Thermal analysis of a microchannel heat sink cooled by two-phase flow boiling of Al$_2$O$_3$ HFE-7100 nanofluid

Ali Soleimani, Amirmohammad Sattari*, Pedram Hanafizadeh*

School of Mechanical Engineering, College of Engineering, University of Tehran, Iran

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ABSTRACT

According to advances in high heat flux devices, sufficient thermal management methods shall be developed to dissipate high heat fluxes, especially for small-scale devices. In this research, highly subcooled flow boiling of HFE-7100 containing different concentrations of alumina nanoparticles are simulated in a microchannel heat sink using Computational Fluid Dynamics (CFD). The alumina nanofluids are considered homogenous and their equivalent properties are obtained using correlations. The operating conditions of the current study include Reynolds numbers in the range of 530 to 2000 and boiling numbers in the range of $2.3 \times 10^{-3}$ to $7.1 \times 10^{-3}$. The Volume of Fluid (VOF) model is employed in two-phase flow simulations. Two sets of experimental data are used to confirm the numerical model of two-phase boiling flows in microchannel heat sinks. The results indicate that disturbance in the thermal boundary layer caused by gas-phase bubble movements near microchannel walls enhances heat transfer in the flow boiling in comparison with the single-phase flow. Therefore, the local heat transfer coefficient increases after the onset of nucleate boiling. Additionally, it is observed that the wall temperature distribution is significantly affected by boiling nucleation and bubble movements. It is concluded that the heat transfer enhancement obtained from adding alumina nanoparticles to the base fluid is not impressive in comparison with the heat transfer enhancement obtained by two-phase flow boiling instead of single-phase flow in thermal management systems.

1. Introduction

Recent developments in heat dissipative devices, such as processors, have led these devices to have higher heat generation and smaller dimensions than older models. Therefore, these devices need to be cooled using a sufficient thermal management approach. This sufficient approach must be able to absorb a high heat flux and also maintain a uniform temperature distribution through the device. Flow boiling in microchannels is an adequate cooling approach for such high heat flux devices. Hence, two-phase flow and heat transfer in microchannels is a field of interest for many researchers.

One importance of studying flow boiling is to determine the flow patterns in microchannels and their impact on the heat transfer mechanism. Kandlikar [1] classified flow patterns of subcooled flow boiling inside a microchannel into three main zones. Based on this classification, after the onset of nucleate boiling point, there exists bubbly flow. Bubbles coalesce and they form bigger bubbles with dimensions in the order of magnitude of the microchannel and they are called confined bubbles. Afterward, if the channel length is long enough, the slug flow pattern can be observed.

Tuckerman and Pease first suggested the idea of using a microchannel heat sink in heat transfer applications. [2], As it is well known, microchannels have considerably higher heat transfer coefficients than conventional channels [3,4]. Flow and heat transfer of microchannel heat sinks with different channel widths was studied numerically by Hatami and Ganji [5]. They observed higher Nusselt numbers for microchannels with lower widths. Prajapati [6] investigated the influence of fin height on heat transfer and fluid flow characteristics of the microchannel heat sink. The fin heights varied from 0.4 mm to 1.0 mm which the 1.0 mm fin height corresponds to the completely closed heat sink. They reported that the heat sink with 0.8 mm fin height has the best heat transfer performance.

Akbari et al. [7] investigated the effect of employing ribs in a microchannel on single-phase flow and heat transfer. They showed that the ribs improved local heat transfer by mixing the flow and scrambling the thermal boundary layer. In addition, the ribs increased the penalty of pressure drop in the microchannel. Therefore, a low-pressure-drop solution must be developed to prevent the growth of the thermal boundary layer thickness in microchannels of thermal management systems. In order to interrupt the thermal boundary layer growth, Wang

*Corresponding authors.
E-mail addresses: amirmsattari@ut.ac.ir (A. Sattari), hanafizadeh@ut.ac.ir (P. Hanafizadeh).

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et al. [4] also added vertical and spanwise ribs to the conventional microchannels of the heat sink. Fluid flow mixing and boundary layer regeneration were observed after each rib, and the heat transfer coefficient was enhanced up to 100% using the bidirectional ribs. Meanwhile, the ribs increased the pressure drop penalty considerably. Japar et al. [8] studied the thermal performance of novel microchannel heat sink designs with single-phase flow, numerically. Their novel designs include adding rectangular ribs, triangular cavities, and secondary channels to the conventional microchannel heat sink design. It was concluded that the design with the most flow mixing has the best heat removal performance.

The idea of using metallic nanoparticles to improve the thermal properties of liquids was first developed by Choi and Eastman [9]. They observed an increase in the convective heat transfer coefficient of nanofluid single-phase flow using a theoretical approach. Thereafter, single-phase heat transfer of nanofluids became the field of interest of many researchers. Nanoparticles are prepared by suspending nanoparticles in a base fluid. Alfaryjat et al. [10] investigated the influence of nanoparticle material on the single-phase fluid flow and heat transfer numerically. They used a heat sink with hexagonal microchannels and water as the base fluid. Alumina nanofluids were reported as the nanofluid with the highest heat transfer coefficient. Lelea [11] studied the thermal performance of water and alumina nanofluid flow in a microchannel heat sink using a numerical approach. He employed nanoparticles with 13 nm, 28 nm, and 47 nm diameters and nanofluid volumetric concentrations from 1% to 9%. He concluded that without considering the viscous heating effects, there might be 20% to 30% error in the heat transfer coefficient. Thermal performance and irreversibility of nanofluids with different nanoparticle shapes, including brick, blade, platelet, cylinder, and oblate spheroid, were studied in a microchannel heat sink by Bahiraei et al. [12]. The equivalent nanofluid properties were obtained using empirical relations. They showed that the platelet nanoparticle shape has the lowest thermal resistance value.

