AIR-SIDE HEAT TRANSFER ENHANCEMENT BY SELF-AGITATORS

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ABSTRACT

Airfoil-based self-agitators (AFAs), bio-inspired rectangular-shaped self-agitators (RSAs), and caudal-fin inspired hourglass-shaped self-agitators (CHSAs) were installed inside plate-fin heat exchanger. The heat transfer enhancement and pressure drop characteristics of these AFAs, RSAs, CHSAs design were experimentally investigated and compared with the clean channel case. We found that the self-agitators vibrate periodically and generate vortices, which enhance flow mixing and thus heat transfer performance. For the chosen heat sink and assigned working conditions, the best heat transfer performance was obtained with four rows AFAs, which caused an 80% increase in overall Nusselt Number over the clean channel at same Reynolds Number, and a 50% rejected heat increase at the same pumping power due to the strong longitudinal vortices generated by the presence of the AFAs. Experiments were conducted at a wide range of Reynolds numbers from 400 to 10000, which covered laminar-transitional-turbulent regime with CHSAs. Experimental correlations of the pressure drop as a function of dimension parameter and friction factor and Nusselt number as functions of dimensionless ones have been proposed. Mutual coupling motions and effects of multiple-row flapping CHSAs in parallel and tandem configurations were studied by using a high-speed camera. A stereo Particle Image Velocimetry (PIV) system was used to conduct detailed flow field measurements to quantify the flow mixing level. For the chosen plate-fin heat exchanger and assigned working conditions, the best heat transfer performance was obtained with six-row CHSAs with a pitch of 25mm, which caused a 200% increase in the Nusselt number over the clean channel at the same Reynolds number. However, the best overall performance was obtained with twelve-row CHSAs with a pitch of 12.5mm, which caused

a 68% enhancement in thermal-hydraulic characteristic compared to the clean channel at the same Reynolds number.

Chapter 1 Air-side heat transfer enhancement in plate-fin channel with an airfoil-based self-agitator.

NOMENCLATURE

AFA	= airfoil-based self-agitator	(-)
A_s	= conduction area of heating block	(m ²)
A_1	= convection heat transfer area of channel	(m ²)
A_2	= cross section area of channel	(m ²)
b	= length of beam	(mm)
c	= chord length	(mm)
c_p	= specific heat	(J kg ⁻¹ K ⁻¹)
d	= spacing between agitator	(mm)
DAFA	= delta airfoil-based self-agitator	(-)
D_h	= hydraulic diameter	(m)
h	= heat transfer coefficient	(W m ⁻² K ⁻¹)
H	= height of heatsink channel	(mm)
k_1	= thermal conductivity of heating block	(W m ⁻¹ K ⁻¹)
k_2	= thermal conductivity of air	(W m ⁻¹ K ⁻¹)
L	= length of heatsink channel	(mm)
Nu	= overall Nusselt number	(-)
RAFA	= rectangular airfoil-based self-agitator	(-)
Re	= Reynolds number	(-)
Δp	= pressure drop	(Pa)
P	= pumping power	(W m ⁻²)
T	= temperature	(K)
q	= conduction heat rate through heating block	(W)
Q	= rejected heat of plate-fin heatsink	(W m ⁻²)
U_{in}	= inlet flow velocity	(m/s)
s	= span of airfoil	(mm)
t	= thickness of airfoil	(mm)
$\dot{V}$	= volume flow rate	(m ³ s ⁻¹)
W	= width of heatsink channel	(mm)
ν	= kinematic viscosity of air	(m ² s ⁻¹)
θ	= angle of attack of AFA	(°)
φ	= inclined angle between agitators	(°)
ρ	= air density	(kg m ⁻³)

1.1 Introduction

Plate-fin heat exchangers can be used for a wide range of industrial applications, such as power plants, chemical engineering, heating and air conditioning, electronics

cooling, and air-cooled solar collectors [1]. Enhancement of thermal performance in heat exchangers allows for a more efficient design, resulting in material and energy savings. Heat transfer in plate-fin channels has been intensively studied using theoretical, numerical, and experimental methods. Placing vortex generators in a plate-fin channel is one of the most widely used heat transfer augmentation methods. This method is typically classified into two categories: passive and active. Ahmed et al. [2] and Sheikholeslami et al. [3] conducted a comprehensive review on passive vortex generators. Passive methods include modifying the surface of the channel and adding an additional upstream device to generate a vortex. Active vortex generators have limited practical application because they need external power input and they are difficult to place inside the channel. The use of a piezofan is one of the most popular active techniques. Yu et al. [4] studied active agitator plates in an air-cooled heat sink and reported the effects of operational mode, tip gap size, operational amplitude, and frequency of heat transfer enhancement. An alternative to traditional passive vortex generation methods involves placing flexible materials such as plates, flags, and reeds inside the channel. As these flexible structures are immersed in channel flow, they engage in periodic flapping motions, and this fluid-structure interaction (FSI) generates vortices. This method combines the highlighted advantages of both the passive and active methods described above.

A number of numerical studies which employed FSI of flexible materials coupled with heat convection have been published. Shoele et al [5] conducted modeling studies on a flexible reed in a heated channel, which showed that heat transfer is maximized when the interaction between the channel walls and the wake of the reed creates large modulations in the boundary layer. Park et al. [6] demonstrated that an inverted flag embedded in a sur-

rounding flow can improve thermal performance through the generation of vortical structures by FSI. The investigation conducted by Khanafer et al. [7] presented the flow and heat transfer characteristics around a flexible microcantilever in a fluidic cell, which included the effect of fluid velocity, flow direction, and the height of the bluff body on the deflection of the microcantilever. Shi et al. [8] investigated vortex-induced vibration through a flexible plate attached to lee side of a stationary cylinder. The results show that vortex-induced vibration, which is considered a promising way to augment the heat transfer of an air-cooled heat sink with acceptable pressure drop, can dramatically increase the average Nusselt number by up to 90.1% compared to a clean channel. Similar results were observed by Soti et al. [9] through their simulation analyses for a rigid cylinder attached to a deformable plate in a channel flow. The analyses showed that larger Prandtl and Reynolds numbers improved heat transfer, but reduced efficiency. Further studies on the effects of channel blockage by a cylinder conducted by Gallegos et al. [10] indicated that there exists an optimum blockage that maximizes the frequency and amplitude of the vibrations in a flexible structure behind the cylinder. Ali et al. [11] demonstrated that 134% overall heat transfer enhancement was obtained by inserting free elastic flaps instead of rigid flaps in the channel. Lee et al. [12-13] conducted an investigation on heat transfer enhancement by flags in an asymmetric configuration (FAC) and in a symmetric configuration (FSC). They reported that the FAC significantly augmented the thermal efficiency compared to the FSC, and enhanced heat transfer by 207% compared to the base flow.

However, few experimental investigations were conducted on FSI coupled with convective heat transfer [14]. Herrault et al. [15] and Hidalgo et al. [16-17] reported beneficial effects on heat transfer by inserting elastic fluttering reeds in heatsink fins, which

are driven by the air flow. The experimental results reported by Hidalgo et al. [16], demonstrated a coefficient of performance enhancement of about 2.4-fold with a reduction in flow velocity of 50% and an increase of about 190% in Nusselt number relative to the base flow in single channel testbed. However, for the tests conducted in a 5-channel heatsink, the heat transfer enhancement in Nusselt number varies between a minimum of 53% and a maximum of 69% at lower Reynolds number ranges. The test results shown by Hidalgo et al. [17] reported that Nusselt number enhancement increases 43% at $Re = 2000$ to 88% at 5000 and the pressure drop increment increases from 37% at $Re = 2000$ to nearly 100% at $Re = 4300$. Both fluttering reed and AFA are passively driven by air flow through the fluid-structure-interaction process for inducing vortices within the channel to enhance the heat transfer coefficient at the heated wall surface. However, the oscillation frequency of fluttering reed can be varied from about 100– 600 Hz, which is much higher than the frequency of AFA at 37 Hz in the present work. The fluttering reed generally has a larger aspect ratio to create strong disturbance on wall flow, which makes a loud noise during the high-frequency vibration. The overwhelming advantage of AFA compared with elastic reed is that the shape and structure of AFA can be optimized easily to create favorable longitudinal vortices.

Wendt et al. [18] investigated the geometry and flow characteristics of the vortex generator, which is a symmetric airfoil having a NACA 0012 cross-sectional profile. In the present study, the design of both RAFAs and DAFAs was based on the NACA 0012 cross-sectional profile but not exactly the same. AFAs lose stability and start to vibrate at a critical velocity, and then oscillate periodically due to fluid-structure-interaction. Airfoils

that exhibit large deformation due to FSI increase fluid mixing, and thus heat transfer, significantly.

To investigate the convective heat transfer performance of the plate-fin heatsink with AFAs inside, an experimental setup was established and first evaluated by a numerical simulation in the clean channel case. The thermal performance and pressure drop characteristics of AFAs were examined in different shapes, chord lengths, and arrangements. The results of this investigation indicate that the use of AFAs is a potential way to enhance heat transfer rate without significant increases in pumping power and fin density.

1.2 Experimental Description

1.2.1 Experimental apparatus and instrumentation

All experiments were conducted in an open transparent airflow duct. A schematic diagram of the setup is shown in Figure 1-1, and a photo of the experimental system can be seen in Figure 1-2(a) [19-21]. The test bed consists of an airflow chamber (AMCA 210-99, Airflow Measurement System), a heating block, a test section, a seven-channel plate-fin heatsink (518-95AB, Wakefield), AFAs, and devices for monitoring temperature and pressure difference. A high-speed camera (Miro 310, Vision Research) is set to capture the flapping motion of the AFAs to get a good understanding of structure vibration. Air is drawn in by a variable speed fan and passes through the test section of the airflow duct as shown in Figure 1-2(b). The customized aluminum heating block carrying two resistor heaters (LPS 300, Vishay Americas) is wrapped by fiber-glass thermal insulation and supported by a lab jack through a 3D-printed stage. Fourteen T-type thermocouples are

inserted into the heating block to help collect temperature gradient data and monitor heat flux as shown in Figure 1-2(c). A seven-channel plate-fin heat sink is mounted on the heating block using eight screws. Thermal paste is added at the interface between the heat sink and heating block to reduce contact thermal resistance. Heat is generated in the heating block by two resistor heaters. The data acquisition (DAQ) device is comprised of a 4-slot Ethernet chassis (NI cDAQ-9184, National Instrument), a 16-differential voltage input module (NI 9205, National Instrument), and a 16-channel thermo- couple input module (NI 9213, National Instrument).

Figure 1-1 Schematic diagram of the experiment setup

Figure 1-2 Photo of the experimental setup: 1. airflow chamber; 2. pressure transducer; 3. 3.7-channel heatsink and AFA; 4. tubes connected to pressure sensor; 5. high-speed camera; 6. heating block wrapped by thermal insulation; 7. resistance heaters; 8. six T-type thermocouples for monitoring temperature of heatsink base; 9. eight T-type thermocouples for monitoring heat flux; 10. computer and DAQ devices; 11. power supply

The black anodized seven-channel heat sink is made of aluminum, and each channel has a length, width, and height of 121.92 mm, 6.60 mm, and 21.84 mm, respectively. A high-accuracy differential pressure transducer (220DD, MKS Instruments) with a range of 0–20 torr is used to monitor pressure difference inside the airflow chamber in order to

correlate the volume flow rate with the system curve. Another differential pressure sensor (698A11TRB, MKS Instruments) with a range of 0–10 torr is set to collect the pressure drop through the heat sink channels. Six T-type thermocouples are installed at the interface of the heating block and the base of the heat sink to measure the bottom surface temperature of the heat sink, one T-type thermocouple is installed at the inlet of the duct to measure bulk inlet temperatures, and eight T-type thermocouples are installed uniformly throughout the heating block to monitor the heat flux through the heating block. All T-type thermocouples were calibrated by a calibrator (ThermCal400, Accurate Thermal Systems) with an accuracy of ± 0.25 °C.

Figure 1-3 Airfoil-based self-agitator assembly composed of a beam, connector, airfoil, shaft and a support.

An AFA design is developed for heat transfer enhancement, as shown in Figure 1-3. The AFA assembly consists of a support, a beam, a connector, a shaft, and an airfoil. The shaft, beam, and support are fabricated using carbon steel for vibration stability. The airfoil and connector are 3D printed. The beam is bonded to the support structure by epoxy, and epoxy is also used between the connector and the beam. The beam is 15 mm long and 0.5

mm thick. The thickness and span of the airfoil profile (based on, but not identical to, NACA 0012) are 1 mm and 10 mm, respectively, and the chord length is varied from 3 mm to 9 mm.

1.2.2 Experimental procedure

First, a thermal test for the clean channel was conducted to obtain a baseline for heat transfer performance. The heat flux was monitored through the aluminum heating block using the eight thermocouples at assigned locations. The base temperature (T_{base}) of the heat sink was estimated using the six thermocouples embedded directly underneath the heat sink. The air temperature at the inlet ($T_{\text{air,in}}$) and outlet ($T_{\text{air,out}}$) of the heat sink were also monitored. The temperature difference between the incoming air-flow and the average temperature of the base of the heat sink was maintained at 30 K. Tests were repeated for increasing airspeed values from 2 m/s to 6 m/s.

The temperature measurements given by the thermocouples embedded in the heating block were used to find the conduction heat rate, q , according to Fourier's Law for 1-D heat conduction.

$$q = k_1 A_s \frac{\Delta T}{d} \quad (1 - 1)$$

In Eqn. (1) k_1 is thermal conductivity of the aluminum block, A_s is the cross-sectional area of the block, d is the distance between thermocouples in the direction of heat conduction, and ΔT is the temperature difference between the thermocouples.

The Reynolds number of the airflow at the channel inlet is given by

$$Re = \frac{U_{in} D_h}{\nu} \quad (1 - 2)$$

where U_{in} is the inlet velocity, D_h is the hydraulic diameter of the single channel, and ν is the kinematic viscosity of air. The inlet velocity, U_{in} , is given by

$$U_{in} = \dot{V}/A_2 \quad (1 - 3)$$

where A_2 is the cross-sectional area of the plate-fin channel and $\dot{V}$ is volumetric flow rate. A fan system curve which reflects the relationship between volume flow rate and pressure difference inside airflow chamber is used to estimate $\dot{V}$.

Pumping power, P , is determined by the pressure drop and flow velocity, which can be expressed by

$$P = U_{in}\Delta p \quad (1 - 4)$$

where Δp is the pressure drop through the plate-fin heat sink channel.

