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# On conjugate heat transfer in microchannel heat sinks

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## ABSTRACT

This work aims to study the anisotropic thermal impacts of employing liquid metal fluids including Na and NaK as coolants in two metallic microchannel heat sinks. The two studied microchannel heat sinks have aspect ratios of 1 and 2 while their hydraulic diameters are 267  $\mu\text{m}$  and 400  $\mu\text{m}$ . In the first stage, the accuracy and reliability of the developed numerical model are verified against benchmark problems and experimental data. In the second stage, the utilized verified and validated numerical model is extended to forced convection of Na and NaK in metallic microchannel heat sinks. The investigated metallic microchannel heat sinks are assumed to be out of stainless steel and Nickel with thermal conductivities of 17 W/mK and 90 W/mK, respectively. Obtained results revealed that NaK provides higher Nusselt numbers than Na regardless of the microchannel aspect ratio and hydraulic diameter. The effect of the microchannel heat sink solid substrate thermal conductivity on the temperature and heat flux distributions over the microchannel walls are also investigated comprehensively. For the investigated steady state Na and NaK flow within metallic microchannel heat sinks, the thermal conductivity of the heat sink solid substrate is observed to have a significant effect on the local temperature and heat flux distributions over microchannel walls.

## 1. Introduction

Micro-scale heat sinks are efficient at dissipating heat in confined spaces due to their high surface area-to-volume ratios. Since their conceptualization in the early 1980s, considerable research efforts have been made to advance knowledge and enhance the overall thermal and hydrodynamic performance of microchannel heat sinks. An overview of some of these studies is also presented in review papers [1,2,3]. The heat dissipation capabilities of miniature heat sinks have been explored using fluids like water, ethanol, and dielectric liquids [4]. These fluids are preferred for their high specific heat capacities and compatibility with the materials commonly used in compact and micro-scale heat exchangers. Despite their benefits, such liquids exhibit low thermal conductivity, ranging from 0.07 to 0.7 W/mK, and have boiling points between 34 °C and 100 °C. This limitation has largely confined their use

to managing heat in microelectronic components [5]. Conversely, liquid metals like Na, NaK, and GaInSn, with thermal conductivities between 16 and 79 W/mK and only slightly higher dynamic viscosities compared to water, have emerged as superior alternatives. Na and NaK, in particular, are suitable for a wide range of operational temperatures, from 150 °C up to 900 °C, due to their high boiling points. Gallium-based liquid metals are non-toxic, have melting points ranging from 19 °C to 29.7 °C [6], and exhibit thermal conductivities of about 16 to 30 W/mK [7]. These properties have made gallium-based liquid metals a topic of interest for researchers studying their flow and heat transfer characteristics in miniature heat sinks. Luo and Liu conducted studies to understand the heat transfer properties of GaInSn in compact heat sinks, observing a gradual increase in Nusselt numbers with the Peclet number in laminar flow conditions. Their findings led to a new correlation for estimating the GaInSn Nusselt number based on the Peclet number [8]. Additional research into laminar forced convection of water and liquid

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| Nomenclature |                                                           |                  |                                      |
|--------------|-----------------------------------------------------------|------------------|--------------------------------------|
| $C_p$        | specific heat (J/kgK)                                     | $u$              | fluid velocity (m/s)                 |
| $D$          | diameter ( $\mu\text{m}$ )                                | $W$              | microchannel width ( $\mu\text{m}$ ) |
| $D_h$        | Equivalent hydraulic diameter ( $\mu\text{m}$ )           | $x, y, z$        | coordinate system                    |
| $H$          | microchannel height ( $\mu\text{m}$ )                     | <i>Subscript</i> |                                      |
| $h$          | convective heat transfer coefficient (W/m <sup>2</sup> K) | ave              | average                              |
| $k$          | thermal conductivity (W/mK)                               | b                | bulk                                 |
| $Kn$         | Knudsen number, $X \cdot D_h$                             | in               | microchannel inlet                   |
| $L$          | microchannel length (mm)                                  | f                | fluid                                |
| $n$          | normal vector                                             | s                | solid                                |
| $Nu$         | Nusselt number, $hD_h/k$                                  | <i>Greeks</i>    |                                      |
| $q$          | heat flux (W/m <sup>2</sup> )                             | $\Delta P$       | pressure difference (kPa)            |
| $P$          | pressure (Pa)                                             | $\alpha$         | microchannel aspect ratio, $H^*W$    |
| $Pr$         | Prandtl number, $\mu C_p/k$                               | $\beta$          | slip length ( $\mu\text{m}$ )        |
| $Pe$         | Peclet number, $RcPr$                                     | $\lambda$        | fluid mean free path (m)             |
| $Re$         | Reynolds number, $\rho D_h u/\mu$                         | $\mu$            | dynamic viscosity (kg/m·s)           |
| $T$          | temperature (K)                                           | $\rho$           | density (kg/m <sup>3</sup> )         |

metals in miniature heat sinks showed that models predicting thermal performance align well with experimental data for water, but less so for liquid metal flows [9]. Heat sinks utilizing liquid metals as coolants demonstrated lower overall thermal resistance compared to those using water. In particular, experiments by Zhang et al. [10] on GaInSn flow and heat transfer within a copper mini-channel heat sink revealed that the Nusselt numbers for laminar flow were significantly lower than the theoretical predictions for fully developed laminar flows [11]. Furthermore, numerical simulations by Muhammad et al. highlighted the differences in pressure loss and thermal efficiency among various liquid metal alloys in miniature heat sinks, with GaIn showing the best performance due to its superior thermophysical properties [12]. Finally, Miner and Ghoshal's experimental investigation into the flow and heat transfer of GaInSnZn liquid metal in a micro-scale heat sink underscored the potential of liquid metals in enhancing heat dissipation efficiency, despite the challenges associated with their use [13].

Liquid metals are playing a pivotal role in the development of next-generation nuclear reactors, offering significant advancements in safety, efficiency, and sustainability. These reactors, often referred to as Generation IV reactors, aim to address the limitations of current nuclear technology by using innovative materials and designs. Liquid metals, such as sodium (Na), lead (Pb), lead-bismuth eutectic (LBE), and molten salt, are among the key materials being explored for their exceptional qualities as coolants and, in some cases, as fuel carriers. Liquid metals, such as sodium (Na), lead (Pb), lead-bismuth eutectic (LBE), and gallium, offer high thermal conductivity. This property is critical for efficient heat transfer from the reactor core to the energy conversion system, making the reactor more compact and efficient. The high boiling points of liquid metals allow the reactor to operate at high temperatures without the need for pressurization. This reduces the risk of pressure-related accidents and simplifies the reactor design, which is especially important for portable applications where robustness and safety are paramount. The natural circulation capability of liquid metals can be harnessed for passive cooling, reducing reliance on active safety systems. In the event of a shutdown or malfunction, the reactor can cool down naturally without external power, enhancing its safety profile in varied and potentially challenging environments. Some portable reactor designs utilizing liquid metals can operate on a fast neutron spectrum, which improves fuel efficiency and reduces waste. They can be designed to use a variety of fuel types, including enriched uranium, depleted uranium, and even spent nuclear fuel, offering flexibility in fuel choice and supply.

Numerous studies have explored the dynamics of liquid metal flow and heat transfer across a wide range of applications, from nuclear

reactor cooling to solar power systems, focusing primarily on large-scale devices with hydraulic diameters exceeding 5 mm [13,14]. However, research on the convective heat transfer characteristics of gallium-based liquid metals in miniature heat sinks is less common. Notably, detailed investigations into the behavior of sodium (Na) and sodium-potassium alloy (NaK) in micro-scale heat sinks, especially in contexts requiring high operational temperatures, appear to be scarce. Na and NaK, with boiling points around 800–900 °C, are promising as high-temperature coolants for applications such as micro-reactors and fission batteries, where they can facilitate efficient heat transfer from the reactor core to the power conversion units in systems generating from a few kilowatts to up to 1 megawatt. These systems might utilize lead or sodium as the primary coolant [15]. Moreover, the development of engineered heat transfer structures could significantly reduce the required surface area by 30 to 50 %, thus benefiting the design of lighter and more compact micro-reactors [16]. Despite the extensive body of research on the fluid dynamics and heat transfer of Na and NaK in larger-scale and macro-scale heat exchangers, the literature on their use in forced convection within microchannel heat sinks is notably limited. Therefore, there is a clear need for further research to thoroughly investigate the flow and conjugate heat transfer processes in micro-scale heat sinks cooled with Na and NaK liquid metals [17].

In recent studies, miniature heat sinks have been explored as critical components in the cooling systems of devices operating under extreme conditions, characterized by very high heat fluxes and elevated temperatures. These devices include high-power laser diode arrays, MW-class gyrotron resonators, and Na-cooled thermal-spectrum fission batteries. Notably, miniature heat sinks have been successfully integrated into the cooling loops of MW-class gyrotron resonators over the past two years [18]. Another noteworthy investigation introduced a mini-channel heat sink, cooled by the liquid metal GaInSn, to manage heat in a high-power laser diode array [19]. The demanding nature of these applications requires novel cooling strategies. Heat transfer enhancement can be achieved through different methods including creating flow disruption, altering the shape of the channel or its curvature, introducing secondary flows, channel roughness and attachments to increase surface area and turbulence [20,21,22]. Bhandari and Prajapati, for example, examined the effects of fin on the performance of open microchannel heat sinks where the Reynolds number varies from 100 to 800 [23]. It was observed that increasing the fin height up to 1.5 mm augmented the heat transfer rate and beyond this height heat dissipation capacity of the heat sink dropped.