Kalteh et al. [13] investigated the single-phase flow of nanofluids in a microchannel heat sink numerically and experimentally. They reported an increase in heat transfer coefficient for higher nanofluid concentrations and lower nanoparticle diameters. In addition, they considered the assumption of nanofluid homogeneity reasonable. Lee and Mudawar [14] conducted some experiments to assess the effectiveness of nanofluids’ single-phase flow. They considered aluminum oxide nanoparticles with 36 nm diameter and volumetric concentrations of 1% and 2%. They also reported that the heat transfer coefficient increases for higher nanofluid concentrations. Furthermore, they concluded that the developing flow region has a better heat transfer enhancement caused by nanoparticles than the fully developed flow region. Shi et al. [15] studied single-phase flow and heat transfer of nanofluids considering geometrical parameters, including the aspect ratio of the microchannel and the ratio of the microchannel width to the wall thickness. The equivalent thermal and physical properties of water and alumina nanofluids with a volumetric concentration of 1% were used in the numerical simulations. It was reported that the microchannels with a higher aspect ratio and a greater ratio of the microchannel width to the wall thickness, have better heat transfer characteristics.

According to various applications of flow boiling, many researchers are interested in studying this phenomenon. Aranzabal et al. [16] investigated the capabilities of single- and two-phase flows to cool high power devices numerically. The devices’ temperatures in the two-phase flow boiling cooling approach were remarkably lower than the single-phase flow cooling method. There was 43% average enhancement in the reported heat transfer coefficients of two-phase flow in comparison

### Nomenclature

- \( A \): Cross-sectional area of flow
- \( Bo \): Boiling number
- \( C_p \): Specific heat in constant pressure
- \( Co \): Courant number
- \( d_h \): Hydraulic diameter
- \( E \): Energy
- \( F \): General external forces
- \( F_s \): Surface tension force
- \( G \): Mass flux
- \( g \): Acceleration of gravity
- \( H \): Enthalpy
- \( h \): Heat transfer coefficient
- \( h_{m} \): Microchannel height
- \( h_{s} \): Heat sink height
- \( k \): Thermal conductivity
- \( Kn \): Knudsen number
- \( L \): Microchannel length
- \( m \): Fin parameter
- \( m \): Mass flow rate
- \( n \): Empirical shape factor
- \( Nu \): Nusselt number
- \( P \): Pressure
- \( Per \): Wet perimeter of flow
- \( q' \): Heat flux
- \( Re \): Reynolds number
- \( S_h \): Volumetric heat source
- \( t_{w} \): Half of microchannel wall thickness
- \( u \): Longitudinal velocity component
- \( \nabla \): Velocity vector
- \( w_{m} \): Microchannel width

### Greek symbols

- \( \pi \): Volume fraction
- \( x \): Curvature of phase interface
- \( \lambda \): Molecular characteristic length
- \( \mu \): Dynamic viscosity
- \( \rho \): Density
- \( \sigma \): Surface tension
- \( \phi \): Nanofluid volumetric concentration

### Subscripts

- \( B \): Microchannel bottom
- \( bf \): Base fluid
- \( ch \): Channel
- \( in \): Inlet
- \( l \): Liquid phase
- \( L \): Averaged in microchannel length
- \( lv \): Between liquid and vapor phase
- \( out \): Outlet
- \( p \): Nanoparticle
- \( q \): \( q^{th} \) phase
- \( m \): Fluid bulk
- \( nf \): Nanofluid
- \( sat \): Saturation
- \( sub \): Subcooled
- \( v \): Vapor phase
- \( w \): Microchannel wall
- \( x \): Local
with the heat transfer coefficients of single-phase flow. In an experimental research, Agostini et al. [17] studied the flow boiling of R236fa refrigerant. In their experiments, saturated liquid entered the microchannel heat sink with a constant heat flux on its base surface. They reported an optimum for heat transfer coefficient at a specified vapor quality for each heat flux. Lee and Pan [18] researched the flow boiling of water in a shallow microchannel. They studied two-phase characteristics in microchannels with diverging and uniform cross-sections and reported a more stable boiling in the diverging cross-section. Additionally, they showed that the two-phase flow pressure drop along the microchannel length increases for higher heat fluxes.

Recently, Soleimani et al. [18] studied submerged flow boiling in a microchannel heat sink numerically. According to the maximum base temperature value of the microchannel, which was below 70 °C with the heat flux of 173.3 W/m², they concluded that the two-phase flow could be efficient in thermal management applications. Besides, it was observed that the velocity profile was not symmetric in the flow boiling, and the vapor velocity magnitude was higher than the liquid bulk velocity magnitude. Mathew et al. [19] used a hybrid microchannel heat sink to conduct experiments on flow boiling of water. The hybrid heat sink consisted of a microchannel array in the upstream region for low vapor quality flow and a microgap in the downstream region for high vapor quality flow. It was reported that the microgap has a lower heat transfer coefficient but stabilizes the flow boiling compared to the flow boiling in conventional microchannels.