The heat transfer over the heating surface is found using the overall Nusselt number

$$Nu = \frac{hD_h}{k_2} \quad (1 - 5)$$

where k_2 is the thermal conductivity of air and h is the average heat transfer coefficient, which is obtained by

$$h = \frac{Q}{\eta\Delta T_{lm}} \quad (1 - 6)$$

where η is the fin efficiency, ΔT_{lm} is the log mean temperature difference (LMTD), and Q is the actual rejected heat. Q is defined as the average heat flux of the plate-fin heat sink, and is found by

$$Q = Q_1 - Q_2 \quad (1 - 7)$$

where Q_1 is rejected heat calculated from experimental data and Q_2 (estimated in the following section) is the heat loss to the environment. Q_1 is given by

$$Q_1 = q/A_1 \quad (1 - 8)$$

where A_1 is convection heat transfer area of plate-fin heat sink. ΔT_{lm} is obtained by

$$\Delta T_{lm} = \frac{\Delta T_o - \Delta T_i}{\ln(\Delta T_o/\Delta T_i)} \quad (1 - 9)$$

where ΔT_i is the difference between the channel surface temperature and the mean air temperature at the inlet, and ΔT_o is the difference between the channel surface temperature and the mean air temperature at the outlet. ΔT_i was maintained at 30K, and ΔT_o is obtained by

$$\Delta T_o = \Delta T_i - \Delta T_c \quad (1 - 10)$$

where ΔT_c is temperature difference through the channel. ΔT_c is given by,

$$\Delta T_c = \frac{q}{\rho c_p \dot{V}} \quad (1 - 11)$$

where ρ and c_p are the density and specific heat of air, respectively.

Next, tests were carried out by placing RAFAs and DAFAs into the plate-fin heatsink channel in a one-row double arrangement to investigate the effect of AFA shape as shown in Figure 1-4. As shown in Figure 1-5, a double arrangement is when two AFAs are attached to the same support structure. The chord length for each airfoil shape was the same for this test. The experiments were then repeated with RAFA chord lengths of 3 mm, 4 mm, 5 mm, 6 mm, 7 mm, and 9 mm, and DAFA chord lengths of 5 mm, 7 mm, and 9 mm.

Three different arrangements of DADAs were compared: a one-row single arrangement, a one-row double arrangement, and a two-row single arrangement. All three of these arrangements are depicted in Figure 1-6.

Finally, tests were performed with multiple rows DAFAs installed in the plate-fin channel in order to maximize heat transfer as shown in Figure 1-7. As shown in Figure 1-8, the spacing between neighboring rows is 300 mm.

Figure 1-4 (a) Rectangular airfoil-based self-agitators (RAFAs) and (b) Delta airfoil based self-agitators (DAFAs) one-row double arrangement.

Figure 1-5 Two airfoil-based self-agitators were placed in a double arrangement.

Figure 1-6 Delta airfoil based self-agitators (DAFAs) deployed in (a) one row double, (b) two rows single and (c) one row single.

Figure 1-7 Delta airfoil based self-agitators (DAFA) arrayed in (a) one row, (b) two rows, (c) three rows, and (d) four rows.

Figure 1-8 Spacing between each row of delta airfoil based self-agitators (DAFAs).

1.2.3 Uncertainty analysis and validation of experimental setup

Experimental results were compared with numerical simulation results for the clean channel in order to validate this experimental setup. A simulation was conducted to obtain the heat loss, Q_2 , to the environment. Figure 1-9 depicts the simplified model for the numerical simulation, where only half of the test section is considered because the test section is symmetric. Most of the exposed area of the heating block is insulated with fiberglass material, while a small portion is surrounded by the bottom wall of the channel duct, which is made of 3D printed material. In the numerical simulation conducted in ANSYS CFX, the computational domain includes the heating block, thermal insulation, 3D printed material support, and heatsink; the conjugate heat transfer is modeled. Ambient temperature is prescribed at the outer surfaces of the thermal insulation and supporting material. The heat flux is tuned at the heating block base to maintain the average temperature of the heatsink base 30°C higher than the inlet temperature in both numerical and experimental investigations. A more detailed numerical and experimental comparison of a clean channel can be found in our recent work [22]. Rejected heat is a critical parameter in the evaluation of heat sink performance. Rejected heat from the whole heat sink is lower than from the bottom of the heating block because of heat loss to the environment. The fin surface temperature is not constant, and it is lower than the heatsink base temperature because of imperfect fin efficiency. Table 1-1 shows the fin efficiency, η , and heat loss, Q_2 , for different inlet velocity conditions. Heat loss for all velocities was about 300 W/m². As overall heat transfer improves, so does the heat transfer coefficient, which decreases calculated fin efficiency and rejected heat improvement. Therefore, we need to eliminate

the fin efficiency and heat loss effects in order to make a fair evaluation of the thermal performance.

To calculate the constant fin temperature results based on the experimental data, we need to calculate the heat transfer coefficient for the experiment without considering heat loss and fin efficiency. This is accomplished according to the following steps: (1) calculate the rejected heat from the heating block; (2) use the rejected heat and inlet velocity to obtain the outlet temperature; (3) estimate the LMTD by assuming the wall temperature is equal to the heating block average base temperature; (4) divide the rejected heat by LMTD to obtain the heat transfer coefficient. We can then use the calculated heat transfer coefficient to estimate fin efficiency [23]. Note that, although rejected heat in this calculation is found by assuming that no heat is lost to the environment, this assumption results in less than 3% error for all experimental results.

Figure 1-9 Simplified model for numerical simulation is comprised of a heatsink, supporting part, insulation and heating block.

Table 1-1 Fin efficiency and heat loss for clean channel

U_{in} (m/s)	2	3	4	5
Q_2 (W/m ²)	308.9	305.1	307.0	310.5
η (%)	98.2	97.6	97.4	97.3

For the numerical results, the realistic rejected heat from the heat sink, Q , is calculated using Eq. (7). Q is divided by the fin efficiency, η , to estimate the rejected heat with constant wall temperature. The constant fin temperature condition is also solved numerically for different conditions.

Clean channel tests were repeated and averaged more than ten times in order to obtain reliable data. The uncertainty for the pre- sent experimental setup is estimated using the standard approach proposed by Moffat et al. [24], and the analytic definition of standard uncertainty is

$$\delta R = \left[\sum_i^N \left(\frac{\delta R}{\delta X_i} \delta X_i \right)^2 \right]^{1/2} \quad (1 - 12)$$

Each term in Eqn. (12) represents the contribution made by the uncertainty in one variable, δX_i , to the overall uncertainty in the result, δR .

The uncertainty estimations range from 2.79% to 5.04% for the rejected heat and 3.18% to 6.05% for the overall Nusselt number. The highest uncertainties were associated with the lowest flow velocity and Reynolds number, as shown in Figure 1-10 Uncertainty of experimental setup for (a) rejected heat and (b) overall Nusselt number. Figure 1-10. These results are also tabulated in Tables 2 and 3.

Figure 1-10 Uncertainty of experimental setup for (a) rejected heat and (b) overall Nusselt number.

Table 1-2 Relative Uncertainty of Rejected Heat.

U_{in} (m/s)	2.02	2.65	3.15	3.81	4.39	4.81	5.51	5.79
Q_{un} (%)	5.04	4.48	4.25	3.74	3.37	3.17	2.91	2.79

Table 1-3 Relative Uncertainty of Overall Nusselt Number.

Re	1310	1717	2043	2463	2742	3114	3568	3745
Nu_{un} (%)	6.05	5.25	4.90	4.30	3.87	3.63	3.31	3.18

Fig. 11(a) shows the numerical and experimental rejected heat comparison under the constant fin temperature condition. The two results generally agree, with an average difference of 10.45%. In addition to the rejected heat, pressure loss between the inlet and outlet of the heatsink is another critical parameter in the evaluation of thermal performance. Fig. 11(b) shows the numerical and experimental results for pressure drop at different flow velocities, and general agreement is observed between two sets of results. Agreement between these numerical and experimental results indicates that our experimental setup is reliable.

Figure 1-11 Validation results between numerical and experiment for (a) rejected heat and (b) pressure drop.

1.3 Results and discussion

This research was conducted in order to find an effective mechanism to enhance heat transfer using AFAs. First, experiments were conducted to optimize AFA shape and chord length. With the optimal shape and desired chord length decided, the effect of AFA arrangement was studied to find an efficient way to install AFAs in a certain length of the plate-fin channel. The experimental results on heat transfer performance in a heat sink channel equipped with AFAs are presented in the form of an overall Nusselt number and rejected heat.

1.3.1 Shape comparison of AFA

Thermal tests were conducted for RAFAs and DAFAs in a one row double arrangement with a chord length of 5 mm to compare heat transfer enhancement performance. As shown in Figure 1-12(a), for the same flow velocity, channels with DAFAs exhibited about 50% more rejected heat than clean channels, and channels with RAFAs exhibited about 25% more rejected heat than clean channels. As shown in Figure 1-12(d), under the same conditions, the overall Nusselt number when using DAFAs is significantly higher than when using RAFAs. For DAFAs, the overall heat transfer enhancement is around 70%, while the heat transfer enhancement is around 33% for RAFAs. This difference in heat transfer enhancement is because the strong longitudinal vortex generated by a delta airfoil results in better fluid mixing than the transverse vortex produced by a rectangular airfoil [25]. Longitudinal vortices have their axes parallel to the main flow direction and transverse vortices have their axes perpendicular to the main flow direction. Moreover, the longitudinal vortex exhibits less damping along the channel length. The comparison of

pressure drop is shown in Figure 1-12(c). DAFAs result in less pressure loss than RAFAs. Therefore, DAFAs result in better heat transfer performance at the same pumping power. This result is shown in Figure 1-12(b). These experiments were repeated with a 9 mm chord length. As shown in Figure 1-12(a), the heat transfer enhancement for DAFAs is higher than RAFAs with the 9 mm chord length, but this difference is much less pronounced than with the 5 mm chord length. As shown in Figure 1-12(c), the pressure drop for RAFAs is higher than DAFAs. The pressure drop for both DAFAs and RAFAs is higher with a 9 mm chord length than with a 5 mm chord length, possibly because the longer chord allows for more motion of the airfoil, resulting in more blockage in the channel. These experiments indicate that using DAFAs results in better heat transfer performance over RAFAs at a lower pressure loss penalty.

Figure 1-12 Heat transfer and pressure drop characteristics of different shapes.

The oscillating motion of the airfoil was captured by a high-speed camera for a RAFA and a DAFA with a chord length and a span of 5 mm and 6 mm, respectively. Different shaped AFAs have different oscillating modes, and the oscillating mode of a single AFA depends on flow velocity. The vibration of an AFA is mainly heave and pitch. Heave motion is a linear vertical motion, in which the AFA moves up and down without any rotation. Pitch motion is a rotation motion, wherein the AFA rotates around its rotation axis. At lower flow velocity ranges, the force applied on the AFA is mostly from the coming airflow. The AFA vibrates in both heave and pitch motion due to the fluid-solid coupling force, while the AFA mainly exhibits heave motion at higher flow rates. At higher flow rates, the trailing edge of the airfoil touches the side wall, which applies force on AFA

perpendicular to the flow direction, suppressing the pitch of the AFA. However, the noise generated is very limited because of the low frequency of vibration, about 37 Hz.

The Figure 1-13 shows a typical snapshot of oscillating motion for a RAFA. Motion I was captured at flow velocity 3.1 m/s (the lowest velocity at which stable vibration is observed). The heave motion and pitch motion can be seen very clearly. The angle of attack, θ , varies periodically between -60° and 60° . The RAFA reaches maximum deformation when θ reaches its maximum, and in this position the RAFA blocks the most cross-sectional area of the channel and make the instantaneous pressure drop reach its maximum. Oscillating motion II can be observed for a flow velocity of 4.2 m/s, as shown in Figure 1-13. In motion II, the heave becomes stronger and the pitch becomes weaker, and hence the maximum of θ reduces to 30° , which results in less channel blockage and lower pressure drop.

Figure 1-13 One cycle motion for DAFs, θ is the angle of attack of the RAFA; motion I is captured at flow velocity 3.1 m/s and motion II is captured at flow velocity 4.2 m/s.

Figure 1-14 shows a typical snapshot of oscillating motion for a DAFA. Motion III is observed at a flow velocity 3.3 m/s (the lowest velocity at which stable vibration is observed). The heave and pitch motions are quite similar to those of motion I for a RAFA, and θ also ranges from -60° to 60° . However, when the DAFA reaches its position of maximum deformation, and angle of attack, θ , reaches its maximum, airflow blockage is much lower than with a RAFA. As the flow velocity is increased to 4.5 m/s, the DAFA oscillates with motion IV, also shown in Figure 1-14. The heave becomes the dominant motion as the pitch motion becomes weaker, and hence the maximum in θ reduces to 15° .

When the DAFA oscillates with motion IV, flow blockage and pressure drop are significantly reduced.

Figure 1-14 One cycle motion for DAFAs, θ is the angle of attack of the RAFA; motion III is captured at flow velocity 3.3 m/s and motion IV is captured at flow velocity 4.5 m/s.

1.3.2 Effects of the chord length of double RAFA

Figure 1-15(d) shows the overall Nusselt numbers for RAFAs with different chord lengths compared with the clean channel test. The channel with RAFAs achieves the highest heat transfer enhancement at around 40% with a 7 mm chord length, compared with the clean channel. For a given flow velocity, the overall Nusselt number increases with increasing chord length up to a chord length of 7 mm. As shown in Figure 1-15(a), a similar trend is observed for rejected heat. This can be explained by the stronger transverse vortex generated by a larger oscillating amplitude due to the longer chord. With chord lengths greater than 7 mm, the motion of the oscillating RAFA starts to change due to the width confinement of the channel and contact with the wall. When the trailing edge of the airfoil touches the side wall of the channel, the amplitude of the agitator's vibration can no longer increase. At the same time, the force applied on the RAFA from the wall perpendicular to the flow direction suppresses the pitch motion of RAFA, leaving the heave mode as the dominant motion of the self-agitator. As shown in Figure 1-15(c), the flapping-induced pressure drop for longer chords is no higher than for shorter chords because the dominant heave motion allows for less blockage of the flow.

RAFAs with short chord lengths need high flow velocity in order to vibrate, so they do not provide higher heat transfer performance at low flow velocities. For example, the RAFA with 3 mm chord length cannot vibrate at low flow velocity, which results in low pressure drop. This result is depicted in Figure 1-15(c). Because of its better heat transfer performance and lower pressure loss, the RAFA with a 7 mm chord length results in the best rejected heat performance at the same pumping power, as shown in Figure 1-15(b).

Figure 1-15 Heat transfer and pressure drop characteristics of different chord length of RAFA.

