



# Single-phase thermal and hydraulic performance of embedded silicon micro-pin fin heat sinks using R245fa

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## ABSTRACT

Aggressive thermal management strategies such as liquid cooling have become essential for high-performance three-dimensional (3D) integrated circuit (IC) chips. Micro-pin fin arrays integrated between stacks can provide superior thermal performance with relatively less pumping power compared to microchannel cooling. In this work, we experimentally studied the single-phase heat transfer and pressure drop characteristics of micro-pin fin arrays. Three different samples consisting of 31–131 rows of cylindrical micro-pin fins with pin diameters  $D_h = 45$ – $100\ \mu\text{m}$ , center-to-center pin spacings  $S = 74$ – $298\ \mu\text{m}$ , and pin height  $H_f \sim 200\ \mu\text{m}$  were tested. Dielectric fluid R245fa was used as the working fluid with mass flow rates  $\dot{m} = 14.7$ – $181.6\ \text{g/min}$  and corresponding Reynolds numbers  $Re = 35$ – $481.3$ . The heat fluxes ranged from  $2.5\ \text{W/cm}^2$  to  $48.7\ \text{W/cm}^2$ , and the inlet fluid temperature was maintained at ambient temperature in the range of  $22.2$ – $25.3\ ^\circ\text{C}$ . The local heater temperature distributions, average heat transfer characteristics, and pressure drops for various geometries of the embedded microfluidic pin–fin arrays were determined. The experimentally determined heat transfer coefficient varied with both the mass flow rate and pin spacing with an averaged heat transfer coefficient of up to  $18.2\ \text{kW/(m}^2\cdot\text{K)}$ . Full-scale conjugate simulations with a turbulence model were conducted using ANSYS Fluent to validate the experimental results for the three cases. A comparison with the numerical model showed mean absolute errors of 9.1% for the heat transfer and 14.3% for the pressure drop.

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## 1. Introduction

As the demand for high-performance microprocessor computing increases, three-dimensional (3D) stacked integrated circuit (IC) chips have received a considerable amount of attention because of their broad potential benefits, such as multifunctionality, increased performance, reduced power, small form factor, reduced packaging, increased yield and reliability, flexible heterogeneous integration, and reduced overall costs [1]. However, the decreased global wire length and increased wire-limit clock frequency achieved by through-silicon via (TSV) technology [2–4] have been achieved at the cost of very high heat dissipation rates,

which limit the performance according to thermal management constraints and requirements [5].

There have been many studies on microfluidic concepts using a variety of cooling configurations including microchannels [6,7], jet impingement [8,9], spray cooling [10,11], and 3D manifold [12] since Tuckerman and Pease [13] introduced the concept of microchannels in 1981. A micro-pin fin structure embedded in a stacked chip is a favorable solution for thermal management in 3D stack integration as it can provide higher heat transfer performance by enhanced fluid mixing, eliminate the use of thermal interfacial material (TIM), and requires relatively less pumping power compared to microchannel-type cooling configuration [14]. In the early 2000s, Moores and Joshi [15] investigated the heat transfer performance of relatively small-scale pin fin arrays, which are applicable for the thermal management of electronics. They used staggered pin fins with pin diameters  $D_f = 3.67$ – $3.84\ \text{mm}$  and pin heights  $H_f = 2$ – $4\ \text{mm}$ , and investigated the

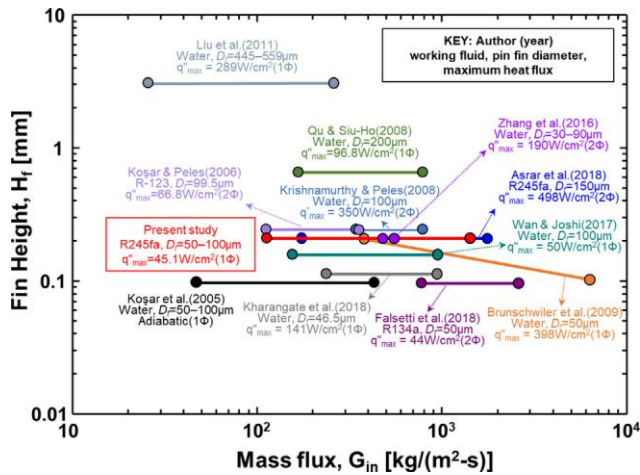
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**Fig. 1.** Relevant previous studies for a wide range of pin-fin heights, mass velocities, and working fluids for micro-pin fin arrays.

of flow rates and heat fluxes. The experimental results were compared with computational fluid dynamics (CFD) simulations performed using commercially available software, ANSYS Fluent.

## 2. Experimental methods

### 2.1. Experimental apparatus

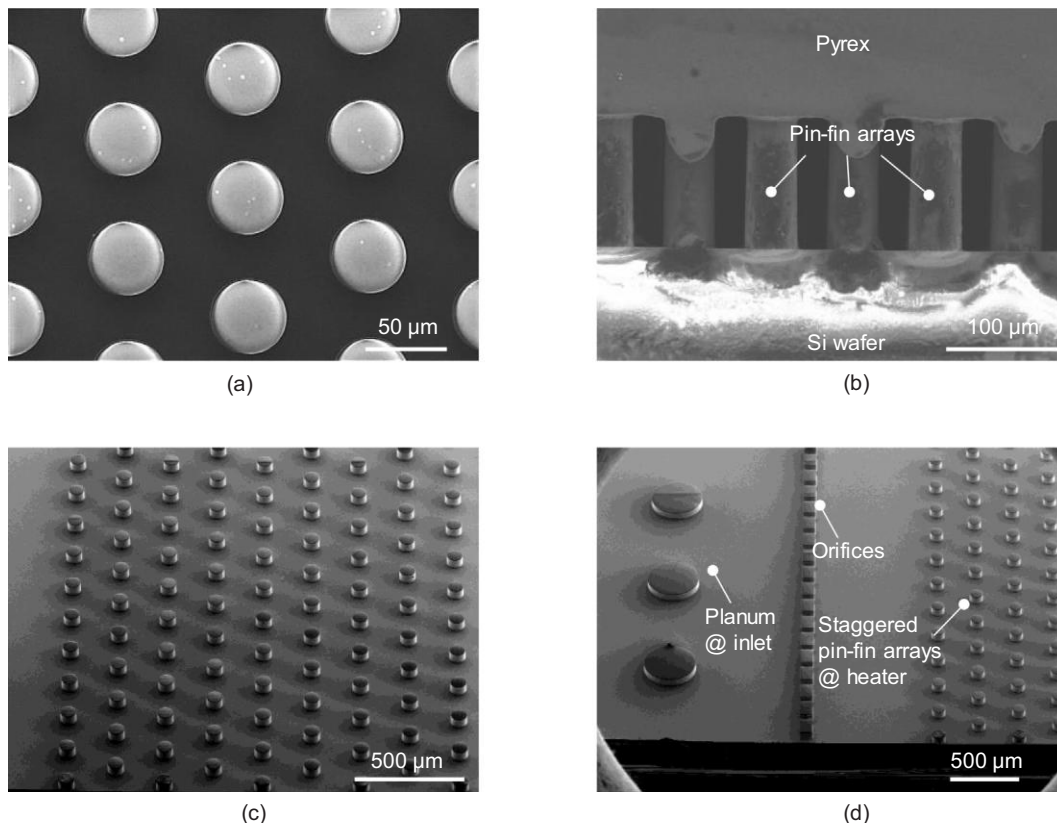
#### 2.1.1. Fabrication of a micro-pin fin test module