In a recent research, Hedau et al. [20] studied subcooled flow boiling of water in a microchannel heat sink using experimental and numerical approaches. They presented flow patterns of the flow boiling in different working conditions which included bubbly flow and annular flow. It was reported that bubbles moving near the surface of microchannel enhance the heat transfer from the wall to the fluid bulk. Additionally, they showed that a microchannel heat sink with higher number of channels has a greater heat transfer coefficient than a microchannel heat sink with lower number of channels but the same footprint area. The thermal performance of highly subcooled flow boiling in microchannel heat sinks was investigated experimentally by Lee and Mudawar [21]. The working fluid was HFE-7100 which its saturation temperature is 60 °C at atmospheric pressure, with inlet temperatures of 0 °C and −30 °C. They concluded that the highly subcooled flow boiling can be efficiently applied to the thermal management of high heat flux devices. Additionally, they reported that an increase in the degree of inlet fluid subcooling delays the onset of nucleate boiling point in microchannel and enhances the critical heat flux. Drummond et al. [22] studied saturated flow boiling of HFE-7100 in a manifold microchannel heat sink experimentally. The manifold consisted of several layers that lead the fluid into the microchannel heat sink. The microchannels with the highest aspect ratio had the lowest thermal resistance and the highest pressure drop. In the highest mass flow rate, these microchannels dissipated heat with a rate of 910 W/cm² and had a pressure drop value of 162 kPa.

Mostly, the review papers focus on the research gaps and provide suggestions for the future studies. Fang et al. [23] investigated correlations related to subcooled flow boiling heat transfer and used water flow boiling database to evaluate these correlations. The reviewed correlations were included for conventional channels, minichannels, and microchannels. It was reported that the boiling number (Bo) is also an important dimensionless parameter in presenting the subcooled flow boiling correlations. A review of the experimental boiling flow researches in microscale channels was carried out by Cheng and Xia [24]. Correlations for calculating the heat transfer coefficient and the friction factor of microchannel flow boiling were presented. They reported that the available flow boiling database doesn’t match with the correlations and more experimental microchannel flow boiling data are needed. Additionally, the nucleate boiling mechanism was considered the dominant heat transfer mechanism when the heat transfer coefficient is principally a function of system pressure and heat flux. Guichet et al. [25] studied pool boiling in two-phase closed thermosyphons and classified correlations related to the bubble departure diameter and frequency and the pool boiling heat transfer coefficient. They also examined different stages of the bubble formation and departure from a nucleation site and thermal characteristics of each stage. The bubble was considered an energy mover and its movement improved the heat transfer from the hot surface to the bulk fluid. It was reported that the bubble departure diameter and frequency have a key role in the pool boiling heat transfer coefficient. Leong and Wong [26] reviewed researches relating to pool and flow boiling of pure dielectric fluids and investigated effects of surface enhancement on boiling process and heat transfer. They reported that the surface wettability has a considerable influence on the heat transfer coefficient and the critical heat flux. In addition, they investigated some correlations relating to the critical heat flux of the pool boiling phenomenon.

Numerical simulation of the boiling phenomenon has high computational costs. Therefore, this phenomenon is studied by fewer numerical researches in comparison with single-phase flow simulation. Recently, Xu et al. [27] investigated saturated flow boiling in a microchannel in different gravity conditions using a numerical approach and the VOF model. The continuum surface force (CSF) model was used to formulate the surface tension force. They reported a stratified flow pattern in the extreme gravity condition. This flow pattern causes surface dry-out and therefore, a decrease in local heat transfer coefficient was observed. Huang et al. [28] deliberated about subcooled flow boiling of R-141b in a minichannel using a numerical approach. They used a two-dimensional geometry and a uniform heat flux boundary condition. The VOF model with the geometric reconstruction scheme was used to solve the two-phase flow. They reported that the heat transfer coefficient is enhanced by decreasing the channel width. In two separate researches, Zhan and Wang [29,30] studied subcooled flow boiling in a microchannel numerically using the VOF model. They employed the geometric reconstruction interpolation scheme within the VOF model to capture a better interface boundary. Additionally, the physical properties of vapor and liquid phases were considered temperature-dependent. In the first research, the working fluid was HFE-7100 and bubbly flow pattern was reported. Because of the subcooled flow, they observed that the bubbles condensed in the flow. In the second research, the working liquids are R-22 and R-134a, and the bubbly and slug flow patterns were reported. They concluded that in fluids with higher surface tension, bubbles coalesce easier and slugs are formed faster.

Flow and heat transfer of a single slug inside a microtube was researched numerically by Magnini and Thome [31]. In this research, the VOF model and R245fa fluid were employed. They also used the continuum surface force model and calculated the local interface curvature parameter of this model, implementing a User-Defined Function called the Height Function algorithm. A variable time-stepping method is used for the solver by considering a maximum courant number of 0.5. The movement of the slug near the microtube wall increased the local heat transfer coefficient. Recently, Parida and Chainer [32] compared heat transfer and fluid flow characteristics of two-phase flow boiling of four different refrigerants inside a microchannel numerically. The refrigerants included R-1234ze, R-245fa, R-134a, and R-600a, and the microchannel consisted of micro pin-fins. The Eulerian multiphase model and a wall boiling model were employed for two-phase flow boiling and the fluid flow was considered turbulent and the wall nucleation was simulated using a wall boiling model. It was observed that the R-134a refrigerant maintained the lowest temperature distribution along the microchannel. In another research, Prajapati et al. [33] investigated the subcooled flow boiling of water in a finned microchannel computationally. They used the VOF multiphase model with the CSF model in a two-dimensional computational domain. The phase change criterion was considered the saturation temperature of the fluid. They observed the bubbly flow pattern for the subcooled flow boiling. Also, they reported that in high heat fluxes, the local surface temperature
increased significantly because of dry-out.

