On the Heat Transfer Characteristics of Heat Sinks: With and Without Vortex Generators

Kai-Shing Yang, Jhih-Hao Jhong, Yur-Tsai Lin, Kuo-Hsiang Chien, and Chi-Chuan Wang

***Abstract*—This paper examines the airside performance of heat sinks having fin patterns of delta, semi-circular vortex generators, plain fin and their combinations. Test results indicate that the heat transfer performance is strongly related to the developing and fully developed flow characteristics. The augmentations via vortex generator are relatively effective when the flow is in the developing region whereas they become quite less effective in the fully developed region. This is especially pronounced when the fin pitch is small or operated at a lower frontal velocity. Actually, the plain fin geometry outperforms most of the fin patterns at the fully developed region. This is because a close spacing prevented the formation of vortex, and the presence of interrupted surface may also suffer from the degradation by constriction of conduction path. The results suggest that the vortex generators operated at a higher frontal velocity and at a larger fin pitch are more beneficial than that of plain fin geometry. The semi-circular vortex generator possesses the highest heat transfer coefficients and pressure drops at developing region, suggesting the mechanism of blockage of conduction path cannot be overlooked. The performance of dense or loose vortex generator is moderate either in a developing or fully developed region. In association with the VG-1 criteria (same pumping power and same heat transfer capacity), the asymmetric design (VG+plain) reveals the best results. The design could reduce 31.1% required heat dissipation area at a frontal velocity of 5 m/s within a developing region. Yet, it is still applicable in a fully developed region with an area reduction of 1.8–11.5% at a frontal velocity 3–5 m/s.**

***Index Terms*—Fully developed, heat sink, plain, vortex generator.**

NOMENCLATURE

A	Heat transfer surface area (m^2).
A_{ct}	Cross sectional area at the test section (m^2).
A_{front}	Frontal area of fins (m^2).
C_p	Specific heat at constant pressure of air (J/kg·K).
D_h	Hydraulic diameter (m^2).
F_p	Fin pitch (m).
F_s	Fin spacing (m).
f	Friction factor (dimensionless).
H	Fin height (m).
$\bar{h}$	Average convective heat transfer coefficient (W/m^2·K).
j	Colburn factor (dimensionless).
k	Thermal conductivity (W/m·K).
L	Duct length (m).
$\dot{m}$	Mass flow rate (kg/s).
N	Number of fins (dimensionless).
P	Pitch (m).
P_h	Wetted perimeter of the fin channel (m).
Pr	Prandtl number (dimensionless).
$\dot{Q}_{\mathrm{conv}}$	Convention heat transfer rate (W).
Re_{dh}	Duct Reynolds number (dimensionless).
St	Stanton number (dimensionless).
$T_{\mathrm{air,in}}$	Average temperature of the inlet test section (K).
$T_{\mathrm{air,out}}$	Average temperature of the outlet test section (K).
T_{avg}	Average temperature of the air (K).
T_w	Average surface temperature (K).
t	Fin thickness (m).
$\dot{V}$	Volumetric flow rate (m^3/s).
V_c	Mean velocity in the flow channel (m/s).
V_{front}	Frontal velocity (m/s).
x^+	Inverse Graetz number (dimensionless).

Greek Symbols

ΔP	Total pressure drop (Pa).
ρ	Density of air (kg/m^3).

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I. INTRODUCTION

AIR COOLING IS the most popular thermal management of electronics for its simplicity and low cost. The low thermal conductivity of air inevitably results in a very low heat transfer coefficient. As a consequence, the general approach for heat transfer improvement is via exploitation of smaller fin spacing to accommodate more fin surface. However, a limitation is imposed on this conventional approach when the fin spacing is small or when the operation speed is low. This is made clear from Yang *et al.* [1] who examined the thermal-hydraulic performance of heat sinks having plain, slit, and

