Experimental Investigation of Embedded Micropin-Fins for Single-Phase Heat Transfer and Pressure Drop

Three-dimensional (3D) stacked integrated circuit (IC) chips offer significant performance improvement, but offer important challenges for thermal management including, for the case of microfluidic cooling, constraints on channel dimensions, and pressure drop. Here, we investigate heat transfer and pressure drop characteristics of a microfluidic cooling device with staggered pin-fin array arrangement with dimensions as follows: diameter $D = 46.5 \mu m$; spacing, $S = 100 \mu m$; and height, $H = 110 \mu m$. Deionized single-phase water with mass flow rates of $\dot{m} = 15.1–64.1 g/min$ was used as the working fluid, corresponding to values of $Re$ (based on pin fin diameter) from 23 to 135, where heat fluxes up to $141 \text{ W/cm}^2$ are removed. The measurements yield local Nusselt numbers that vary little along the heated channel length and values for both the Nu and the friction factor do not agree well with most data for pin fin geometries in the literature. Two new correlations for the average Nusselt number ($\overline{Nu} = Re^{1.04}$) and Fanning friction factor ($\overline{f} = Re^{-0.52}$) are proposed that capture the heat transfer and pressure drop behavior for the geometric and operating conditions tested in this study with mean absolute error (MAE) of 4.9% and 1.7%, respectively. The work shows that a more comprehensive investigation is required on thermofluidic characterization of pin fin arrays with channel heights $H_f < 150 \mu m$ and fin spacing $S = 50–500 \mu m$, respectively, with the Reynolds number, $Re < 300$. [DOI: 10.1115/1.4039475]

Keywords: embedded cooling, micropin fins, staggered, Nusselt number, friction factor

1 Introduction

Three-dimensional (3D) stacked integrated circuit (IC) chips that stack multiple planer ICs in the vertical direction utilizing through-silicon-via (TSV) technology [1,2] provide significant advantages in terms of decreasing the global wire length [3] and increasing the wire-limited clock frequency [4]. However, utilizing such a technology directly increases the density of heat generation and adds to the thermal management constraints and requirements. As the number of chips in the 3D-chip stack is increased, the heat dissipation and thermal conduction resistance to the heat sink (usually at the top) also increase resulting in significant temperature rise and degradation in performance and reliability. The intrachip microfluidic single- or two-phase cooling emerged as a viable solution for 3D-chips architecture and high heat flux applications [5–13].

Research in microfluidic cooling has been focused on developing miniature heat sinks with microstructures (channels or pin fins) that both enhance the surface area and heat transfer coefficients. Parallel microchannels configuration has received much attention [14–16] due to ease of microfabrication process but more recently pin-fin microstructures have been looked more favorably due to its compatibility with the TSV technology. Pin fins can further enhance heat transfer coefficients due to higher flow turbulence and fluid mixing [17]. It is important to understand that utilizing short pin fins (<100 μm) for 3D-chips cooling...
application leads to a small hydraulic diameter and a larger pressure drop that limits the flow rates to laminar (or transition) flow regime with the Reynolds number to less than 1000 [18]. In the work by Peles et al. [17], we can see that high pressure drops (>200 kPa) are unavoidable when utilizing these pin-fin microstructures. Some work has been done toward reducing the pressure drops by designing innovative heat sinks. For 3D-chip stacks, Colgan et al. [19,20] and Wächli et al. [21] showed how a multi-layer heat sink design could provide pressure drop <65 kPa for two layers stack and <40 kPa for three layers stack, respectively. Escher et al. [22] also proposed a novel manifold design with perpendicular inlet and exit manifolds that could achieve a pressure drop <10 kPa.

Fig. 1 Overall microfabrication process of embedded liquid cooling device with micropin-fin arrays

application leads to a small hydraulic diameter and a larger pressure drop that limits the flow rates to laminar (or transition) flow regime with the Reynolds number to less than 1000 [18]. In the work by Peles et al. [17], we can see that high pressure drops (>200 kPa) are unavoidable when utilizing these pin-fin microstructures. Some work has been done toward reducing the pressure drops by designing innovative heat sinks. For 3D-chip stacks, Colgan et al. [19,20] and Wächli et al. [21] showed how a multi-layer heat sink design could provide pressure drop <65 kPa for two layers stack and <40 kPa for three layers stack, respectively. Escher et al. [22] also proposed a novel manifold design with perpendicular inlet and exit manifolds that could achieve a pressure drop <10 kPa.

