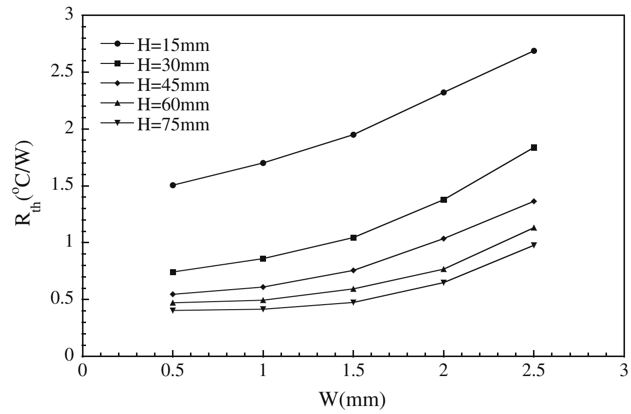

Fig. 4. Effect of fin dimensions on thermal resistance of heat sink for Re = 10,000.

Fig. 5 shows the thermal resistance for an increased Reynolds number of 20,000. The variation of the thermal resistance due to the fin width effect is small when $W \leqq 1.5$ mm. It remains almost constant as the fin width increases from 0.5 to 1.5 mm – especially for $H \geqq 45$ mm – indicating that the cooling air can enter the inter-fin channels to exchange heat when the Reynolds number is slightly higher. Hence, the thermal performance does not clearly vary with the fin width up to a certain value. However, when the fins are too wide ($W \geqq 2.0$ mm), the cooling air is impeded and the thermal resistance is considerably increased. The thermal resistance declines as the fin height increases. Additionally, when W = 0.5 mm, the thermal performance is not substantially improved as the fin height increases above 45 mm. Therefore, when the fin width is minimal the main heat transfer area is close to the low part of the fins. The increased area associated with excessive fin height increase does not improve the heat transfer.

When the Reynolds number is increased to 30,000, the thermal resistance is as plotted in Fig. 6. Notably, when $W \leqq 1.5$ mm, the variation in the thermal resistance differs from that at the lower Reynolds number given above. The thermal resistance may drop

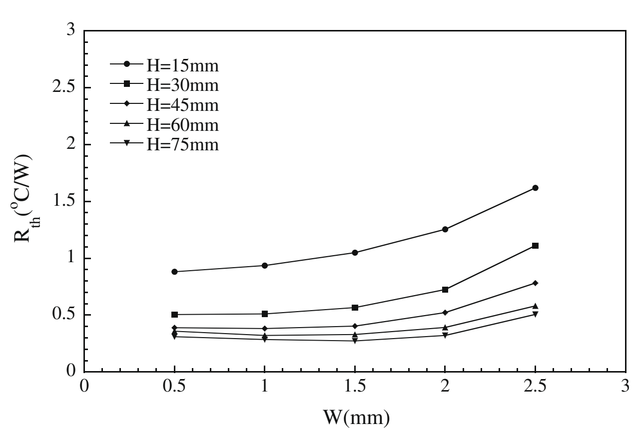

Fig. 5. Effect of fin dimensions on thermal resistance of heat sink for Re = 20,000.

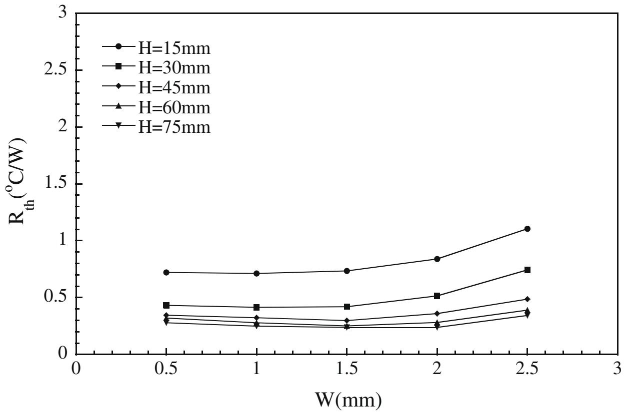

Fig. 6. Effect of fin dimensions on thermal resistance of heat sink for Re = 30,000.