Flow boiling of nanofluids is a research field with fewer researches than the flow boiling of base fluids field. Flow boiling of water-based nanofluids was investigated in a tube by Sarafraz et al. [34]. Experiments were carried out for nanofluids with zirconia nanoparticle mass concentrations ranged from 0.1% to 0.3%. The heat transfer coefficient and the pressure drop increased with increasing the concentration of nanofluid. Also, they considered bubble interactions after the onset of nucleate boiling as one of the causes of pressure drop increase. Vafaei and Wen [35] conducted experiments to study subcooled flow boiling of aqueous alumina nanofluids inside a microtube, considering low concentrations of nanofluids. They observed particle deposition with thicker structures near the outlet region. The deposition of nanoparticles was reported as the reason for heat transfer enhancement caused by nanofluids. In addition, the critical heat flux of nanofluid boiling was enhanced in comparison with the base fluid boiling. Li Xu and Jinliang Xu [36] studied the subcooled flow boiling of water-based nanofluids in a microchannel experimentally. They used 40 nm diameter alumina nanoparticles and a 0.2% concentration nanofluid. They reported bubbly flow and slug flow patterns. They did not observe any particle deposition after boiling tests; thus, it was concluded that not only the nanofluid flow boiling enhances the heat transfer coefficient, but also it stabilizes flow boiling inside a microchannel. Dong et al. [37] researched the flow boiling of aqueous alumina nanofluids in swirl microchannels with volumetric concentration ranged from 0.03% to 0.1% in various gravity and acceleration conditions. It was found that the heat transfer coefficient decreases with an increase in acceleration magnitude and with an increase in the aspect ratio of the microchannel cross-section. An optimum value of 0.07% was reported for the volumetric concentration of the nanofluid.

Abedini et al. [38] investigated the subcooled flow boiling of a water-based nanofluid inside a tube with a numerical approach. Nanoparticles with 30 nm diameter and nanofluids equivalent properties at 1%, 2%, and 4% volumetric concentrations were used. They used the mixture multiphase model, but they did not report the boiling flow pattern. Nevertheless, they concluded that their numerical model could predict the wall temperature and the vapor volume fraction adequately. An increase in the local heat transfer coefficient after the onset of nucleate boiling was also reported. Besides, they reported a decrease in vapor volume fraction for high nanofluid concentrations. Recently, Zhang et al. [39] studied non-boiling heat transfer and fluid flow of a slug moving in a vertical minitube numerically. The VOF model was used, and aqueous copper oxide and nitrogen were considered as the primary and the secondary phases, respectively. The equivalent properties of the nanofluid were calculated in volumetric concentrations ranging from 0% to 3%. The computational domain was axisymmetric two-dimensional, and a User-Defined Function was employed for fully developed gas–liquid flow in the inlet and outlet. It was reported that the heat transfer coefficient and pressure drop are maximum for the nanofluid with 3% volumetric concentration. In one of the few researches that simulated flow boiling of a nanofluid in a microchannel, growth of a single bubble was studied by Khalighi et al. [40]. The nanofluid was aqueous alumina with volumetric concentrations of 1% and 2%. They computed the nanofluid equivalent thermal and physical properties using empirical relations. The level-set multiphase model was employed in simulations. Because of the higher heat transfer rate in higher concentrations of nanofluids, they observed an increase in the bubble growth rate.

To the best of authors’ knowledge, a few of the researches in nanofluids flow boiling inside microchannels are numerical. In this research, subcooled flow boiling of a pure fluid and various concentrations of nanofluids in a microchannel heat sink are studied numerically. The dielectric HFE-7100 is used as the working fluid and the base fluid for nanofluids. Alumina nanofluids with volumetric concentrations of 1%, 2%, and 4% are employed. The numerical model is verified using experimental data. The effects of two-phase flow boiling on the flow and heat transfer characteristics are investigated. Additionally, the heat transfer enhancement obtained from adding nanofluids to the base fluid in flow boiling is inspected and compared with two-phase flow heat transfer enhancement.

2. Methodology

2.1. Governing equations

Fluid flows in this study are assumed to be incompressible and Newtonian. The VOF multiphase model is employed in the simulations. This model solves a single set of conservation equations. In each computational cell, at least one phase is assigned, and if two phases appear in one cell, that cell represents the interface of those phases. The volume fraction equation is based on the continuity of the phase fraction, and it is presented as follows [41],

\[
\frac{1}{\rho_{\text{eff}}}rac{\partial}{\partial t}(\alpha_{p}\rho_{p}) + \nabla \cdot (\alpha_{p}\rho_{p}\mathbf{v}_{p}) = \sum_{q=1}^{n} \left( n_{pq} - n_{qp} \right) 
\]

(1)

here \( n_{pq} \) is the mass transfer from phase \( p \) to phase \( q \), and \( n_{qp} \) is vice versa. This equation is solved for the volume fraction of the \( q^{\text{th}} \) phase, \( \alpha_{q} \), and the volume fraction of the other phase is derived from this relation:

\[
\sum_{q=1}^{n} \alpha_{q} = 1
\]

(2)

In the VOF model, the momentum equation is solved for each computational cell and the resulting flow field is applied to all phases of that cell. This equation is formulated as,

\[
\frac{\partial}{\partial t} (\rho \mathbf{v}) + \nabla \cdot (\rho \mathbf{v} \mathbf{v}) = -\nabla p + \nabla \cdot \left( \mu \left( \nabla \mathbf{v} + \nabla \mathbf{v}^{T} \right) \right) + \rho g + \mathbf{F}_{\text{ext}}
\]