Fig. 2 (a) A bonded Si-Pyrex wafer after anodic bonding process. (b) Location of five RTDs in the 1 cm x 1 cm heated region (on the backside of the pin fins) marked on the Si sample. (c) Microscopic top-view image of current pin fins sample.

Fig. 3 (a) Isometric view of the test module and its components, (b) cross-sectional side view of the test module (not to scale), and (c) photo of the assembled test module.
Past research in single-phase flows included flow and heat transfer characterization of pin-fin array structures at high Reynolds numbers \((\text{Re} > 1000)\) \([23–28]\) and considerably lower Reynolds numbers \((175 < \text{Re} < 1000)\) \([29,30]\). More recently, Kosar et al. \([31]\) investigated staggered and inline pin fins with diameters of 50 \(\mu m\) and 100 \(\mu m\) and Reynolds number ranging from 5 to 128. Kosar and Peles \([32]\) tested staggered pin fins with a diameter of 99.5 \(\mu m\) and Reynolds number ranging from 14 to 112. Prasher et al. \([33]\) experimentally investigated staggered pin fins in Si with diameters ranging from 55 to 155 \(\mu m\). Adewumi et al. \([34]\) numerically investigated pin fins with diameters ranging from 40 to 60 \(\mu m\) and heights ranging from 30 to 160 \(\mu m\). Mohammadi and Peles \([35,36]\) in two recent numerical studies investigated staggered and inline pin fins with diameters ranging from 50 to 200 \(\mu m\) and height of 100 \(\mu m\). Some researchers have also investigated other geometries for the pin fins, like square and diamond \([18,31,33,37]\).

This study is the first part of the long-term study addressing thermal performance and pressure drop in Si and SiC-based microfabricated pin-fin arrays. In the current study, we provide an experimental investigation of single-phase heat transfer and pressure drop characteristics of a staggered pin-fin array configuration. The pin-fin geometry used for the study has a diameter of 46.5 \(\mu m\), spacing of \(\sim 100 \mu m\), and height of \(\sim 110 \mu m\). De-ionized water was used as the working fluid. Experimentally obtained averaged Nusselt number \((\text{Nu})\) and friction factor \((f)\) over the 1 \(\times\) 1 cm\(^2\) heated area are reported. We also report local Nusselt number variations along the 1 cm longitudinal length of the channel, which has not been reported extensively in the literature due to the difficulty in accurate local temperature measurement. Some recent work by Falsetti et al. \([13]\) used IR camera measurements to study local heat transfer variation for micropin fins during flow boiling of R1234ze(E). Experimental data for Nusselt number and friction factor are also compared to existing correlations for pin fins in single-phase flow.

2 Experimental Methods

2.1 Microfabrication of Si Sample. The micropin-fin array is fabricated on one side of a 4-in Si wafer. On the other side of the wafer, a serpentine heater and resistance temperature detectors (RTDs) are deposited to simulate heat flux conditions and measure surface temperatures. The overall microfabrication process of the embedded cooling module is described in Fig. 1. In the first step, Si is etched by deep reactive ion etch process to define the inlet...
and outlet plenums, orifices, micropin-fin array, and pressure taps. Pyrex substrate is also machined to define the fluid inlets/outlet ports and the pressure ports. Figure 1 shows etched micropin-fin arrays on a Si substrate, as well as drilled holes for fluid inlet/outlet, and pressure ports in a Pyrex wafer. Silicon oxide layer is then grown on both surfaces of the etched Si wafer up to 500 nm to generate an electric passivation layer underneath the metal heater/RTD layer. In the next step, pairs of Si-Pyrex wafers are bonded by anodic bonding process. The bonding temperature, voltage, and time are 350 °C, 900 V, and 6 min, respectively. After the anodic bonding process, two metal layers, Ti (20 nm) and Au (500 nm), are deposited on the top surface of bonded wafer pairs and the following lift-off process defines heaters and RTDs in those bonded wafer pairs. Figure 2(a) shows the image of the front side of the bonded Si test wafer. Each Si wafer has five pinfin-array samples on it, which are later cut. Seven RTDs were deposited along the centerline of the heated section of the sample in the longitudinal direction; however, two RTDs were not functioning properly. So, the temperature data obtained from only the five functional RTDs are recorded and presented in this study. Figure 2(b) shows the location of those working RTDs. We want to make it clear that the Si sample has a separate deposited heater and separate deposited RTDs. There is no impact of the missing RTDs on the heater heat flux as the serpentine heater lines are not electrically connected to the RTD lines. Therefore, missing RTDs do not have any impact on the heat flux distribution across the device, and therefore have no impact on the temperature distribution. The local temperature at the missing RTDs is also calculated using linear interpolation and therefore, the error in heat flux distribution and heat transfer estimation is significantly minimized. The current test sample has a staggered pin-fin array with pin average diameter, spacing, and height of 46.5 μm, 100 μm, and 110 μm, respectively, as shown in Fig. 2(c).