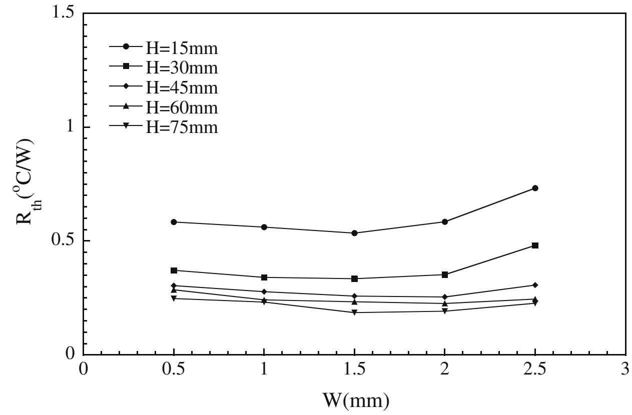

Fig. 8. Effect of fin dimensions on thermal resistance of heat sink for Re = 50,000.

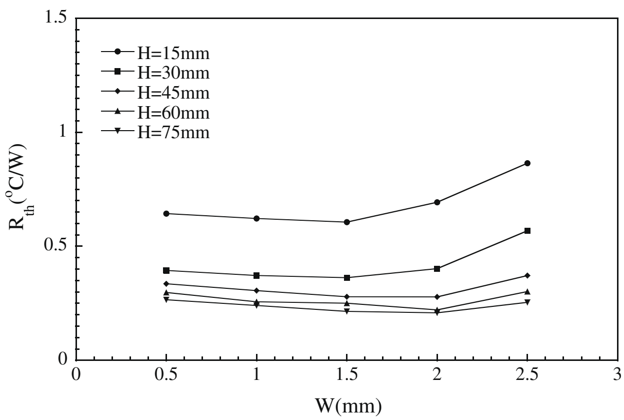

Fig. 7. Effect of fin dimensions on thermal resistance of heat sink for Re = 40,000.

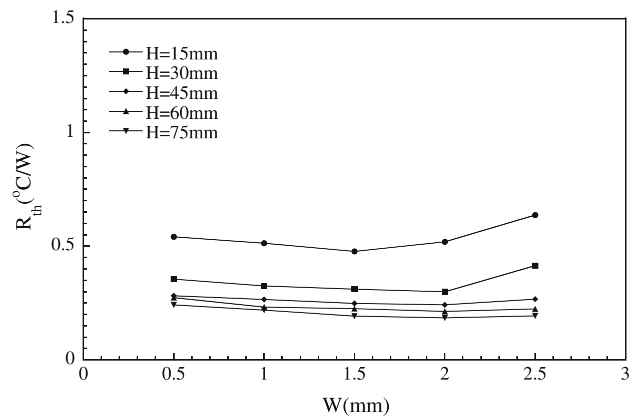

Fig. 9. Effect of fin dimensions on thermal resistance of heat sink for Re = 60,000.

as the fin width increases. The thermal resistance is minimal as the fin width increases to 1.5 mm for heat sinks with fin heights of 45, 60 and 75 mm. The thermal resistance increases as the fin width continues to increase. This fact indicates that a heat sink with a larger heat transfer area has better thermal performance if the cooling air has sufficient kinetic energy to flow into the heat sink. Moreover, if the fins are too wide, the cooling air is obstructed from entering the heat sink to exchange heat. Therefore, the increase in the fin width to improve heat transfer is limited. Similarly, increasing the fin height reduces the thermal resistance. However, the decline is not great as at Re = 20,000.

Fig. 7 displays the thermal resistance when the Reynolds number is increased to 40,000. The thermal resistance is minimal at $W = 1.5$ mm for heat sinks with $H = 15$ and 30 mm; it is minimal at $W = 2.0$ mm for heat sinks with $H = 45$, 60 and 75 mm. A comparison between Figs. 6 and 7 indicates that the optimal fin width for a particular fin height increases with the Reynolds number, because the cooling air enters the heat sink more quickly to exchange heat as the Reynolds number increases. Therefore, the heat sink with a larger heat transfer area performs better thermally. However, the thermal resistance clearly increases because the inter-fin channels are too narrow when $W = 2.5$ mm. The variation of the thermal resistance of the heat sinks with fin height at fixed fin width is similar for $W \leqq 1.5$ mm. The increase in the thermal resistance becomes greater as the fin height increases for the widest fin ($W = 2.5$ mm), because this fin width is associated with the smallest inter-fin channels. Therefore, increasing the heat transfer area can improve the thermal performance.