1.3.3 Effects of the chord length of double DAFA

Heat transfer performance of one-row double DAFAs was examined with 5 mm, 7 mm, and 9 mm chord lengths. As shown in Figure 1-16(a), (b), (d), chord length has little effect on Nusselt number, rejected heat, or pressure drop. A heat transfer enhancement of around 40% was obtained for all three of these cases. The heat transfer enhancement can be attributed to the longitudinal vortex generated by delta airfoil at the tip of the trailing edge, which efficiently carries cool fluid from the center of the channel to the channel wall. The longitudinal vortex produced by the DAFA enhances heat transfer more effectively than the transverse vortex produced by the RAFA. As shown in Figure 1-16(c), the pressure drop for the agitator with a 5 mm chord length is slightly higher than the pressure drop for the agitator with 7 mm and 9 mm chord lengths. This is because the heave motion becomes dominant and hence the flow blockage is reduced for oscillating agitators with a long chord length. The results discussed above reveal that a DAFA with a 5 mm chord length has similar heat transfer enhancement and a minor difference in pressure drop but can significantly reduce the materials cost compared with DAFAs with 7 mm and 9 mm chord length. Therefore, a DAFA with a 5 mm chord length is an optimal choice for a delta-shaped agitator.

Figure 1-16 Heat transfer and pressure drop characteristics of different chord length of DAFA.

1.3.4 Arrangement effect for DAFA

First, single and double arrangements of DAFAs in a channel were investigated. Both cases were examined in one-row installations, and relative position in the channel is shown in Figure 1-6(a) and (c). As shown in Figure 1-17(a) and (d), the double arrangement has better thermal performance, with 40% heat transfer enhancement obtained for double configuration and only 31% for the single arrangement. This may be because the vortices produced in a channel with a double agitator interact more strongly with the heated wall boundary layer than the vortices produced by a single agitator, resulting in better flow mixing. Motion captured by a high-speed camera shows that out of phase vibration for a double arrangement is an effective way to reduce the flow blockage. As shown in Figure 1-17(c), the pressure drop for single deployment is much lower than for a double design due to less flow blockage. The pressure drop is almost the same as the baseline test at high velocity because the heave motion of the small agitator results in very little flow blockage. Because the double DAFA design has better heat transfer performance with lower pressure loss, it has the best rejected heat performance at the same pumping power, as shown in Figure 1-17(b). This indicates that the double design is the most effective method to enhance heat transfer in a heat exchanger.

To investigate the details of a double arrangement of 14 DAFAs oscillation in the channel, we used a high-speed camera to capture one cycle, as shown in Figure 1-18. The motion is shown at a flow velocity of 3.5 m/s. The angle of attack becomes small, but both heave and pitch motions can be clearly observed. The vibrations of the upper and lower DAFAs are out of phase, which minimizes pressure drop in the channel. When the DAFAs

reach their position of maximum deformation, the angle φ between the upper and lower DAFAs becomes 90° , which increases flow mixing in the channel.

Figure 1-17 Heat transfer and pressure characteristics of different DFAA arrangements.

Figure 1-18 One cycle motion for double DFAA: φ is the inclined angle between upper and lower DFAA and motion is captured at flow velocity 3.5 m/s.

1.3.5 Heat transfer enhancement of multiple row double AFAs

Multiple rows of double DAFAs were installed in the channel as shown in Figure 1-7. Only four rows were installed due to space limitations. Because of the results of the previous studies, all DAFAs in this experiment had a span and chord length of 10 mm and 5 mm, respectively. Oscillating frequency was recorded by the high-speed camera, and the videos show that disturbances in the flow field caused by upstream agitators have little effect on the vibrational frequency of downstream agitators. The recorded vibration frequency for all agitators was around 37 Hz. As shown in Figure 1-19(d), heat transfer enhancement increases with the number of rows of DAFAs for any Reynolds number and a maximum enhancement of around 80% is obtained with a four-row design. However, when we consider rejected heat and pumping power, as shown in Figure 1-19(b), we see similar heat transfer enhancement for agitators in two, three, and four rows. This is because pressure loss increases proportionally with heat transfer as the number of rows increases. These results are shown in Figure 1-19(c) and (a). The results of this study indicate that no more than two rows of DAFAs per channel are needed if pressure drop is a concern. Additional investigation is necessary to determine the optimal arrangement and spacing of multiple rows of AFAs.

Figure 1-19 Heat transfer and pressure up characteristics of multiple row DAFA.

1.4 Summary

In this study, we first validated our experimental setup with a numerical simulation. We then investigated the thermal and pressure drop characteristics of a plate-fin heat sink using both RAFAs and DAFAs with different spacing and row arrangements. We drew the following conclusions based on our results:

1. The delta-shaped AFA generates strong longitudinal vortices, thus enhancing heat transfer more than a rectangular shaped AFA and also resulting in a lower pressure drop.

2. The optimal chord length for a RAFA is 7 mm. This chord length maximizes heat transfer.

3. Increasing the chord length to pursue higher heat transfer performance for DAFA comes with diminishing returns. A DAFA with 5 mm chord length is optimal, enhancing heat transfer significantly but costing fewer materials than one with higher chord length.

4. The double arrangement for DAFAs has the best heat transfer performance with the lowest pressure penalty, so out of the arrangements examined in this study, it is the best design for heat transfer enhancement in a heat exchanger.

5. Heat transfer continues to improve with more than two rows of DAFAs, but at the cost of additional pressure drop. No more than two rows of agitators are necessary if pressure loss is a concern.

Chapter 2 Bio-inspired rectangular-shaped self-agitator for convective heat transfer enhancement

2.1 Introduction

As the need for highly efficient convective heat transfer rates for electronic component cooling and power generation continues to grow, bio-inspired vortex generation with fluid- structure interaction (FSI) [14], [26-31] has been highlighted as a promising method to improve the overall convective heat transfer without additional pumping power requirements. It is based on the notion that ages of biological evolution could provide effective inspiration for the design of useful technologies. Examples of FSI in nature, such as grass vibrating in the wind or a bird flying, show that the FSI process can generate fluid vortices which can be used to improve fluid mixing. Therefore, bio-inspired designs can be developed to improve convective heat transfer based on the enhanced mixing processes. Two categories of FSI exist, and the driving force of structural vibration distinguishes them, an active one with external power and a passive one without external power requirement. Active FSI vortex generators have been employed to enhance heat transfer in various applications [32-35]. These can be applied to current heat exchangers where their oscillating frequencies directly affect the thermal performance. However, they require an additional power supply system to drive the structural oscillation, limiting the range of practical applications. Passive FSI vortex generators for heat transfer enhancement [15-17], [36-37] have also been investigated. However, a method to improve the vortex field has not yet been identified. This article focuses on passive methods where we develop a bio-inspired self-agitator for convective heat transfer enhancement and explore the principles underlying its performance.

Currently, a range of methods exist for achieving convective heat transfer enhancement, such as offset strip fins, T-mixers, Dean flow in curved pipes, and stationary vortex generators. In general, they all create vortices to enhance mixing for convective heat transfer enhancement. For various designs, transverse or longitudinal vortices can be generated for heat transfer enhancement [3], [38-48]. Compared to a stationary vortex generator, the FSI process has been shown to achieve better performance [49]. Thermal performance is maximized when large modulations are created at the boundary layer of the heat transfer surface while avoiding the creation of strong vortices in other regions [5]. Although many methods exist to introduce vortices for heat transfer enhancement, what type of vortex shows the best performance is still unclear.

In this work, we first investigate the fundamental relation between vorticity modes and thermal performance. To explore this, dynamic mode decomposition (DMD) [50-52] is performed with the transient numerical results in the vorticity field. Subsequently, various modes with fixed frequencies can be achieved. Then, the vorticity modes introduced by the structural motions can be distinguished by matching the frequency of the vorticity mode to that of the structural motion. Several other modes (including the dominant one) exist concurrently with frequencies different from that of the structural motion. Another interesting phenomenon is that the first and second fluid modes' frequencies do not change significantly for different self-agitator conditions for a specific geometry for heat transfer. This indicates that the heat exchanger has its "preferred" frequencies for the vorticity field in convective heat transfer enhancement. It is well accepted that vortices near the boundary with different frequencies can break the thermal boundary layer to improve convective heat transfer. However, no direct guidance has been

achieved before to reveal how to refine the vorticity field for further improving convective heat transfer. From the analysis part of this work, we find that synchronizing the preferred frequency to the structural motion can improve convective heat transfer. This conclusion is used to guide the experimental structural design, and the experimental results demonstrate that the self-agitator can improve the thermal performance significantly. We choose a commercially available plate fin heat exchanger (6.6mm gap, 22.0mm height, and 120.0mm length) as an example to test the self-agitator performance. The experimental results demonstrate that heat transfer can be enhanced by 120% without additional pumping power requirements compared to the clean channel while achieving a Nusselt number of up to 30 within the laminar flow region by matching with the preferred frequency.

2.2 Test structure and measurements

Inspired by biological processes observed in nature, a self-agitator design is proposed where a thin-film material is attached to a heat exchanger with an especially designed shape as shown in Figure 2-1. As airflow is introduced, FSI causes the structure to vibrate, generating vortices that will improve the overall heat transfer rates in the channels of the heat sink. The enhanced heat sink performance with self-agitators is measured in terms of increased rejected heat over a range of flow rates compared to the baseline flow in the absence of the self-agitators.

To guide the prototype design, numerical investigations are performed to study the dynamics of the self-agitator. Verification of the numerical tools for solving fluid structure interactions and heat transfer problems has been reported in our recent papers [19], [22].

Figure 2-1 Experimental setups: 150 CFM chamber; airflow measurement systems are employed to provide cooling air.

2.3 Results and discussion

2.3.1 Dynamics of the self-agitator

First, numerical simulations are performed with the same geometry as the experimental setups: the height and the length of the channel are 6.6mm and 120 mm, respectively. The initial inclined angle α of the self-agitator is 45° , which is motivated by other groups' results [11]. The Young's Modulus of the self-agitator is 2.5 GPa, and the density (ρ_0) is 1420 kg/m^3 ; these properties are obtained from the commercially available

Kapton film. The inlet velocity is 4 m/s, and the temperatures T_{wall} and T_{in} are 308K and 298 K, respectively.

This FSI problem is solved numerically, and the Fourier analysis for the structural motion is shown in Figure 2-2(b). Transient results of vorticity fields are employed to perform the modal analysis with DMD [112]. The two dominant frequencies are found to be 352 Hz and 704 Hz. Figure 2-2(c) shows one snapshot of the vorticity fields in the region downstream of the surface agitator and the corresponding modes obtained from modal analysis. A vortex street exists through the channel as shown with instantaneous vorticity contours, and it can be decomposed into various modes with DMD, while each mode has an impact in the whole channel. In our previous work [49], it has been demonstrated that the thermal performance is related to the steady modes (not time dependent) directly. However, no efforts have been made to guide how to refine the steady mode for better thermal performance. In this article, we explore to create resonance between flow and structure modes to improve the thermal performance. Besides the steady mode, the first and second unsteady modes (172 Hz and 271 Hz) play important roles in the vorticity field, and their frequencies are different from the structure ones. These can be the targeted frequencies to resonate with the structure for obtaining heat transfer enhancement. The third and fourth modes (355Hz and 721 Hz) have frequencies similar to the structural motions. Therefore, they could be introduced by the motion of the structure. Assuming that the dominant flow modes do not change in different FSI processes, we need to decrease the structural resonance frequencies by 24% to reach 271Hz or by 51% to meet 172 Hz, so that the structure can resonate with the dominant flow modes.

Figure 2-2 Numerical results and modal analysis: (a) 2-D assumption is applied to simplify the FSI process of the self-agitator; (b) to evaluate the structural oscillation frequency, Fourier analysis is performed on the maximum displacement of the self-agitator; (c) a series of snapshots of the transient vorticity field are used for DMD analysis. Only the first few low-frequency modes are included here [112].

2.3.2 Modal analysis and preferred frequency

To decrease the structure's resonant frequency, eigenfrequencies of the self-agitator are analyzed for different structure densities as shown in Figure 2-3(a), where the original density, ρ_0 , for the structure is 1420 kg/m^3 . The eigenfrequency of the original density self-agitator is 300 Hz, which is very close to the condition found in the FSI process. Therefore, the eigenfrequencies for different densities can provide guidance for obtaining the final structure's resonant frequency in the FSI process. As shown in Figure 2-3(a), multiplication factors of 1.7 and 4.0 times the original density can decrease the resonant frequencies by 24% and 51%, respectively, to achieve a structural resonance with dominant flow modes in FSI.

Figure 2-3 Comparison of results for different densities [112]

Two FSI cases are solved numerically with $1.7\rho_0$ and $4.0\rho_0$, while the other parameters are the same as the original density case. Fourier analysis of the structural motions is shown in Figure 2-3(b), where it is shown that a higher density results in a lower resonant frequency. The multiplication factors of $1.7\rho_0$ and $4.0\rho_0$ can decrease the structure frequency by 23% and 49% compared to the original density condition, which fulfills the goals set in the last section. Figure 2-3(c) shows one snapshot of the vorticity fields in the region downstream of the surface agitator and the corresponding modes from modal analysis for the $1.7\rho_0$ condition. Besides the steady mode, the first and second unsteady modes (181 Hz and 287 Hz) play the dominant roles in the flow fields. The second mode is enhanced because its frequency is close to the resonant frequency of the structure. Therefore, with the refined $1.7\rho_0$, the specific mode is enhanced as we expected. Figure 2-3(d) shows one snapshot of the vorticity field in the region downstream of the self-agitator and the corresponding modes from modal analysis for the $4.0\rho_0$ condition. The dominant unsteady modes have frequencies of 183 Hz and 186 Hz. The flow mode at 186 Hz is enhanced significantly as previously expected. Therefore, enhancing the targeted mode by adjusting the structure's density is promising. After adjusting the structure design, the results are shown in Figure 2-3(c) and Figure 2-3(d). In contrast to the original design in Figure 2-2(c), the enhanced modes [highlighted in Figure 2-3(c) and Figure 2-3(d)] show better vorticity preservation through the channel, even though the vortices still decay in the stream-wise direction. To evaluate the performance of the refined self-agitators, rejected heat rates are compared for the different conditions as shown in Figure 2-3(e). For the original density condition, the averaged rejected heat is 634W/m^2 with a pressure loss of 198 Pa. For the $1.7\rho_0$ case, the average rejected heat is increased to 687W/m^2 and the

pressure loss is decreased to 175 Pa. With $4.0\rho_0$, the average rejected heat increases to 698W/m^2 , while the pressure loss changes to 203 Pa. Compared to the original density condition, heat transfer performance is improved with self-agitators of redefined density. These results agree well with our previous work [49] by comparing the steady modes in different cases in Figure 2-2 and Figure 2-3. In the highest-performance case, the steady mode of vorticity has the most discrete patterns compared to the steady mode in the other cases, which indicates more effective breaking of the thermal boundary layer. Therefore, matching the structural resonant frequencies with the first few dominant flow modes can improve the thermal performance of the self-agitator. First dominant mode frequencies are preferred for convective heat transfer enhancement. Currently, the frequencies of the first dominant modes for different density conditions are quite close to each other (about 180 Hz). Therefore, one existing heat exchanger with a given geometry has its preferred frequency of the vorticity field. These results can guide the future design of self-agitators.