2.2 Test Module. The experiments are conducted in a highly instrumented test module. Figures 3(a)–3(c) show the test module isometric view, not-to-scale schematic of the cross-sectional view, and a photo of the assembled test module, respectively. As shown in Figs. 3(a)–3(c), the module is made up of transparent polycarbonate plastic (Lexan) material. The embedded sample is placed on the top side of the Lexan holder and compressed down with a copper support plate. The transparent material allows visual access to the microchannel features in the embedded samples. Each embedded sample has two pressure ports, which connect to the test module to measure inlet and outlet fluid pressure using Omega absolute pressure transducers. Fluid temperature measurements are made at the inlet and outlet of the test module with K-type thermocouples. The embedded samples have RTDs spread out over the surface to get both local and average temperature measurements.

2.3 Fluid Conditioning Loop. The required operating conditions to the test module are provided by a fluid conditioning loop. A system-level schematic of the fluid loop is provided in Fig. 4(a). In the main loop, the working fluid is circulated with the aid of a variable speed pump. A micropump GA-series gear pump is used to circulate the working fluid in a closed-loop circuit. Exiting the pump, the flow is passed through a regulating valve followed by a micromotion Coriolis flow meter, one in-line electrical preheater, and a filter before entering the test section. The inline heater is used to fine-tune the liquid temperature at the inlet to the test module. Exiting the test module, the heated fluid is routed first through a filter and then through a brazed-plate water-to-water heat exchanger (condenser) to convert the mixture back to atmospheric temperature. One filter before the test section ensures clean fluid delivery to the test specimens, while a second filter after the test section enables detection of test specimen break-up. A nitrogen-pressurized accumulator between the condenser and pump compensates for any expansion or contraction of the working fluid throughout the loop as well as provides a stable reference pressure point for the loop’s operation. A detachable accessory also used as a fluid reservoir for the two-phase loop is used to deaerate the working fluid before performing tests. Pressure and temperature measurements are made at different locations of the flow loop as shown in Fig. 4(a). The flow boiling module, heater and temperature controls, and video camera system are mounted on a rigid optical table as shown in Fig. 4(b).

2.4 Operating Conditions, Operating Procedure, and Measurement Uncertainty. The operating conditions for the study are as follows: Water inlet pressure of \( p_{in} = 144.4–354.6 \text{kPa} \) (20.9–51.4 psi), inlet temperature of \( T_{in} = 24.3–25.6 \text{ °C} \), exit temperature of \( T_{out} = 24.5–84.3 \text{ °C} \), mass flow rate of \( m = 15.1–64.1 \text{ g/min} \), and heat flux of \( q^* = 0–141.4 \text{ W/cm}^2 \). Experiments are performed by setting the desired flow rate on the pump and a heat flux of \( \sim 25 \text{ W} \) on the heater surface of the Si test sample. Once steady-state is reached, temperature and pressure measurements are made for the fluid and the surface of the test sample. The data are recorded using a LabVIEW code in conjunction with a NI CompactDAQ modular data acquisition (DAQ) systems. Once the data are recorded, the heat flux is increased in increments of \( \sim 25 \text{ W} \) and the procedure is continued. Type-K thermocouples with an accuracy of \( \pm 0.7 \text{ °C} \) are used to measure fluid temperatures. Calibrated RTDs also have an uncertainty of \( \pm 0.7 \text{ °C} \).
Pressure measurements in the test module are made with absolute pressure transducers, which have an accuracy of ±0.25%. The Coriolis mass flow meter has an accuracy of ±0.01%. Heat loss, which is primarily caused by natural convection from the sample and sample holder assembly, is estimated to be <3% for all test cases, based on the difference between the input heater power and sensible heating of liquid water as it travels across the test module. The wall heat input is measured with an accuracy of ±0.5 W. Heat transfer coefficients have a maximum uncertainty of ±14.4%, and pressure drop measurements have an uncertainty of ±2.5–25.9% based on the pressure drop for the test case.