The test module consists of a microfabricated silicon substrate with a Pyrex cover, a polyether ether ketone (PEEK) plastic sample holder, and a copper plate for compressing the sample on the

holder. We fabricated the micro-pin fins on a 4-in silicon wafer as a substrate. An isotropic reactive-ion etching process was performed twice to define shallow alignment marks on the top and bottom surfaces of the substrate, followed by an anisotropic deep reactive-ion etching process to fabricate staggered pin-fin arrays, inlet and outlet plenums, and pressure ports over an area of  $2.4 \times 1.9 \text{ cm}^2$ . Fig. 2 shows scanning electron microscopy (SEM) images of the microfabricated staggered pin-fin arrays. The tops of the micro-pin fin arrays were closed using a 500- $\mu\text{m}$ -thick Pyrex wafer, which was anodically bonded to the silicon substrate to provide optical access to the flow while avoiding any fluid leakage during the test. Openings for the fluid inlet, outlet, and pressure measurements were drilled through the Pyrex cover with a photoresist layer having a thickness of 3–5  $\mu\text{m}$  for passivation. Anodic bonding was performed under operating conditions of 350  $^\circ\text{C}$  and 900 V for 6 min. After the bonding process, a 500-nm-thick silicon dioxide ( $\text{SiO}_2$ ) layer was deposited onto the backside of the silicon substrate for electrical insulation; then, a titanium/gold (20 nm/500 nm) layer was deposited to fabricate two gold serpentine heaters and 14 resistance temperature detectors (RTDs). The temperature distribution of the heated surface was observed using either the 14 RTDs (for CASE 1) or a FLIR A655sc infrared (IR) thermal imaging camera (for CASE 2 and CASE 3) with a resolution of  $480 \times 640$  pixels. Both the IR and RTD measurements were calibrated using the electrical resistance thermometry of the patterned gold layer. Details of the fabrication steps are shown in Fig. 3.

#### 2.1.2. Flow loop facility

A system-level schematic of the flow apparatus and test module is depicted in Fig. 4. The working fluid was delivered to the micro-pin fin test module using a magnetically driven Micropump GJ-N23 gear pump. A Micromotion CMFS010 Coriolis mass flow meter and customized in-line immersion preheater were used to measure the



**Fig. 2.** SEM images of a staggered pin-fin array: (a) top view ( $D_f \sim 46.5 \mu\text{m}$ ), (b) side view ( $H_f \sim 110 \mu\text{m}$ ), and (c, d) 45° view.

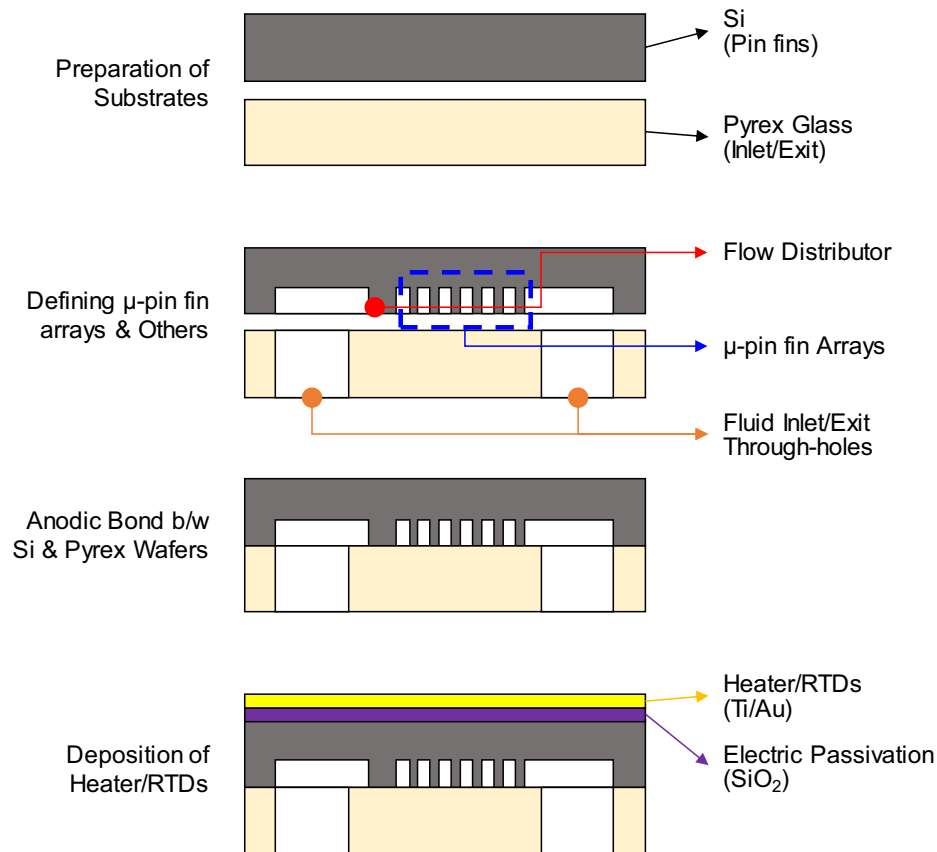


Fig. 3. Fabrication steps of the test samples.