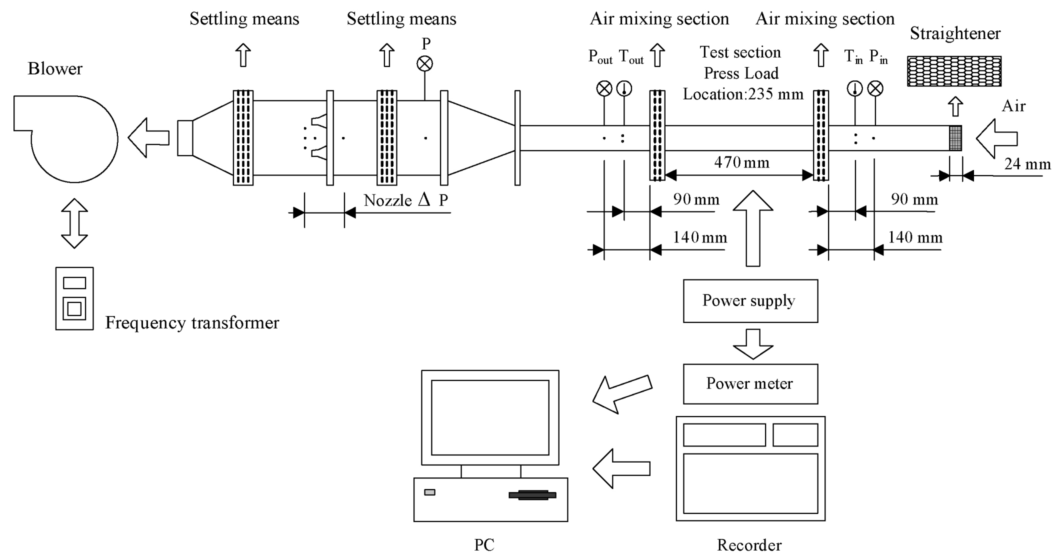

Fig. 1. Experimental setup.

louver fin configurations. Their results indicated a significant drop of heat transfer performance at a low Reynolds number and at small fin spacing. This is because fully developed flow prevails. In this sense, one would resort to interrupted fin geometry to reduce the thermal resistance. The general concept is via periodical renewal of boundary layer. Unfortunately, as pointed out by Yang *et al.* [1] and Webb and Trauger [2], typical interrupted surfaces, like louver fin, show appreciable degradation in low velocity region pertaining to the "duct flow" phenomenon [1], [2]. At a higher velocity, the air flow can be easily directed by the louver and results in good mixing amid adjacent channels. By contrast, for a lower frontal velocity, the majority of the entering air just flows alongside the channels without generating noticeable mixing [1]. This is regarded as "duct flow," and is identified at low Reynolds number by Webb and Trauger [2]. As a result, the improvement of heat transfer performance is rather small in the low frontal velocities region. The results imply a difficult situation of heat transfer augmentation occurring at a low velocity having smaller fin spacing. As explained earlier, the poor heat transfer performance is expected for its fully developed nature. Yet, in the low Reynolds number region the interrupted surface suffers from the "duct flow" phenomenon. In summary of the foregoing results, it is found that significant augmentation is hard to achieve in this region (small fin spacing and low Reynolds number). One of the alternatives to tailor this problem is to introduce vortex shedding and destabilized flow field, and the common way for doing this is using vortex generators [3].

Vortex generators in early research were used to delay boundary layer separation on aircraft wings. For example, Schubauer and Spangenberg [4] exploited streamwise vortices in boundary layer control and measured the effects of a number of mixing and vortex-generating surface elements on boundary layer development. Recently vortex generators have been considered to be effective means for reducing thermal resistance. Fiebig *et al.* [5], [6] reported the influence of longitudinal vortex generators on the heat transfer improvement of a channel flow; the configurations were limited to a single delta or rectangular wing, a single delta or rectangular winglet, or a single pair of delta or rectangular winglets. The heat transfer enhancement is as high as 50% with the pressure drop penalty being approximately 45%. The delta wings were found to be the most effective amid the tested vortex generators. Tanaka *et al.* [7] have studied the enhancement of heat transfer by vortex generators. The experimental results showed that the local Nusselt number in the vicinity of a single rectangular vortex generator peaks at an inclination angle at 60° and an attack angle to the flow direction at 45°. Gentry and Jacobi [8] reported average enhancements of 20–50% with accompanying pressure drop penalty of 50–110% for Reynolds numbers ranging from 400 to 2000.