2.3.3 Self-agitator motions and thermal performance

As we previously discussed, the designed heat sink with a certain channel gap has a preferred frequency from the 2-D study, and it is reasonable to hypothesize that a 3-D case has similar behavior. To validate the performance of the optimized self-agitator with the preferred frequency, we designed a rectangular self-agitator using a 0.025mm thickness Kapton film. Here, the resonant frequency of the Kapton film based self-agitator is designed to be close to the predicted channel preferred frequency in the simulation (about 180 Hz). To investigate the vibration characteristics of the self-agitator in the channel, we use a high-speed camera to monitor the motion of the self-agitator as shown in Figure 2-1(experimental setup), which has been built and verified in our lab [21], [53-55]. As the channel height is 21.8 mm, we placed two rectangular self-agitators as a pair to generate stronger vortices. The vibration characteristics of the designed self- agitators are shown in Figure 2-4(a), while the results show that the self-agitator near the top and near the bottom vibrate with the same frequency but in different phases. Figure 2-4(b) (Multimedia view) shows the motion for the rectangular shape self-agitator; the slow-motion results indicate that the two self-agitators in one row have the out of phase phenomenon [56-58], which can reduce the pressure drop of the self-agitator.

Figure 2-4 Self-agitator motions. (a) Two self-agitators are set in one row, and they move out-of-phase. Fourier analysis is performed with the vibration amplitude. (b) Out-of-phase motions of self-agitators in one period are presented.

The thermal tests were conducted for clean channel, rectangular shaped self-agitators and a rigid longitudinal vortex generator (LVG), which can be viewed as the state of the art, [5], [38-40], [42], [46], [59-63] to evaluate the thermal performance. As shown in Figure 2-5, under the same conditions, the overall Nusselt number for the self-agitators is significantly higher than those with the LVGs and clean channel cases. The Nusselt number (Nu) and the Reynolds number (Re) are widely used to evaluate the performance

of convective heat transfer [49]. As shown in Figure 2-5(a), the self-agitator can improve Nu by 200% at the same Re. Figure 2-5(b) shows that the self-agitator can enhance the heat transfer coefficient by 120% at the same pumping power compared to the clean channel. The above experimental results indicate that the thin-film based self-agitator can obtain better heat transfer performance than the LVG, which also proves that a self-agitator oscillating at the channel's preferred frequency can enhance the heat transfer performance of the heat sink.

Figure 2-5 Comparison of results. (a) Nu-Re; two rows pf the self-agitator can achieve better performance of four rows of LVGs. (b) Pumping power is calculated with pressure loss multiply velocity.

2.4 Summary

This work includes a self-agitator design to improve convective heat transfer and explores the fundamental relationship between heat transfer and the vorticity field, which has never been discussed before. From modal analysis of the transient vorticity fields, the dominant modes and modes introduced by the self-agitator motions can be distinguished. Matching the frequency of structural motion to that of the dominant mode can create resonance to enhance the fluid mixing. These findings show that the FSI process of the self-agitator can improve the convective heat transfer significantly and that further enhancement can be achieved by approaching the preferred frequency of the vorticity field. The self-agitators were fabricated and experimentally tested for convective heat transfer enhancement. Self-agitator oscillation at the preferred frequency is achieved, and experimental results demonstrate that the heat transfer can be enhanced by 120% without additional pumping power requirements while the Nusselt number can be improved by 200% with the same Reynolds number compared to the clean channel.

Chapter 3 A Novel Caudal-Fin-Inspired Hourglass-Shaped Self-Agitator for Air-Side Heat Transfer Enhancement in Plate-Fin Heat Exchanger

NOMENCLATURE

A_1 (m ²)	= cross section area of channel	
A_2	= convection heat transfer area of heat exchanger	(m ²)
B (mm)	= width of TSB platform	
c_p (K ⁻¹)	= specific heat	(J kg ⁻¹)
CHSA	= caudal-fin-inspired hourglass-shaped self-agitator	(-)
d (mm)	= gap between dual CHSAs	
D_h (m)	= hydraulic diameter	
f	= friction factor	(-)
h (W m ⁻² K ⁻¹)	= heat transfer coefficient	(W m ⁻² K ⁻¹)
I	= turbulent intensity	(-)
k (W m ⁻¹ K ⁻¹)	= thermal conductivity of air	(W m ⁻¹ K ⁻¹)
L (mm)	= length of CHSA	
L_p	= distance of test section between pressure taps	(m)
Nu)	= overall Nusselt number	(-)
P (W m ⁻²)	= separation distance of multiple-row CHSA	(W m ⁻²)
ΔP (Pa)	= pressure drop	
Q (W)	= rejected heat of plate-fin heatsink	
Q_1 (W)	= heat loss in the test section	
Q_2	= overall electrical power added to the heating section	(W)
R (-)	= number of rows	
Re (-)	= Reynolds number	
t (s)	= time	

T (K)	= temperature	
TSB)	= Triangle supporting bar	(-
u' (m/s)	= turbulent velocity fluctuations	
$\bar{u}$ (m/s)	= mean velocity in PIV measurement	
U_{in}	= inlet flow velocity in thermal-hydraulic test	(m/s)
$\dot{V}$ (m ³ s ⁻¹)	= volume flow rate	
W (mm)	= width of CHSA	
<i>Greek symbols</i>		
α (°)	= inclined angle of CHSA	
δ (mm)	= thickness of CHSA	
ρ (kg m ⁻³)	= air density	
ν (m ² s ⁻¹)	= kinematic viscosity of air	
θ (-)	= air effectiveness	
η (-)	= overall performance	
<i>Subscripts</i>		
a (-)	= air	
c (-)	= augmented channel with CHSA	
i (-)	= inlet	
o (-)	= outlet	
0 (-)	= equivalent clean channel value	

3.1 Introduction

Heat exchangers are widely used in energy conversion and management systems, such as power plants, automobiles, heating and air conditioning, electronics cooling, smart

buildings, high-performance data centers [64-69],[23],[70-75]. Enhancement of thermo-hydraulic performance in heat exchangers allows for more efficient design, leading to material and energy savings. Plate-fin channels are one of the most often used heat exchanger modules [76-87]. Heat transfer enhancement in plate-fin heat exchangers has been intensively investigated using theoretical, numerical, and experimental methods. Rigid vortex generators placed in a plate-fin channel are one of the most widely used air-side cooling technologies, and significant effort has been made to improve the overall performance of plate-fin heat exchangers using this method in the last few decades. Delta winglet [42],[20],[88-91] is one of the most commonly designed vortex generators to create longitudinal vortices, and trapezoidal tap [92-95] is one of the most used vortex generators to induce hairpin vortices. These vortices increase flow mixing and enhance thermo-hydraulic performance.

Vortices generated by flow-induced vibration have become an important method in increasing flow mixing, and hence enhancing convective heat transfer performance. Compared to active vortex generators that use a piezoelectric fan [4], [61], [96] to drive the structural oscillation, passive technologies that create vortices through flow-structure-interaction. (FSI) which have a more practical application due to there not being a need for external energy and easy installation. Oscillating airfoil structures, flapping flags, fluttering reeds or strips, and vibrating flexible plates are most often used as self-excited vortex generators to enhance heat transfer performance.

Airfoil-shaped structures were also explored as vortex generators for heat transfer enhancement. Li *et al.* [53][21][97] studied convective heat transfer enhancement by placing a novel airfoil-shaped self-agitator in plate-fin heat sink channel, and their

numerical results showed a 53% Nusselt number increase at the same Reynolds number and 27% heat transfer coefficient enhancement at same pumping power. However, their experimental results reported only a 34% Nusselt number increase at the same Reynolds number and 15% heat transfer coefficient enhancement. Li *et al.* [98] experimentally investigated thermo-hydraulic performance by installing rectangular and delta shaped airfoil-based self-agitators in a plate-fin channel. The optimal design was determined by conducting parameter studies on shape, chord length, and arrangement, and the best performance was obtained with four rows of dual airfoil-based self-agitators with a delta shape during out-of-phase FSI vibration, which resulted in an 80% increase in overall Nusselt number at the same Reynolds number and a 50% rejected heat increase at the same pumping power.

Flapping rectangular flags have been investigated in many efforts to increase the thermal performance and reduce the drag. Kim *et al.* [99] experimentally studied the dynamics of an inverted flag, and they found two quasi-steady regimes: straight and fully deflected modes, and a limited-cycle flapping mode with a large amplitude depending on a non-dimensional bending stiffness. Their analysis also demonstrated that the flapping dynamics are closely related to the periodic formation and shedding of vortices at the leading edge. Sader *et al.* [100] used mathematical theory to analyze the large-amplitude flapping motion of an inverted flag in a uniform steady flow. The results revealed that the onset of flapping is due to divergence instability and reducing the flag's height-length ratio increased the critical flow velocity for divergence, which ultimately eliminated flapping. Huang *et al.* [101] investigated the interaction of two tandem inverted flags, with an in-line and a staggered arrangement, in a uniform flow by a direct numerical simulation at a

Reynolds number of order 200. For infinite flags with a periodic boundary condition, it was found that the flapping amplitude or bending energy of the rear flag is enhanced and significantly higher than those of an isolated flag. Additionally, results for finite inverted flag groups showed that antiphase synchronization is preferred. Kim *et al.* [102] theoretically and experimentally studied the stability and coupled dynamics of three-dimensional dual inverted flags with finite height. Their analysis demonstrated that the critical velocity is reduced monotonically if the flag height increases and the separation distance decreases. Furthermore, a novel attached mode, in which the leading edges of the two flags meet and the flags maintain their deformed shapes statically was found. This novel mode was mainly observed when the flag height and the gap distance were small. Gallegos *et al.* [14] reported a brief review of flag vortex generators for vortex dynamics and heat transfer performance, and provided a list of existing gaps, challenges, and potential research areas for the use of flag vortex generators for heat transfer enhancement. This review guides future research directions in the fluid-structure-thermal-interaction problem. Argentina *et al.* [103] presented a theoretical explanation for the onset of fluttering in a flag that can be used to predict a critical velocity for the onset of fluttering and the frequency of flapping. Ristroph *et al.* [104] studied pairs of flags in a tandem layout and generalized to larger aggregates; they reported that the leader of a group of flapping flags enjoys a significant drag reduction, while the downstream flag suffers a drag increase. Huertas-Cerdeira *et al.* [105] investigated the coupled motion of two inverted flags in a side-by-side layout. In-phase, anti-phase, staggered, alternating and decoupled flapping modes were observed, with the anti-phase mode being predominant in small flag separation and low flow velocity. Flapping amplitude and frequency were also enhanced

in the two-flag system with compared to an isolated flag. In addition, coupling between flags of different lengths was studied, and the observation showed that the longer flag could induce flapping in the shorter one when the shorter one was outside of its flapping range. Yu *et al.* [106-107] experimentally studied the vortex dynamics and heat transfer performance behind an inverted flag of various length-to-channel-width ratios ($C^* = L/W$), 0.125, 0.250 and 0.375 in a rectangular channel. The velocity field was measured by time-resolved particle image velocimetry (TR-PIV) for three systems, and the mixing between the core flow and the near-wall flow was reflected from turbulence kinetic energy (TKE) and the wall-normal velocity fluctuation intensity. The results showed that the increase in C^* significantly promotes wall heat removal due to a stronger disruption of the boundary layer, however, the most efficient heat transfer removal occurred at an intermediate C^* value of 0.250. Abderrahmane *et al.* [108] experimentally studied the flapping dynamics of flags in turbulent flow, and they found that two parameters that might impact the flapping dynamics, which are the flexural rigidity and the aspect ratio of the flag. In addition, they presented for the first time a phenomenon of switching-oscillations, including the back and forth alternation between two modes of the flag, stretched straight and flapping modes. Alben *et al.* [109-110] investigated some of the fundamental effects of channel confinement on the flutter instability of flag and large-amplitude flapping dynamics. The results showed that increasing channel confinement destabilizes the flag and enlarges the regime of unstable flapping. Ali *et al.* [111] numerically investigated the flow mixing efficiency and thermal performance by installing elastic flaps inside a channel in a 2D laminar flow. They showed that vibration of flexible vortex generators enhances the mixture quality up to 98%, with a 134% increase in the overall heat transfer, compared

to rigid vortex generators. Li *et al.* [19] presented a 2D numerical study of a flapping vortex generator mounted on a plate-fin heatsink for air-side heat transfer enhancement, and the results showed a 200% average Nusselt number improvement compared with a clean channel at the same Reynolds number. Inspired by blades of grass vibrating in the wind, Li *et al.* [112] developed a rectangular self-agitator to improve the heat transfer performance of a plate-fin heat exchanger. They conducted a 2D FSI simulation and a dynamic modal decomposition (DMD) analysis and found a preferred frequency of the vorticity mode in a given channel. This analysis guided the design of an experimental prototype. The experimental results demonstrated that heat transfer can be enhanced by 120% without additional pumping power input while the global Nusselt number can be improved by 200% at the same Reynolds number compared to the baseline. Lee *et al.* [13] numerically modeled a pair of elastic flags clamped vertically in a heated plate-fin channel to figure out the dynamics of the flags and their impacts on heat transfer enhancement and the effects of the channel height, bending rigidity of flag, and the Reynolds number on the thermal efficiency were studied to find an optimal parameter set. The results reported that the net heat flux was enhanced by 185% and thermal efficiency factor was augmented by 106% by using the flexible flags with an optimal parameter.