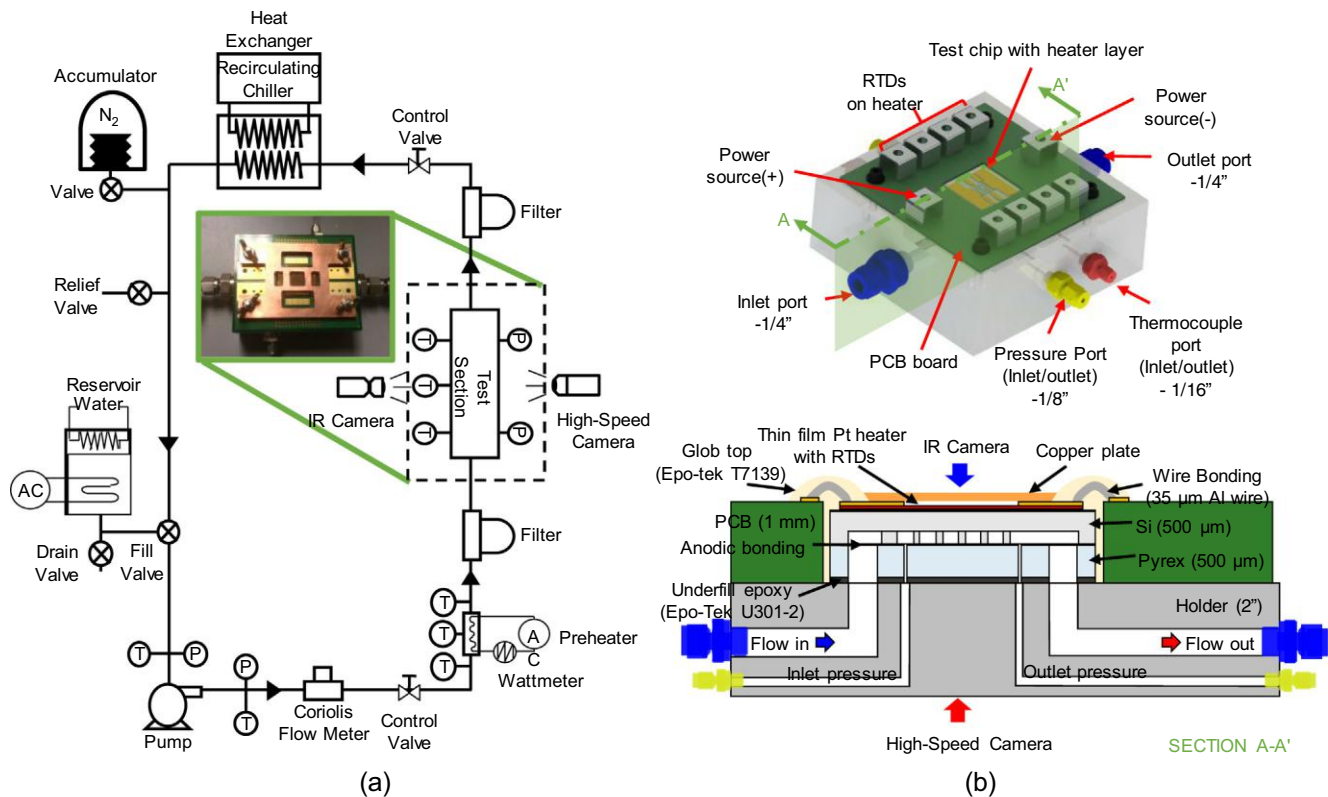


Fig. 4. (a) Schematic of the cooling loop and (b) construction and cross section of the assembled test module.

flow rate and set the desired inlet temperature for the test module, respectively. Two filters were installed before and after the test module to ensure clean fluid delivery during the test. The heated working fluid exiting the outlet filter was cooled to atmospheric temperature with the aid of a brazed-plate liquid-to-liquid heat exchanger connected to an external recirculating liquid chiller (NESLAB ThermoFlex 3500). The inlet and outlet fluid pressures of the test module were measured through two pressure ports connected to Omega absolute pressure transducers. The fluid temperature at both the inlet and outlet of the test module were measured using K-type thermocouples.

### 2.1.3. Test procedure, operating conditions, and uncertainty

The flow rate was adjusted by setting the rotational speed of the pump. Then, power was applied to the serpentine heaters using two regulated direct-current (DC) power supplies to achieve the desired operating conditions. Once steady state has been attained, the pressure and temperature of the fluid and heated surface were measured using a LabVIEW code connected to an NI Compact DAQ modular data acquisition system. IR images of the heated surface were also recorded simultaneously to measure the distribution of the heater temperature. After all data have been obtained, the heat flux was increased, and the procedure was repeated.

The operating conditions and micro-pin fin geometries used in this study are summarized in Table 1. The flow rate  $\dot{m} = 14.7$ – $181.6$  g/min with a range of heat fluxes  $q'' = 2.5$ – $48.7$  W/cm<sup>2</sup>. The inlet fluid temperature was set to the ambient temperature in the range of 22.2–25.3 °C. The inlet pressure  $P_{in} = 205.6$ – $351.6$  kPa. Three different pin–fin diameters  $D_f = 45$ , 90, and 100  $\mu\text{m}$  with  $S = 74$ , 200, and 298  $\mu\text{m}$  and  $H_{ch} = 200$ – $208$   $\mu\text{m}$  were tested. A total of 30 data points were obtained from the three geometries. This includes the results for the wall temperatures and pressure drop measurements. The pressure drop results for CASE 2 were intentionally excluded from the analysis since the inlet pressure could not be measured owing to transducer malfunction. However, the measurements of the exit pressure were used to estimate the properties of the fluid for experimental data reduction and the boundary conditions of the CFD simulations as all the properties are not sensitive to the pressure in an incompressible flow [42].

The fluid temperatures were measured using Type-K thermocouples with an accuracy of  $\pm 0.4$  °C. Calibrated RTDs and IR measurements were used for the heated surface, which have uncertainties of  $\pm 0.7$  and  $\pm 0.3$  °C, respectively. The fluid pressures were measured using two absolute pressure transducers with an accuracy of  $\pm 0.25\%$ . The Coriolis mass flow meter has an accuracy of  $\pm 0.1\%$ , whereas the preheater and test module power supplies have an accuracy of  $\pm 0.5$  W. Consequently, according to all measured parameters, the heat transfer coefficients and pressure drops have mean uncertainties of  $\pm 13.6\%$  and  $\pm 4.5\%$ , respectively.

## 2.2. Computational model

### 2.2.1. Governing equations and computational domain

We conducted CFD simulation to estimate the thermal and hydraulic performance in a micro-pin fin heat sink. Fig. 5 shows the computational domain used in the simulations, which

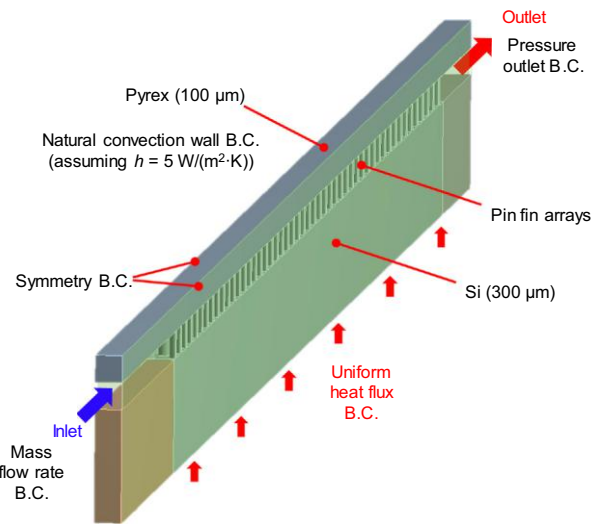


Fig. 5. Computational domain of a unit cell of the micro-pin fin channels.

