1. Introduction

The trend of miniaturization has led to the increased generation of heat flux in many engineering devices, such as electronic chips and micro electromechanical systems. It is certain that it will reach 10,000 kW/m² in the immediate future [1]. It is possible for such devices to fail permanently if heat generation is not properly managed. This quantity of heat cannot be removed...
Nomenclature

\[ \begin{align*}
A & \quad \text{area, mm}^2 \\
c_p & \quad \text{specific heat, J/(kg K)} \\
D_h & \quad \text{hydraulic diameter, mm} \\
H & \quad \text{height, mm} \\
k & \quad \text{thermal conductivity, W/(m-K)} \\
L & \quad \text{length, mm} \\
N & \quad \text{total number of pin fins} \\
\bar{N}_\text{u} & \quad \text{average Nusselt number} \\
\Delta p & \quad \text{pressure drop, Pa} \\
q & \quad \text{heat flux, kW/m}^2 \\
Re & \quad \text{Reynolds number} \\
T & \quad \text{temperature, K} \\
\bar{T}_\text{bw} & \quad \text{average bottom wall temperature, K} \\
TPF & \quad \text{thermal performance factor} \\
U & \quad \text{velocity, m/s} \\
V & \quad \text{velocity matrix, m/s} \\
W & \quad \text{width, mm} \\
\mu & \quad \text{dynamic viscosity, Pa s}
\end{align*} \]

Subscripts and Superscripts

\[ \begin{align*}
\text{avg} & \quad \text{average} \\
b & \quad \text{bottom} \\
bulk & \quad \text{bulk fluid} \\
ch & \quad \text{channel} \\
\text{eff} & \quad \text{effective} \\
f & \quad \text{fin} \\
in & \quad \text{inlet} \\
l & \quad \text{liquid} \\
o & \quad \text{base configuration} \\
s & \quad \text{substrate} \\
wl & \quad \text{solid liquid interface}
\end{align*} \]

Abbreviations and Acronyms

- HS: heat sink
- MCHS: microchannel heat sink
- MEMS: micro-electromechanical system
- MPFHS: micro pin fin heat sink
- STEP: stepped
- UO: uniform

Two dissimilar pin-fin arrangements, i.e. inline and staggered were often considered in MPFHS designs. An experimental comparison [15] revealed that staggered configurations possess greater heat transfer coefficients and friction factors compared to inline arrangements under similar flow rates and packaging densities. However, prevailing differences in performance diminish with the increasing packaging density. Keshavarz et al. [16] examined the performance of circular and drop shaped MPFHS having inline and staggered arrangements. The authors perceived that the staggered arrangement has a higher outlet temperature for lesser fin density, while the inline arrangement has more outlet temperature for moderate pin-fin density. However, for all pin-fin density configurations, the
staggered arrangement has a higher pressure drop compared to the inline arrangement. In recent work by Bhandari et al. [17], different pin fin shapes were analyzed. They found that four-side arrangements have the best performance among different cases. The fluid flow mixing is one of the important factors that affects the thermo-hydraulic performance [18].

The micro pin fin heat sink represents a crucial innovation with diverse applications across several domains [19–21]. Primarily, it plays a pivotal role in battery thermal management systems, ensuring the efficient dissipation of heat generated during charging and discharging processes. This is particularly relevant in the context of advancing battery technologies, where effective thermal control is imperative for enhancing overall performance and longevity. Moreover, the utilization of micro pin fin heat sinks extends to hydrogen storage systems, contributing to the optimal management of thermal conditions in the storage process. In the realm of solar photovoltaic (PV) technology, these heat sinks find application in cooling systems, addressing the challenge of excess heat accumulation during solar energy conversion [22–26]. Additionally, in electronics cooling, microchannel heat sinks play a vital role in addressing thermal challenges associated with modern, compact electronic devices, enhancing their reliability and longevity [27, 28]. This underscores the versatility of micro pin fin heat sinks in facilitating enhanced thermal regulation across diverse technological domains. Recent advancements in materials and design methodologies further underscore the ongoing development in this field, emphasizing the contemporary relevance of micro pin fin heat sinks in addressing evolving thermal management needs.

In the present work, the directionality of stepped MPFHS configurations has been numerically studied in two different fashions. The total of six different configurations have been compared on the basis of the average Nusselt number (\(\overline{Nu}\)), pressure drop (\(\Delta p\)) and thermal performance factor (TPF). It is the primary objective of the existing work to investigate out-of-plane mixing in MPFHS.