Flexible plates, oscillating strips, and fluttering reeds with large aspect ratios have been recently investigated to guide to augmentation of the thermal performance of plate-fin heat exchangers. Tang *et al.* [113] numerically investigated the dynamics of an inverted plate in a uniform flow by a finite element method for the plate deformation and an immersed boundary-lattice Boltzmann method for the fluid flow. They identified several distinct modes of the plate deformation and motion including straight, flapping, deflected,

deflected-flapping, and asymmetric-flapping states, and which state to occur depends dominantly on the aspect ratio and bending stiffness of the plate. Moreover, they found that the flapping mode increases the conversion of fluid kinetic energy to elastic strain energy. Yadykin *et al.* [114] studied the effect of some parameters such as the number of modes, the length of the strip, and the flow velocity in a fluid-strip subsystem for linear and nonlinear modes. Their simulation work showed that the strip oscillates regularly with a standing-wave type of motion at a low velocity. However, the oscillation changes into a nonregular traveling wave in the direction of the trailing edge at higher flow velocities. Schouveiler *et al.* [115] conducted an experimental visualization analysis for the coupled flutter of an assembly of two, three, and four flexible parallel cantilevered plates immersed in an axial uniform flow. The results showed that different coupled modes were observed depending on the flow velocity, interpolate distance and plate length. Similar work was carried out numerically by Tian *et al.* [116] through placing three filaments in a viscous flow with a side-by-side arrangement. They reported that, besides the in-phase mode, the symmetrical mode, and the out-of-phase mode, three additional flapping modes, the half-frequency mode, irrational-frequency mode, and erratic flapping mode were identified by varying the gap between the filaments. In addition, four typical vortex structures were observed in the wake of the filaments and are described as the coalesced vortices, symmetrical vortices, erratic vortices, and independent vortex streets, which are quite related to the coupling states and the dynamics of the filaments. Shoele *et al.* [5][117] installed a flexible reed inside a heated channel and conducted a coupled flow-structure-thermal simulation by developing an immersed boundary method. The investigations showed that the thermal performance was significantly enhanced by using a vibrating reed,

which was also found to depend more strongly on the reed inertia than its bending stiffness. Furthermore, it was found that the heat transfer performance was maximized when the reed created large modulations in the boundary layer of the channel according to their studies on the effect of both the Reynolds number and the channel confinement. Herrault *et al.* [15] and Hidalgo *et al.* [16-17] presented beneficial effects on thermal performance by installing fluttering reeds in plate-fin channels, which were driven by the air flow. The experimental results from Hidalgo *et al.* [16] showed a coefficient of performance (COP) increment of around 2.4-fold with a reduction in airflow velocity of 50% and an enhancement of about 190% in Nusselt number compared to the base flow in a single channel setup. However, for the experiments tested in a 5-channel plate-fin heatsink, the heat transfer augmentation in global Nusselt number varies from 53% to 69% at lower Reynolds number ranges. The test results presented by Hidalgo *et al.* [17] reported that averaged Nusselt number augmentation increases 43% at $Re = 2000$ to 88% at $Re = 5000$ and the pressure drop increment arises from 37% at $Re = 2000$ to nearly 100% at $Re = 4300$. Jha and *et al.* [118] experimentally investigated the thermal-hydraulic performance of a single channel equipped with autonomously fluttering reed. Their results show the Strouhal number primarily affects the incremental pressure loss but weakly on Nusselt number. Therefore, the pressure drop can be minimized to yield high heat transfer enhancement by optimizing the structure design.

In the present study, inspired by the caudal-fin of a fish [119], an hourglass-shaped self-agitator was developed to enhance the thermal-hydraulic performance of plate-fin heat exchanger. The caudal fin is the tail of a fish, located at the end of the caudal peduncle. A CHSA comprises the leading fin and the trailing peduncle connected by a narrow neck, as

shown in Fig. 1(a). The shape of the leading part is directly inspired by the caudal-fin, but the trailing part holds an inverted structure to simulate a caudal peduncle. In contrast with an active fishtail to generate locomotor force, the passive CHSA is installed inside the plate-fin channel with a forward-swept configuration and a 35° angle of attack to create vortices during the FSI process and hence enhance flow mixing and heat transfer performance. The edges of a CHSA are comprised of a single first-type leading edge (FLE), double first-type trailing edges (FTEs), double second-type leading edges (SLEs) and a single second-type trailing edge (STE). The leading fin of CHSA can oscillate due to FSI, but the trailing peduncle stays almost static. Compared to rectangular flag clamped at the leading edge or trailing edge, a CHSA features many advantages in heat transfer augmentation, pressure loss reduction, and noise control. First, a CHSA vibrates more easily at a low airspeed, which means CHSA can improve the thermal performance by generating intensive vortices during the FSI mechanism at lower Reynolds numbers. Additionally, benefitted by the smart bionic structure designing, the CHSA extends its oscillating axes from trailing edge to narrow neck which is an essential difference from the traditional square flag with whole-body oscillation, whose has its axes are located at trailing edge or leading edge. Therefore, the leading fin is the dominant flapping part to enhance flow mixing, but the trailing peduncle is kept nearly static acting as a rigid vortex generator to reduce pressure penalty, which is an efficient way to increase the thermal-hydraulic performance. As well known, the trailing edge of a rectangular flag or leading edge of an inverted flag is the primary effective vortex shedding location, however, CHSA with multiple edges can create more intensive three-dimensional vortices than rectangular plates. Based on the reported works of heat transfer enhancement by using a rigid

longitudinal vortex generator (LVG), the longitudinal vortices also can be created at double second leading edges of CHSA' trailing peduncle. Moreover, less airflow blockage because of an hourglass shape, is another important tactic for CHSA to significantly decrease pressure loss. Finally, benefited by the partial-vibration mechanism of CHSA, the noise produced during oscillation is lower than that generated by the conventional rectangular flag.

To investigate the convective heat transfer performance of a plate-fin heat exchanger with CHSAs, an experimental setup equipped with a commercial seven-channel heatsink was established. CHSAs were fabricated and attached to the plate-fin of the heatsink in a single row and in multiple rows with variable pitch. The coupled motion of multiple-row CHSAs was recorded and analyzed by using a high-speed camera. A transparent single channel was designed and built so that the flow field could be analyzed using a PIV system. The thermal performance and pressure drop characteristics of CHSAs were examined with different numbers of row and different pitches in laminar-transitional-turbulent regimes. Turbulence intensity was calculated based on the measured velocity field. The results of this investigation indicate that the use of CHSAs is a potential way to enhance thermal-hydraulic performance without significant increases in pumping power.

3.2 Experimental Method

This section describes the design and fabrication of CHSAs, the experimental setup for thermal testing and PIV measurement, and the methods for data reduction and uncertainty analysis.

3.2.1 Geometry and Fabrication of CHSAs

In the present study, a CHSA is designed and developed to implement the proposed novel concept of air-side heat transfer enhancement. Figure 3-1(b) shows the geometrical details and structural compositions for a CHSA. The leading fin and trailing peduncle are symmetric about the narrow neck with the width of $C = 1\text{mm}$. The CHSAs are fashioned from larger sheets of Kapton film, with Young's modulus of 2.5GPa, Poisson's ratio of 0.34, and density of 1420kg/m^3 . A CHSA has a length, width and thickness of $L=6\text{mm}$, $W= 6\text{mm}$, and $\delta=0.0254\text{mm}$ respectively. Double CHSAs were set in one row with a gap, $d = 4\text{mm}$ as shown in Figure 3-1(c). Multiple-row CHSAs were installed in the heated channel with an inclined angle of $\alpha = 35^\circ$, and separation distance $P = 12.5\text{mm}$, $P = 25\text{mm}$ or $P = 50\text{mm}$. A triangle supporting bar (TSB) was designed and 3D-printed to help attach CHSAs to the fin surface as shown in Figure 3-2(a) and Figure 3-2(b). A TSB was comprised of a handle, a supporting platform and a narrow groove between them. The supporting platform, with a width of $B = 1\text{mm}$, was separated by a streamline convex to guarantee the 4mm gap between upper and lower CHSA. A pair of CHSAs was first attached on the inclined plane of supporting platform as shown in Figure 3-2(c) and Figure 3-2(d), and then the TSB was mounted on the heatsink fin using epoxy. The handle of TSB was stuck at tip of the fin to confine the double CHSAs in the center of the channel and the groove made it easy to break and separate the handle from supporting platform as shown

in Figure 3-3(a). Finally, multiple rows of double CHSAs attached to supporting platform were mounted on the heatsink fin as shown in Figure 3-3(b).

Figure 3-1 A caudal-fin-inspired hourglass-shaped self-agitator (CHSA) attached to the fin of the heatsink.

Figure 3-2 Dual CHSAs attached to a triangle supporting bar (TSB).

Figure 3-3 Multiple rows of dual CHSAs installed in the plate-fin heat exchanger.

3.2.2 Experimental Setup for Thermal-hydraulic Tests

To determine the thermal-hydraulic performance of clean channel and channel equipped with CHSAs, an open-circuit testbed was built. A schematic diagram of the setup is shown in Figure 3-4, and a photo of the experimental system can be seen in Figure 3-5. Each component is also tabulated in Table 3-1. The testbed is comprised of an airflow bench (AMCA 210-99, Airflow Measurement System), an airflow duct with a seven-channel plate-fin heatsink (518-95AB, Wakefield), an aluminum heating block, sensors for monitoring temperature and pressure difference, a high-speed camera, and a data acquisition (DAQ) system.

Figure 3-4 Schematic diagram of the experimental setup for thermal-hydraulic tests.

Figure 3-5 Photo of experimental setup for thermal-hydraulic tests.

Airflow duct consists of a contraction section, honeycomb, upstream duct, downstream duct, and test section, and each part was manufactured by 3D-printing. Moreover, an upstream duct, a downstream duct, and a test section are covered by three transparent acrylic plates respectively, which makes it easy to do visualization analysis. The test section was designed and 3D-printed to hold heatsink, which was wrapped by thermal insulation enclosed inside chambers. Air is drawn in by a variable speed blower and passes through the airflow duct and plate-fin channels. The customized heating block carrying three heaters (FPA250, Mouser Electronics) was wrapped by fiberglass thermal insulation material of 25.4mm thickness and thermal conductivity 0.04 W/mK to minimize heat losses, which was supported by two adjustable lab jacks. A commercial black anodized seven-channel heatsink was mounted on the heating block by twelve screws, and every single channel has a length, width, and height of 182.88mm, 6.60mm, and 21.84mm, respectively. Heat is generated in the heating block by resistor heaters through a power supply. Thermal paste was painted at the interface between the heatsink and heating block, and the interface between heaters and heating block to reduce contact thermal resistance.

Table 3-1 Components of the experimental setup.

Number	Component of the experimental setup
1	Airflow chamber
2	Counter blower
3	Speed controller
4	Differential pressure transducer to monitor the pressure difference through the nozzle
5	Differential pressure transducer to monitor the pressure drop through the test section
6	Pressure taps at the nozzle plate
7	Test section equipped with a heatsink
8	Contraction section and air straightener
9	Nine thermocouples for monitoring temperature of the heatsink base
10	Thermal insulation enclosed inside chambers
11	Pressure probe at the inlet
12	Pressure probe at the outlet
13	Thermocouple for monitoring inlet bulk temperature
14	Thermocouple for monitoring outlet bulk temperature
15	Heating block and heaters wrapped by thermal insulation
16	Power supply wired to heaters
17	DAQ devices
18	LabVIEW code running at the computer

Nine T-type thermocouples are inserted into the basement of the heatsink to help collect the bottom surface temperature of the heatsink. Two T-type thermocouples were installed at the inlet and outlet of the test section to measure bulk inlet temperature and outlet temperature. All thermocouples were calibrated by a calibrator (ThermCal400, Accurate Thermal Systems) with the reliability of $\pm 0.05^{\circ}\text{C}$. A high-accuracy differential pressure transducer (220DD, MKS Instruments) with a range of 0-20 torr is used to monitor pressure difference inside the airflow chamber to correlate the volume flow rate with the fan curve. Another differential pressure sensor (698A11TRB, MKS Instruments) with a range of 0-10 torr was set to collect the pressure drop through the test section. The DAQ device is composed of a 4-slot Ethernet chassis (NI cDAQ-9184, National Instrument), a 16-differential voltage input module (NI 9205, National Instrument), and a 16-channel thermocouple input module (NI 9213, National Instrument).

3.2.3 Experimental Setup for PIV Measurements

To measure the velocity field and quantify the dynamic mixing process inside the channel by using the PIV system, a single transparent duct was designed and built. In the present study, the time-resolved PIV technique was used to characterize the instantaneous and time-averaged flow field in the flow passage of the test section. Figure 3-6 shows the schematic of the experimental setup for the PIV measurement. The channel built for PIV test has a width and height of 6.60mm and 21.84mm which is exactly same with the size of every single channel of plate-fin heatsink used in the thermal-hydraulic test. The length of the channel is longer than the heatsink channel, but identical to full-length of airflow duct built in thermal-hydraulic test in order to eliminate the differences between two channel flow as much as possible.

Figure 3-6 Schematic diagram of the PIV test system.

The air was driven by the airflow bench and pass through the contraction section, air straightener, test section with CHSA and diffusion section of the flow duct. The flow was seeded with 1 μ m olive oil particles generated by the droplet generator. Illumination was provided by a double-pulsed Nd: YAG laser with a wavelength of 527nm. Two high-speed CCD cameras with a high-resolution of 1280 pixel $\times$ 800 pixel were used for the PIV image acquisitions. The two cameras were arranged with an angular displacement configuration of about 30° to minimize the light distortion. Using a set of optical modules, the laser beam was shaped into a thin light sheet with a thickness of about 3mm in the measurement interest. A general stereo in-situ calibration procedure was conducted to obtain the mapping functions between the image plane and object planes for the PIV measurements.

3.2.4 Experimental Procedure

The thermal-hydraulic test was first conducted for the clean heatsink to obtain a baseline of heat transfer performance and pressure characteristics. The total rejected heat was monitored through the reading of power supply. The base temperature of the heatsink was estimated using the nine thermocouples inserted into the basement of the heatsink. The air temperature at the inlet and outlet of the heatsink channels were recorded as well. Electric power input to heaters was adjusted depending on flow velocity and the temperature difference between the incoming airflow and the averaged base temperature of the heatsink was maintained at 30K, which may take about 30 min to reach the steady-state conditions. Experimental tests for baseline were repeated for increasing the Reynolds number from 400 to 10000 to cover laminar-transitional-turbulent flow condition, however, the tests for channels with CHSAs were started at $Re = 400$ and terminated at $Re = 7000$ to prevent the CHSAs from being swept away by the high-speed flow.

A calibration test was carried at zero flow velocity to estimate heat loss in the test section Q_1 . Heat Q rejected by heat exchanger was calculated by subtracting heat loss to the total electrical power input in the test section Q_2 , which is obtained by

$$Q = Q_2 - Q_1 \quad (3 - 1)$$

The Reynolds number of the airflow at the channel inlet is given by

$$Re = \frac{U_{in} D_h}{\nu} \quad (3 - 2)$$

where U_{in} is the inlet velocity, D_h is the hydraulic diameter of the single channel, and ν is the kinematic viscosity of air. The inlet velocity, U_{in} , is given by

$$U_{in} = \dot{V}/A_1 \quad (3 - 3)$$

where A_1 is the cross-sectional area of the plate-fin channels and $\dot{V}$ is volumetric flow rate. A fan system curve which reflects the relationship between volume flow rate and pressure difference inside the airflow chamber is used to estimate $\dot{V}$.

The heat transfer over the heating surface is found using the overall Nusselt number,

$$Nu = \frac{hD_h}{k} \quad (3-4)$$

where k is the thermal conductivity of air and h is the average heat transfer coefficient, which is obtained by

$$h = \frac{Q}{\Delta T_{lm} A_2} \quad (3-5)$$

where A_2 is convection heat transfer area of the plate-fin heatsink, Q is the rejected heat, and ΔT_{lm} is the log mean temperature difference (LMTD) is obtained by

$$\Delta T_{lm} = \frac{\Delta T_o - \Delta T_i}{\ln(\Delta T_o / \Delta T_i)} \quad (3-6)$$

where ΔT_i is the difference between the channel surface temperature and the mean air temperature at the inlet, and ΔT_o is the difference between the channel surface temperature and the mean air temperature at the outlet. ΔT_i was maintained at 30K, and ΔT_o is obtained by

$$\Delta T_o = \Delta T_i - \Delta T_a \quad (3-7)$$

where ΔT_a is air temperature difference through the channel. ΔT_a is given by,

$$\Delta T_a = \frac{Q}{\rho c_p \dot{V}} \quad (3-8)$$

where ρ and c_p are the density and specific heat of air, respectively.