indicates the unit-cell width and the total length of the flow channel. The flow length of the domain was set longer than the length of the test module to minimize entrance and end effects due to the inlet and outlet boundary conditions. A mass flow inlet condition was used as the inlet boundary condition, and a pressure outlet was applied as the outlet boundary condition. A constant heat flux boundary condition was imposed on the bottom heated surface. A Pyrex cover with a thickness of 100  $\mu\text{m}$  was adopted with an adiabatic boundary condition at the top of the cover to minimize the computational cost. The heat loss to the top of the Pyrex cover was estimated to be approximately 3.2% of the total supplied heat based on a simplified model (assuming that  $h = 5$  W/(m<sup>2</sup>·K)). Both side walls of the unit cell were set to have symmetric boundary conditions. Commercial CFD software, ANSYS FLUENT 17.0, was used to compute the conservation equations of the unit-cell simulation. The constitutive equations are as follows:

Continuity equation:

$$\nabla \cdot \mathbf{u} = 0 \quad (1)$$

Conservation of momentum:

$$\rho(\mathbf{u} \cdot \nabla \mathbf{u}) = -\nabla P + \mu \nabla^2 \mathbf{u} + \rho \mathbf{g} \quad (2)$$

Conservation of energy:

$$\rho C_p (\mathbf{u} \cdot \nabla T) = \nabla \cdot (k \nabla T) + \mu \Phi \quad (3)$$

### 2.2.2. Boundary conditions and solution technique

Temperature-dependent properties were adopted for R245fa and silicon to capture the property variations. Constant properties were used for Pyrex since its property variations are insignificant for the temperatures employed in this study. The thermophysical properties of R245fa, Pyrex, and the silicon wafer are provided in Appendix A of the Supplementary Material.

Conformal hexahedral elements were employed in ANSYS Meshing to generate a mesh. Finer meshes were generated near solid-fluid interfaces to resolve sharp changes in the temperature,

**Table 1**  
Operating conditions and micro-pin fin geometries used in the present study.

| Case | $D_f$ [ $\mu\text{m}$ ] | $S_L$ [ $\mu\text{m}$ ] | $S_T$ [ $\mu\text{m}$ ] | $H_f$ [ $\mu\text{m}$ ] | $S_L/D_f$ | $S_T/D_f$ | $\dot{m}$ [g/min] | $q''$ [W/cm <sup>2</sup> ] | $T_{f,in}$ [°C] | $P_{in}$ [kPa] |
|------|-------------------------|-------------------------|-------------------------|-------------------------|-----------|-----------|-------------------|----------------------------|-----------------|----------------|
| 1    | 45                      | 74                      | 74                      | 207                     | 1.64      | 1.64      | 14.7–45.6         | 2.5–20.0                   | 24.2–25.3       | 270.5–351.6    |
| 2    | 90                      | 298                     | 298                     | 208                     | 3.31      | 3.31      | 60.2–181.6        | 9.8–42.7                   | 22.2–22.6       | 205.6–231.7    |
| 3    | 100                     | 200                     | 200                     | 200                     | 2         | 2         | 89.8–90.2         | 9.7–48.7                   | 24.9–25.1       | 298.1–310.3    |

pressure, and velocity near the wall, including resolving the turbulence. Mesh independence was verified using mesh sizes up to 5.6 million cells. It was observed that the pressure drop and average wall temperature reached asymptotic values for only the finest mesh size. Thus, a computational domain with 5.6 million meshes was adopted in all presented simulations.

Single-phase steady-state simulations were carried out using a pressure-based solver. The Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) algorithm was used for pressure–velocity coupling. The least-squares cell-based formulation was adopted for gradient discretization. The second order was used for pressure discretization, and the second upwind scheme was applied for both momentum and energy discretization. The first-order upwind scheme was used for the specific dissipation rate and turbulent kinetic energy. The shear stress transport (SST)  $k$ – $\omega$  turbulence model for a micro-pin fin was employed to consider the effects of turbulence due to the pin fin as indicated by other researchers [24]. The convergence criteria were set to  $10^{-6}$  for continuity, velocities,  $k$ , and  $\omega$ , and  $10^{-12}$  for energy. The grid independence test and details of the solution technique and controls are presented in Tables B1 and B2 of the Supplementary Material, respectively.

### 2.3. Data processing

The Reynolds number  $Re$  is defined as a function of  $D_f$  as

$$Re = \frac{\dot{m}D_f}{\mu A_{min}} \quad (4)$$

where  $\dot{m}$  is the mass flow rate,  $\mu$  is the dynamic viscosity, and  $A_{min}$  is the minimum cross-sectional area of the channel.  $A_{min}$  always occurs at  $S_T$  since the longitudinal spacing  $S_L$  and transverse spacing  $S_T$  are the same for all the geometries tested in the present study, and can be determined as

$$A_{min} = H_f W \left(1 - \frac{D_f}{S_T}\right) \quad (5)$$

where  $H_f$  is equal to the channel height  $H_{ch}$  since there is no tip clearance for the fin in the present study.

The Fanning friction factor  $f$  of a micro-pin fin array is defined as

$$f = \frac{\Delta P}{2\rho u_{max}^2 N_L} \quad (6)$$

where  $\Delta P$  is the total pressure drop across the flow channel,  $N_L$  is the number of pin-fin rows in the flow direction, and  $u_{max}$  is the maximum fluid velocity calculated using

$$u_{max} = \frac{\dot{m}}{\rho A_{min}} \quad (7)$$

The average heat transfer coefficient  $h$  is calculated using the fin analysis method as follows:

$$h = \frac{q'' A_h}{(T_b - T_f) \left[ \left( L \times W - N_{tot} \frac{\pi D_f^2}{4} \right) + \left( \pi D_f H_f N_{tot} \eta_f \right) \right]} \quad (8)$$

where  $T_b$  is the average temperature at the bottom wall of the flow channel,  $T_f$  is the average fluid temperature along the flow channel, and  $N_{tot}$  is the total number of pin fins.  $\eta_f$  is the fin efficiency range, which is 87–97% in the present study and can be determined by

$$\eta_f = \frac{\tanh(mH_f)}{mH_f} \quad (9)$$

where  $m$  is the fin constant defined as

$$m = \sqrt{\frac{4h}{k_s D_f}} \quad (10)$$

where  $k_s$  is the thermal conductivity of the silicon substrate. The built-in MATLAB function, `fzero` is used to iteratively solve for the heat transfer coefficient using Eqs. (8)–(10). The Nusselt number  $Nu$  is calculated as follows:

$$Nu = \frac{hD_f}{k_f} \quad (11)$$

## 3. Results and discussion

### 3.1. Temperature measurement and heat transfer performance

Here, we present the experimentally obtained heat transfer performance for the three cases using the results for  $Nu$  and the total thermal resistance  $R_{tot}$ . Fig. 6 shows the temperature distributions of the heated surface captured by the IR thermal images. The fluid flow is from the left to the right with a linearly increasing temperature along the flow direction but a more-or-less uniform temperature in the transverse direction. The slight variations at the edges are caused by heat spreading to the substrate. With an increase in the heat flux in the range  $q'' = 5.1$ – $20.0$  W/cm<sup>2</sup>, the temperature also gradually increases, as expected. Further, for the case where  $q'' = 20$  W/cm<sup>2</sup>, the surface seems to show a serpentine pattern in the downstream region. The reason for this will be discussed later when we plot the surface temperatures.  $T_f$  is obtained from the inlet ( $T_{tc,in}$ ) and outlet ( $T_{tc,out}$ ) fluid temperatures, whereas the heater temperature  $T_h$  is obtained from area-averaging of the IR thermal images.  $T_b$  is calculated from  $T_h$ , accounting for conduction through the silicon substrate.  $R_{tot}$  is obtained from the temperature difference between  $T_h$  and  $T_{tc,in}$  divided by the total heat input to the fluid. The detailed temperature calculation and thermal resistance network can be found in Appendix C of the Supplementary Material.

Fig. 7 shows variations in  $T_f$  and  $T_h$  for different heat fluxes at each mass flow rate for the three cases. As expected, the fluid temperature increases as the total heat flux increases, and the variations are less for higher mass flow rates owing to the higher thermal capacity of the fluid. The heater temperature also gradually increases as the heat flux increases owing to the increase in the fluid temperature and the corresponding decrease in the thermal conductivity of Si substrate.

Fig. 8 shows a comparison of variations in  $Nu$  for different heat fluxes (a–c) and  $Re$  (d–f).  $Nu$  varies in the ranges of 2.2–9.4, 8.2–14.1, and 19.0–20.4 for CASE 1–CASE 3 with  $Re = 35.0$ – $117.0$ , 155.3–481.3, and 378.3–432.1, respectively. Minimal variations occurred in  $Nu$  as the heat flux changes; however, it decreases slightly as the heat flux increases owing to the decrease in the thermal conductivity of the fluid as the temperature increases [25,26]. At the same flow rate, a smaller spacing yields a higher  $Nu$  due to the higher flow velocity and enhanced mixing of the flow. An increase of 50% in the fin spacing yields a 50% decrease in  $Nu$  at  $\dot{m} = 90$  g/min, as shown in Fig. 8(b) and (c). As shown in Fig. 8 (d–f),  $Nu$  increases as the mass flow rate increases for the same geometry. However, a higher  $Nu$  is achieved for a relatively small mass flow rate using a denser pin-fin configuration (CASE 1). Overall, the experimental values of  $Nu$  are proportional to  $Re^{0.65}$ , which falls within the range of  $Re$  exponents of 0.6–1.04 obtained from relevant existing correlations [5].

Fig. 9 shows a plot of  $R_{tot}$  versus  $Re$ . For all cases,  $R_{tot}$  decreases as  $Re$  increases owing to the increase in convective heat transfer. However, this decrease is weakened at higher  $Re$ , and  $R_{tot}$  reaches asymptotic values for  $Re > 500$ . This is due to two main reasons: the portion of parasitic conduction resistance increases and the rate of increase in convection heat transfer decreases as the flow rate increases. Thus, it seems difficult to achieve a lower  $R_{tot}$  with

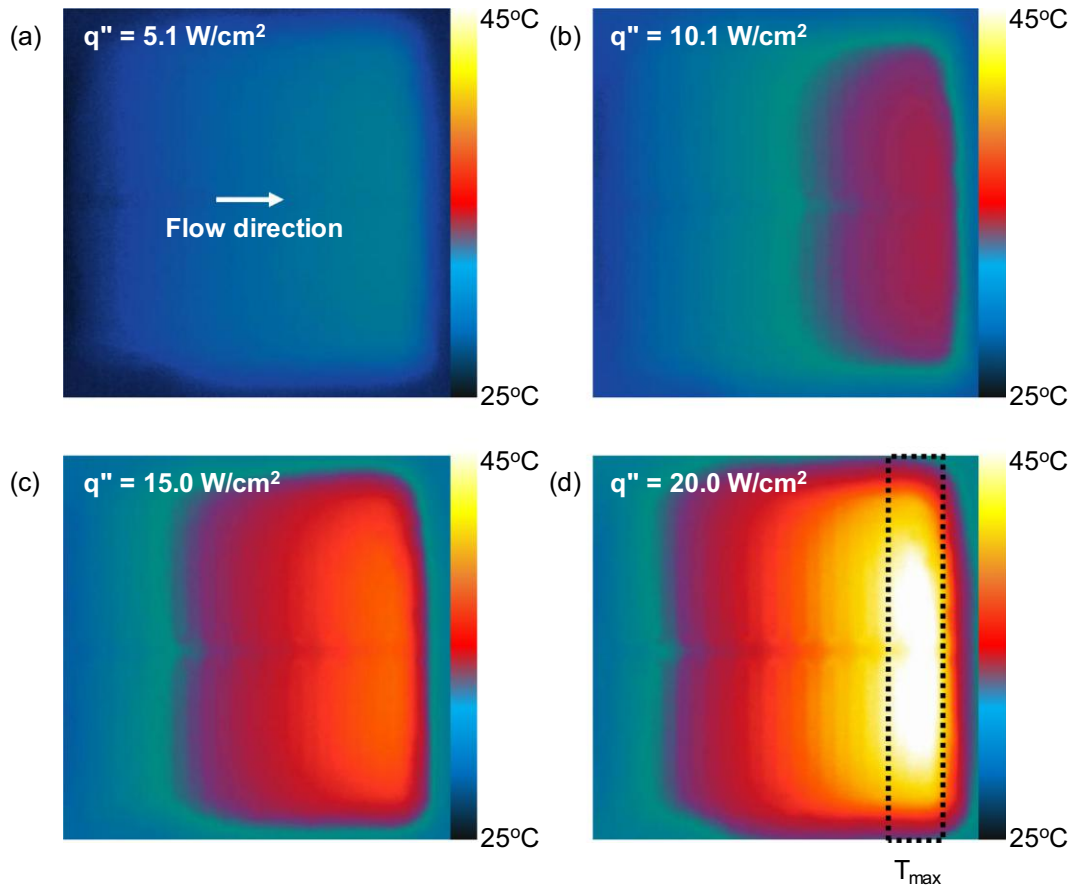


Fig. 6. IR thermal images of the heated surface operating at a fixed mass flow rate of 45.2–45.6 g/min for increasing power input at increments of 5 W/cm<sup>2</sup>.