3 Results and Discussion

3.1 Heat Transfer Results. The heat transfer coefficient is determined as follows:

\[ h(z) = \frac{q' A_b}{(A_{uf} + NA_f \eta_f) (T_{wall}(z) - T_f(z))} \]  (1)

where \( T_{wall}(z) \) is the local fin base temperature, \( T_f(z) \) is the local bulk fluid temperature, \( A_b \) is the area of the base, \( A_{uf} \) is the unfinned area, and \( A_f \) is the surface area of a pin fin, respectively. \( \eta_f \) is the fin efficiency given by

\[ \eta_f = \frac{\tan(h(m(z) H_f))}{m(z) H_f} \]  (2)

where \( m \) is given by

\[ m(z) = \sqrt{\frac{4h(z)}{k_f D_f}} \]  (3)

Equations (1)–(3) are solved iterative to calculate the heat transfer coefficient. The fin efficiencies ranged between 82.2% and 95.3% for our test cases. The corresponding Nu number is determined as follows:

\[ Nu(z) = h(z) \frac{D_f}{k_f(z)} \]  (4)
Table 1 Correlations of single-phase heat transfer for pin fin arrays and the corresponding MAE

<table>
<thead>
<tr>
<th>Authors</th>
<th>Correlation</th>
<th>Geometry</th>
<th>Operating conditions</th>
<th>MAE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Present study</td>
<td>$Nu = 0.0282 Re^{0.04} Pr^{1/3}$</td>
<td>Experimental, staggered, circle, silicon</td>
<td>DI water, $Re = 23–135$, $m = 15.1–64.1$ g/min, $P_{in} = 144.4–340.1$ kPa, $P_{out} = 132.0–201.0$ kPa, $T_{in} = 25$ °C, $T_{out} = 31.5–84.3$ °C, $q'' = 24.0–141.4$ W/cm²</td>
<td>4.9%</td>
</tr>
<tr>
<td>Kosar and Peles [32]</td>
<td>$Nu = 0.0423 Re^{0.09} Pr^{0.21} (\frac{Pr}{Pr_s})^{0.25}$</td>
<td>Experimental, staggered, circle, silicon</td>
<td>R-123, $Re = 134–314$, $m = 1.6–4.4$ g/min, $P_{out} = 267$ kPa, $T_{in} = 22$ °C, $T_{out} = 30–125$ °C, $q'' = 3.5–65.5$ W/cm²</td>
<td>5.3%</td>
</tr>
<tr>
<td>Prasher et al. [33]</td>
<td>$Nu = 0.132 \left(\frac{S_f - D_f}{D_f}\right)^{-0.256} Re^{0.84} (Re &lt; 100)$</td>
<td>Experimental, staggered, circle, silicon</td>
<td>DI water, $Re = 40–1000$, $m = 15–200$ g/min, $T_{in} = 50$ °C</td>
<td>20.7%</td>
</tr>
<tr>
<td>Qu and Siu-Ho [18]</td>
<td>$Nu = 0.281 \left(\frac{S_f - D_f}{D_f}\right)^{-0.63} Re^{0.73} (Re &gt; 100)$</td>
<td>Experimental, staggered, square, copper</td>
<td>DI water, $Re = 45.9–179.6$, $m = 36.7–84.5$ g/min, $T_{in} = 30–60$ °C, $q'' = 7.5–98.0$ W/cm²</td>
<td>(1) 37.7%, (2) 36.6%</td>
</tr>
<tr>
<td>Tulius et al. [38]</td>
<td>$Nu = 0.08 \left(\frac{S_f}{D_f}\right)^{0.2} \left(\frac{S_f}{D_f}\right)^{0.2} (\frac{H_f}{D_f})^{0.25} \left(1 + \frac{dh}{D_f}\right) Re^{0.6} Pr^{0.36} (\frac{Pr}{Pr_s})^{0.25}$ for circular fin</td>
<td>Computational, staggered, circle, square</td>
<td>Aluminum, carbon nanotubes, copper, silicon, $Re = 90–1500$, $m = 56.3–900$ g/min, $P_{in} = 0$ kPa, $T_{in} = 25$ °C, $q'' = 10.0–150.0$ W/m²</td>
<td>23.4%</td>
</tr>
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*Values are estimated from other presented parameters.
The heat flux over the heater surface, \( q^0 \), is assumed to be constant in our calculations. Figures 5(a) and 5(b) show local fluid and wall temperature distributions along the 1 cm channel for heat fluxes of \( q^0 = 45.1–48.1 \) W/cm\(^2\) and \( q^0 = 66.3–71.1 \) W/cm\(^2\) at four mass flow rates, respectively. Using the wall temperature on the Si surface, one-dimensional conduction is used to estimate the fin base temperature, \( T_{wall} \), along the flow distance. As shown in Fig. 5, the local bulk fluid temperature, \( T_l \), is obtained by assuming a linear increase in temperature along the flow distance of the fluid from the inlet to the exit of the channel. This assumption is valid as the specific heat of water changes only by <0.6% over the maximum change in fluid temperature across the heated pin fins section as investigated in this study.