For the laminar flow of Na and NaK within microchannel heat sinks, small flow Peclet number as well as conjugate heat transfer can

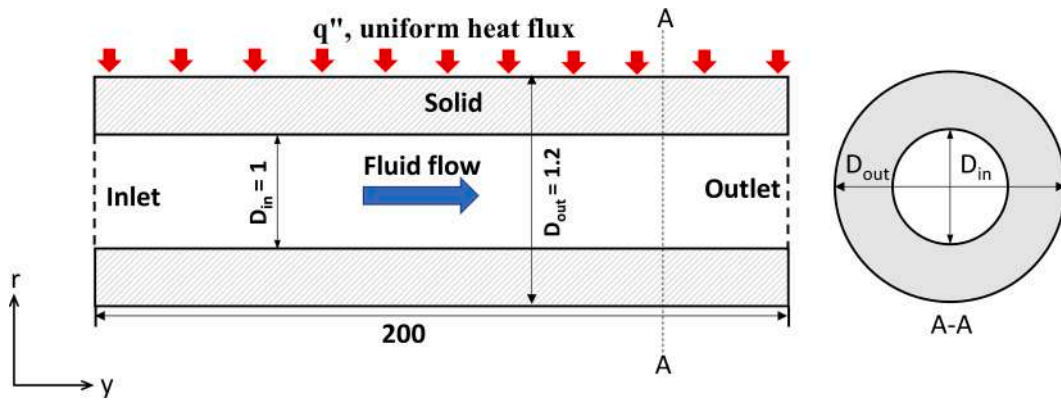


Fig. 1. Schematic of the investigated tube geometry and the applied boundary conditions (all dimensions are in mm).

significantly affect the thermal-hydraulic performance of small-scale heat sinks [10,24]. Therefore, in this research work, we perform a comprehensive 3-D numerical study of the Na and NaK flow and heat transfer within microchannel heat sinks of different aspect ratios and hydraulic diameters. It is notable that the focus of this article is heat transfer without any phase change. However, multiphase analysis for sodium (Na) and sodium-potassium (NaK) alloys, when used as coolants in nuclear reactors, is a critical area of study within nuclear engineering. This analysis involves understanding the interactions between different phases (solid, liquid, gas) of these metal coolants under various operational conditions. Sodium and sodium-potassium alloys are considered for their excellent thermal conductivity, high boiling points, and low vapor pressure at operating temperatures, which make them effective for heat transfer in nuclear reactors. The multiphase analysis of sodium and sodium-potassium coolants in nuclear reactors is a sophisticated field that requires a deep understanding of thermal physics, fluid dynamics, and materials science. Despite the challenges, such analysis is crucial for the safe and efficient design of nuclear reactors, particularly those aiming to utilize these high-performance liquid metals as coolants. The ongoing research and development efforts aim to enhance the predictive capabilities of multiphase models and to ensure that future nuclear reactors can leverage the benefits of Na and NaK coolants while mitigating their risks [25]. Solid-liquid phase change plays an important role in metallurgy, building energy saving, solar thermal energy storage, and thermal management of electronic equipment [26]. Ye and Arici performed comprehensive modeling of solid-gallium phase change using enthalpy-porosity methodology [27,28]. The simulation was performed using 2D and 3D domains and the results were validated against experimental data. It was observed that both 3D and 2D simulations satisfactorily predicted the results such as temperature and solid-liquid interface position, suggesting the preferability of 2D simulation due to its considerably lower running time [28]. The authors also provided correlations for parameters such as Nusselt number, volumetric melting rate and mean liquid layer thickness [27].

The main objective is to evaluate the effects of two concurrent parameters of flow Peclet number and conjugate heat transfer, on the thermal-hydraulic performance of the investigated microchannel heat sinks. These two concurrent parameters are here investigated for the first time for Na and NaK flow and heat transfer within micro-scale heat sinks for applications involving high working temperatures. As the first step in this research work, the accuracy and reliability of the developed numerical model are first verified against benchmark problems as well as available experimental data in the literature. In the second section of this research work, the effect of coolant flow Peclet number and conjugate heat transfer on the convective heat transfer and pressure drop within microchannel heat sinks is checked. To fulfill this objective, the established and verified numerical procedure in the first section of this work is utilized to assess the thermal performance of micro-scale heat sinks with different aspect ratios and hydraulic diameters having Na and

NaK as the coolants with an operating temperature range of 300–600 K. The investigated operating temperature range in this work can include a wide range of applications such as high-power laser diodes and fission batteries [29]. The obtained Nusselt numbers are presented and discussed as a function of inlet Peclet numbers. Local heat flux and temperature distributions along the microchannel walls are studied in detail to examine the effect of conjugate heat transfer on Nusselt numbers within miniature heat sinks. The impact of heat sink thermal conductivity on local temperature, heat flux, and Nusselt number distributions are also investigated and discussed in detail.

## 2. Numerical approach

To perform the computational analysis, the finite volume method was applied for a three-dimensional domain model using Ansys Fluent to solve simultaneously the governing equations in fluid and solid regions. Discretized equations are solved using SIMPLE solver schemes while a second-order upwind interpolation scheme is applied to convection terms. Note that the continuum assumption and no-slip boundary condition hold when the Knudsen number of the flow is less than 0.001 [30, 31]. The Knudsen number ( $Kn = \lambda/L$ ) is a dimensionless number that represents the ratio of the fluid particles' mean free path ( $\lambda$ ) to the characteristic length of the flow ( $L$ ).  $\lambda$  is defined as the average distance between particle collisions and is usually on the order of  $10^{-10}$  m (1 Angstrom) for most liquids such as water, as well as liquid metals. It is noted that the nature of the mean free path in liquid metals due to ionization is different from those for conventional coolants such as water. However, their length scale is on the order of Angstroms. For example, sodium becomes a positive ion when it melts, while the mean free path of the electrons in liquid metals is on the order of 1 to 8 Angstrom [32]. The  $Kn$  number can be used to categorize the flow regime and validity of the Navier-Stokes equation, as well as the no-slip boundary condition in a small-scale device [30,31]. For liquid flows within the investigated small-scale heat sinks in this work, the largest  $Kn$  is almost  $10^{-6}$ ; this occurs for the heat sink with a hydraulic diameter of 267  $\mu\text{m}$ , which is the smallest hydraulic diameter investigated in this work.

The governing equations for incompressible and steady-state fluid flow are presented by Eqs. (1) to (3), namely, the continuity, Navier-Stokes, and energy equations for an incompressible laminar flow. The steady-state heat conduction equation with a temperature-dependent thermal conductivity, given by Eq. (4) is also solved for the solid substrate of the heat sink. Moreover, steady-state numerical simulations are performed in this work with a no-slip boundary condition at the solid-fluid interfaces (walls) within investigated miniature heat sinks. The fluid flow within the investigated microchannel heat sinks is considered laminar since the maximum Reynolds number in the performed numerical simulations is 1900. The coolant enters the investigated heat sinks with a uniform velocity and a constant temperature in all

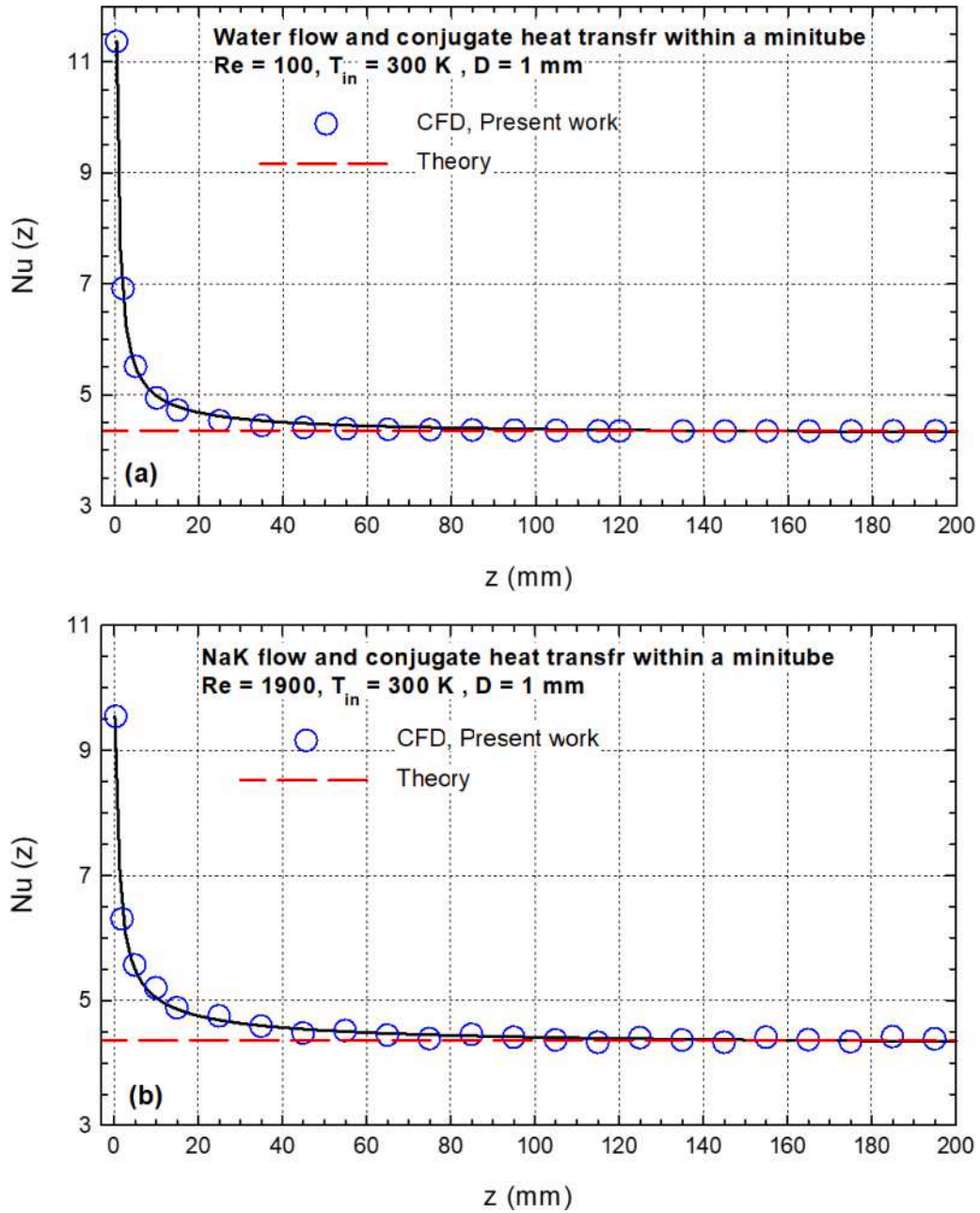


**Fig. 2.** Local Nusselt numbers of (a) water and (b) NaK along a tube with an inner diameter of 1 mm and thickness of 0.2 mm under a uniformly applied heat flux.

performed numerical simulations in this work.

$$\nabla \cdot (\rho u) = 0 \quad (1)$$

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

$$\nabla \cdot (\rho c_p u T) = \nabla \cdot (k_f \nabla T) \quad (3)$$

$$\nabla \cdot (k_s \nabla T) = 0 \quad (4)$$

Laminar NaK and water flow and conjugate heat transfer within a tube for two different Reynolds numbers of 1900 and 100 at an applied constant heat flux of  $10,000 \text{ W/m}^2$  are investigated as a case study in this section to examine the accuracy and validity of the conducted numerical simulations in the present work. The tube has outer and inner diameters of 1.2 mm and 1 mm, respectively with a length of 200 mm as illustrated in Fig. 1. Liquid metal thermophysical properties were assigned to vary with temperature in all performed numerical

simulations [33].