(3)

where \( \mathbf{v} \), \( \mu \), and \( p \) are flow field, dynamic viscosity, and pressure field, respectively. The term \( \mathbf{F}_{\text{ext}} \) represents external force. This external force includes the surface tension force. In this model, surface tension is considered based on the continuum surface force (CSF) model, which was developed by Brackbill et al. [42]. In the CSF model, surface tension force (\( F_{\text{surf}} \)) is derived from

\[
F_{\text{surf}} = \frac{2\sigma_{x}}{\alpha_{l} + \alpha_{g}}
\]

(4)

where \( \sigma \) and \( \alpha \) are surface tension and curvature of the phase interface, respectively. According to this formulation, the surface tension force is not calculated for computational cells in which there is only one phase. The energy equation can be written as,

\[
\frac{\partial}{\partial t} (\rho E) + \nabla \cdot (\rho \mathbf{v} E + p) = \nabla \cdot \left( k_{\text{eff}} \nabla T + (c_{\text{eff}} T \mathbf{v} \cdot \nabla) \right) + S_{h}
\]

(5)

where \( k_{\text{eff}} \) and \( p \) are effective conductivity and density, respectively. These physical properties are shared by all phases. The terms \( E \) and \( T \) are energy and temperature. Volumetric heat source and stress tensor are shown by \( S_{h} \) and \( \mathbf{F}_{\text{ext}} \), respectively. The second term on the right-hand side of Eq. (5) indicates viscous heating effects. The physics of the boiling process consists of a mass transfer from the liquid phase to the vapor phase. In this numerical model, this mass transfer is calculated using the Lee model [43]. Based on this model, mass transfer from a liquid phase to the vapor phase occurs if the temperature of the liquid is higher than the saturation temperature (\( T_{\text{sat}} \)). The mass transfer from the vapor phase to the liquid phase occurs if the temperature of the vapor is lower than the saturation temperature. The former mass transfer is the evaporation mass transfer and the latter one is the condensation mass transfer. The evaporation mass transfer (\( m_{\text{evap}} \)) is formulated as,
\[ m_{li} = C \cdot \alpha_{li} \frac{(T_{li} - T_{wall})}{T_{sat}} \]  \hspace{1cm} (6)

In this equation, the term \( C \) is an empirical constant that can be calculated based on the physical properties of the fluid [41].

### 2.2. Non-dimensional numbers

In this section, some non-dimensional numbers used in the context are introduced. Reynolds number is the ratio of inertial forces to the viscous forces. It is formulated as,

\[ Re = \frac{\rho U_{in} d_{h}}{\mu} \]  \hspace{1cm} (7)

where \( U_{in} \) is the inlet velocity of the microchannel and the hydraulic diameter \( (d_{h}) \) is calculated from,

\[ d_{h} = \frac{4A}{Per} \]  \hspace{1cm} (8)

where \( A \) and \( Per \) are the cross-sectional area of flow and wet perimeter of flow, respectively. The Knudsen number is defined as the ratio of molecular characteristic length \( (\lambda) \) to the characteristic dimension of the channel in a gaseous flow [3],

\[ Kn = \frac{\lambda}{w_{ch}} \]  \hspace{1cm} (9)

in which the characteristic dimension is considered the width of the microchannel \( (w_{ch}) \), which is the lowest dimension of the microchannel. In unsteady simulations, the numerical time step \( (\Delta t) \) can be constrained by the courant number \( (Co) \) using the courant number definition:

\[ Co = \frac{u \Delta t}{\Delta x} \]  \hspace{1cm} (10)

In flow boiling, the ratio of evaporation mass flux to the flow mass flux \( (G) \) is called boiling number \( (Bo) \). It can be calculated from this equation:

\[ Bo = \frac{q'}{G \cdot H_{bo}} \]  \hspace{1cm} (11)

Here \( H_{bo} \) is the enthalpy of vaporization. Nusselt number which can be defined as a measure of the intensity of the heat flux [44] is calculated from,

\[ Nu = \frac{h d_{h}}{k_f} \]  \hspace{1cm} (12)

where the terms \( h \) and \( k_f \) are heat transfer coefficient and conduction coefficient of the fluid.

### 2.3. Heat transfer coefficient

The local heat transfer coefficient of the microchannel in the numerical results is calculated based on an energy conservation equation, which was presented by Lee and Mudawar [14]. This equation is presented as

\[ q \cdot (w_{ch} + 2t_{ch}) = h_{ch}(T_{w(ch)} - T_{Tin}) \left[ w_{ch} + 2 \left( \frac{\tan h m_{x} h_{ch}}{m_{x} h_{ch}} \right) r_{ch} \right] \]  \hspace{1cm} (13)

where \( w_{ch}, t_{ch}, \) and \( h_{ch} \) are geometrical parameters presented in Fig. 1(b) and Table 1. The local fin parameter \( (m_{x}) \) is formulated as

\[ m_{x} = \sqrt{\frac{h_{ch}}{k_{f} t_{ch}}} \]  \hspace{1cm} (14)

All parameters of Eq. (13) except \( h_{ch} \) are known or calculated and therefore, this equation can be solved to find the local heat transfer coefficient \( (h_{ch}) \).