Though some studies had been conducted for vortex generator in association with heat exchanger application, its applicability into the electronic cooling system is rarely investigated. Note that electronic cooling applications usually incorporate a very dense fin accommodation. The forgoing drawbacks for conventional fin designs like low heat transfer performance and higher pressure drop motivate this paper to introduce the vortex generator into electronic cooling applications. Efforts are made toward sufficient enhancements at an affordable pressure drop penalty.

II. Experimental Apparatus

The experiment apparatus is based on American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) wind tunnel setup to measure the heat transfer and the pressure drop characteristics of the heat sinks. Two main parts of the experimental apparatus are described below.

As seen in Fig. 1, experiments were performed in an open-type wind tunnel. The ambient air flow was forced across the test section by a centrifugal fan with an inverter. To avoid and minimize the effect of flow maldistribution in the experiments, an air straightener-equalizer and a mixer were

TABLE I
DETAILED GEOMETRY OF THE HEAT SINK (UNIT: MM)

Fin Type	Pitch (F_p)	Number of Fins (N)	VG Pitch (L_{VG})	Opening Angle
Plain	1.00	50	–	–
Plain	1.85	27	–	–
Plain	2.63	19	–	–
Delta VG	1.00	50	2	60°
Delta VG	1.85	27	2	60°
Delta VG	2.63	19	2	60°
Delta VG	1.00	50	4	60°
Delta VG	1.85	27	4	60°
Delta VG	2.63	19	4	60°
Delta VG+plain	1.00	50	2	60°
Delta VG+plain	1.85	27	2	60°
Delta VG+plain	2.63	19	2	60°
Semi-circular VG	1.00	50	2	60°
Semi-circular VG	1.85	27	2	60°
Semi-circular VG	2.63	19	2	60°

provided. The inlet and the exit temperatures across the sample were measured by two T-type thermocouple meshes. The inlet measuring mesh consists of four thermocouples while the outlet mesh contains eight thermocouples. The sensor locations inside the rectangular duct were established following ASHRAE [9] recommendation. These data signals were individually recorded and then averaged. During the isothermal test, the variation of these thermocouples was within 0.2 °C. In addition, all the thermocouples were pre-calibrated by a quartz thermometer having 0.01 °C precision. The accuracies of the calibrated thermocouples are of 0.1 °C. The pressure drop of the test sample and nozzle was detected by a precision differential pressure transducer, reading to 0.1 Pa. The air flow measuring station was a multiple nozzle code tester based on the ASHRAE 41.2 standard [10]. All the data signals are collected and converted by a data acquisition system (a hybrid recorder). The data acquisition system then transmitted the converted signals through the Ethernet interface to the host computer for further operation.

A total of 15 heat sinks were made and tested; the corresponding fin patterns are: 1) plain fin; 2) delta vortex generators fin; 3) delta vortex generators+plain fin; and 4) semi-circular vortex generators fin. The delta vortex generators are of equilateral triangle. The heat sinks are made from copper with a thermal conductivity of 398 W/m·K. The fabricated vortex generators are punched from copper sheet, leaving holes alongside the fin. Detailed geometries of heat sink are shown in Fig. 2, and their detailed dimensions are also tabulated in Table I. The base plates of the heat sinks are of square configuration with a length/width of 50 mm and a thickness of 2 mm. The corresponding fin pitches are 1.0, 1.85, and 2.63 mm, respectively, with a constant fin thickness of 0.2 mm. In addition, the height of the heat sinks is 10 mm. A film heater with the same size of base plate is attached to the bottom of heat sink. During the tests, electric power supply provided 25 W power input to the heater. Five temperature sensors were placed below the heat sink to measure the average temperature of the heat sink. The bakelite board is installed beneath the film heater in order to minimizing the heat loss. The heat sinks were loaded to a constant force of 11 N for all experiments. This provided consistent thermal contact resistance between the heat sinks and heater.