Na:

$$\rho \left( \frac{\text{Kg}}{\text{m}^3} \right) = 1016.9 - 0.23914 T (\text{K}) \quad (5)$$

$$k \left( \frac{\text{W}}{\text{m} \cdot \text{K}} \right) = 99.5 - 3.91 \times 10^{-3} T (\text{K}) \quad (6)$$

$$C_p \left( \frac{\text{J}}{\text{Kg} \cdot \text{K}} \right) = 1611.3 - 0.75564 T (\text{K}) + 3.9773 \times 10^{-4} T (\text{K})^2 \quad (7)$$

$$\mu (\text{Pa} \cdot \text{s}) = 0.007034 \exp[-0.00811 T (\text{K})] + 0.0004864 \exp[-0.001007 T (\text{K})] \quad (8)$$

NaK:

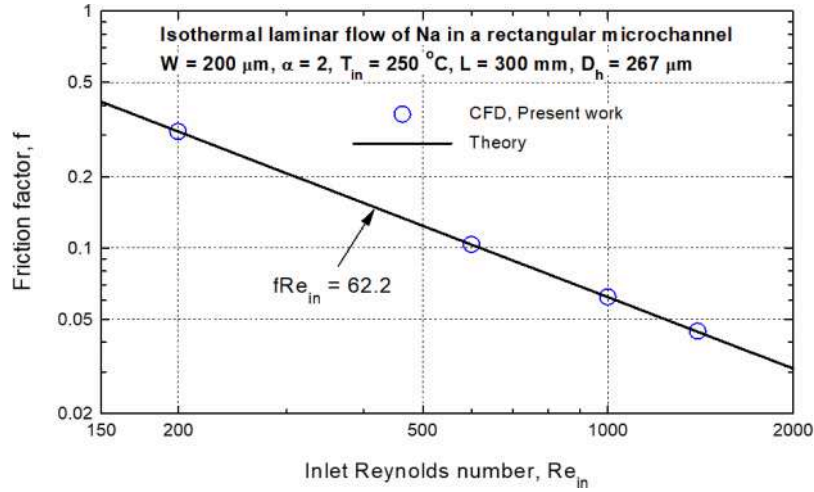


Fig. 3. Comparison between numerical results and the theoretical values of friction factors for isothermal laminar flow of Na in a rectangular microchannel of aspect ratio 2 at different Reynolds numbers.

$$\rho \left( \frac{\text{Kg}}{\text{m}^3} \right) = 930.523 - 0.237642 T (\text{K}) \quad (9)$$

$$k \left( \frac{\text{W}}{\text{m} \cdot \text{K}} \right) = 15.0006 + 0.0302877 T (\text{K}) - 2.08095 \times 10^{-5} T^2 (\text{K}) \quad (10)$$

$$C_p \left( \frac{\text{J}}{\text{Kg} \cdot \text{K}} \right) = 1074.68 - 0.454098 T (\text{K}) + 1.8816 \times 10^{-4} T^2 (\text{K}) + 7.80983 \times 10^{-8} \times T^3 (\text{K}) \quad (11)$$

$$\mu (\text{Pa} \cdot \text{s}) = 0.00235203 \exp[-0.00603235 T (\text{K})] + 0.000251016 \exp[-0.000650538 T (\text{K})] \quad (12)$$

NaK and water enter the tube through the inlet at a constant temperature of 27 °C while their uniform velocity values at the tube inlet result in Reynolds numbers of 1900 and 100, respectively. The pressure outlet boundary condition is applied to the tube exit, while a no-slip boundary condition is imposed on the tube's inner wall. The thickness of the tube is 0.1 mm as shown in Fig. 1.

Figs. 2(a) & (b) present the obtained numerical results for local Nusselt numbers along the investigated minitube for water and NaK. It can be observed that the obtained local Nusselt numbers are high near the tube inlet due to the effect of the entrance region while they approach the theoretical fully developed value of 4.36 as NaK and water

move toward the tube exit. This result assures the accuracy and reliability of the implemented numerical method in this work to investigate conjugate heat transfer in small-scale devices.

In the third case study, the isothermal laminar flow of Na at different Reynolds numbers is investigated within a microchannel with a width of 400 μm, an aspect ratio of 2, a hydraulic diameter, and a length of 300 mm. The obtained numerical results for fully developed pressure drop as well as Na average velocity are inserted into Eq. (13) to calculate the Darcy friction factor at each Reynolds number.

$$\Delta P = \frac{f L \rho V^2}{2 D_h} \quad (13)$$

where  $f$  is the Darcy friction factor,  $L$  is the length of the heat sink,  $\rho$  is the coolant density,  $V$  is the coolant average velocity, and  $D_h$  is the microchannel hydraulic diameter.

The calculated Darcy friction factors in Fig. 3 are compared against the theoretical ones for laminar flow within a rectangular channel with an aspect ratio of 2,  $fRe_{in} = 62$  [34]. Excellent agreement between obtained numerical results and theoretical values proves the accuracy and reliability of obtained results of pressure drops in the conducted numerical simulations in this work.

Besides the investigated case studies above for water and NaK forced convection within a uniformly heated tube, authors have successfully studied in their previous work the conjugate heat transfer in water and

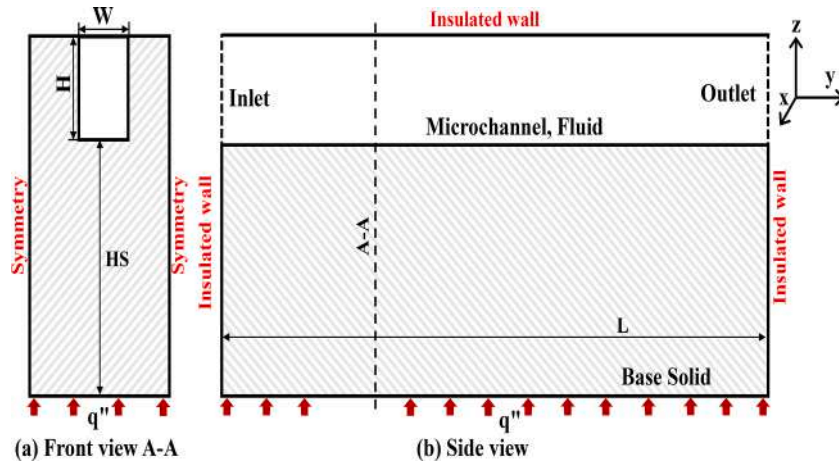


Fig. 4. Schematic of the numerical domain for a miniature heat sink investigated in the present numerical simulations. The origin of the coordinate is placed at the microchannel inlet.

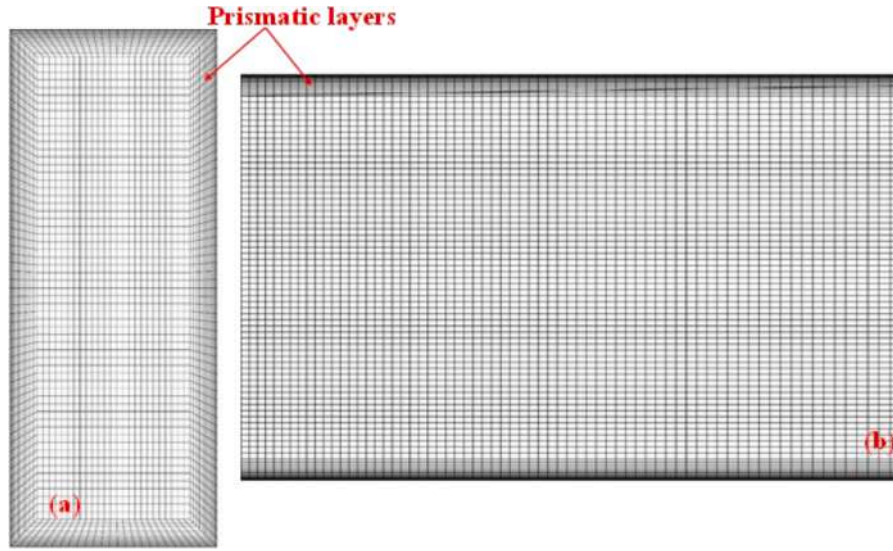


Fig. 5. Mesh grid configuration for (a) channel cross-section and (b) in a section along the channel.

Table 1

Sensitivity of the obtained Nusselt numbers of GaInSn flow in the investigated rectangular miniature heat sink to mesh grid refinement at the Reynolds number of 1500.

| Mesh Grid    | Total mesh elements (Millions) | $Nu_{ave}$ |
|--------------|--------------------------------|------------|
| Coarse       | 0.92                           | 3.934      |
| Intermediate | 1.9                            | 4.566      |
| Fine         | 3.85                           | 4.597      |
| Finest       | 7.15                           | 4.588      |

Na/K-cooled small-scale rectangular heat sinks using the same numerical procedure utilized in this work. More details on the implemented numerical procedure and utilized mesh grid configurations can be found in the authors' previous publications [29,35,36].