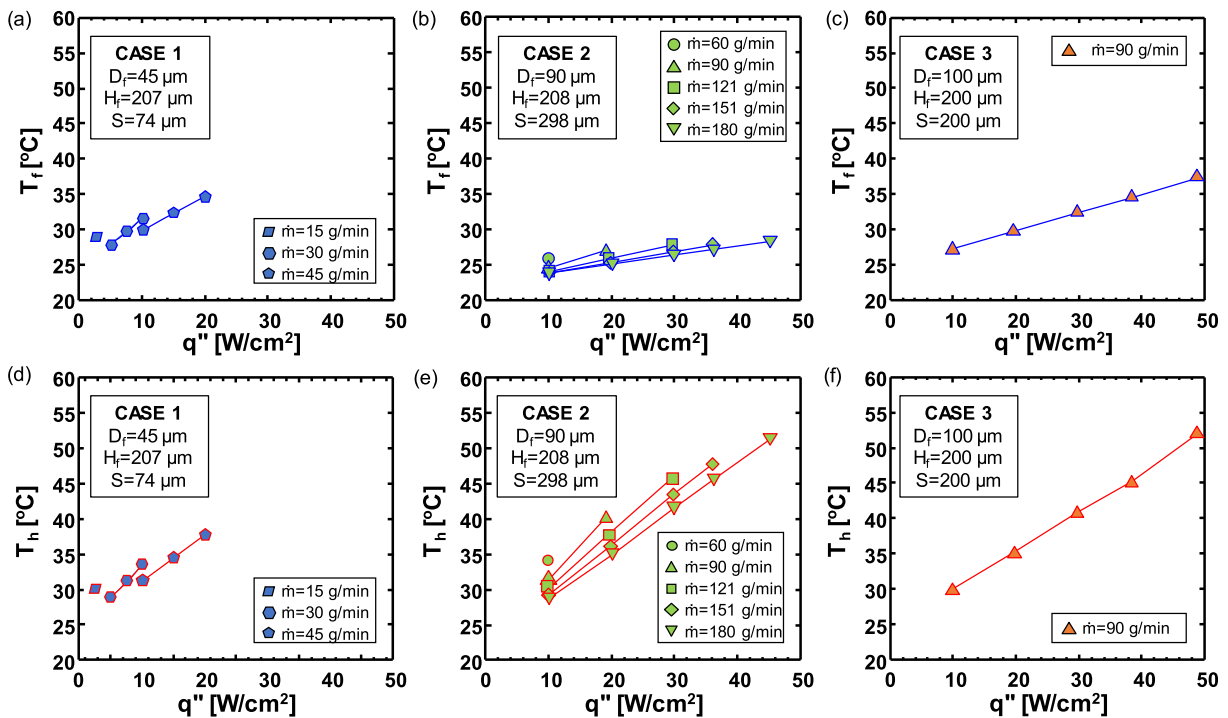


Fig. 7. Variations in the experimentally determined (a–c) average fluid temperature and (d–f) average heater temperature as a function of the heat flux for three different micro-pin fin geometries: (a, d) CASE 1, (b, e) CASE 2, and (c, f) CASE 3.

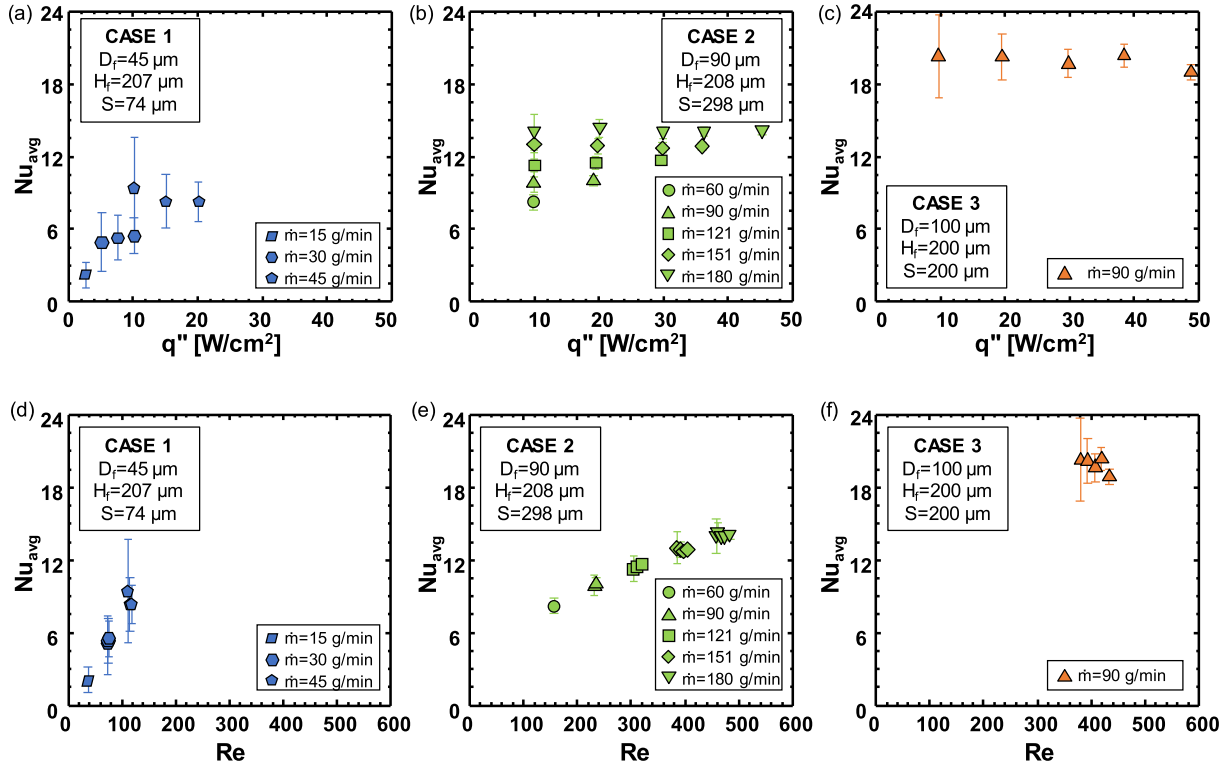


Fig. 8. Experimentally determined average Nusselt number for different (a–c) heat fluxes and (d–f) Reynolds numbers with three different micro-pin fin geometries: (a, d) CASE 1, (b, e) CASE 2, and (c, f) CASE 3.