III. ANALYSIS OF HEAT SINK

The airside performances of the test heat sinks are in terms of pressure drop and heat transfer performance characteristics. For determination of the friction factor of the test samples, an adiabatic test is performed to obtain the total pressure drops. Hence, the measured friction factor can be obtained from the following equation:

$$f = \frac{\Delta P}{4\left(\frac{L}{D_h}\right)\cdot\left(\frac{\rho V_c^2}{2}\right)} \tag{1}$$

where L, P_h, and ρ are the duct length, wetted perimeter, and density of air, respectively. The hydraulic diameter (D_h) is defined by height of fin (H) and fin spacing (F_s), and can be obtained from the following equation:

$$D_h = \frac{4A_c}{P} = \frac{4\times(H\times F_s)}{2\times(H+F_s)} \tag{2}$$

Yet, the characteristic velocity is calculated by flow rate and cross sectional area at the test section as

$$V_c = \frac{\dot{V}}{A_{\mathrm{ct}} - A_{\mathrm{front}}} \tag{3}$$

where $\dot{V}$, A_{ct}, and A_{front} represent the volumetric flow rate, cross sectional area at the test section, and the frontal area of the heat sink. The total heat transfer surface area (A) is the surface in contact with work fluid, and the cross sectional area at the test section of fin (A_{ct}) is the whole flow channel of test section, which can be calculated as