Nusselt number is computed (Eqs.(1)–(4)) at the five wall thermocouple locations along the heated section. Figure 6 shows the local Nusselt number variation for four sets of mass flow rates in the range \( \dot{m} = 15.1–64.1 \) g/min, heat flux of \( q^0 = 24.0–141.4 \) W/cm\(^2\). For all flow rates, \( \dot{m} = 15.1–64.1 \) g/min, the local Nusselt number is observed to be almost uniform along the channel in the inlet and middle sections, with a slight enhancement near the exit section of the channel. The increase in Nusselt number near the exit is partially caused by inaccuracies in estimating the local bulk fluid temperature and the heat losses to the manifold sections. For a 2.5 cm x 3 cm highly conductive Si test section, a portion of the total heat is transferred to the fluid in the inlet and outlet manifold. This results in fluid temperature rise at the entrance and exit of the pin fins section. The modified equation for heat transfer at the inlet and outlet thermocouples based on this change is shown as follows:

\[
\begin{align*}
\hat{h}_{in}(z) &= \frac{(q^0 - q_{loss, in}^0)A_b}{(A_{eff} + N\lambda\eta_f)(T_{wall, in} - T_{in}(z) - \Delta T_{in})} \\
\hat{h}_{out}(z) &= \frac{(q^0 - q_{loss, out}^0)A_b}{(A_{eff} + N\lambda\eta_f)(T_{wall, out} - T_{out}(z) + \Delta T_{out})}
\end{align*}
\]

where \( q_{loss, in}^0 \) is the heat transferred to the manifold regions, \( \Delta T_{in} \) is the temperature rise in the inlet manifold section, and \( \Delta T_{out} \) is the temperature rise in the outlet manifold section. A simple numerical simulation was run to quantify this effect. We assumed an effective heat transfer coefficient in the pin fins section to be 10 kW/m\(^2\) K, and effective heat transfer coefficients in both the inlet and outlet manifolds as 0.5 kW/m\(^2\) K and 2 kW/m\(^2\) K for mass flow rates of 15.1 and 64.1 g/min, respectively. For 15.1 g/min and 64.1 g/min, the simulations showed that the fluid temperature rise in the inlet manifold and the temperature drop in the outlet manifold are 1.3 °C and 1.1 °C, respectively, and the heat transferred in the manifold sections are 5.1 W/cm\(^2\) and 7.1 W/cm\(^2\), respectively. The net effect is that we see a 3.6–6.4% underestimation of heat transfer coefficient in the inlet region and an 18.6–22.7% overestimation in the exit regions for the two mass flow rates. Even though this trend is expected, the exact variation is not quantifiable because it is challenging to accurately estimate the local heat transfer and the corresponding local value of bulk fluid temperatures at the inlet and exit locations of the pin fin section without a full CFD simulation, which is beyond the scope of the present study but will be pursued in the near future.