2. Mathematical modelling

2.1. Heat sink geometry

A numerical analysis of different MPFHS configurations is carried out in the present work. The total of six MPSHS configurations were chosen, including three in inline arrangement and the others in staggered arrangement. All MPFHS geometrical constructions are as follows:

1) Uniform inline MPFHS (UO inline): In this configuration, pin fins have uniform height throughout the heat sink length and width. Pin fin height of 0.375 mm is kept in channel of height 0.5 mm. Further, fins are arranged in inline style. This configuration is kept as a base configuration to assess the total performance of modified sinks.

2) Uniform staggered MPFHS (UO staggered): A staggered arrangement of pin-fins in this configuration is characterized by equal height and even spacing.

3) Three-stepped inline unidirectional MPFHS (3 step inline): The pin-fin height differs in three consecutive rows, i.e. along the channel length only.

4) Three-stepped staggered unidirectional MPFHS (3 step staggered): Similar configuration as described in (c) but arranged in staggered fashion.

5) Three-stepped inline bidirectional MPFHS \((3 \times 3\) step inline): Pin-fin height variation takes place along both longitudinal direction and transverse direction in inline fashion.

6) Three-stepped staggered bidirectional MPFHS \((3 \times 3\) step staggered): Similar to 5), but arranged in staggered fashion. The perspective view and top view of \(3 \times 3\) step staggered configuration are depicted in Fig. 2.

As depicted above, a 3-D model of the heat sinks (HS) has been designed to conduct the current simulation work. The footprint area and overall height of the considered cases are equal, namely \(10.13 \times 7.875 \times 1.5\) mm. In all cases, pin fins have a square cross section, which measures 0.375 mm by 0.375 mm. The pin fins are positioned differently across nine columns, each with twelve fins along the channel length. So, the total of 108 pin fins are there in every heat sink. The thorough geometrical specifications of the open MPFHS are tabulated in Table 1.

![Fig. 2. Three-stepped staggered bidirectional MPFHS (3 × 3 step staggered): (a) Perspective view; (b) Top view with its geometrical specification in mm.](image)

<table>
<thead>
<tr>
<th>Table 1. MPFHS geometrical specifications.</th>
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<tbody>
<tr>
<td>Parameter</td>
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<tr>
<td>MPFHS length ((L))</td>
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<tr>
<td>MPFHS width ((W))</td>
</tr>
<tr>
<td>Micro Fin height variation ((H_h))</td>
</tr>
<tr>
<td>total height of Heat sink ((H = H_a+H_b))</td>
</tr>
<tr>
<td>Bottom wall thickness ((H_b))</td>
</tr>
<tr>
<td>Footprint dimension of fin ((W))</td>
</tr>
<tr>
<td>Pitch between two successive pin-fins along channel length</td>
</tr>
<tr>
<td>Pitch between two successive pin-fins along channel width</td>
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<tr>
<td>Total fins in each configuration</td>
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2.2. Solution methodology

Fluid flow and thermal characteristics of projected MPFHS cases under different flow rates were examined using the same operating conditions. Simulations were performed using ANSYS Fluent Version 18.0 commercial code to study the different
heat sink geometries, where geometry and meshing have been carried out in Workbench. Copper is the substrate of the heat sink, while liquid water is the coolant.

The steady-state basic governing equations, i.e. continuity Eq. (1), momentum Eq. (2), and energy Eq. (3) applicable for the fluid flow zone are:

\[
\nabla \cdot (\rho_i \vec{V}) = 0, \\
\vec{V} \cdot \nabla (\rho_i) = -\nabla p + \nabla \cdot (\mu_i \nabla \vec{V}) + \rho_i \vec{g}, \\
\vec{V} \cdot \nabla (\rho_i c_{p,i} \vec{T}_T) = \nabla \cdot (k_i \nabla \vec{T}_i).
\]

In the above equations \( \vec{V} \) is the velocity matrix. The energy equation for solid substrates is:

\[
\nabla \cdot (k_s \nabla T_s) = 0.
\]

The thermo-physical properties of single phase liquid water vary with temperature. For more accurate results, the same has also been incorporated into the simulation. The variation of liquid thermophysical properties is a polynomial function of temperature [28, 30], while the substrate material has constant properties. Both properties of substrate and working fluid are listed in Table 2.