$$A_{\mathrm{ct}} = W \times H \tag{4}$$

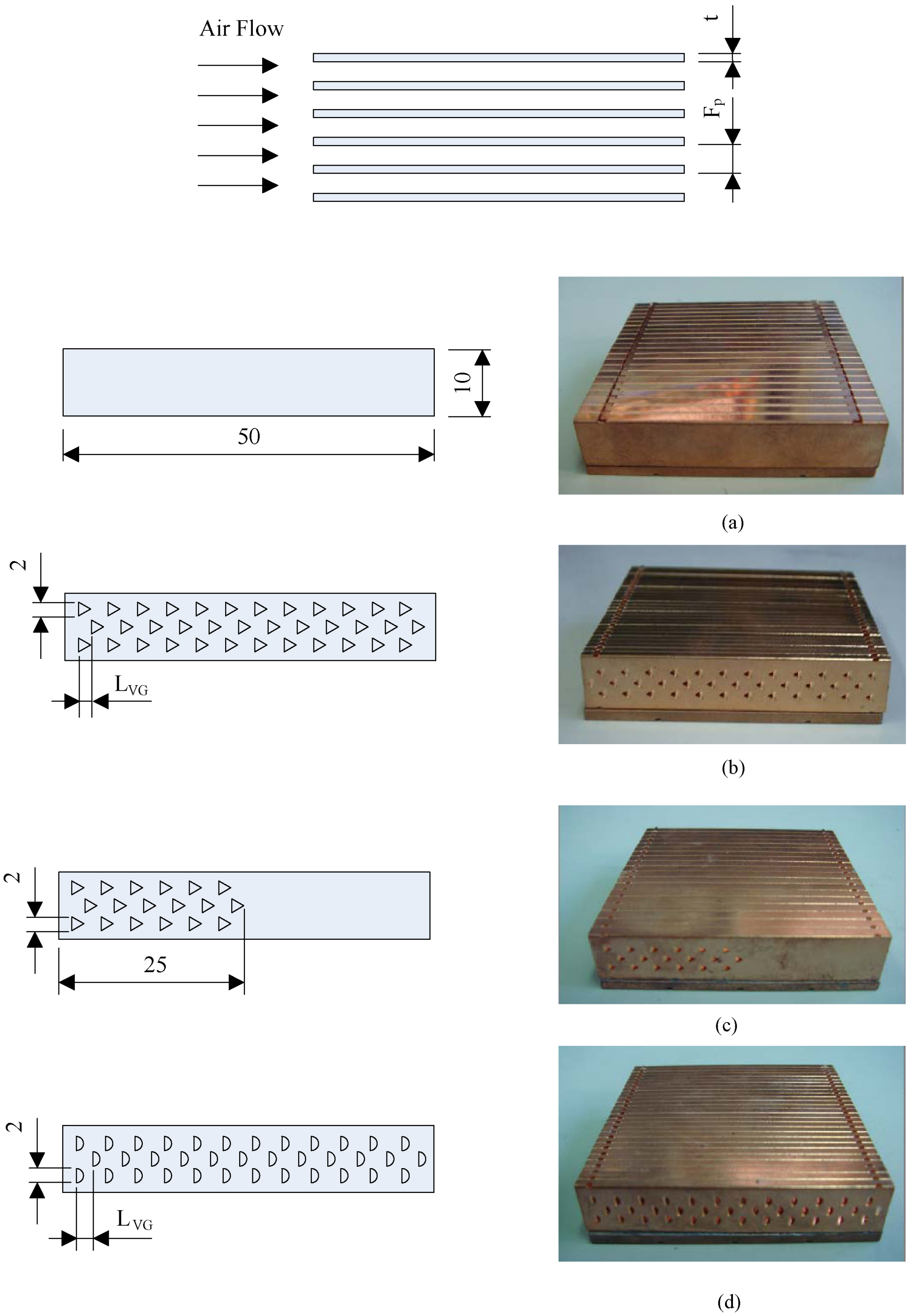

Fig. 2. Schematic of heat sinks geometry. (a) Top view and right view of plain fin. (b) Delta vortex generators. (c) Delta vortex generators+plain. (d) Semi-circular vortex generators fin (unit: mm).

The frontal area of fins (A_{front}) can be calculated by number of fins (N), thickness of fin (t), and height of fin (H) as flow

$$A_{\text{front}} = N \times t \times H \tag{5}$$

The convective heat transfer rate of the experimental system can be obtained from the following equation:

$$\dot{Q}_{\text{conv}} = \dot{m} c_p \left(T_{\text{air,out}} - T_{\text{air,in}}\right) \tag{6}$$

where $\dot{m}$, c_p, $T_{\text{air,out}}$, and $T_{\text{air,in}}$ represent the mass flow rate, specific heat, average temperature of the inlet test section, and the average temperature of the outlet test section, respectively.

The heat transfer coefficients are evaluated from the measured wall and air temperature

$$\bar{h} = \frac{\dot{Q}_{\text{conv}}}{A\left(T_w - T_{\text{air,avg}}\right)} \tag{7}$$

where T_w denotes the average surface temperature.

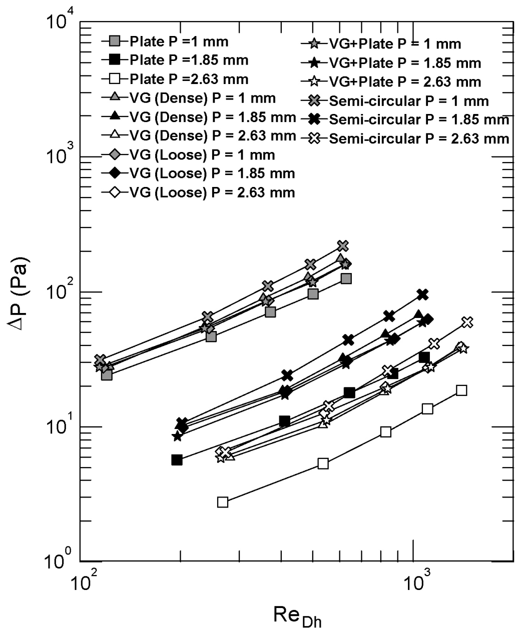

Fig. 3. Pressure drops versus Reynolds number for the test heat sinks.