Figure 7 shows the axially averaged Nusselt number for the four sets of mass flow rates in the range \( \dot{m} = 15.1–64.1 \) g/min plotted versus the Reynolds number. Heat transfer increases with flow rate due to the increase in Reynolds number and expected decrease in the thickness of the boundary layer. As shown in Fig. 7, axially averaged Nusselt number is observed to be
approximately proportional to \( Re^{1.04} \). It is important to note that relevant correlations developed for pin fins as described in Table 1 have an exponent rating for Nusselt number with \( Re \) ranging from 0.6 to 0.99.

To further explore the range of validity and consistency of the present experimental data, we compare those with the predictions of the existing correlations in literature for staggered pin-fins configuration. There are a large number of correlations so we had narrowed down the comparison to those with partial overlap in important geometrical and flow characteristics, see Table 1. We selected four correlations \([18,32,33,38]\) that meet the following criterion: \( D_f < 1 \text{ mm}, H_f < 1 \text{ mm} \) and include test cases with \( 22 < Re < 1500 \). The correlation by Tullius et al. \([38]\) is included because it also encompassed a subset of test cases with \( w_f < 1 \text{ mm} \) and \( H_f < 1 \text{ mm} \). Kosar and Peles \([32]\) and Qu and Siu-Ho \([18]\) correlations had been developed based on a single geometry as such only require fluid properties and operating condition for estimation of the Nusselt numbers. The correlations by Prasher et al. \([33]\) and Tullius et al. \([38]\) are a function of geometrical configurations such as fin spacing, and fin diameter, along with the fluid properties and operating conditions.

In Fig. 8, we compare the experimentally determined Nusselt number with those predicted utilizing the correlations described in Table 1. The predictive accuracy of a correlation is determined by mean absolute error (MAE), which is expressed as

\[
\text{MAE} = \frac{1}{n} \sum \left| \frac{\text{Nu}_{\text{pred}} - \text{Nu}_{\text{exp}}}{\text{Nu}_{\text{exp}}} \right| \times 100\% \quad (6)
\]

The correlation by Kosar and Peles \([32]\), Fig. 8(b), agrees well with our experimental data, with \( \text{MAE} = 5.3\% \). This can be attributed to the correlations being developed for circular pin fins closest to the tested geometric conditions in our study. Qu and Siu-Ho \([18]\), Fig. 8(d), did not do so well with \( \text{MAE} = 37.7\% \) and 36.6\%. This is mainly because of the correlation was developed based on different shape and size of pin fins. The correlation provided by Prasher et al. \([33]\) gives the best predictions, with an \( \text{MAE} = 20.7\% \) out of correlations developed utilizing multiple geometric configurations, followed by the correlation by Tullius et al. \([38]\) with an \( \text{MAE} = 26.0\% \). Out of the correlations, the new correlation proposed in this study, Fig. 8(a), gives the best predictions, with \( \text{MAE} = 4.9\% \). The discrepancy between the present data and the existing correlations is somewhat expected due to

![Fig. 9](image-url)  
**Fig. 9** (a) Comparison of Nusselt number predictions based on correlations from present study with experimental data from Wan and Joshi \([39]\); comparison of Nusselt number predictions based on correlations from present study with computational fluid dynamics simulations data from Mohammadi and Peles \([35]\) for (b) \( S_T = 75 \mu\text{m}, S_L = 75 \mu\text{m} \), (c) \( S_T = 75 \mu\text{m}, S_L = 150 \mu\text{m} \), and (d) \( S_T = 150 \mu\text{m}, S_L = 75 \mu\text{m} \), and (e) \( S_T = 150 \mu\text{m}, S_L = 150 \mu\text{m} \)