Pressure drop ΔP measurement was carried out along the pressure test section ($L_p = 260\text{mm}$) by means of a high-accuracy differential pressure transducer. The hydraulic diameter of the channel was used as the reference diameter to calculate the friction factor. The Fanning coefficient f was determined from flow velocity and pressure drop by means of

$$f = \frac{2\Delta P D_h}{\rho L_p U_{in}^2} \quad (3 - 9)$$

Another criterion, namely the overall thermal-hydraulic performance factor η , is calculated to compare the heat transfer enhancement at constant pumping power, which is defined as the ratio of thermal enhancement in the CHSA case to the baseline according to the relation,

$$\eta = \left(\frac{Nu_c}{Nu_0} \right) \left(\frac{f_c}{f_0} \right)^{-\frac{1}{3}} \quad (3 - 10)$$

where Nu_c and f_c are the heat transfer and friction factor obtained with the CHSAs, and Nu_0 and f_0 are the heat transfer and friction factor obtained with the clean channel. Meanwhile, Nu_c/Nu_0 and f_c/f_0 are described as the Nusselt number augmentation and the friction factor augmentation respectively.

Tests were carried out by placing CHSAs into the plate-fin heatsink channels in a one-row double arrangement. Then, multiple rows of CHSAs were installed in the heatsink with a pitch 50mm to investigate the thermal-hydraulic performance. To determine the saturation level of the convective heat transfer, the air effectiveness θ is calculated by

$$\theta = \frac{\Delta T_a}{\Delta T_i} \quad (3 - 11)$$

For the cases with $P = 50\text{mm}$, the heat transfer was far away from saturation. Therefore, another series of tests were conducted for CHSAs with a reduced pitch from $P = 25\text{mm}$ and $P = 12.5\text{mm}$, which allowed for up to 12 rows of CHSA to be installed to further improve heat transfer performance.

In the PIV measurement, the spatial distributions of the turbulence intensity were calculated to evaluate the flow mixing level, which is defined as the root mean square of the turbulent velocity fluctuations divided by the average flow velocity at a particular location over a specified period of time,

$$I = \frac{u_{\text{rms}}}{\bar{u}} \quad (3 - 12)$$

where u_{rms} is the root-mean-square of the turbulent velocity fluctuations, which is obtained by,

$$u_{\text{rms}} = \sqrt{\frac{1}{N} \sum_{i=1}^N (u'_i)^2} \quad (3 - 13)$$

and $\bar{u}$ is the mean velocity, which is given by,

$$\bar{u} = \frac{1}{N} \sum_{i=1}^N u_i \quad (3 - 14)$$

3.2.5 Uncertainty Analysis

Tests were conducted and repeated with an interval of $Re = 100$ at the range $300 < Re < 5000$ and interval of $Re = 500$ at the range $5000 < Re < 10000$. The experimental uncertainty for the present test bed is estimated by following the root-sum-square method published by Moffat *et al.* [24], and the analytic definition of overall standard uncertainty is

$$\delta R = \left[\sum_i^N \left(\frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right]^{1/2} \quad (3 - 15)$$

Each term in Eq. (15) refers to the contribution made by the uncertainty in one independent variable δX_i to the overall uncertainty in the result δR . Table 3-2. shows the measurement accuracies associated with sensors and parameters. Relative uncertainty calculated for Reynolds number is below 1% as shown in Figure 3-7. The Nozzle inside the airflow chamber was switched to a larger one with a diameter of 1.0 inch, at $Re = 5500$, to obtain high volume flow rate. Therefore, there is a slight drop at $Re = 5500$ for the uncertainty of Reynolds number. Results shown in Figure 3-8 are Nusselt numbers and corresponding uncertainty of the clean channel in laminar, transitional and turbulent flows. The relative uncertainty of Nusselt number decreases as Reynolds number increases and sharp decline occurs especially at pure laminar regime at the Reynolds numbers below 500. The maximum value of 10.78% is associated with the lowest Reynold number, and the averaged uncertainty of Nusselt number is 3.77% for the full flow regime. Figure 3-9 shows the friction factor and corresponding uncertainty of the smooth channel in different flow conditions. The relative uncertainty of friction factor is below 2.8% and slight drop occurs at $Re = 5500$ due to nozzle switch as mentioned above.

Table 3-2 Accuracies of the sensors.

Sensors and parameters	Accuracies
T-type thermocouples, T	$\pm 0.5^{\circ}\text{C}$
Pressure transducer, ΔP	$\pm 0.08\%$ of Reading
Differential pressure transducer, ΔP_1	$\pm 0.15\%$ of Reading
Voltage measurement	1%+2 counts
Current measurement	0.5%+2 counts
Temperature calibrator	$\pm 0.4^{\circ}\text{C}$

Measurement of the velocity field in the PIV technique is based on a statistical approach, which is subject to inherent errors resulting from a finite number of tracer particle, image resolution, and duct size. The uncertainty of PIV tests mainly arises from deficient data due to insufficient particle seeding and non-uniform particle distribution, and correlation anomalies because of unmatched tracer images in the interrogation window. The uncertainty level for the measurement of the velocity is estimated to be less than 3% by following the three rules of thumb aforementioned.

Figure 3-7 The uncertainty of the Reynolds number.

Figure 3-8 The Nusselt number and corresponding uncertainty for the clean channel in the laminar-transitional-turbulent flow.

Figure 3-9 Friction factor and corresponding uncertainty for the clean channel in the laminar-transitional-turbulent flow.

3.3 Measurements Results and Discussion

The CHSA first was installed in a single channel to study the motion and vortex dynamics by a high-speed camera and PIV system. The vorticity profile was captured and plotted in three classical modes, straight mode around $Re = 1600$, large-amplitude flapping mode around $Re = 4600$, and steady deflected mode around $Re = 6000$. Then, the multiple rows of CHSAs, with variable pitch of 50mm, 25mm and 12.5mm, were placed in a seven-channels plate-fin heat exchanger and were examined to obtain the heat transfer and pressure loss characteristics. Because of the entrance disturbance and the coupling effect between neighboring rows, the CHSA inside multiple-channel heatsink would vibrate at little lower flow speed around $Re < 1600$ than when it was in a single channel.

3.3.1 Motion Analysis and Vortex Dynamics of CHSAs

To investigate the motion details of a pair of CHSAs, a high-speed camera was used to capture the vibration mode of CHSAs. Similar to the behavior of an inverted flag with clamped trailing edge and free leading edge, CHSA exhibits two quasi-steady regimes, straight mode, and deflected mode, and a periodical flapping mode appearing between the two quasi-steady regimes as shown in Figure 3-10. At a low flow velocity, the CHSA was unmoving and keeps an initial inclined angle. At an intermediate flow velocity, the CHSA flapped periodically with large amplitude because of the interactions of the aerodynamic force and the solid elastic force during the fluid-structure-interaction process, which destabilized the CHSA and induced its deformation. At high flow velocity, larger aerodynamic force pushed the CHSA to sweep back and remained almost stationary. Figure 3-11 depicts a half cycle of flapping motion of a dual-CHSA around $Re = 4600$ which clearly shows that the vibrations of the closer one and further one are out-of-phase. A cycle of flapping comprises two strokes, half-cycle bending stroke and half-cycle rebounding stroke. If one of CHSA vibrates in the bending stroke, and meanwhile the other one vibrates in the rebounding stroke. The two CHSAs meet at a meeting position where each CHSA has the same displacement but vibrates in different stroke as shown in Figure 3-11 at $t + 0.0020s$. In the out-of-phase flapping state, the vibration amplitude for each CHSA is very large and leads to the largest airflow blockage at their maximum displacement of bending stroke. However, the out-of-phase flapping of dual-CHSA improves the cross flowing between the windward side of one CHSA and leeward side of the other one, which results in a decrease of the streamwise pressure drop and an improvement of the crosswise flow mixing.

Figure 3-10 Stroboscopic images of the CHSA motion as a function of increasing airflow speed; flow direction is left to right.

Figure 3-11 A half cycle of out-of-phase flapping motion for a dual-CHSA captured at $Re = 4600$; flow direction is right to left.

To make a comparison between the natural frequency of CHSA structure and vortex shedding frequency of flow field, a 3D structural simulation was carried out by CATIA V5 and fast Fourier transform(FFT) analysis was conducted based on PIV measurement data. In the structural simulation, STE of CHSA is clamped and FLE of CHSA is free, and the analysis reported that the natural frequency of the CHSA is 129.1 Hz. In addition, the neck of the CHSA bears the largest stress which means the neck probably is the easiest broken part as shown in Figure 3-12(a), and the leading part has a much larger vibration amplitude than trailing part as shown in Figure 3-12(b). Figure 3-13 depicts the FFT results for the velocity variation at 5mm downstream of mid-span of CHSA trailing edge in three different modes, the straight mode around $Re = 1600$, the largest-amplitude flapping mode at $Re =$

4000, and steady deflected mode around $Re = 4600$. A dominant frequency of 121Hz appears as shown in Figure 3-13(d) is quite close to the natural frequency of the structure, which indicates that the resonance occurs between structure and flow mode and results in large-amplitude vibration.

Figure 3-12 The natural frequency is calculated through structural analysis with (a) stress and (b) displacement contour.

Figure 3-13 Fourier analysis results for the velocity variation at 5mm downstream of CHSA trailing edge in (a-b) straight mode at $Re = 1600$, (c-d) flapping mode at $Re = 4000$, and (e-f) deflected mode at $Re = 4600$.

Figure 3-14 and Figure 3-15 depict the vorticity profile and turbulence intensity distribution of the flow field for clean channel and channel with CHSA in straight mode, flapping mode, and deflected mode. The vorticity of clean channel becomes stronger with an increase of flow velocity, but the turbulence intensity is quite low as expected as shown in Figure 3-14. Figure 3-15 shows that the vorticity is highly enhanced by the installation of CHSA. In straight mode around $Re = 1600$, the vorticity enhancement is very limited and focuses on the core flow around the tip of unmoving CHSA. However, in the flapping mode around $Re = 4000$, the large-amplitude vibration induces a much stronger vorticity than the clean channel either for core flow or near-wall flow, and turbulence intensity increases to 5.48% at the CHSA downstream, which highly improves the mixing of secondary flow. When the CHSA switches to deflected mode at a high flow speed of $Re = 4600$, the vorticity strength is further augmented and the highest turbulence intensity of 6.46% appears behind the neck of the CHSA.

Figure 3-14 Contour plots of vorticity strength and turbulence intensity of the flow field for the clean channel at (a) $Re = 1600$, (b) $Re = 4000$, and (c) $Re = 4600$.

Figure 3-15 Vorticity profile and turbulence intensity distribution of the flow field with CHSA in (a) straight mode at $Re = 1600$, (b) flapping mode at $Re = 4000$, and (c) deflected mode at $Re = 4600$.

To obtain a detailed understanding of the vorticity enhancement induced by flapping CHSA, the contour is plotted for a cycle of flapping CHSA around $Re = 4600$ as shown in Figure 3-16. During the bending stroke, the flow kinetic energy converses to elastic strain energy, which also correlates with the evolution of the vortex at leading edges of CHSA. The fully developed vortex starts to shed from the leading edges when the CHSA finishes bending stroke and starts to rebound from the stage of maximum strain energy. Therefore, the flapping CHSA in rebounding stroke induces stronger flow mixing and produces a larger pressure drop than it in bending stroke. Unlike a traditional square flag that creates the vortex at the tip, the CHSA with multiple leading edges and trailing edges allows more vortex to be generated at the same time, which further improve the flow mixing. In addition, because of the narrow neck, the airflow can travel through the gap between leading part and trailing peduncle, which is beneficial to reduce the pressure drop.

Figure 3-16 Vorticity profiles in a cycle of flapping of a CHSA at $Re = 4600$.

3.3.2 Thermal-Hydraulic Performance of CHSAs with $P = 50\text{mm}$

To illustrate the thermal and hydraulic performance of the CHSA, we started with a specific CHSAs configuration with $P = 50\text{mm}$. The Nusselt number and friction factor have been obtained in a continuous Reynolds number range from 300 to 7000. The performance of the CHSAs in one row, two row and three row arrangement were compared with the performance of the clean channel of the plate-fin heat exchanger.

The results shown in Figure 3-17 are the comparison of pressure drop for different cases. The pressure loss for clean channel increases slowly with the Reynolds number at the laminar regime, and the slope gradually becomes larger in transitional and turbulent flow. As expected, the more rows CHSAs were installed inside channels, the larger pressure drop was generated. The results indicate that the introduction of CHSA blocks the airflow and triggers flow disturbance at low Reynolds number ($Re < 1600$) and results in slight faster raising of pressure loss compared to smooth channels. The abrupt increasing occurs around $Re = 1600$ when the CHSA started to vibrate, probably because the large-amplitude flapping produced intensive turbulence and results in a further faster increase of pressure penalty. When the motion of CHSA transferred to deflected mode around $Re = 4800$, the growth rate of pressure drop slightly decreases and approaches a constant.

Figure 3-17 Pressure drop characteristic of CHSAs with $P = 50\text{mm}$. The straight lines represent the critical Reynolds number of onset and ending of flapping mode.

Figure 3-18(a) shows the friction factor of the clean channel, and the channel equipped with CHSA. For the clean channel case, the friction factor decreases with Reynolds number raising in full flow condition. At $Re < 500$, friction factor values are proportional to Re^{-1} , which means a pure laminar flow. The transition to turbulent regime occurs in a gradual way with the range $500 < Re < 2800$. In a fully turbulence regime ($Re > 2800$), the decline rate of friction factor become much slower and more constant than it in laminar and transitional flow. For the channel equipped with CHSAs, the friction factor characteristics changes a lot in the full Reynold number range, which mainly depends on the vibration motion. The friction factor decreases sharply at the Reynolds numbers below 1600 before the onset of CHSA flapping. However, the steep decline is stopped around the $Re = 1600$ at the beginning of flapping, and then the friction factor goes from decreasing to increasing within the range $1600 < Re < 4800$. Moreover, the friction factor returns to decline after the motion transition to deflected mode and tends to

liner decrease. Similar with pressure loss characteristics, the friction factor of more rows of CHSAs are uniformly higher than less rows of CHSAs. In addition, installation of more rows of CHSA in channel moves the crest and trough of the curve to lower Reynolds number range, which means the coupling between multiple-row CHSAs excites them to vibrate at lower flow speed and also result in higher turbulence intensity in early flow stage. Figure 3-18(a) shows that the critical Reynolds number is not clearly marked, which suggests that the transition from laminar to turbulent flow is produced steadily without being possible to determine the point where the transition takes place. The friction factor augmentation is defined by the ration between f_c and f_0 at the same Reynolds number. Figure 3-18(b) shows friction factor augmentation produced by multiple rows of CHSAs. The friction factor augmentations go up as flow velocity increasing but turn down at high Reynolds number within deflected vibration of CHSAs. The maximum augmentations occur around $Re = 5330$ for one-row layout, $Re = 5496$ for two-row layout, and $Re = 5590$ for three-row layout with 2.21, 3.30 and 4.41 respectively. Additionally, the decline slope is quite constant at a pure turbulent flow at $Re > 6000$.