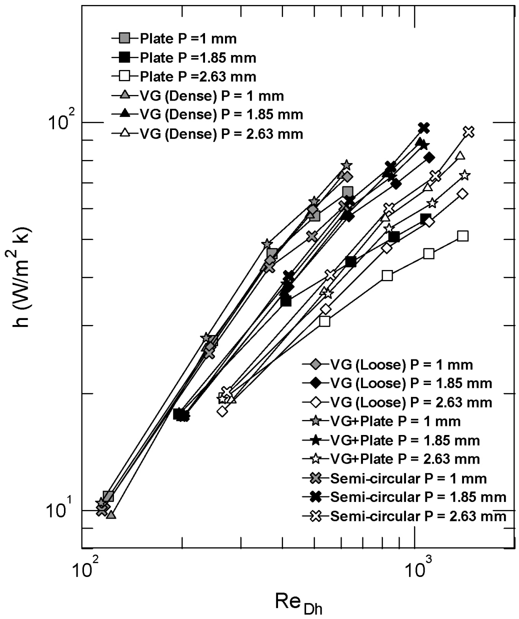

Fig. 4. Heat transfer coefficients versus Reynolds number for the test heat sinks.

Uncertainties in the reported experimental values were estimated by the method suggested by Moffat [11]. The highest uncertainties are 3.71% for the heat transfer coefficient and 2.02% for *f*. The highest uncertainties were associated with lowest Reynolds number.

IV. Results and Discussion

Test results of pressure drops and heat transfer coefficients versus Reynolds number for all the test samples are plotted in Figs. 3 and 4. As expected, both heat transfer coefficients and the pressure drops increase with the rise of Reynolds number. For the increase of pressure drop among the test fin patterns, it can be found that the pressure drops increase significantly when the fin spacing is reduced. The heat sinks with semi-circular vortex generators show the highest pressure drop than other fin types. Also, the pressure drop of dense delta vortex generators fins is higher than that of loose delta vortex generators and that of delta vortex generators+plain fin. In contrast to the pressure drop, the heat transfer coefficients pertaining to various fin configurations depend on the fin pitch. For a larger fin pitch like 1.85 mm or 2.63 mm, the heat transfer coefficients for plain fin always fall below others. At a rather dense fin pitch of 1.0 mm, the plain fin outperforms most of the enhanced fin patterns such as semi-circular, VG (loose), and VG (dense) and is second to VG+plain. This phenomenon becomes even more conspicuous at a lower frontal velocity. The results of heat transfer performance are quite unexpected for one might expect augmentation to take control. For further explanation for this unusual phenomenon, one can examine the corresponding reciprocal of the inverse Graetz number x^+, which is defined as

$$x^+ = \frac{L/D_h}{Re_{D_h}\,\mathrm{Pr}} \tag{8}$$

where L is the streamwise duct length and Pr is the Prandtl number. The flow may be considered to be fully developed when $x^+ > 0.1$ [12]. With a further comparison about the influence of developing flow on the heat transfer performance, test results are plotted in terms of h/h_{plain} versus the inverse Graetz number as depicted in Fig. 5. The augmentation levels shown in the figure are apparently divided into two categories. For a lower inverse Graetz number ($x^+ < 0.1$) where the entrance effect plays a significant role, one can see substantial improvements of heat transfer through heat transfer augmentation. This is applicable to all the enhanced fin patterns being tested. Among the tested fin patterns at $x^+ < 0.1$, the heat transfer performance for semi-circular VG outperforms other augmentations. On the other hand, for a fully developed situation where $x^+ > 0.1$, a clear level-off of the enhanced level for all the enhanced fin patterns is seen, and most of the augmentations fail. The test results suggest that the airside enhancements highly depend upon the developing characteristics. Conventional augmentation is effective only in developing regions. Yet, in a fully developed region, one must seek alternative enhancement using different mechanisms of enhancements.