![Fig. 10](image-url)  
**Fig. 10** Fanning friction factor versus Reynolds number
<table>
<thead>
<tr>
<th>Authors</th>
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<th>Geometry</th>
<th>Operating conditions</th>
<th>MAE (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Present study</td>
<td>$f = 2.5\text{Re}^{-0.52}$</td>
<td>Experimental, staggered, circle, silicon</td>
<td>DI water</td>
<td>1.7%</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$D_f = 46.5~\mu m$</td>
<td>$Re = 23–135$</td>
<td></td>
</tr>
<tr>
<td>Kosar et al. [31]</td>
<td>$f = \left(\frac{H_f}{D_f} \right)^{1.14} \left(\frac{S_L}{A_c} \right)^{-0.3} + \frac{345}{\text{Re}} \left(\frac{1}{H_f/D_f + 1} \right)^{2} \left(\frac{S_L}{A_c} \right)^{-0.3}$</td>
<td>Experimental, staggered, circle, diamond, silicon</td>
<td>DI water</td>
<td>43.7%</td>
</tr>
<tr>
<td>Prasher et al. [33]</td>
<td>$f = \left(\frac{H_f}{D_f} \right)^{-0.64} \left(\frac{S_L}{D_f} \right)^{-0.258} \left(\frac{S_f}{D_f} \right)^{0.283} \text{Re}^{-1.35}$ (Re &lt; 100)</td>
<td>Experimental, staggered, circle, square, silicon</td>
<td>DI water</td>
<td>69.0%</td>
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<td></td>
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<td>$D_f = 55–153~\mu m$</td>
<td>$Re = 40–10000$</td>
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<tr>
<td>Brunschwiler et al. [40]</td>
<td>$f = 11.827 \left(\frac{H_f}{D_f} \right)^{0.368} \left(\frac{S_L}{D_f} \right)^{-4.102} \text{Re}^{-0.774}$ (Re &lt; 100)</td>
<td>Experimental, computational staggered, inline, circle, silicon</td>
<td>DI water</td>
<td>26.6%</td>
</tr>
<tr>
<td>Tullius et al. [38]</td>
<td>$f = 0.74 \left(\frac{S_f}{D_f} \right)^{-0.2} \left(\frac{H_f}{D_f} \right)^{-0.2} \left(1 + \frac{dh}{D_f} \right)^{0.2} \text{Re}^{-0.435}$ for circular fin</td>
<td>Computational, staggered, circle, square</td>
<td>Water</td>
<td>32.1%</td>
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<tr>
<td></td>
<td></td>
<td>$H_f = 100–200~\mu m$</td>
<td>$Re = 90–1500$</td>
<td></td>
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<tr>
<td></td>
<td></td>
<td>$H_f = 100–200~\mu m$</td>
<td>$Re = 90–1500$</td>
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</tbody>
</table>

*Values are estimated from other presented parameters.*
differences in geometry, working flow, and pin-fin array geometry. To check the validity of the current correlation, we used our correlation to predict Nusselt number obtained in some recently conducted experiments by Wan and Joshi [39] and obtained an MAE of 43.6% as shown in Fig. 9(a). We also predicted Nusselt number obtained in some recently conducted computational fluid dynamics simulations by Mohammadi and Peles [35] and obtained an MAE ranging from 60.5% to 72.4% depending on the fin spacing as shown in Figs. 9(b)–9(e). In general, it is observed that the correlation cannot accurately predict the heat transfer and pressure drop behavior in those studies due to differences in test and operating conditions, and therefore, additional work is needed in future studies to develop more robust Nusselt number correlations that can cover wider geometrical and operational conditions with multiple fluids.

### 3.2 Pressure Drop Results

The pressure drop along the pin-fin area is measured by subtracting the pressure measured by the absolute pressure transducers located at the inlet and outlet of the heated area of the Si sample. The experimentally determined mean Fanning friction factor is obtained as follows:

$$f = \frac{1}{2N_f} \frac{\Delta \rho}{\mu_{\text{max}} \bar{v}}$$

where $N_f$ is the number of pin fins in the longitudinal direction, $\Delta \rho$ is the pressure drop along the pin fin section, and $\mu_{\text{max}}$ is the maximum velocity of fluid.