Figure 3-18 (a) Friction factor and (b) its augmentation of CHSAs with $P = 50\text{mm}$.

Figure 3-19(a) shows the Nusselt number results vs. the Reynolds number for clean channel and channel with CHSAs in the laminar, transitional, and turbulent regime. For the clean channel, in a pure laminar flow ($Re < 500$), heat transfer was produced under mixed convection. The Nusselt number increases with a constant rate at $500 < Re < 2300$, and a smooth transition to turbulent flow was observed at Reynolds numbers above 2300. For the cases with CHSAs, the installation of CHSAs early triggers the transition to turbulent flow and improve the flow mixing, which hence enhances the heat transfer performance. Growth rate of Nusselt number slightly increases at the transition from flapping mode to deflected mode and decreases after the transition with a tendency to clean channel. The Nusselt number augmentation is defined by the ration between Nu_c and Nu_0 at the same Reynolds number. Figure 3-19(b) shows the Nusselt number augmentation for multiple-row CHSAs. Since CHSAs accelerate the transition to turbulent flow, they show a considerable heat transfer enhancement particularly in flapping mode. The highest augmented Nusselt numbers of 1.56 for one-row arrangement, 1.87 for two-row arrangement, and 2.07 for the three-row arrangement of CHSAs, occur around $2100 < Re < 2500$ with moderate flapping amplitude. At $2500 < Re < 4400$, the flapping amplitude gradually increases and up to its largest value before the transition to the deflected mode. However, improved flow mixing induced by large-amplitude flapping in turbulent flow does not further contribute to heat transfer enhancement and the Nusselt number augmentation turns to decrease with Reynolds number. In addition, the Nusselt number augmentation experiences a transient increase at the motion transition stage from the flapping to the deflected mode but keeps decreasing at steady deflected mode.

Figure 3-19 (a) The Nusselt number and (b) its augmentation of CHSAs with $P = 50\text{mm}$.

Figure 3-20 shows the overall thermal-hydraulic performance of the enhancement cases with CHSAs. The performances were enhanced most at 38.98% for the one-row set, 46.79% for the two-row set, and 57.58% for the three-row set around $Re = 2000$, where the CHSAs flapped with moderate amplitude and the clean channel was involved in the transitional flow. The results indicate that heat transfer enhancement cannot offset the friction factor increase at $Re > 2000$, therefore, the augmented overall performance turn to decrease rapidly. The decline of performance experiences a short pause before CHSAs turn to deflected mode at $4100 < Re < 4800$ and continues with a slight rate at steady deflected mode. Furthermore, the performance is approaching to 1 at a high Reynolds number at $Re > 6000$, which means the enhancement is very limited at a fully turbulent flow. The analysis allows one to conclude that the use of CHSAs are not recommended at high Reynolds number range, and the best thermal hydraulic performance can be obtained at the transitional regime with moderate flapping amplitude.

Figure 3-20 Overall thermal-hydraulic performance of CHSAs with $P = 50\text{mm}$.

Air effectiveness is calculated to estimate the saturation level of the plate-fin heat exchanger with and without installation of CHSAs as shown in Figure 3-21. For the clean channel, the results show that the air effectiveness decreases rapidly from 70.43% to 32.84% in laminar regime at $Re < 2300$, and then turns to slowly decreasing from 32.84% to 26.13% in turbulent flow. For the channel equipped with multiple-row CHSAs, the air effectiveness is higher than clean channel as expected, and but it is still far away from saturation at the favorable flow regime around $Re = 2000$. The highest air effectiveness of about 60% was obtained for a three-row installation at $Re = 2000$, therefore, the analysis allows more rows of CHSAs to be installed in the same heatsink with a smaller pitch to further improve the heat transfer enhancement.

Figure 3-21 Air effectiveness of CHSAs with $P = 50\text{mm}$.

3.3.3 Thermal-Hydraulic Performance of CHSAs with $P = 25\text{mm}$

In order to further enhance the heat transfer for the given plate-fin heat exchanger, up to six rows CHSAs were placed in the heatsink with a pitch of $P = 25\text{mm}$. Thermal-hydraulic tests have been conducted at $300 < Re < 7000$. The performance of the CHSAs in the arrangement of one row, two rows, three rows, four rows, five rows and six rows were compared with the clean channel one.

Figure 3-22 shows the pressure drop characteristics for multiple rows of CHSAs inserted in the heatsink. Similar to the cases with a larger pitch, the more rows CHSAs were placed in the heatsink, the larger pressure lost through the channels. However, the increment of pressure drop becomes smaller at the same Reynolds number if more rows CHSA were equipped in channels. As analyzed above, the insertion of flapping CHSAs enhanced the turbulence intensities and, hence, the pressure drop increases quickly at $1450 < Re < 4900$. However, the growth slope becomes a little lower after motion transiting to deflected mode at $Re > 4900$ and approaches to be constant at a steady fully turbulent flow.

Figure 3-22 Pressure drop characteristic of CHSAs with $P = 25\text{mm}$.

The results shown in Figure 3-23(a) are the friction characteristics of CHSAs with a pitch of $P = 25\text{mm}$. Like the CHSAs installed in channels with a larger pitch, the friction factor declines significantly at $Re < 1450$ in straight mode, and gradually turns to increasing when the CHSAs start to flap. However, the friction factor returns to decrease after the mode transition to the deflected mode and the decline rate approach to be constant at $Re > 6000$. Figure 3-23(b) depicts the augmented friction factor for the cases with a different row of CHSAs. The friction factor augmentation grows with Reynolds number but turn to decreasing at the vibration transition to deflected mode. The maximum augmentation of 7.98 was obtained for six-row CHSAs at $Re = 4900$. The decrease rate is approaching a constant at the fully turbulent flow at $Re > 5600$. In addition, the decline rate becomes higher if more rows of CHSAs were installed in the channels and the six-row configuration holds the highest decline rate.

Figure 3-23 (a) Friction factor and (b) its augmentation of CHSAs with $P = 25\text{mm}$.

Figure 3-24 shows the heat transfer results and the thermal enhancement of CHSAs with a pitch of $P = 25\text{mm}$. The intensive installation of multiple CHSAs further improved the flow mixing and hence augmented the heat transfer performance. Similar to the cases with a pitch of 50mm , the Nusselt number increases smoothly in the flapping mode and a short rapid increase occurs at the stage of motion transition to deflected mode around $Re = 4900$. Figure 3-24(b) depicts the Nusselt number augmentation for multiple-row of CHSAs with a pitch of 25mm . The best thermal performance for each case was obtained around $2100 < Re < 2600$, and more rows of CHSAs were installed in heatsink along streamwise extending the Reynolds number range at which ultra-high augmentation was acquired. The largest Nusselt number augmentations of 1.93 for two-row layout, 2.14 for the three-row layout, 2.30 for the four-row layout, 2.41 for the five-row layout and 2.57 for the six-row layout were observed in the flapping mode with moderate vibration amplitude. Additionally, a transient increase of the augmented Nusselt number was observed at the stage of motion transition to the deflected mode, and since then the enhancement keeps decreasing at a steady deflected status.

Figure 3-24 (a) The Nusselt number and (b) its augmentation of CHSAs with $P = 25\text{mm}$.

Figure 3-26 shows the overall thermal-hydraulic performance of the augmented channels with CHSAs arranged in the pitch of $P = 25\text{mm}$. The overall capabilities were enhanced at most by 50.16% for the two-row set, 57.60% for the three-row set, 59.76% for the four-row set, 61.83% for the five-row set, and 67.64% for the six-row set of CHSAs around $Re = 1900$, where the CHSAs vibrated with moderate amplitude. The results indicate that thermal enhancement cannot offset the increase of the friction factor at $Re > 2000$, therefore, the enhancement turns to decrease rapidly. Moreover, the decline of the performance experiences a short pause before CHSAs turning to a deflected flutter around $4000 < Re < 4900$ and continues to decrease slowly in fully turbulent flow.

Figure 3-25 Overall thermal-hydraulic performance of CHSAs with $P = 25\text{mm}$.

Air effectiveness was also estimated to determine if the augmented heat exchanger with CHSAs is approaching saturation as shown in Figure 3-26. The results depict that, around $Re = 1900$, the highest value of 67% was obtained for a six-row layout, which still provides potential room to further improve the heat transfer performance of heat exchanger at a favorable flow regime.

Figure 3-26 Air effectiveness of CHSAs with $P = 25\text{mm}$.

3.3.4 Thermal-Hydraulic Performance of CHSAs with $P = 12.5\text{mm}$

In order to exploit the best performance of the heatsink with CHSAs, the three rows, six rows, nine rows, and twelve rows CHSAs were placed in the channels with a pitch of $P = 12.5\text{mm}$. Thermal tests and pressure measurement have been conducted at $300 < Re < 7000$, and the thermal-hydraulic performance was compared with the clean channel. Based on the observation, the CHSAs can vibrate at a lower flow velocity of $Re = 1300$ than they are arranged in the larger pitch, but turn to deflected mode at a higher flow speed of $Re = 5000$.

Figure 3-27 shows the pressure drop characteristics for multiple rows of CHSAs installed in the heat exchanger. Twelve-row CHSAs produced the largest pressure drop of up to 770 Pa around $Re = 7000$. Similarly, the increment becomes smaller at the same Reynolds number if more rows CHSA were placed in the heatsink. The insertion of oscillating CHSAs enhanced the flow mixing and hence the pressure drop increased very fast at $1300 < Re < 5000$, however, the rate of increase becomes a little lower after the mode switching to steady mode at $Re > 5000$, and approaches being constant.

Figure 3-27 Pressure drop characteristic of CHSAs with $P = 12.5\text{mm}$.

The results shown in Figure 3-28(a) are the friction characteristics of CHSAs with a pitch of $P = 12.5\text{mm}$. The friction factor decreases very fast at $Re < 1300$ when the CHSAs stayed in the straight mode, and gradually turns to increase as the CHSAs start to vibrate. However, the friction factor returns to decline after the mode transition to deflected mode. Figure 3-28(b) depicts the friction factor augmentation for the cases with different rows of CHSAs. The augmented friction factor increases with the increasing of Reynolds number in the flapping mode of CHSAs, but turn to decrease when the mode was switched to steady deflected. The maximum augmentation of 11.60 was obtained for twelve-row CHSAs at $Re = 5550$. The decreasing rate is approaching to a constant at the fully turbulent flow at $Re > 6000$. Furthermore, the rate of declining becomes higher if more rows of CHSAs are installed in the channels and the twelve-row configuration holds the highest one.

Figure 3-28 (a) Friction factor and (b) its augmentation of CHSAs with $P = 12.5\text{mm}$.

Figure 3-29 shows the Nusselt number result and its augmentation of CHSAs with a pitch of $P = 12.5\text{mm}$. CHSAs arranged in a smaller pitch still can improve the thermal performance. The best heat transfer performance for multiple row of CHSAs was obtained around the range of $2000 < Re < 2600$ which is similar to the cases with $P = 25\text{mm}$. The largest Nusselt number augmentations of 1.96 was obtained in flapping mode with moderate vibration amplitude for the three-row arrangement, 2.31 for the six-row arrangement, 2.73 for the nine-row arrangement, and 2.95 for the twelve-row arrangement.

Figure 3-30 plots the overall thermal-hydraulic performance of the heatsink equipped with CHSAs installed in a pitch of $P = 12.5\text{mm}$. The performances were augmented at most by 46.17% for the tree-row layout, 53.0% for the six-row layout, 63.44% for the nine-row layout, and 66.36% for the twelve-row layout around $Re = 1900$, where the CHSAs flapped with moderate amplitude. The results indicate that the heat transfer enhancement cannot offset the increase of pressure loss around $Re > 2000$, therefore, the overall performance decline fast at a high flow speed.

Figure 3-29 (a) The Nusselt number and (b) its augmentation of CHSAs with $P = 12.5\text{mm}$.

Figure 3-30 Overall thermal-hydraulic performance of CHSAs with $P = 12.5\text{mm}$.

Air effectiveness was calculated to evaluate the saturation level of the heat exchanger equipped with CHSAs arranged in a pitch of $P = 12.5\text{mm}$ as shown in Figure 3-31. The results showing the highest value of 70% was obtained for twelve -row CHSAs at $Re = 1900$, which is slightly higher than the case with $P = 25\text{mm}$.

Figure 3-31 Air effectiveness of CHSAs with $P = 12.5\text{mm}$.

3.3.5 Pitch Effects on Thermal-Hydraulic Performance of CHSAs

To compare the difference of coupling between neighboring CHSAs, the vibration motion was recorded for CHSAs with a different pitch. In addition, to figure out the optimal arrangement pitch of CHSAs, the heat transfer and friction characteristics of three cases with a pitch of 50mm, 25mm, and 12.5mm are plotted and compared together.

From the visualization results, it was found that the stronger coupling effects between neighbor rows of CHSAs with a smaller pitch enlarge the flapping range. The range of flapping mode is $1600 < Re < 4800$ for CHSAs with $P = 50\text{mm}$, $1450 < Re < 4900$ for CHSAs with $P = 25\text{mm}$, and $1300 < Re < 5000$ for CHSAs with $P = 12.5\text{mm}$. Another attractive phenomenon is that the CHSAs downstream can vibrate at a lower flow velocity than those upstream, and because of the flow disturbance induced by CHSAs upstream. In the flapping mode, the CHSAs downstream have a larger amplitude than those upstream. Additionally, the motion of CHSAs downstream switch to a deflected mode at a higher flow velocity than those upstream. The smaller pitch of CHSAs are arranged with, the more obviously this phenomenon is observed.

The pressure drop of CHSAs with the smallest pitch produces the largest pressure drop as expected, and the increasing rate is also larger than the CHSAs with a larger pitch as shown in Figure 3-32. Figure 3-33 shows the friction factor and its augmentation compared to the clean channel. The friction factor of CHSAs with smaller pitch is larger than those with the larger pitch at all range of Reynolds. The Nusselt number of CHSAs with smaller pitch is higher than those with a larger pitch, but the case with $P = 12.5\text{mm}$ does not show significant augmentation compared to the cases with $P = 25\text{mm}$ as shown in Figure 3-34. Therefore, the overall performance of CHSAs with $P = 12.5\text{mm}$ is quite

close to those with $P = 25\text{mm}$ as shown in Figure 3-35. Figure 3-36 depicts the saturation of heatsink equipped with CHSAs in a different arrangement. The results show that there is still plenty of room to improve the heat transfer at a high Reynolds number, but it may be difficult to get further improvement by reducing the pitch in transitional flow at low Reynolds number.