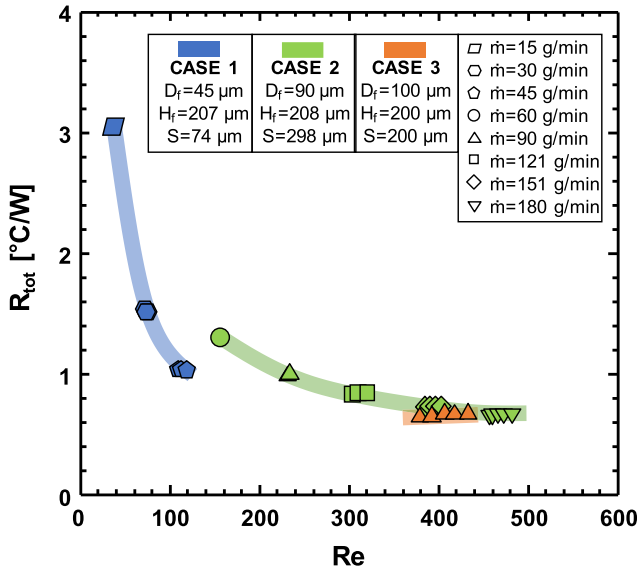


Fig. 9. Variations in the experimentally determined Nusselt number with the Reynolds number for different flow rates and geometries.

an additional increase in the flow rate. A sharp increase occurred in  $R_{tot}$  for a relatively small  $Re$  due to the rapid increase in the advection thermal resistance arising from the increase in the fluid temperature.

### 3.2. Results for the pressure drop

The pressure drop across a micro-pin fin array is determined by the difference between the pressures measured by the absolute pressure transducers installed at the inlet and outlet plenums.

Once the pressure drop is determined, the equivalent Fanning friction factor is calculated using Eq. (6).

Fig. 10(a) shows the variations in the pressure drop for different heat fluxes and mass flow rates. The pressure drop monotonically decreases with increasing heat flux for a constant mass flow rate because the dynamic viscosity decreases for a higher-temperature fluid. This is observed more clearly at a lower flow rate owing to the small heat capacity and the corresponding large temperature increase. The decrease in the pressure drop is relatively small for CASE 3 for the same reason. The variations in  $f$  for different values of  $Re$  are shown in Fig. 10(b). For CASE 1,  $f$  is in the range of 0.255–0.512 with  $Re = 35$ –117, whereas  $f = 0.125$ –0.131 with  $Re = 378$ –432 for CASE 3.  $f$  for the present data is proportional to  $Re^{-0.56}$ , which is in the range of  $Re$  exponents of  $-1.35$  to  $-0.435$  obtained from relevant existing correlations [5].

### 3.3. Validation with computational fluid dynamics

Fig. 11 shows a comparison of experimentally determined local temperature variations in the heated surface along the flow direction with that of simulation predictions. The not-so-linear variation on the heater surface is due to the serpentine-patterned heater, where peaks occur on the gold heater and short troughs in the open area. The computational results slightly underpredict the local temperature in most areas, especially for high heat fluxes. In addition, there is a slight decrease in the temperature near the outlet region of the channel. This trend of decrease in the heat transfer in the outlet manifold was explained in the study by Kharangate et al. [5] and occurs owing to the heat transferred from the silicon substrate to the manifold walls through the working fluid. However, since the adiabatic side wall boundary conditions were set in the computational domain, the effect of the heat losses cannot be captured by simulation. The predictive accuracy is determined by the mean absolute error (MAE) calculated by

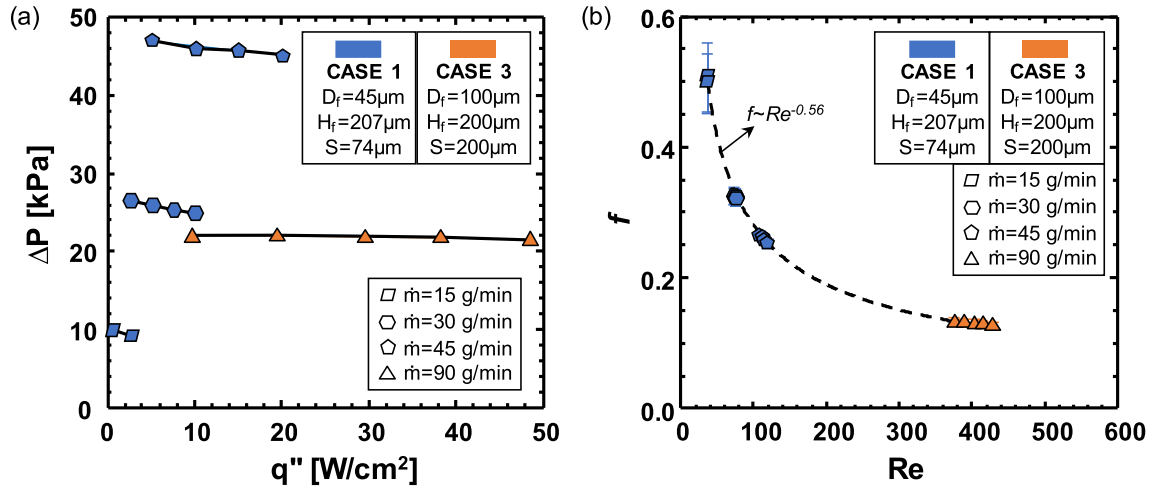


Fig. 10. Variations in the (a) pressure drop versus the heat flux and (b) friction factor versus the Reynolds number.

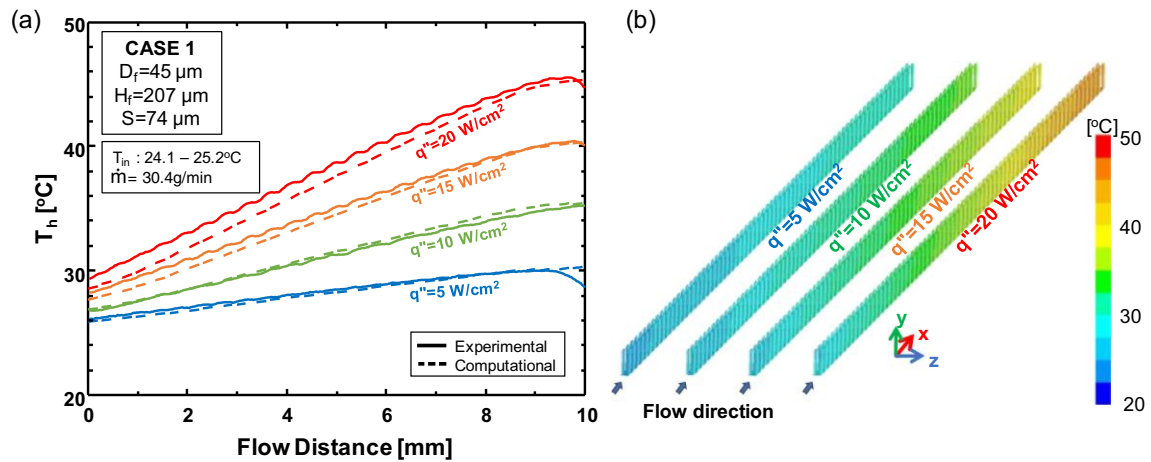


Fig. 11. (a) Comparison of experimental local heater temperature with numerical results, and (b) temperature plot of a micro-pin fin from the numerical model.

$$MAE = \frac{1}{n} \sum_n \frac{|T_{EXP,i} - T_{CFD,i}|}{T_{EXP,i}} \times 100(\%) \quad (12)$$

Overall, the MAE for the present data is 1.1% with 0.74% for  $q'' = 5$  W/cm² and 2.4% for  $q'' = 20$  W/cm².