### 3. Liquid metal flow and heat transfer within miniature heat sinks

In this section, the objective is to investigate the concurrent effects of flow Peclet number and conjugate heat transfer on the thermal-

hydraulic performance of Na and NaK-cooled microchannel heat sinks in the laminar flow regime. Another set of numerical simulations is first conducted in this section to further verify the accuracy and reliability of the applied numerical procedure for laminar forced convection of small Prandtl number fluids. To fulfill this task, GaInSn within a copper-based miniature heat sink is examined. The copper-based mini channel heat sink has a height of 5 mm, a width of 1 mm, an aspect ratio ( $\alpha = H/W$ ) of 5, and a length of 34 mm as shown schematically in Fig. 4. A uniform mass flow rate boundary condition is assigned for the coolant at the heat sink inlet while a pressure outlet is considered for the heat sink exit. A uniform heat flux of  $300 \text{ W/cm}^2$  is applied to the heat sink while the no-slip boundary condition is imposed on the minichannel walls [35]. As represented in Fig. 5, 15 prismatic layers with a growth multiplier of 1.15 are added adjacent to the channel walls. This mesh configuration shown in Fig. 5 was already examined, utilized and verified in the authors' previous work for conjugate heat transfer modeling in rectangular miniature heat sinks [24,29].

Four different mesh configurations are investigated to assess the sensitivity of the obtained numerical results to the mesh grid refinement. Table 1 summarizes the sensitivity of the obtained numerical results for average Nusselt to mesh grid refinement in the rectangular heat sink at

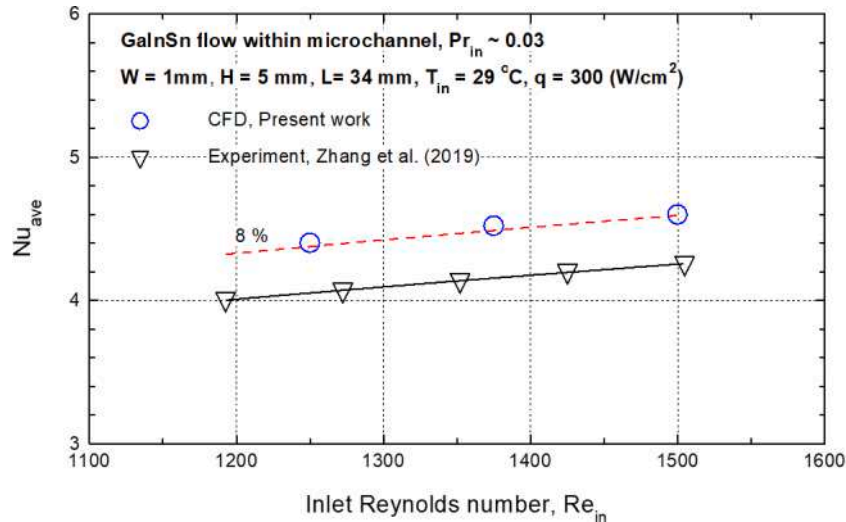


Fig. 6. The obtained numerical data and reported experimental data for GaInSn flow and heat transfer within a copper heat sink at an applied heat flux of  $300 \text{ W/cm}^2$ .

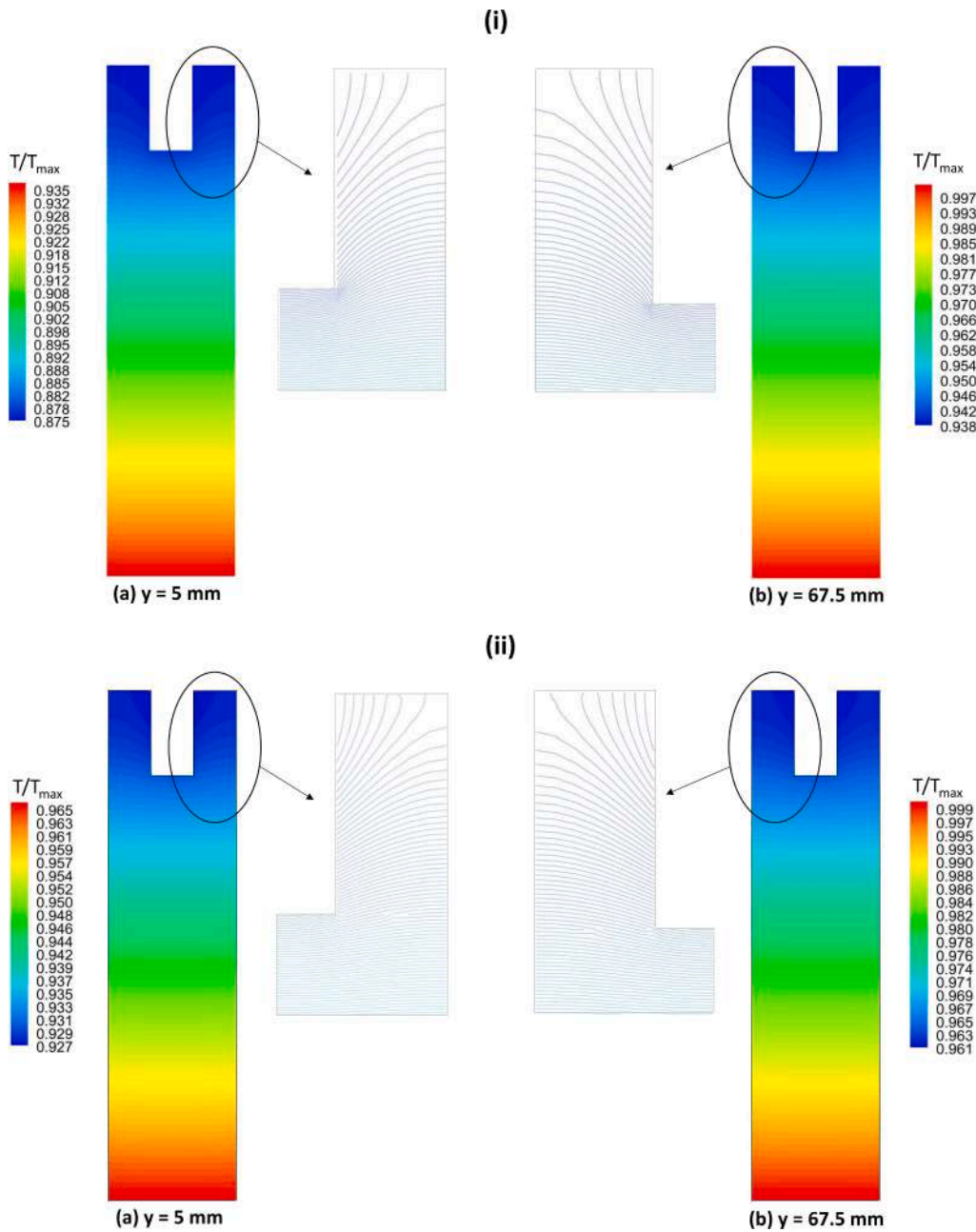


**Fig. 7.** Normalized temperature distribution at two axial locations ( $y = 5$  mm, and  $67.5$  mm) along the solid substrate of the investigated stainless steel microchannel heat sink of aspect ratio 2 at Reynolds number of 1900. (i) Na flow (ii) NaK flow.

the Reynolds number of 1500. As can be seen from Table 1, the obtained Nusselt numbers change negligibly between intermediate, fine, and finest mesh grid configurations. This is mainly due to the already fine mesh elements used in the intermediate mesh grid configuration. The fine mesh in Table 1 is used to conduct the numerical simulations in this study.

Fig. 6 compares the obtained numerical results and reported experimental data for GaInSn flow within the investigated minichannel heat sink with an applied heat flux of  $300 \text{ W/cm}^2$  [10]. Local Nusselt numbers at several axial locations along the investigated heat sink are first calculated using local wall temperatures and local wall heat fluxes. Secondly, average Nusselt numbers are derived by taking the integral of the obtained local Nusselt number distribution along the heat sink. More information on the utilized numerical procedure is provided and discussed in [24,29].

The reported numerical results increase with the Reynolds number

and follow the same trend as experimental data within an error margin of 8 %. It has been shown in the literature [36,37] that the conjugate heat transfer in the small-scale heat sinks causes non-uniform temperature and heat flux distributions along the micro-scale heat sink walls. This was reported in the literature to cause deviation between numerical results and experimental data on average Nusselt numbers within rectangular micro-scale heat sinks by up to 15 % [36,38]. Overall, the two test cases and the reported results in Figs. 2,3 and 6 verify the accuracy of the implemented numerical approach in this work to study laminar flow and heat transfer of liquid metals in miniature heat sinks.