### 2.4. Computational domain and boundary conditions

In this study, flow boiling inside a microchannel heat sink is simulated. This heat sink is the same as one of the microchannel heat sinks in the experimental research of Lee and Mudawar [21], and it consists of 24 identical microchannels. The schematic of the microchannel heat sink is shown in Fig. 1(a). The solid material of this heat sink is copper with thermal conductivity of 387.6 W/m.K, and only one of the microchannels is considered as the computational domain. The cross-sectional view of the computational domain is depicted in Fig. 1(b). In the computations, flow, volume fraction, and energy equations are solved for the fluid domain and only the energy equation is solved for the solid domain. Dimensions of the default computational domain and

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**Table 1**

Dimensions of the computational domain and range of boundary conditions.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>( h_{ch} )</td>
<td>304.9</td>
<td>( \mu m )</td>
</tr>
<tr>
<td>( w_{ch} )</td>
<td>123.4</td>
<td>( \mu m )</td>
</tr>
<tr>
<td>( t_{ch} )</td>
<td>42.1</td>
<td>( \mu m )</td>
</tr>
<tr>
<td>( h_{hs} )</td>
<td>404.9</td>
<td>( \mu m )</td>
</tr>
<tr>
<td>( L )</td>
<td>10</td>
<td>mm</td>
</tr>
<tr>
<td>( m )</td>
<td>5.0, 7.5</td>
<td>g/s</td>
</tr>
<tr>
<td>( q' )</td>
<td>210–440</td>
<td>W/cm²</td>
</tr>
<tr>
<td>( T_{in} )</td>
<td>−30</td>
<td>°C</td>
</tr>
</tbody>
</table>

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Fig. 1. a) copper microchannel heat sink. The computational domain is highlighted in light blue. b) Cross-section of the computational domain. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)
ranges of its boundary conditions are tabulated in Table 1. Working conditions of this research correspond to Reynolds numbers in the range of 530 to 2000 and boiling numbers in the range of \(2.3 \times 10^{-3}\) to \(7.1 \times 10^{-2}\). The saturation temperature of the working fluid is 60 °C at atmospheric pressure, and the boiling flows are highly subcooled with −30 °C inlet temperature.

As displayed in Fig. 2, the flow enters the microchannel heat sink from one end and exits through the other end. The fluid domain is initially stationary, and both fluid and solid domains are initialized with the inlet flow temperature. Constant and uniform heat flux is considered for the base surface of the heat sink. The maximum heat flux of this study is 440 W/cm², which corresponds to 220 W heat dissipation from a 10 × 5 mm² surface. Although the gravity can be neglected in two-phase flows through microchannels [1], in this study, the gravity source term is considered in the solution of the momentum equation. A symmetry boundary condition is used on cut surfaces of the heat sink solid domain, which correspond to the sidewalks of the computational domain. The Knudsen number of the vapor phase is calculated using Eq. (9), and it is lower than 0.001 for all working conditions. Therefore, a no-slip boundary condition is considered for all fluid–solid interfaces. The top surfaces of the computational domains, including the fluid domain and the solid domain, are thermally insulated. The cross marks displayed in Fig. 2 are three points where temperature value is recorded. Average base temperature of the microchannel \(T_B\) is derived using the mean temperature value of these three points. According to the experimental data [21], the distances of these points from the inlet surface are 1.2 mm, 5.0 mm, and 8.8 mm. In the results of this research, the local wall temperature \(T_{wall}\) is the mean temperature of the solid–fluid interface located at distance of \(x\) from the inlet surface. The average wall temperature \(T_{avg}\) is the average of local values along the microchannel length.

### 2.5. Working fluid properties

The main working fluid of the current study is HFE-7100. The saturation temperature of this fluid is 60 °C at atmospheric pressure. The freezing temperature of HFE-7100 is −130 °C; hence, this fluid can be employed in applications with a high degree of subcooling. This thermal fluid is dielectric and can be used in direct contact cooling and immersion cooling applications [45]. The thermal and physical properties of the vapor and liquid phases of HFE-7100 used in the present computations are shown in Table 2. Temperature-dependent properties are used in the current research in order to achieve more accurate simulations. Data for HFE-7100 vapor properties are limited in the literature, and the available data are used. Material properties for temperatures between the listed values are determined by linear interpolation, and material properties for temperatures out of bounds are determined by linear extrapolation. Due to the lack of adequate data for the three phase contact angle, it is considered to be 90°, which is reported to be a reasonable assumption in microchannel nucleate boiling [46].

In a part of this study, the two-phase flow heat transfer is compared with the single-phase flow heat transfer. So, for performing this comparison, another fluid with properties close to the HFE-7100 properties with a high saturation temperature is needed, so that the other fluid maintains single-phase flow while HFE-7100 is boiling in the same operating condition. The HFE-7700 liquid is used for this purpose, and its thermophysical properties are tabulated in Table 3. As it is shown in this table, the physical properties are similar to those of the HFE-7100 liquid, while the saturation temperature is 167 °C at atmospheric pressure.

### 2.6. Nanofluid properties

In a part of this study, nanofluid flow boiling in microchannels is studied. As it was shown in the introduction section, nanofluids can be assumed homogeneous, and their equivalent properties can be calculated using empirical or theoretical equations. Density, specific heat, dynamic viscosity, and thermal conductivity of nanofluid are derived using the following equations, respectively [49–52]:

\[
\rho_{nf} = \phi \rho_p + (1 - \phi) \rho_{nf}
\]

\[
c_{p,nf} = \phi (\rho C_p)_p + (1 - \phi) (\rho C_p)_bf
\]

\[
\frac{\mu_{nf}}{\mu_p} = 1 + 2.5 \phi
\]

\[
k_{nf} = \frac{k_p + (n - 1)k_{bf} - (n - 1)\phi(k_{bf} - k_p)}{k_p + (n - 1)k_{bf} + \phi(k_{bf} - k_p)}
\]

Here \(\phi\) is the volumetric concentration of the nanoparticles. The term \(n\) is the empirical shape factor and is equal to three for spherical particles. In these equations, particle, base fluid, and nanofluid properties are subscripted by \(p\), \(bf\), and \(nf\), respectively. The presented equation for equivalent dynamic viscosity (Eq. (17)), was originally derived by Einstein [53]. Bergman [54] and Lee and Mudawar [14] also used these equations to calculate equivalent properties of nanofluids with HFE-7100 base fluids. Additionally, Helvaci and Khan [55] used

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Table 2: Thermophysical properties of pure HFE-7100 in liquid and vapor phase [21,45,47].