There are some explanations why most of the enhanced fin patterns fail in the fully developed region. The objective of the vortex generator is to provide vortex shedding by which better mixing is achieved. However, the formation of longitudinal vortex is constrained when the fin spacing is reduced. The argument of vortex suppression can be found from a 3-D numerical investigation of a plain fin-and-tube heat exchanger performed by Torikoshi *et al.* [13]. Their investigation showed that the vortex forms behind the tube can be suppressed and the entire flow region can be kept steady and laminar when the fin pitch is rather small. In this sense, it explains part of the reason that the vortex generator is restrained. However, for a very

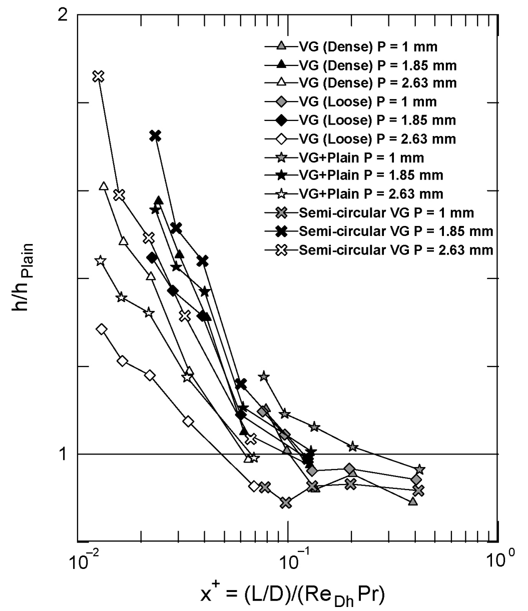

Fig. 5. Inverse Graetz number x^+ versus $h/h_{\rm plain}$ for the test heat sinks.

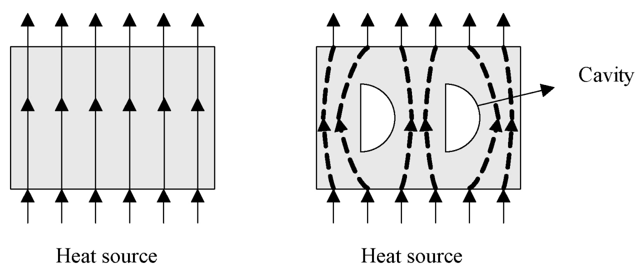

Fig. 6. Schematic of the conduction path with and without the interrupted blockage.

low operation velocity, there is another cause for heat transfer degradation which is the blockage of conduction path of the interrupted surface. Fig. 6 is a schematic showing the heat flows from the base of the surface toward the secondary fin surface. With the presence of interrupted configuration like the semi-circular VG, the conduction path is constricted, yielding a performance drop. This phenomenon becomes more pronounced when the influence of conduction becomes more prominent. That is why at a frontal velocity of 1 m/s and a fin pitch of 1 mm, the heat transfer coefficient for plain fin exceeds all the fin patterns being tested. In fact, this effect cannot be overlooked even at a developing region ($x^+ < 0.1$). As can be seen from Fig. 5, the semi-circular VG outperforms other fin geometry when $x^+ < 0.1$. In summary of the test results, the heat transfer augmentation at $x^+ > 0.1$ is very difficult via a conventional interrupted surface (Yang *et al.* [1]) or via a typical vortex generator. A more compromised design is the VG+plain fin design where the resistance at the downstream is lifted, giving more free space for the vortex development. As a consequence, a small enhancement of this design is seen. The test results suggest it would be made possible from different mechanisms, e.g., unstable vortex shedding or asymmetric fin design (like VG+plain).