The performed numerical simulations are extended to evaluate in detail the Na and NaK flow and heat transfer in microchannel heat sinks of different aspect ratios and hydraulic diameters. The first microchannel heat sink has an aspect ratio of 1, a width of  $400 \mu\text{m}$  and a hydraulic diameter of  $400 \mu\text{m}$ , while the second microchannel heat sink has an aspect ratio of 2, a width of  $200 \mu\text{m}$ , a height of  $400 \mu\text{m}$  and

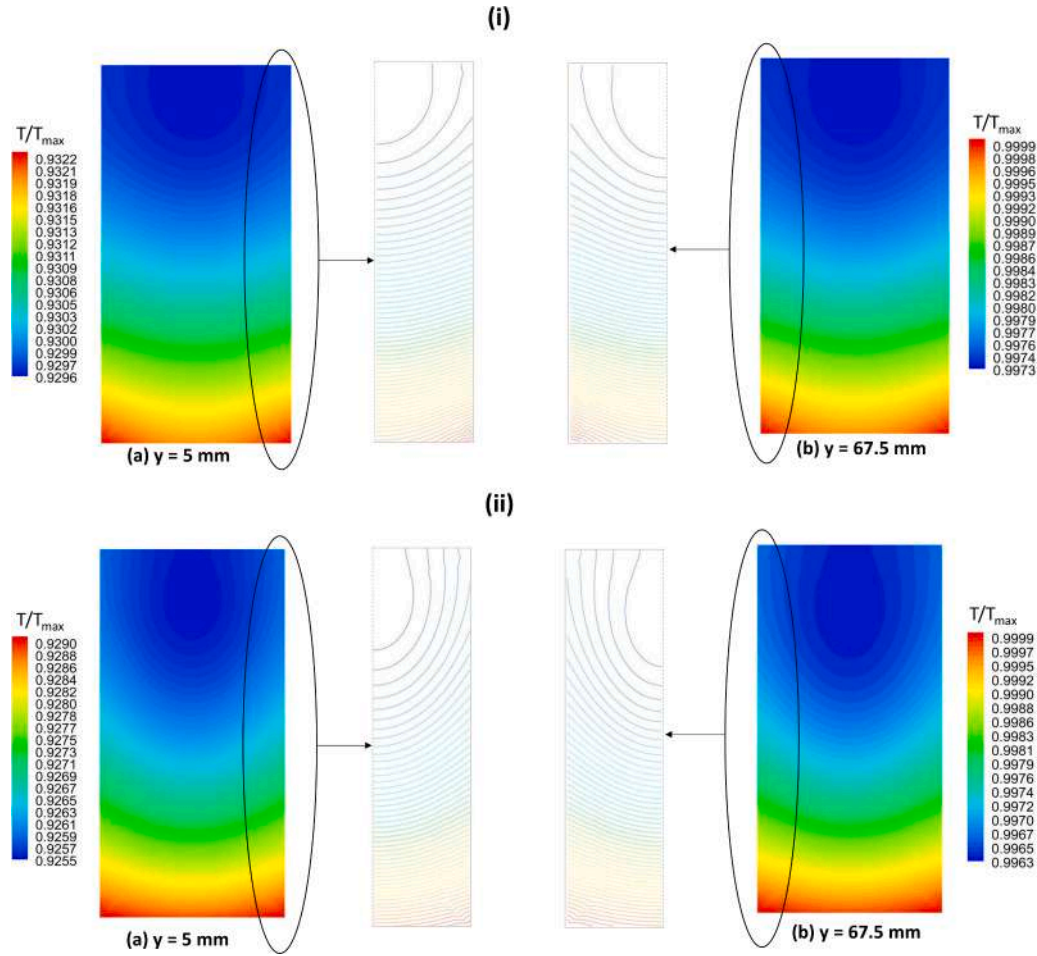


**Fig. 8.** Normalized temperature distribution at two axial locations ( $y = 5$  mm, and 67.5 mm) along the channel of the investigated stainless steel microchannel heat sink of aspect ratio 2 at Reynolds number of 1900. (i) Na flow (ii) NaK flow.

hydraulic diameter of 267  $\mu\text{m}$ . The NaK inlet temperature is set at 27  $^{\circ}\text{C}$ , while the inlet temperature for Na flow is set to 250  $^{\circ}\text{C}$ . Pressure outlet boundary condition is assigned for the heat sink outlet. Flow is considered laminar and incompressible while no-slip boundary condition is applied to the fluid-solid interfaces (channel walls). All performed numerical simulations are for steady-state cases while the convergence criteria (residuals) for continuity and energy balance equations are set as  $10^{-8}$  and  $10^{-15}$ , respectively. All the thermophysical properties of the working fluids, Na and NaK, vary with temperature [33], (Eqs. (5-12)).

Figs. 7 to 9 illustrate normalized temperature and velocity distribution at two axial locations of  $y = 5$  mm and 67.5 mm along the investigated stainless steel microchannel heat sink of aspect ratio 2 for both Na and NaK flow at a Reynolds number of 1900. The normalization has been performed with respect to maximum temperature and velocity in the fluid domain within investigated heat sinks. As expected, the temperature in fluid and solid regions increases from the axial location of 5 mm near the inlet to the axial location of 67.5 mm near the heat sink outlet. The velocity of the coolant also shows its maximum value in the center of the rectangular cross-section in compliance with a fully developed flow in axial locations near the heat sink outlet.

Fig. 10(a) presents the average Nusselt numbers as a function of the Peclet number for both liquids in two microchannel heat sinks with aspect ratios of 1 and 2. Two different metals of Nickel and Stainless steel have been assigned for heat sink solid substrates in the numerical simulations. The thermal conductivity of Nickel was set at 90 W/mK and Stainless steel was assumed to have a thermal conductivity of 17 W/mK. The Peclet numbers are calculated at the heat sink inlet. As shown in Fig. 10(a), the obtained Nusselt numbers increase slowly with increased

inlet Reynolds number in the laminar flow regime. The investigated Peclet number range in Fig. 10(a) is less than 50 for both Na and NaK. As reported in the literature, the effect of heat conduction becomes important when the flow Peclet number is low in the range of 50 [39]. In compliance with other research endeavors on internal forced convection of liquid metals in macro-scale heat sinks [40,41], the average Nusselt numbers in Na/NaK-cooled miniature heat sinks vary little with the Peclet number in Fig. 10(a). In other words, the Nusselt number is a weak function of the Reynolds number and increases slowly with increased Peclet number for the performed numerical simulations. Moreover, it can be observed that the Nusselt numbers for Na are consistently lower than those for NaK within two investigated microchannel heat sinks for the laminar forced convection of Na and NaK with the investigated Peclet number range in Fig. 10(a). For example, for the Peclet number of 10 in Fig. 10(a), the Na-cooled stainless steel microchannel of aspect ratio 2 has an average Nusselt number of around 2 while an identical NaK-cooled microchannel heat sink provides an average Nusselt number of around 2.6. Local Nusselt numbers for Na and NaK within the investigated stainless steel microchannel heat sink of aspect ratio 2 at the Reynolds number of 1900 are reported in Fig. 11. At the same Reynolds number of 1900, NaK generates higher local convective heat transfer rates than Na along the investigated heat sink. Higher local heat transfer rates for NaK contribute to its higher reported average Nusselt numbers in Fig. 10(a). Local and average Nusselt numbers (heat transfer rates) in a microchannel heat sink are a function of the coolant flow and velocity field, the coolant thermophysical properties, as well as, the heat conduction and temperature distribution within the solid substrate of the heat sink [36,38]. The observed

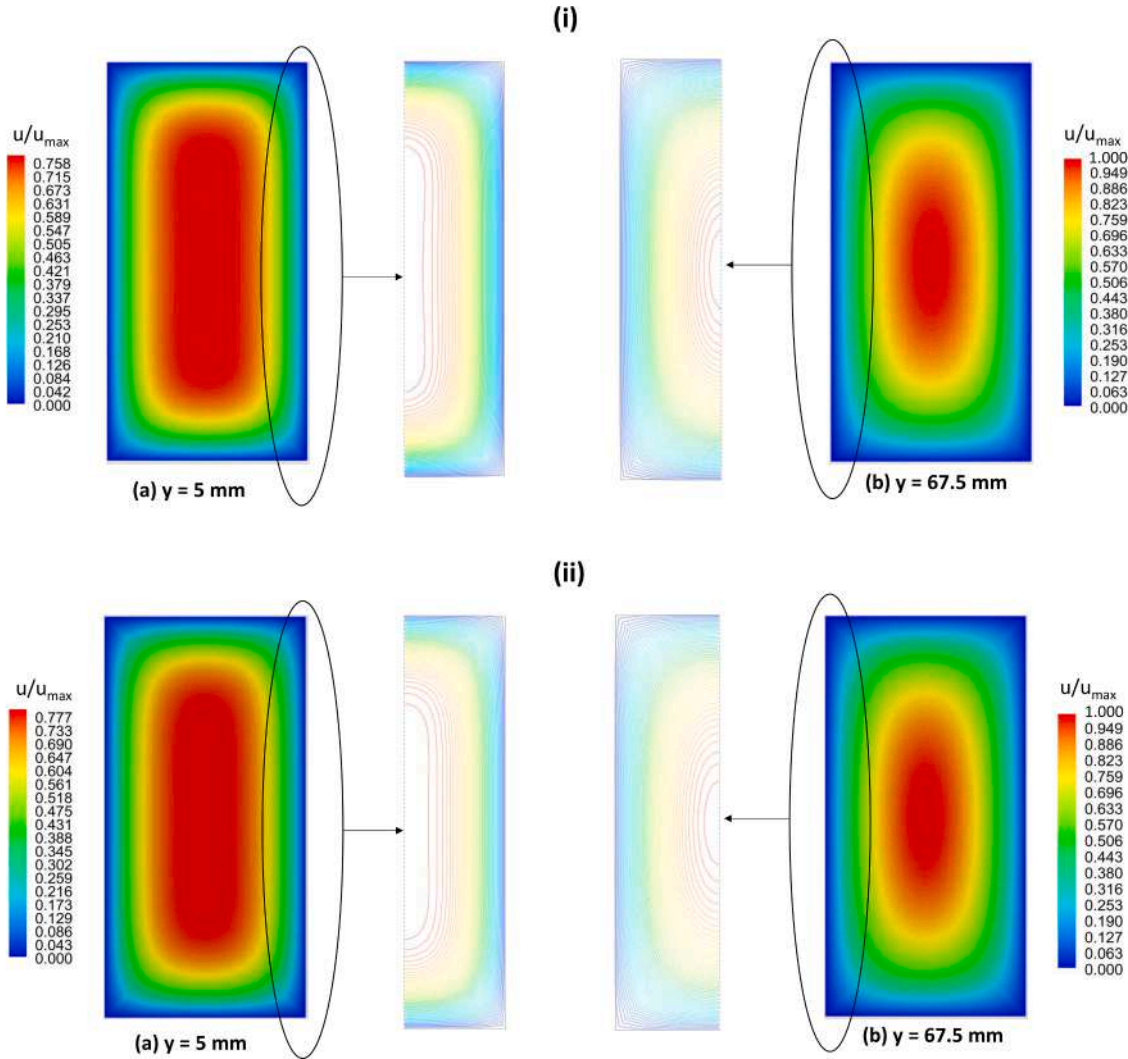


**Fig. 9.** Normalized velocity distribution at two axial locations ( $y = 5$  mm, and  $67.5$  mm) along the channel of the investigated stainless steel microchannel heat sink of aspect ratio 2 at Reynolds number of 1900. (i) Na flow (ii) NaK flow.

differences between local and average Nusselt numbers (heat transfer rates) of Na and NaK in Figs. 10 and 11 are explained by probing local temperature and heat flux distributions within the investigated heat sinks (see Figs. 12–16).