<table>
<thead>
<tr>
<th>Phase</th>
<th>T[°C]</th>
<th>c[N/m]</th>
<th>(\rho[kg/m^3])</th>
<th>(c_p[J/kg.K])</th>
<th>(\mu[kg/m.s])</th>
<th>(k[W/m.K])</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liquid</td>
<td>−30</td>
<td>0.0182</td>
<td>1507</td>
<td>1073.6</td>
<td>0.00147</td>
<td>0.0792</td>
</tr>
<tr>
<td></td>
<td>0</td>
<td>0.0157</td>
<td>1133.2</td>
<td>0.00083</td>
<td></td>
<td>0.0733</td>
</tr>
<tr>
<td></td>
<td>10</td>
<td></td>
<td>0.0067</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>0.0136</td>
<td>1183.7</td>
<td>0.00558</td>
<td>0.0683</td>
<td></td>
</tr>
<tr>
<td></td>
<td>60</td>
<td></td>
<td>1254.7</td>
<td></td>
<td></td>
<td>0.0614</td>
</tr>
<tr>
<td>Vapor</td>
<td>10</td>
<td></td>
<td>9.6</td>
<td>815</td>
<td>9.47 \times 10^{-6}</td>
<td>0.0079</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td></td>
<td></td>
<td>9.39 \times 10^{-6}</td>
<td>0.0085</td>
<td></td>
</tr>
</tbody>
</table>

Table 3: Thermophysical properties of pure HFE-7700 in the liquid phase at 25 °C [48].

<table>
<thead>
<tr>
<th>(\rho[kg/m^3])</th>
<th>(c_p[J/kg.K])</th>
<th>(\mu[kg/m.s])</th>
<th>(k[W/m.K])</th>
</tr>
</thead>
<tbody>
<tr>
<td>1750</td>
<td>1150</td>
<td>0.00125</td>
<td>0.065</td>
</tr>
</tbody>
</table>

---

![Fig. 2. Mid-plane of the computational domain and its main boundary conditions. The microchannel length is not to scale.](image-url)
the same equations to calculate properties of an HFE-7000 based nanofluid.

In this study, spherical alumina (Al₂O₃) nanoparticles with 36 nm diameter are used for nanofluids with 1%, 2%, and 4% volumetric concentrations. The same nanoparticles were used in the experimental research of Lee and Mudawar [14]. Density, specific heat, and thermal conductivity of alumina are 3600 kg/m³, 765 J/kg.K, and 36 W/m.K, respectively [14]. The deposition of nanoparticles during boiling and its effect on the wettability properties of the surface are neglected in this study. Therefore, nanoparticles remain in the liquid phase and they do not affect vapor phase properties. Because there is no agreement on the actual effect of nanoparticle deposition on heat transfer characteristics [56], this assumption is reasonable. Owing to the lack of proper data or correlation for HFE-7100 based nanofluids, the effect of nanoparticles on surface tension is neglected. Additionally, the insignificant influence of nanoparticle concentration on the surface tension of the aqua nanofluids was reported by Das et al. [57].

2.7. Numerical procedure

The governing equations of this study are discretized and solved using ANSYS Fluent software version 18.2 using the finite volume method. According to the nature of the boiling phenomenon, a transient solution is considered in flow boiling simulations. Time step sizing is considered variable and its value is determined using Eq. (10) with a fixed coarsen number of 0.9. The VOF multiphase model alongside the PISO (Pressure-Implicit with Splitting of Operators) pressure-velocity coupling are used. The viscous dissipation effects are also considered in fluid flows. The momentum, energy, and pressure equations are discretized using QUICK (Quadratic Upstream Interpolation for Convective Kinetics), second-order upwind and body force weighted schemes, respectively. The interfaces of each phase are captured using the geometric reconstruction scheme, which applies a spatial interpolation to obtain a linear interphase location in cells near the interface of two phases [41].

All simulations are conducted until a steadiness criterion is reached. In order to check the steadiness criterion in a time step, the percentage of the wall temperature difference with the root mean square of wall temperatures at ten consecutive previous time steps is calculated. If the value of this percentage is below 1% for two successive time steps, then that step is considered the start of the steady time range. All results are reported in the steady time range of calculations.

2.8. Mesh independency

The fluid and solid computational domains are meshed using hexahedral mesh types. In order to capture velocity and temperature gradients accurately, a finer mesh sizing is employed to fluid cells near the wall boundaries. In mesh independency study, a mesh with minimum computational cost and minimum cell sizing error is chosen. In two-phase flows, because of cell sizing effect on the volume fraction of phases, errors caused by cell sizing cannot be removed.

Computations of a case are conducted in five different cell sizing, and a summary of the details of these meshes are presented in Table 4. In this table, the inflation layer thickness, fluid bulk cell size, average skewness of all computational cells, total heat transfer coefficient, and the error percentage of each mesh concerning the last mesh is tabulated. Furthermore, the derived total heat transfer coefficients are displayed in Fig. 3. As it is shown, for a number of cells higher than 68,600, the differences between calculated heat transfer coefficients are less than 0.2%. As a result, the mesh with 68,600 computational cells is considered as the optimal mesh.