Fig. 8 presents the thermal resistance for Re = 50,000. The effect of the fin width on the thermal resistance declines as $H \geqq 45$ mm. At $H = 15$ and 30 mm, the thermal resistance varies similar to that at Re = 40,000. The thermal resistance increases as the fin width increases to 2.5 mm.

Fig. 9 shows the thermal resistance for Re = 60,000. Comparing Fig. 9 with Fig. 8 indicates that the drop in the thermal resistance with an increase in the Reynolds number in this range is very small. Accordingly, in practical applications, a lower Reynolds number can be chosen to reduce the pumping power of the fan without sacrificing the cooling capacity.

In summary, for fixed fin width, the heat sink with the highest fins has the least thermal resistance. For constant fin height, the optimal fin width increases with the Reynolds number. Therefore, the fin width must be combined with the Reynolds number to optimize the thermal performance. The improvement in the thermal performance is limited when the Reynolds number is in a particular range. Hence, an effective Reynolds number should be chosen to meet the heat transfer requirement.

3.3. Effects of fin dimensions and Reynolds number on pressure drop

Figs. 10–14 plot the pressure drops across the heat sinks for various Reynolds numbers. The pressure drops of the heat sinks are

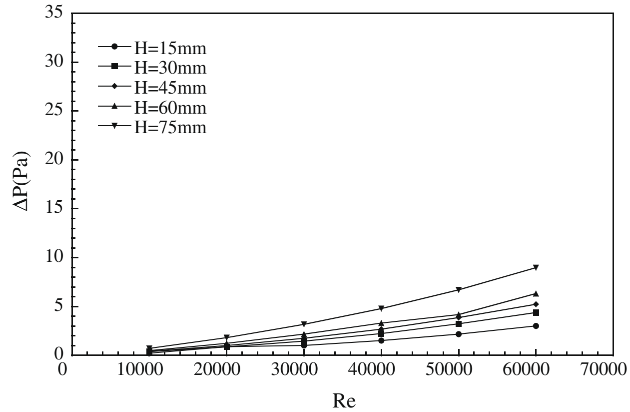

Fig. 10. Effect of fin height and Reynolds number on pressure drop across heat sink for $W = 0.5$ mm.

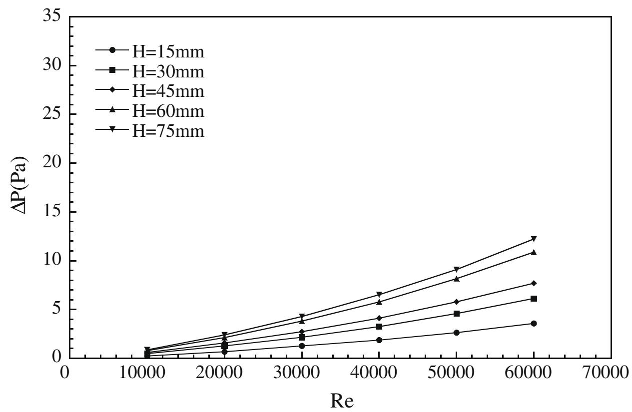

Fig. 11. Effect of fin height and Reynolds number on pressure drop across heat sink for $W = 1.0$ mm.