The computed friction factors are plotted as a function of the inlet Reynolds number in Fig. 10(b). The friction factors are derived by substituting the obtained pressure drops along the microchannel heat sink from the numerical simulations into Eq. (13).

As can be seen in Fig. 10(b), the obtained numerical results on friction factors for Na and NaK flow can be estimated by  $f = (21.9 / Re_{in}^{0.845})$ . Due to the effect of heat transfer adjacent to the walls of microchannels on the Na/NaK viscosities, the obtained friction factors vary with the Reynolds number to the power of 0.845. The reported Reynolds numbers in Fig. 10(a) are calculated at the heat sink inlet,  $Re_{in}$ .

Figs. 12 and 13 illustrate the bottom- and side-wall normalized temperature and the normalized heat flux distributions at two axial locations of  $y = 34$  mm and  $y = 51$  mm away from the inlet of the microchannel heat sink with  $\alpha = 2$  for Na and NaK. The horizontal axes of the figures are normalized using the microchannel width and height for the bottom and side walls, respectively. The origin of the local coordinate system is at the left bottom corner of the microchannel. The heat flux and temperature values are normalized against their local maximum values for the bottom and side walls at the two investigated axial locations, respectively. Figs. 12 and 13 are for Na and NaK flowing

at an inlet Reynolds number of 1900, through the stainless steel (SS-316) microchannel heat sink. The inlet temperatures of the Na and NaK are set to  $250^\circ\text{C}$  and  $27^\circ\text{C}$  while the applied uniform heat fluxes are  $32\text{ W/cm}^2$ , and  $10\text{ W/cm}^2$  for Na and NaK respectively.

As can be seen from Figs. 12 and 13, the Na and NaK wall temperatures and heat fluxes are distributed nonuniformly along the bottom and side walls. For the Na-cooled heat sink in Figs. 12 and 13, both the heat fluxes and the wall temperatures are lowest in the middle of the bottom wall and increase slightly near the left and right walls (i.e., the side walls). The heat flux varies by almost 15 % between its maximum and minimum values along the bottom wall. The local heat flux for the side walls of the microchannel is highest near the bottom wall and decreases continuously toward the top wall. The observed distributions in Figs. 12 and 13 show  $\sim 83\%$  difference between maximum and minimum values of normalized heat fluxes for the side walls. The heat flux distributions in Figs. 12 and 13 are almost identical for the two investigated axial locations since the local Nusselt number ( $Nu(y)$  in Fig. 11) reaches its fully developed value at the two investigated axial locations in the microchannel heat sink with an aspect ratio of 2. For the NaK-cooled heat sink at the Reynolds number of 1900 and both axial locations in Figs. 12 and 13, heat flux is the highest in the middle of the bottom wall and decreases near the left and right walls (i.e., the side walls). However, the wall temperature is the lowest in the middle of the bottom wall and increases near the side walls. The heat flux varies by

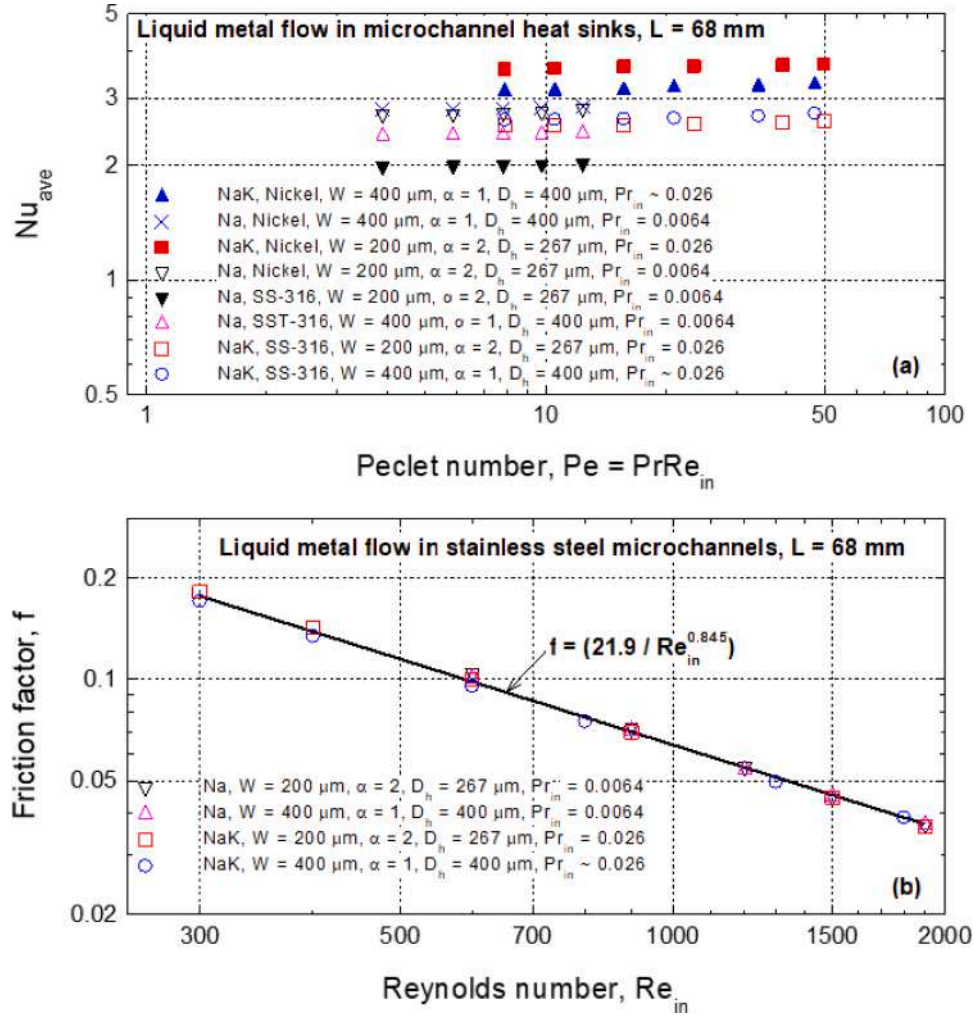


Fig. 10. Na and NaK flow and heat transfer within stainless steel microchannel heat sinks. (a) average Nusselt number (b) friction factor.

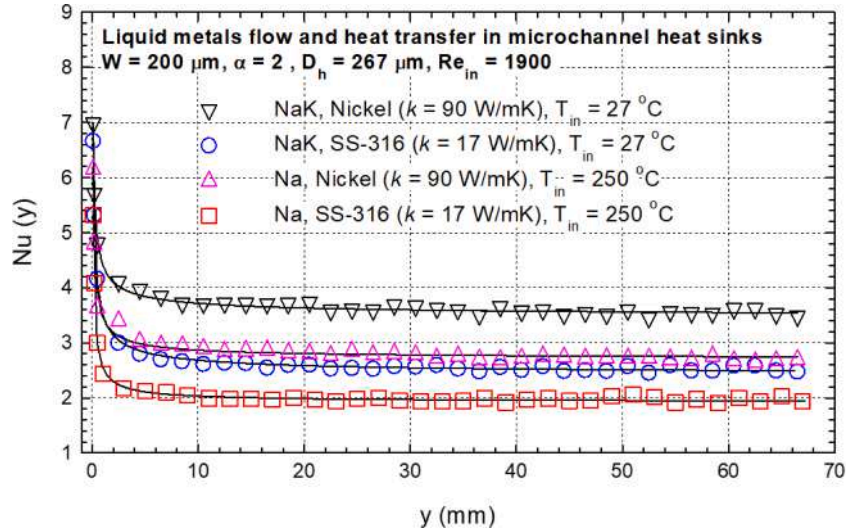


Fig. 11. Local Nusselt number for Na and NaK within the investigated stainless steel microchannel of aspect ratio 2.

almost 40 % between its maximum and minimum values along the bottom wall. Moreover, the local heat flux for the side walls of the NaK-cooled microchannel is the highest in the middle of the side wall and decreases continuously toward the top and bottom walls. The observed

distributions in Figs. 12 and 13 show a  $\sim 32$  % difference between maximum and minimum values of normalized heat fluxes for the side walls of the investigated NaK-cooled heat sink. As expected, wall temperatures are higher at axial locations of  $y = 51$  mm compared to  $y = 34$

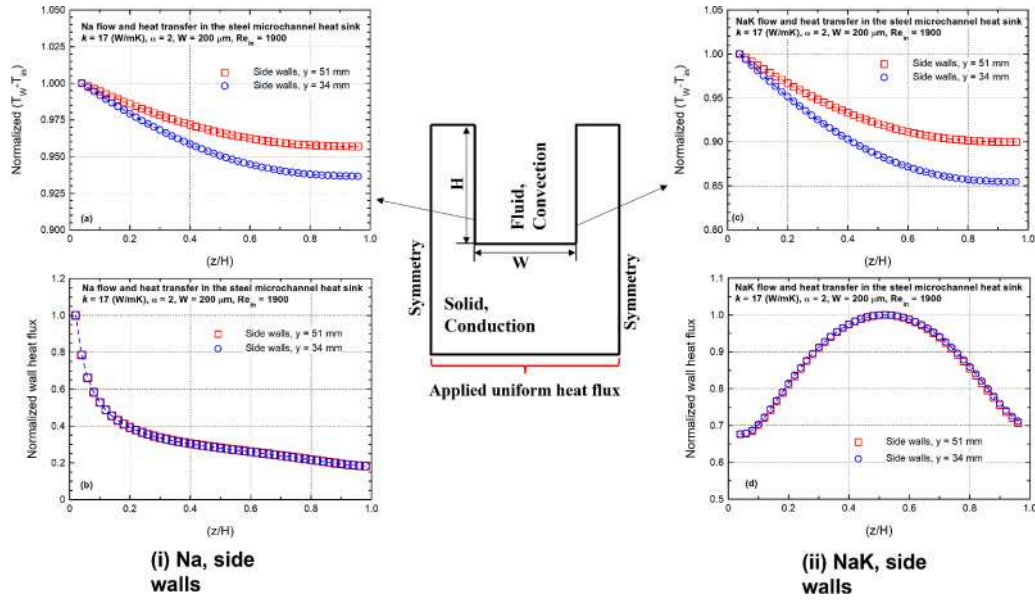


Fig. 12. Side walls temperature and normalized heat flux distributions at two axial locations along the investigated Na/NaK-cooled stainless steel microchannel ( $k = 17$  W/mK) for inlet Reynolds number 1900.