<table>
<thead>
<tr>
<th>Table 2. Thermo-physical property of substrate material and working fluid.</th>
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<tbody>
<tr>
<td><strong>Thermo-physical property</strong></td>
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<tr>
<td>Density</td>
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<tr>
<td>Specific heat</td>
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<tr>
<td>Thermal conductivity</td>
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<td>Viscosity</td>
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The pressure drop was evaluated as a pressure difference between the inlet and outlet. The Reynolds number (Re) was calculated based on the heat sink pin fin hydraulic diameter according to:

\[
\text{Re} = \frac{\rho u D_h}{\mu}.
\]

Based on the expression below, the average Nusselt number \( \text{Nu} \) has been calculated

\[
\text{Nu} = \frac{h D_h}{k_i} = \frac{q_{eff} D_h}{(T_{avg,\text{wl}} - T_{bulk}) k_i},
\]

where the area weighted average temperature of the heat sink’s s-l interface is \( T_{avg,\text{wl}} \), \( T_{bulk} \) is the bulk volumetric working fluid temperature. As the wetted surface area is greater than the foot print area, the effective heat flux \( q_{eff} \) has been opted in \( \text{Nu} \).

The effective heat flux is the heat flux applied on the interface and has been evaluated according to Eq. (7):

\[
q_{eff} = \frac{A_{bw}}{A_{wl}},
\]

where \( A_{bw} \) and \( A_{wl} \) refer to the bottom wall surface area or foot print area and the solid-liquid contact area of the heat sink, respectively. Regardless of the configuration, both areas remain the same.

### 2.3. Boundary conditions

In current numerical work, a uniform heat flux of 500 kW/m² has been applied. Previous studies [6,11] have shown that the heat flux has little influence on microchannel heat sink properties. So, only one heat flux condition has been studied. Except for the bottom wall, all outer walls are adiabatic. Working fluid with a temperature of 298 K was taken with inlet velocity varying from 0.1 to 0.4 m/s. Pressure–velocity coupling was based on the SIMPLE algorithm, and the equations were solved using the Gauss-Seidel iterative technique.

### 2.4. Grid independence

In order to prevent any errors resulting from coarse mesh, a grid independence test was performed before the broad analysis. The uniform inline (UO) MPFHS case having unit grid size variation was considered. In total, three different cases were simulated with \( Re = 110 \) and \( q = 500 \text{ kW/m}^2 \). Table 3 enlists the complete details of the grid independence test. Two parameters – pressure drop (Δp) and average bottom wall temperature (\( T_{bw} \)) were compared for a varying number of elements. It is observed that the differences in results for fine and very fine grid structures are less than 3%, but time taken for the simulation is doubled. So, the fine mesh type is chosen in simulation of all MPFHS cases.

<table>
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<tr>
<th>Table 3. Grid independence test performed on uniform inline (UO) configuration for ( Re = 110 ).</th>
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<tbody>
<tr>
<td><strong>Mesh Type</strong></td>
</tr>
<tr>
<td><strong>Coarse</strong></td>
</tr>
<tr>
<td><strong>Fine</strong></td>
</tr>
<tr>
<td><strong>Very Fine</strong></td>
</tr>
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### 2.5. Model validation

Using the methodologies of the proposed model, the work of Ali and Arshad [29] has been reproduced to validate the present numerical work. Identical boundary conditions and similar micro pin fin geometry have been used to predict the results. Under a heat flux of 37.2 kW/m², the bottom wall temperature was compared for different flow rates. Furthermore, it is compared
to the numerical work by Ambreen and Kim [30], as they have also used a similar geometry [29]. Based on the present simulation and the literatures, the predicted average bottom wall temperature is shown in Fig. 3. Observations of the present results show decent agreement with both literatures, with an extreme deviation of 5%.

The average bottom wall temperature \( \bar{T}_{bw} \) is calculated as the area-weighted normal temperature of the lower wall of the heat sink where the heat flux is applied. Across various configurations of the heat sink, a consistent trend in the bottom wall temperature with respect to Reynolds number was observed. The general understanding is that as the Reynolds number increases, the coolant flow rate also increases, leading to more effective heat transfer from the heat sink and a decreased bottom wall heat sink temperature. A significant difference was observed between inline and staggered arrangements due to better fluid flow through the channel. Among all configurations, 3 step (staggered) configuration has shown the lowest value of \( \bar{T}_{bw} \) for \( Re > 70 \) and for \( Re < 70 \), it is UO staggered configuration. This is due to a higher heat transfer rate in this case. It can be concluded that heat sink usage at low Reynolds numbers is not justified as there is less heat transfer than.