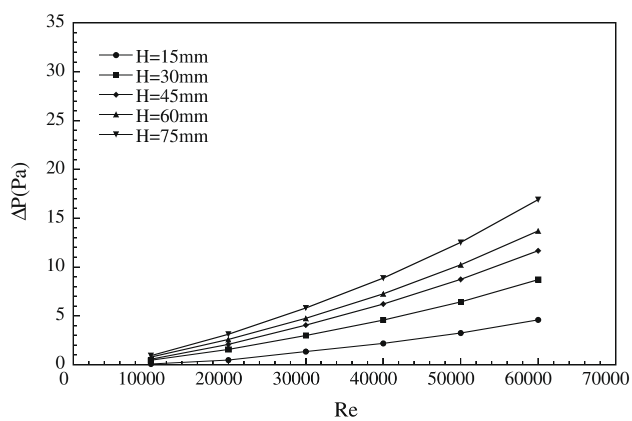

Fig. 12. Effect of fin height and Reynolds number on pressure drop across heat sink for $W = 1.5$ mm.

relatively small at Re = 10,000. The variation among the pressure drops is also not large. The pressure drop increases gradually as the Reynolds number increases. The increase in the pressure drop becomes large as the fins become high, because the cooling air is more obstructed as it flows through a heat sink with higher fins. Increasing the fin width increases the pressure drop, because wider fins reduce the inter-fin channels and increase the flow resistance of the cooling air.

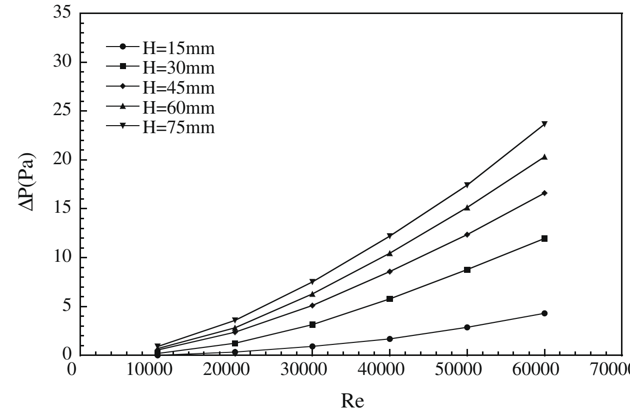

Fig. 13. Effect of fin height and Reynolds number on pressure drop across heat sink for $W = 2.0$ mm.

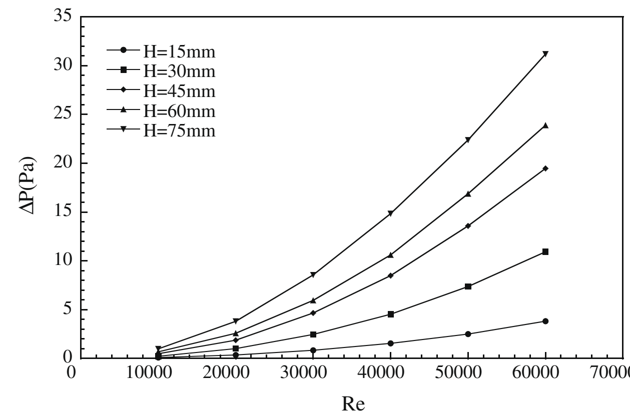

Fig. 14. Effect of fin height and Reynolds number on pressure drop across heat sink for $W = 2.5$ mm.

In summary, the pressure drop increases as the Reynolds number, the fin width and the fin height increase. The increase in the pressure drop becomes greater as the Reynolds number increases, particularly when the fin height and the fin width are large.

4. Conclusions

The performance of plate-fin heat sinks under cross flow cooling has been studied experimentally for various Reynolds numbers, fin heights and fin widths. Based on the results of this study, we conclude the following.

1. The temperature of the heat sink declines from the bottom to the top and from the middle to the exterior. Additionally, it is lower in the front than in the rear of the heat sink.
2. For constant fin width, the heat sink with the highest fins has the best thermal performance.
3. For constant fin height, the fin width that provides the best thermal performance increases with the Reynolds number. Further increasing the fin width will degrade the thermal performance.
4. The thermal performance improves as the Reynolds number increases until a certain value, at which the improvement becomes limited.
5. The pressure drop increases as the Reynolds number, the fin width and the fin height increase.

Acknowledgements

The authors thank the National Science Council of the Republic of China for financially supporting this research under Contract No. NSC 96-2221-E-211-017.

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