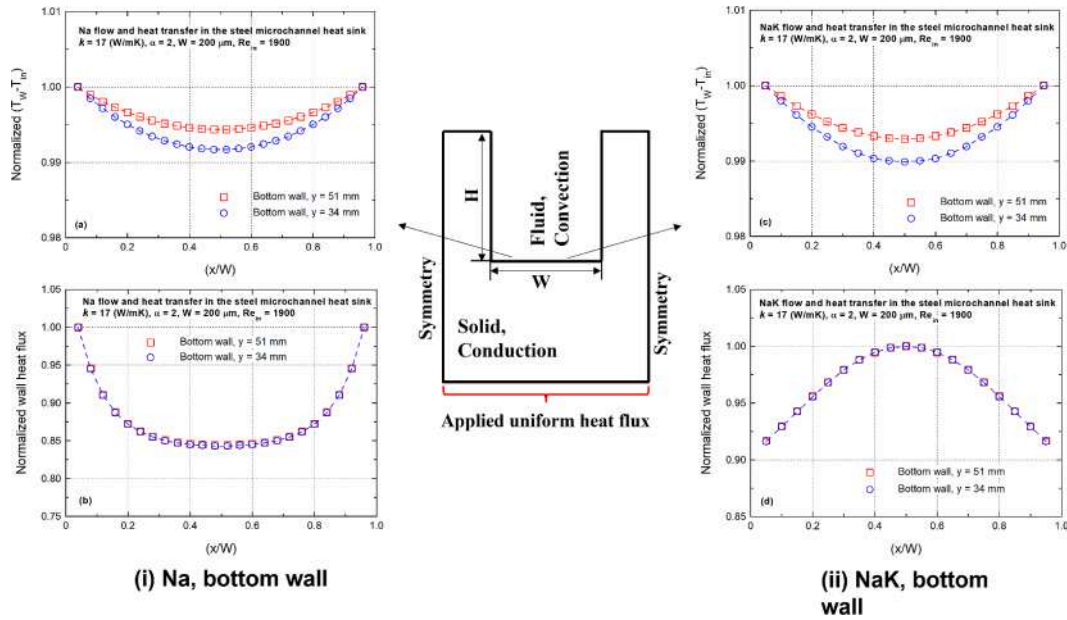


Fig. 13. Bottom wall temperature and normalized heat flux distributions at two axial locations along the investigated Na/NaK-cooled stainless steel microchannel ( $k = 17$  W/mK) for inlet Reynolds number 1900.

mm. However, heat flux distributions over microchannel walls are almost identical at the two investigated axial locations. Local Nusselt numbers almost lie in the fully developed region beyond the axial location of  $y = 10$  mm (see Fig. 11). Therefore, heat flux distributions over microchannel walls at the axial location of 34 mm and 51 mm are almost identical in Figs. 12 and 13.

Moreover, local wall temperatures and heat fluxes are used to calculate local Nusselt numbers shown in Fig. 11. Therefore, the different local temperature and heat flux distributions over the microchannel walls resulted in different local Nusselt numbers ( $Nu(y)$ ) for Na and NaK as shown in Fig. 11. The average Nusselt numbers for the investigated heat sinks are calculated by taking the integral of local Nusselt numbers over the heat sink length. Consequently, different local Nusselt numbers ( $Nu(y)$ ) for Na and NaK reported in Fig. 11 resulted in

different average Nusselt numbers in Fig. 10(a).

A new set of numerical simulations for the Na flow within a nickel microchannel heat sink having the same geometrical parameters ( $\alpha = 2$ ,  $W = 200$   $\mu$ m, and  $D_h = 267$   $\mu$ m) was done to investigate the effect of the heat sink thermal conductivity on microchannel wall temperature and heat flux distributions. The thermal conductivity of nickel is 90 W/mK, which is five times higher than the thermal conductivity of stainless steel. Local temperatures and normalized heat fluxes for side and bottom walls at two axial locations along the investigated heat sink are shown in Figs. 14 and 15.

Comparing Figs. 13(b) and 14 (b) reveals that the heat flux distribution along the bottom wall of the Na-cooled microchannel changes dramatically. In the nickel microchannel heat sink with a higher thermal conductivity, the maximum heat flux occurs in the middle of the bottom

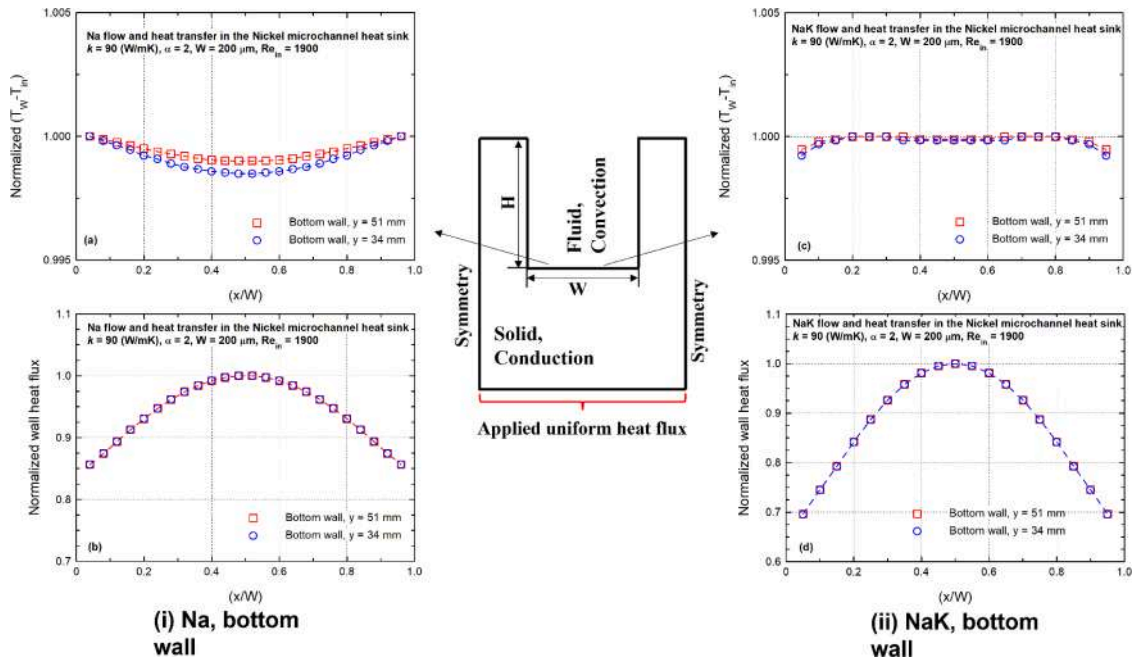


Fig. 14. Bottom wall temperature and normalized heat flux distributions at two axial locations along the investigated Na/NaK-cooled nickel microchannel ( $k = 90$  W/mK) for an inlet Reynolds number 1900.

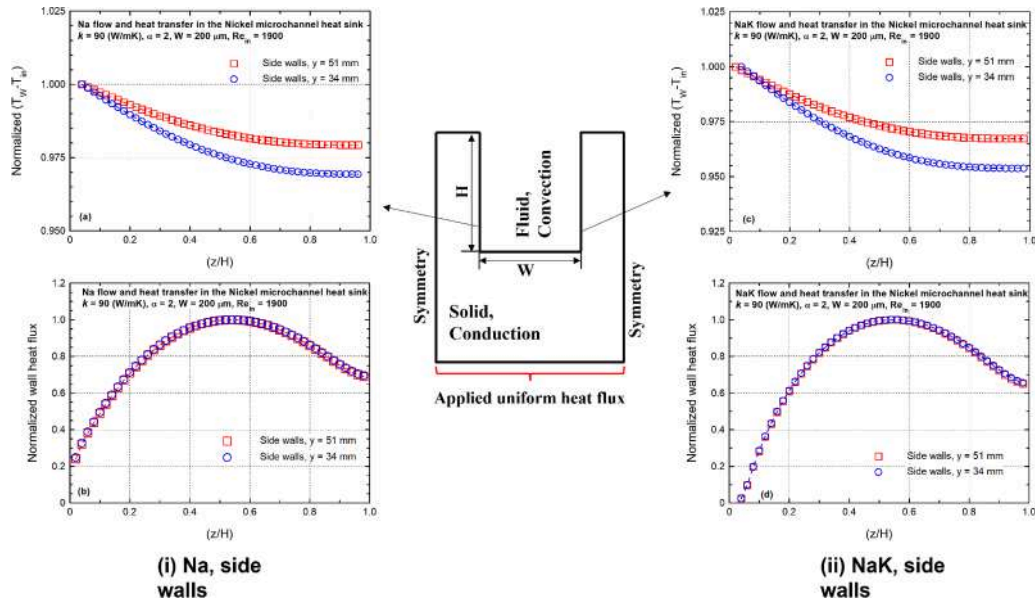


Fig. 15. Side walls temperature and normalized heat flux distributions at two axial locations along the investigated Na/NaK-cooled nickel microchannel ( $k = 90$  W/mK) for an inlet Reynolds number 1900.

wall, while it decreases near the side walls. This is the opposite behavior compared to the stainless steel heat sink in Fig. 13(b), in which the maximum heat flux occurs at the endpoints of the bottom wall and the heat flux reaches a minimum at the middle point of the bottom wall. The heat flux distribution for the side walls of the nickel microchannel heat sink is also completely different than the stainless steel heat sink as shown in Figs. 15(b) and 12 (b). As can be seen from Fig. 15(b), the maximum heat flux occurs in the middle section of the side walls, and it decreases in the regions near the bottom and top walls. This is different than the observed trend in the Na-cooled stainless steel microchannel heat sink in Fig. 12(b) where the highest heat flux occurs near the heat sink bottom wall and decreases sharply toward the heat sink top wall. Also, the side and bottom wall temperature distributions for the Nickel

microchannel heat sink in Figs. 15 (a & c) and Fig. 14 (a & c) are more uniform than the ones for the identical stainless steel microchannel heat sink shown in Figs. 12 (a & c) and 13 (a & c).