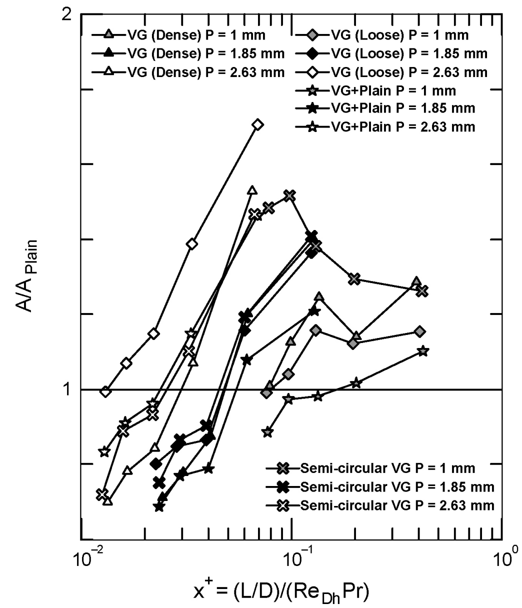

Fig. 7. Inverse Graetz number x^+ versus required heat-dissipation surface area subject to VG-1 criteria.

For further performance evaluation of the tested heat sinks, comparisons are made subject to the VG-1 [14] criteria. The VG-1 criterion seeks for area reduction at the same heat transfer capacity and pumping power. As shown in Fig. 7, the ordinate of the figure is $A/A_{\rm ref}$. A value above unity indicates that the required surface area for the interrupted fin surface design exceeds that of the plain fin surface to fulfill the same heat duty at a fixed pumping power. The results shown in this figure suggest that the vortex generators fin operated with a higher frontal velocity and with a larger fin spacing is more beneficial. The results show that when frontal velocity as 3–5 m/s and fin pitch as 1.85, 2.63 mm effectively reduce required surface area. Despite semi-circular fin geometry possesses the highest heat transfer coefficient when $x^+ < 0.1$, its significant pressure drops lift it from the top choice of vortex generator subject to the VG-1 criteria. When the Reynolds number is decreased or when the fin pitch is decreased, the required heat transfer area of vortex generators would gradually surpass that of plain fin. The result from the present experiment suggests that the asymmetric combination using delta vortex generators+plain fin can be quite effective. A fin pitch of 1.65 mm is regarded as the optimum enhancement design for it could reduce 31.1% required heat dissipation area at a frontal velocity of 5 m/s. The asymmetric VG+plain design is still applicable even when the fin pitch is reduced to 1 mm for it still can reduce required heat dissipation area 1.8%–11.5% at a frontal velocity around 3–5 m/s.

V. Conclusion

This paper conducted an experimental study concerning the airside performance of heat sinks under longitudinal flow condition. Various heat sinks, including delta, semi-circular vortex generator, plain fin pattern, and their combinations, are tested and compared. It was found that the heat transfer performance is strongly related to the developing/fully developed flow characteristics. The augmentations via vortex generator are relatively effective when the flow is in the developing region whereas in the fully developed region they become less effective. This phenomenon becomes especially evident when the fin pitch is small or at a lower frontal velocity. Actually, the plain fin pattern outperforms most of the enhanced fin patterns at the fully developed region. This is because a close spacing prohibited the formation of vortex, in the meantime the presence of interrupted surface may also jeopardize heat conduction path due to constriction. The results indicate that the vortex generators fin operated at a higher frontal velocity and at a larger fin pitch is more beneficial than that of plain fin geometry. The semi-circular vortex generator possesses the highest heat transfer coefficients and pressure drops at the developing region. The performance of dense or loose vortex generator is moderate either in the developing or fully developed region. In association with the VG-1 criteria (same pumping power and same heat transfer capacity), the asymmetric design (VG+plain) reveals the best overall consequences. The design could reduce 31.1% required heat dissipation area at a frontal velocity of 5 m/s at the developing region. Yet, it is still applicable in the fully developed region with an area reduction of 1.8%–11.5% at a frontal velocity 3–5 m/s.

In summary of the test results, it is therefore concluded that augmentation via various fin patterns like interrupted or vortex generator is quite effective only at the developing region or at a larger fin pitch (>1.8 mm). However, the conventional enhanced fin patterns lose its superiority at the fully developed region. To tackle this problem, some techniques employing vortex shedding or unstable flow field accompanied with the asymmetric design show potential to resolve this problem.

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