Figure 10 shows the experimentally determined Fanning friction factor for the four sets of mass flow rates in the range $\dot{m} = 15.1$–64.1 g/min plotted versus Reynolds number. The Fanning friction factor for the present data scales with $Re^{-0.52}$ compared to that for the existing correlations (Table 2) where the Reynolds number exponent spans over a wide range of $-1.35$ to $-0.0435$. We have shortlisted four correlations in Table 2 for the Fanning friction factor that are closest to the operating and geometric conditions of the present study. In Fig. 11, we compare the experimentally determined Fanning friction factor of the present study with those predicted utilizing the correlations listed in Table 2. The predictive accuracy of a correlation is determined by mean absolute error, which is expressed as

$$\text{MAE} = \frac{1}{n} \sum \left| \frac{f_{\text{pred}} - f_{\text{exp}}}{f_{\text{exp}}} \right| \times 100\%$$

All the Fanning friction factor correlations utilized for model comparisons give relatively poor predictions. Most of the correlations overpredict the data for high Reynolds range of $Re > 100$ while Tullius et al. [38] underpredict the present data. It is observed that the correlation by Brunschwiler et al. [40], Fig. 11(d), shows the best predictions; however, it has an MAE = 24.9%. This correlation was developed utilizing multiple geometries, with the same working fluid, DI water, and had similar pin-fin height and Reynolds number as used in the current study. The correlation by Prasher et al. [33], Fig. 11(c), was predicted with an MAE = 67.8%. This correlation provides relatively accurate predictions of MAE = 29% for $Re = 40$–100 while MAE = 117% for $Re < 40$ and $Re > 100$. The new proposed correlations, Fig. 11(a), give the best accuracy of MAE = 1.7%. To check the validity of the current correlation, we used our correlation to predict Fanning friction factor obtained in some recently conducted experiments and simulations.
conducted experiments by Wan and Joshi [39] and obtained an MAE of 18.7% as shown in Fig. 12(a). We also predicted Fanning friction factor obtained in some recently conducted computational fluid dynamics simulations by Mohammadi and Peles [35] and obtained an MAE ranging from 34.8% to 46.3% depending on the fin spacing as shown in Figs. 12(b)–12(e). Like was observed for Nusselt number, this also shows that additional work is needed in future studies to develop more robust Fanning friction correlations that can cover wider geometrical and operational conditions with multiple fluids.

4 Conclusions

In this study, we investigate heat transfer and pressure drop characteristics of micropin-fin array using single-phase water. The Nusselt number and Fanning friction factor are experimentally determined for mass flow rates of $\dot{m} = 15.1–64.1$ g/min and heat fluxes up to 141 W/cm². The results are compared with predictions of the most relevant existing correlations. Experimentally measured local Nusselt number was observed to remain almost uniform along the channel and averaged Nusselt numbers are scaled with $Re^{1.04}$. We proposed correlations for the Nusselt number and Fanning friction factor with MAE of 4.9% and 1.7%, respectively. In general, the existing correlations for the Nusselt number and Fanning friction factor do not accurately predict the present data indicating that more thorough experimental investigation needs to be conducted over a large range of pin fin array geometry, flow condition, and working fluid. More specifically, the community should focus on thermofluidic characterization of pin-fin arrays with channel heights and fin spacing $H_f < 150 \mu m$, and $S_f = 50–500 \mu m$, respectively, with the Reynolds number, $Re < 300$.

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Nomenclature

- $A_b$: base area of the pin fins heat sink
- $A_f$: surface area of a single pin fin
- $A_{uf}$: unfinned area of the pin fins heat sink
- $D_f$: diameter of fin
- $dh$: clearance of channel
- $f$: friction factor
- $h$: heat transfer coefficient
- $H_f$: height of the fin
- $k_s$: conductivity of silicon
- $L$: heated length
- $m$: fin parameter
- $\dot{m}$: mass flow rate (g/min)
- $n$: number of data point
- $N$: number of pin fins
- $Nu$: averaged Nusselt number
Greek Symbols
\[ \eta \] = fin efficiency
\[ \nu \] = specific volume

Subscripts
\[ \exp \] = experimental (measured)
\[ f \] = fin
\[ in \] = inlet to heated portion of flow channel
\[ l \] = fluid
\[ L \] = longitudinal
\[ out \] = outlet to heated portion of flow channel
\[ \text{pred} \] = predicted
\[ s \] = surface
\[ T \] = transverse
\[ \text{wall} \] = fin base

References