Different heat conduction scenarios are discussed and considered in Fig. 16 to explain in more detail the observed trends for wall temperature and wall heat flux distributions within investigated stainless steel and Nickel microchannel heat sinks shown in Figs. 12 to 15. As presented schematically in Fig. 16, the investigated stainless steel/Nickel heat sink has a base stainless steel section with a thickness of 2 mm, while uniform heat flux is applied at the bottom surface of the heat sink. In the context of the investigated stainless steel heat sink, heat flux diffuses along the  $z$ -axis within the structure, maintaining its initial uniform distribution until it reaches the vicinity of the microchannel

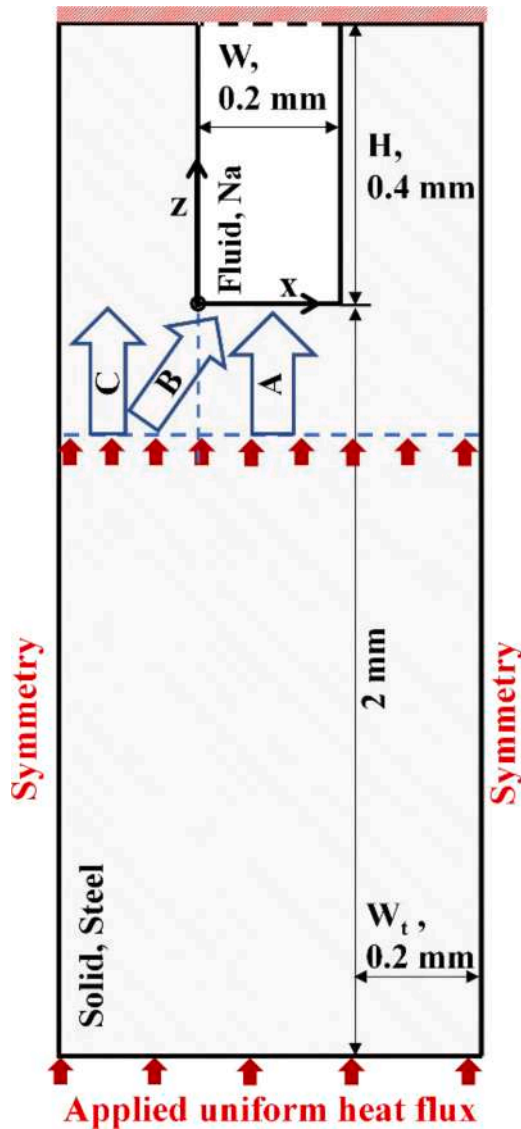


**Fig. 16.** Schematic of the front view of the stainless steel microchannel heat sink and the conduction heat flux distribution along the bottom and side walls.  $\alpha = 2$  and  $D_h = 0.267$  mm.

bottom wall. The section of uniform heat flux beneath the microchannel bottom wall, denoted by arrow 'A,' is directly transferred to the coolant flowing towards the outlet. The heat flux confined within the region between the microchannel side walls and the symmetry boundary conditions ( $W_t = 0.2$  mm) has two potential paths for diffusion: it can either move upward towards the top insulated wall (indicated by Arrow 'C') or laterally towards the microchannel bottom wall (indicated by Arrow 'B'). The height of the microchannel is 400  $\mu\text{m}$ , which is twice its width (200  $\mu\text{m}$ ) and hence acts as a bigger resistance for heat conduction within the solid substrate of the investigated stainless steel heat sink. Consequently, it is expected that more heat will be transferred to the liquid coolant through the bottom wall of the microchannel than from its side walls. In this scenario, higher heat fluxes are anticipated at the endpoints of the bottom wall compared to its middle section, primarily due to the contribution of lateral heat fluxes, as illustrated by Arrow 'B.' This discussed scenario provides insight into the observed results for bottom wall and heat flux distributions within stainless steel Na-cooled microchannel heat sinks, as depicted in Figs. 13(b).

On the other hand, Nickel, with a thermal conductivity of 90 W/mK, imposes significantly less thermal resistance to heat diffusion,

approximately five times less than stainless steel. Consequently, the heat flux represented by Arrow 'C' in Fig. 16 can efficiently diffuse upward, reaching the top wall of the heat sink. As Arrow 'C' ascends, it also diffuses laterally towards the Na flow. The entire diffused heat eventually transfers to the Na flow due to the insulated top wall of the heat sink. This explanation sheds light on the higher observed heat flux for the upper section of the side walls in Fig. 15(b), with the maximum value located in the middle section of the side wall. Furthermore, the heat flux represented by Arrow 'B' in Fig. 16 finds it easier to diffuse laterally towards the middle section of the bottom wall. This ease of lateral diffusion is primarily attributed to the smaller thermal resistance of the nickel heat sink compared to stainless steel. The heat flux laterally diffused in this manner combines with the heat flux region represented by Arrow 'A,' resulting in higher observed heat flux values for the middle section of the bottom wall in Fig. 14(b).

The local and average Nusselt numbers (representing heat transfer rates) in a microchannel heat sink are intricately linked to the coolant flow and velocity field, the thermophysical properties of the coolant, as well as the heat conduction and temperature distribution within the solid substrate of the heat sink [36,38]. The varied local temperature and heat flux distributions within the investigated Na- and NaK-cooled microchannel heat sinks in this study contribute to the diverse trends observed for the reported local and average Nusselt numbers in Figs. 10 (a) and 11.

#### 4. Conclusion

A 3D numerical procedure was developed to investigate the simultaneous effects of coolant Peclet number and conjugate heat transfer on the thermal-hydraulic performance of microchannel heat sinks. The accuracy and reliability of the applied numerical approach were first evaluated by comparing the obtained numerical results with experimental data and analytical solutions. Further numerical analyses were then performed using the verified numerical procedure to study Na and NaK liquid metal flows and heat transfer within micro-scale heat sinks. To the knowledge of the authors, this work is the first scientific endeavor investigating Na and NaK liquid metal flow and heat transfer in microchannel heat sinks at elevated working temperatures in the range of 500–600 K. The investigated operating temperature range in this work can include a wide range of applications such as high-power laser diodes and fission batteries. The Nusselt numbers for forced convection of Na and NaK within microchannel heat sinks with different aspect ratios and hydraulic diameters were observed to increase slowly with the increase of the Peclet number. This phenomenon indicated the importance of the heat conduction mechanism in the laminar flow of liquid metals within microchannel heat sinks. Despite the higher thermal conductivity of the Na, obtained results at the same Reynolds number revealed that NaK flows provided higher Nusselt numbers than the Na flows in the investigated microchannel heat sinks. Local temperature and heat flux distributions were presented and discussed to explain the observed trends for local and average Nusselt numbers for Na and NaK. Furthermore, it was observed that both temperature and heat fluxes were distributed non-uniformly along the small-scale heat sink walls. The thermal conductivity of the solid section of the microchannel heat sink was observed to have a significant effect on the microchannel wall temperature and heat flux distributions. Na and NaK have high thermal conductivities, while their boiling temperatures are in the range of 800–900  $^{\circ}\text{C}$ . These peculiarities can make Na and NaK efficient coolants in miniature heat sinks for applications involving high heat dissipation rates, such as laser diodes and batteries. Liquid metals are at the forefront of innovation in next-generation nuclear reactors, offering solutions to enhance safety, efficiency, and waste management. While challenges remain, ongoing research and development efforts are focused on overcoming these hurdles, paving the way for the broader adoption of Generation IV nuclear technologies. Efforts to integrate liquid metals into portable reactor designs are ongoing, with research focusing on addressing the

challenges of material compatibility, safety, and operational reliability. Several designs are in various stages of development, with the aims to deploy these reactors for civilian energy needs and space missions. This article reflected the thermal capacity of liquid metal sodium and sodium potassium to be used as a coolant in small-scale heat sinks.

### CRediT authorship contribution statement

**Nima Fathi:** Conceptualization, Data curation, Formal analysis, Funding acquisition, Investigation, Methodology, Resources, Software, Supervision, Validation, Visualization, Writing – original draft, Writing – review & editing. **Mahyar Pourghasemi:** Investigation, Software, Writing – original draft. **Seyed Sobhan Aleyasin:** Writing – original draft, Writing – review & editing, Methodology. **Laura Savoldi:** Writing – review & editing, Methodology. **Sal Rodriguez:** Writing – original draft, Writing – review & editing.

### Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. This work was not funded by any agencies and the research team from different universities and a National Lab are investigating a new horizon for potential future proposals and funding. If there are other authors, they declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

### Data availability

No data was used for the research described in the article.

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### Financial Disclosure

None reported

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