Fig. 12 shows plots of  $Nu$  obtained from the simulations versus those from the experiments. The computational results show a good prediction of the experimental data with MAEs of 5.5%, 8.2%, and 17% for CASE 1, CASE 2, and CASE 3, respectively. The MAE is calculated using

$$MAE = \frac{1}{n} \sum_n \frac{|Nu_{EXP} - Nu_{CFD}|}{Nu_{EXP}} \times 100(\%) \quad (13)$$

It is important to note that for CASE 1, the temperature was measured by the IR camera, and we obtained a more accurate representation of the temperature across the heated wall. On the other hand, local RTDs with linear interpolations were used to estimate  $Nu$  for CASE 2 and CASE 3. CASE 2 has an error of 8.2% with computational predictions being slightly overestimated. However, the predictions improve with the flow rate. For CASE 3, all computational results are underestimated with MAE = 17%. The overall MAE for  $Nu$  including all the test cases investigated in this study is 9.1%.

Fig. 13 shows comparisons of the experimental  $f$  with the computational  $f$  with the MAE defined as

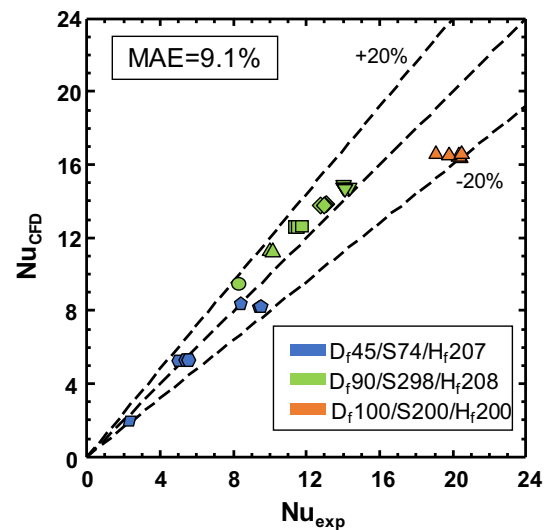


Fig. 12. Comparison of experimental  $Nu$  with numerical  $Nu$  for three configurations.

$$MAE = \frac{1}{n} \sum_n \frac{|f_{EXP} - f_{CFD}|}{f_{EXP}} \times 100(\%) \quad (14)$$

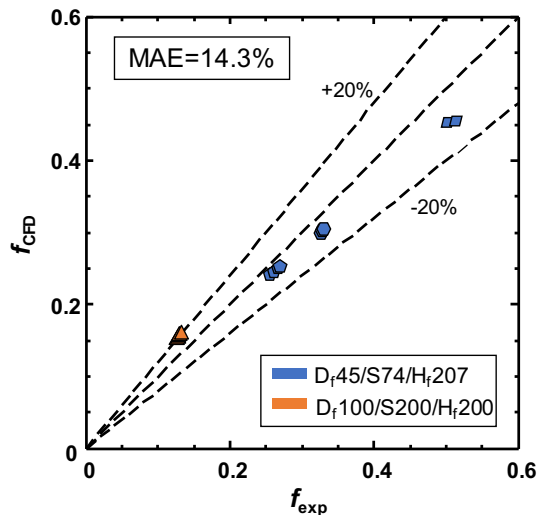


Fig. 13. Comparison of the experimental friction factor  $f_{exp}$  with the numerical friction factor  $f_{CFD}$  for two configurations.

Overall, the computational results show a fair prediction of the data with MAE of 14.3%. However, additional data are essential for a more accurate comparison.

#### 4. Conclusions

In this work, we investigated the heat transfer characteristics and pressure drop using staggered micro-pin fin arrays ( $H_f \sim 200 \mu\text{m}$ ) for single-phase cooling with R245fa as the working fluid. A maximum heat transfer coefficient of  $18.2 \text{ kW}/(\text{m}^2 \cdot \text{K})$  was achieved with CASE1 ( $D_f = 45 \mu\text{m}$ ,  $S = 74 \mu\text{m}$ ,  $H_f = 207 \mu\text{m}$ ) at the maximum flow rate of  $45 \text{ g/min}$  for a corresponding pressure drop of  $47 \text{ kPa}$ .  $Nu$  and  $f$  were expressed under conditions where  $Re = 35\text{--}481$ . A 3D numerical model with symmetric boundary conditions for the unit cell was developed to validate the experimental results. The experimental  $Nu$  and  $f$  were compared with the predicted values using computational modeling. The key observations of this work are as follows:

- (1) The heat transfer coefficient increases as the flow rate increases. However, it reaches an asymptotic value, and no further increase is observed for higher flow rates since a parasitic conduction resistance exists and the increase in convective heat transfer is weakened.
- (2)  $Nu$  was fitted to a power law of  $Re$  and is  $\sim Re^{0.65}$ . The heat transfer coefficient is not influenced by the applied heat fluxes.
- (3) The pressure drop slightly decreases for higher heat fluxes owing to the smaller dynamic viscosity of R245fa at a higher fluid temperature.
- (4) The Fanning friction factor was fitted to a power law of  $Re$  and is  $\sim Re^{-0.56}$ . However, it is necessary to conduct experiments where the range of  $Re$  is  $100\text{--}400$  since the number of data points for a wide range of values of  $Re$  is insufficient to derive an accurate relation.
- (5) The 3D CFD model with the turbulence model shows the good prediction of the experimental data with MAEs of  $9.1\%$  for  $Nu$  and  $14.3\%$  for  $f$ .
- (6) Additional experiments should be conducted in the low  $Re$  range with various geometrical combinations of pin fins for better understanding of the heat transfer and hydraulic performance in micro-pin fin heat sinks.

#### Declaration of Competing Interest

None.

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#### Appendix A. Supplementary material

Supplementary data to this article can be found online at <https://doi.org/10.1016/j.ijheatmasstransfer.2019.05.073>.

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