Figure 3-32 Pressure drop characteristic of CHSAs with a different pitch.

Figure 3-33 (a) Friction factor and (b) its augmentation of CHSAs with a different pitch.

Figure 3-34 (a) The Nusselt number and (b) its augmentation of CHSAs with a different pitch.

Figure 3-35 Overall thermal-hydraulic performance of CHSAs with a different pitch.

Figure 3-36 Air effectiveness of CHSAs with a different pitch.

3.4 Summary

A comprehensive experimental investigation has been carried out to explore a novel air-cooling method with an hourglass-shaped self-agitator to enhance the thermal-hydraulic performance of a plate-fin heat exchanger. Inspired by the unique caudal fin of a fish, the CHSA can flap periodically due to fluid-structure-interaction, which enhances the thermal-hydraulic performance. An experimental setup was built to determine the heat transfer performance in laminar-transitional-turbulent flow, and another single channel testbed was fabricated to investigate the vibration motion and vortex dynamics behind a CHSA in a channel flow by using a PIV system. The effects of the pitch ($P = 50\text{mm}, 25\text{mm}$ and 12.5mm) between the neighboring row of CHSAs on heat transfer, friction factor, and overall performance were evaluated in great detail based on quantitative measurements. The following conclusions can be drawn:

1. With increasing of flow velocity, two quasi-steady regimes, straight mode and deflected mode, and a flapping mode appearing between the two quasi-steady modes were clearly observed. In addition, the out-of-phase phenomenon was observed in the flapping mode of a dual-CHSA vibration.
2. It was found that the range of the flapping mode can be enlarged by reducing the pitch of CHSAs because of the stronger coupling effects between the neighboring rows of CHSAs. Furthermore, the CHSAs placed in the rear rows could lose stability and start to flap in lower flow velocity due to the flow disturbance induced by CHSAs upstream.
3. Based on the PIV measurement, intensive vortices generated by multiple edges of a CHSA highly enhanced the flow mixing and turbulence intensity in the

flapping mode, especially during rebounding stroke. Resonance occurs at large-amplitude flapping where the vortex shedding frequency is quite close to the natural frequency of the structure.

4. The heat transfer in the augmented channels with flapping CHSAs can be enhanced considerably at the transitional regime at the range of $2000 < Re < 2600$. Higher Nusselt number augmentation can be achieved by inserting more rows of CHSAs and arranging with a smaller pitch. The best performance was obtained by twelve-row CHSAs with $P = 12.5\text{mm}$, which is up to about 200% over the clean channel.
5. The overall thermal-hydraulic performance was enhanced by installing CHSAs, but cannot be further improved by reducing the pitch. The best improvement of 68% was obtained with six-row CHSAs in a pitch of 25mm. Therefore, a pitch less than 25mm is not recommended to be used in this given plate-fin channel if pressure loss is a concern.
6. CHSAs operating in deflected mode result in large airflow blockage at high Reynolds number of $Re > 5000$, and the heat transfer enhancement cannot offset the friction increase. Therefore, the overall performance decreases rapidly to 1.0 which means the use of CHSAs in their deflected mode would not improve the performance of heat exchanger at all.

Chapter 4 Shape, Structure and Configuration Optimization of Self-Agitators

4.1 Self-Agitators with Punched Holes

In order to further reduce the pressure loss and improve the overall thermal-hydraulic performance of heatsink equipped with self-agitators, some holes with variable diameter are punched on the conventional rectangular-shaped self-agitators by using hole punch cutter, as shown in Figure 4-1.

Figure 4-1 Hole punch cutter

The diameter of holes is chosen as 3.5 mm in Figure 4-2(a), 2.5 mm in Figure 4-2(b), 1.5 mm in Figure 4-2(c), 1.0 mm in Figure 4-2(d)&(e), or 0.5 mm in Figure 4-2(f). Each design has been installed in the heatsink channel, but not all agitators can vibrate in channel flow. The self-agitators with holes of 3.5 mm, 2.5 mm, 1.5 mm and 1.0 mm can vibrate, but those with holes of 1.0 mm in the parallel setup as shown in Figure 4-2(d) and 0.5 mm cannot vibrate.

Figure 4-2 Self-agitators with punched holes

The thermal-hydraulic test has been carried out for the self-agitators with 2.5 mm-diameter holes to determine the heat transfer performance and pressure drop characteristics. Figure 4-3 shows the pressure drop and friction factor performance for channel flow with hole-punched self-agitators and rectangular self-agitators (RSA). The results indicate that the punched holes reduce the pressure and friction factor at $Re > 3000$, but would not help at low flow velocity.

Figure 4-3 Pressure drop and friction factor performance of self-agitators with holes

Figure 4-4 shows the heat transfer improvement and overall thermal-hydraulic performance of rectangular self-agitators and hole-punched self-agitators. The experimental results indicate the heat transfer augmentation decreases for the case with punched holes because the structure vibration frequency decrease and keeps away from channel preferred frequency. Therefore, the overall thermal-hydraulic performance decreased a lot at $Re < 3000$.

Figure 4-4 Nusselt number and overall thermal-hydraulic performance of hole-punched self-agitators

4.2 Self-Agitators fabricated in multi-thickness

Novel design with multi-thickness is developed to enhance the overall performance as shown in Figure 4-5. The leading part is fabricated with Kapton film 50HN and the trailing part is fabricated with 500 HN. The leading part can vibrate with large amplitude during FSI process, but the trailing part keeps unmoved due to the thicker structure. The two parts connected with each other with super glue.

Figure 4-5 Geometry of multi-thickness self-agitator

Thermal tests have been conducted and pressure loss is measured to evaluate the enhancement performance. Figure 4-6 reports the pressure penalty and corresponding friction factor at the same Reynolds number. The partially vibrating motion can significantly reduce the pressure drop and friction factor compared to single thickness self-agitators at full flow regime.

Figure 4-6 Pressure drop and friction factor characteristics of multi-thickness self-agitators

Figure 4-7 shows the Nusselt number and overall thermal enhancement factor of channel flow equipped with multi-thickness self-agitators. The results show that the multi-thickness self-agitators cannot demonstrate higher heat transfer enhancement, but the overall performance is higher than traditional RSA at $Re > 3000$. The smaller vibrating structure results in high frequency flapping and smaller sweeping area, however, this mechanism would not further improve the flow mixing performance.

Figure 4-7 Heat transfer and overall thermal performance of multi-thickness self-agitators

4.3 Double Pairs of Rectangular Shaped Self-Agitators

Two types of double pairs of RSA have been designed and developed to further improve the flow mixing as shown in Figure 4-8. The larger one with a width of 3mm and the smaller one with a width of 2.5mm.

Figure 4-8 Geometry of Single pair (a), double pairs (b) & (c) of self-agitators

Figure 4-9 shows the comparison of pressure drop and friction factor for the clean channel, single pair self-agitators, larger double-pair self-agitators (DORSA), and smaller double-pair self-agitators. The DORSA vibrates with high frequency, but the airflow blockage is too large so that the friction increases a lot compared to the single pair case. However, the smaller one demonstrates lower friction at $3000 < Re < 5000$. Therefore, due to large pressure loss and limited heat transfer improvement, the overall thermal-hydraulic performance of DORSA is lower than RSA in full flow regime as shown in Figure 4-10.

Figure 4-9 Pressure drop and friction factor of DORSAs

Figure 4-10 Nusselt number and overall thermal performance of DORSAs

4.4 Inner-Outer/Cut-Out Self-Agitator

Inspired by an Inner-Outer beam [120] used in energy harvest application, a novel inner-outer or cut-out self-agitator is designed and fabricated as shown in Figure 4-11. The RSA is punched by a hole cutter mentioned in section 4.1 and reshaped to an inner-outer self-agitators, and installed in a channel of the heatsink. The tips of inner and outer parts are painted by epoxy to increase the mass roughly. The most interesting phenomena observed is that the inner part and outer part vibrate out of phase. The Inner-outer self-agitators are not fabricated massively because of the manufacturing difficulty, so the thermal-hydraulic test has not carried out until the accurate fabrication can be achieved in the future.

Figure 4-11 Inner-Outer self-agitator

4.5 Optimization of Angle of Attack

The angle of attack (AoA) is an important configuration parameter for self-agitator, which has a significant impact on the self-agitator flapping motion. A pair of traditional rectangular-shaped is installed in the transparent single channel and a high-speed camera is used to capture the vibration motion of self-agitators in channel flow as shown in Figure 4-12. Three AoAs, 25° , 35° , 45° are chosen to do parameter studies to figure out the AoA effects on flapping motion. The observation indicates that the smaller AoA makes self-agitators easier to vibrate and enlarges the vibration amplitude.

Figure 4-12 Vibration motion of self-agitators at different airflow velocity

Pressure drop is tested and compared for different cases with variable AoAs as shown in Figure 4-13. The smaller AoA of self-agitators demonstrates lower pressure loss and enlarge the flapping out-of-phase range. The self-agitators installed with 45° AoA hardly vibrate periodically, and their motion changes gradually from straight mode to deflected mode with the increasing flow velocity.

Figure 4-13 Pressure drop of self-agitators with different AoAs

4.5 Summary

In this chapter, four optimizations have been developed on shape, structure, and configuration. The hole-punched self-agitator is designed to reduce pressure drop by decreasing the airflow blockage. The multi-thickness self-agitator is designed to improve heat transfer performance by increasing structure vibration frequency and reduce the pressure drop by using partially vibration structure. The double-pair self-agitator is invented to enhance the thermal performance by adding more vibrating agitators and stronger coupling effect between two pairs of self-agitators. The most promising design would be the inner-outer self-agitator, which probably features many advantages. The inner-outer self-agitator flapping in out-of-phase may enhance the flow mixing and create more 3D vortices to disturb both heated channel walls and reduce the pressure penalty due to lower flow blockage and out-of-phase flapping motion. The above design may be optimized further to make another breakthrough with the rapid development of smart materials and microfabrication techniques.

Chapter 5 Conclusions

In this work, an airfoil-based self-agitator, a bio-inspired rectangular-shaped self-agitator, and a novel caudal-fin-inspired hourglass-shaped self-agitator have been developed to enhance the heat transfer performance and reduce the pressure drop. Comprehensive numerical and experimental investigations have been carried out to explore the heat transfer and pressure drop characteristics of the plain-fin heat exchanger with self-agitators. The self-agitators can flap periodically due to fluid-structure-interaction, which enhances thermal-hydraulic performance. Several experimental setups are built to determine the heat transfer performance and pressure loss in laminar-transitional-turbulent flow. PIV system is used to measure the velocity field and study the vortex dynamics of channel flow with or without self-agitators. The effect of geometry, pitch, and configuration on heat transfer, friction factor, and overall performance is evaluated in great detail based on quantitative measurements. The following conclusions can be drawn:

1. The delta-shaped AFA generates strong longitudinal vortices, thus enhancing heat transfer more than a rectangular shaped AFA and also resulting in a lower pressure drop.
2. The optimal chord length for a RAFA is 7 mm. This chord length maximizes heat transfer.
3. Increasing the chord length to pursue higher heat transfer performance for DAFA comes with diminishing returns. A DAFA with 5 mm chord length is optimal, enhancing heat transfer significantly but costing fewer materials than one with higher chord length.

4. The double arrangement for DAFAs has the best heat transfer performance with the lowest pressure penalty, so out of the arrangements examined in this study, it is the best design for heat transfer enhancement in a heat exchanger.

5. Heat transfer continues to improve with more than two rows of DAFAs but at the cost of additional pressure drop. No more than two rows of agitators are necessary if pressure loss is a concern.

6. Matching the frequency of structural motion to that of the dominant mode can create resonance to enhance the fluid mixing.

7. FSI process of the self-agitator can improve the convective heat transfer significantly and that further enhancement can be achieved by approaching the preferred frequency of the vorticity field.

8. Self-agitator oscillation at the preferred frequency is achieved, and experimental results demonstrate that the heat transfer can be enhanced by 120% without additional pumping power requirements while the Nusselt number can be improved by 200% with the same Reynolds number compared to the clean channel.

9. With increasing flow velocity, two quasi-steady regimes, a straight mode and deflected mode, are observed. An unsteady flapping mode appearing between the two quasi-steady modes is also clearly observed. In addition, the two CHSAs in a dual-CHSA configuration are observed to flap out-of-phase phenomenon in the flapping vibrational mode.

10. Because of the stronger coupling effect between neighboring rows of CHSAs, the range of the flapping mode can be enlarged by reducing the pitch length between rows. Furthermore, the CHSAs placed in the rear rows lose stability and start to flap at a lower flow velocity due to the flow disturbance induced by CHSAs upstream.

11. PIV measurements show that vortices generated by the multiple edges of a CHSA strongly enhance flow mixing and turbulence intensity in the flapping mode, especially during the rebounding stroke. Resonance occurs at large-amplitude flapping where the vortex shedding frequency is quite close to the natural frequency of the structure.

12. The heat transfer in the augmented channels with flapping CHSAs can be enhanced considerably in the transitional regime ($2000 < Re < 2600$). Higher Nusselt number augmentation can be achieved by inserting more rows of CHSAs with a smaller pitch. The best performance is obtained by twelve-row CHSAs with $P = 12.5$ mm, which constitutes a performance improvement of up to about 200% over the clean channel.

13. The overall thermal-hydraulic performance is enhanced by installing CHSAs, but reducing the pitch beyond 25 mm does not result in significant further improvement. The best improvement of 68% is obtained for six-row CHSAs with a pitch of 25 mm. Therefore, if pressure loss is a concern, a pitch length of less than 25 mm is not recommended for this plate-fin channel.

14. CHSAs operating in the deflected mode result in considerable airflow blockage at high Reynolds numbers ($Re > 5000$), and the heat transfer enhancement does not offset the friction increase. In this flow regime, the overall performance decreases rapidly to 1.0 which means that the use of CHSAs in their deflected mode does not improve the performance of heat exchanger.

Based on the analyses and conclusions above, the proposed self-agitators would be a promising method to improve the heat transfer performance of plate-fin heat exchanger in many industrial applications, such as electronics cooling, data center, power plant, battery thermal management, chemical engineering, and HVAC. The rectangular-shaped self-agitator can be employed in some application requiring ultra-high heat transfer performance, but CHSA would be a better alternation if the pressure drop is a big concern. In addition, the current design may be further improved through optimization of structure and material. Future work should be focused on designing a novel structure with smart material which can demonstrate channel preferred frequency to enhance heat transfer and favorable flapping motion to reduce pressure drop.

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