The variation of \( \Delta p \) with Re for different MPFHS cases has been plotted in Fig. 5(a). It is recognized that as Re increases, there is a corresponding rise in the pressure drop (\( \Delta p \)) due to elevated flow resistance. Additionally, the gradient of the \( \Delta p \) curve becomes steeper as the Reynolds number value increases. Among different configurations, 3 step staggered configuration has yielded highest value of pressure drop, while UO inline configuration has shown the lowest \( \Delta p \). This is due to increase in flow obstruction in the heat sink. The gap between inline and staggered arrangements kept on increasing with the rise in Re.

To further evaluate the design efficacy of heat sinks, the thermal performance factor (TPF) was plotted in Fig. 5(b). The enhancement of the average Nusselt number through diverse design modifications invariably involves incurring a pressure drop penalty [22]. Consequently, researchers have endeavored to assess the effectiveness of these designs using various parameters such as thermo-hydraulic performance, figure of merit, and coefficient of performance [22,31–33]. In the current study, the overall performance of various heat sink configurations was evaluated based on the parameter TPF. The thermal performance factor based on Eq. (8) has been calculated to assess the overall performance of MPFHS configurations.

All configurations have shown the TPF value > 1 except at lower values of Re. The design modifications have augmented thermal characteristics with the least pressure drop penalty. Among all configurations, UO stepped configuration has the

3. Results and discussion

On simulating different heat sink configurations under heat flux of 500 kW/m², the parameters \( \bar{Nu} \) and \( \bar{T}_{bw} \) were evaluated for a series of Reynold numbers (Re). The variation of \( \bar{Nu} \) with Re for different MPFHS cases has been shown in Fig. 4(a). As depicted in the figure, with a rise in Re, there is a rise in \( \bar{Nu} \) value but with a decreasing rate. This can be attributed to an increase in entrance length. There is a distinct gap between inline and staggered configurations for all heat sinks, which is quite obvious due to enhanced fluid mixing. It is exciting to note that for inline configurations, \( 3 \times 3 \) step MPFHS has shown top performance followed by \( 3 \) step inline and UO inline configuration. While, for staggered configurations, the differentiation is not clearly visible up to \( Re < 70 \), and afterwards, \( 3 \) step (staggered) configuration has yielded a better \( \bar{Nu} \) value compared to \( 3 \times 3 \) step (staggered) and UO (staggered) configuration. The observed conduct is entirely contingent on how the working fluid moves within the heat sink.

The influence of Reynolds number (Re) on the average temperature of the bottom wall (\( \bar{T}_{bw} \)) was illustrated in Fig. 4(b) for different cases of the heat sink.

![Fig. 3. Present numerical work validation with previous work [29–30].](image)

![Fig. 4. Deviation of: (a) \( \bar{Nu} \); (b) \( \bar{T}_{bw} \) with Re for different MPFHS configuration.](image)

![Fig. 5. Deviation of: (a) \( \Delta p \); (b) TPF with Re.](image)
maximum value of $TPF$ up to $Re < 80$, while for $Re > 80$, $3$ step staggered configuration has depicted the highest $TPF$ value.

The $TPF$ value for UO stepped, $3$ step staggered and $3 \times 3$ step staggered configurations of MPFHS has shown an increasing trend up to $Re = 80$ and afterwards it starts decreasing. This is attributed to exponential enhancement in pumping power at higher $Re$ values. Whereas, the other configurations, i.e. $3$ step inline and $3 \times 3$ step inline have shown an increasing slope with the increase in Reynolds number, and at higher $Re$ values, the curves tend to flatten.

4. Conclusions

In current work, numerical relative analysis has been done for different configurations of open MPFHS. Mainly three design parameters (stepness, arrangement and directionality) are simulated and analysed using six different MPFHS configurations. Using single phase liquid water as a coolant and Cu as a substrate, present cases were compared for a range of Reynolds number and a heat flux of 500 kW/m². It can be concluded that stepness in pin fin configurations has yielded more augmentation in the inline arrangement rather than in the staggered arrangement. Furthermore, the stepped arrangement has less impact at low values of Reynolds number, i.e. for $Re < 70$, its influence kept on increasing with $Re$. Bi-directional pin fin height variation is beneficial only in the inline arrangement while has a negative impact on the staggered arrangement.

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References


Comparative Thermo-hydraulic Analysis of Periodic Stepped Open Micro Pin-fin Heat Sink