2.9. Experiments

As mentioned earlier, the geometry and the working conditions of this research is similar to that of Lee and Mudawar’s [21]. Here, a brief description of the experiments conducted by Lee and Mudawar is presented. The experimental setup of their research contains two loops, including the primary loop and the secondary refrigeration loop. HFE-7100 is circulated by a pump inside the primary loop, and its temperature is reduced using a heat exchanger, another side of which R507 and R508b refrigerants are circulating in the secondary loop. The primary fluid passes through the test section that is the microchannel heat sink, and after that, it is returned to the reservoir, which is connected to the suction line of the pump.

Uniform heat flux is applied on the base of the heat sink, and the inlet temperature of the subcooled primary fluid is maintained at ≈30 °C. The mass flow rate of HFE-7100 is measured using a Coriolis mass flow meter. The data acquisition system controls the input heat flux of the test section and gathers data of the primary fluid mass flow rate, temperature, and pressure of the system. Additionally, the test section is monitored using a high-speed camera and microscope.

Table 4
Summary of the five meshes details used in the mesh independency study.

<table>
<thead>
<tr>
<th>Number of Elements</th>
<th>Inflation Layer Thickness [µm]</th>
<th>Fluid Bulk Cell Size [µm]</th>
<th>Average Skewness</th>
<th>( h_l ) [W/m²K]</th>
<th>( 100 \times \frac{h_l - h_{l, Last}}{h_{l, Last}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>25,200</td>
<td>7.5</td>
<td>38</td>
<td>0.17</td>
<td>74,527</td>
<td>6.29</td>
</tr>
<tr>
<td>43,900</td>
<td>4.5</td>
<td>23</td>
<td>0.30</td>
<td>77,534</td>
<td>2.51</td>
</tr>
<tr>
<td>68,600</td>
<td>3.0</td>
<td>15</td>
<td>0.25</td>
<td>79,437</td>
<td>0.12</td>
</tr>
<tr>
<td>114,400</td>
<td>2.3</td>
<td>11</td>
<td>0.22</td>
<td>79,331</td>
<td>0.25</td>
</tr>
<tr>
<td>181,600</td>
<td>1.9</td>
<td>8.5</td>
<td>0.20</td>
<td>79,531</td>
<td>–</td>
</tr>
</tbody>
</table>

Fig. 3. Overall heat transfer coefficient derived using different number of cells for a case with a mass flow rate of 2.0 g/s and heat flux of 173 W/cm². The mesh with 68,600 cells is chosen as the optimal mesh.
3. Results and discussion

Highly subcooled transient flow boiling is simulated in a three-dimensional microchannel heat sink geometry using the introduced numerical model. Working fluid is HFE-7100 based nanofluid with 0%, 1%, 2%, and 4% volumetric concentrations where the 0% concentration represents the pure fluid. The equivalent properties of nanofluids are measured using the presented correlations.

3.1. Model validation

The importance of verifying the numerical model is to ensure the accuracy of its results. With this purpose, simulations of subcooled flow boiling in microchannels are conducted with operating conditions of experimental research. According to Lee and Mudawar [21], a mass flow rate of 5.0 g/s and uniform heat flux on the base surface of the heat sink are applied. The inlet temperature of the subcooled HFE-7100 liquid is −30 °C. The microchannel heat sink geometry of this study is the same as the microchannel heat sink of the experimental one. The microchannel average base temperature \( T_B \) derived from the experimental study [21] and current computational fluid dynamics simulations are compared in Table 5. The heat fluxes of this table are established based on the experimental data. Error percentages of the CFD results are also reported. As it is shown, the simulations predicted the wall temperature of the microchannel accurately. The maximum error of numerical results with respect to the experimental results is less than 4%.

Additionally, in order to ensure the validity of the heat transfer coefficient derived from simulations, this parameter is also compared with experimental data. Here, HFE-7100 enters the microchannel heat sink with a temperature of −30 °C and a mass flow rate of 3 g/s and the heat sink base uniform heat flux is adjusted according to Lee and Mudawar [58]. As it is observed in Table 6, there is a good agreement between the experimental data and the numerical data derived from computational fluid dynamics. The maximum error in the heat transfer coefficient calculation is below 6% in comparison with the experimental data. Hereby, the numerical simulation with the VOF multiphase model is suitable to simulate flow boiling in microchannels.

3.2. Qualitative results

Subcooled flow boiling in a microchannel heat sink is simulated using the validated numerical model for heat fluxes of 210 W/cm², 290 W/cm², 370 W/cm², and 440 W/cm². Liquid phase volume fraction distribution on the bottom surface of the microchannel and microchannel mid-plane for all four heat fluxes with a mass flow rate of 5 g/s are displayed in Fig. 4. These volume fraction distributions are plotted near the microchannel outlet and in a time step inside the steady-state time range, in which the changes in wall temperature are less than 1% for two successive time steps. It is observed that larger bubbles are formed for higher heat fluxes. In 210 W/cm² and 290 W/cm² heat fluxes, the volume fraction distributions are similar. In these two heat fluxes, boiling nuclei are observed in the form of low volume fraction bubbles. For the higher heat flux, which is 370 W/cm², bubbles at the bottom surface have a higher volume fraction. Additionally, small bubbles are distributed throughout the fluid domain. In the 440 W/cm² heat flux, the bubbles at the microchannel bottom surface coalesce. Hence, in this heat flux, the bottom surface of the microchannel is in contact with a layer of HFE-7100 vapor.

Bubble movement near the microchannel walls interrupts thermal and hydrodynamic boundary layer growth. Therefore, an enhancement in the heat transfer rate is expected in boiling flows. Besides, for higher vapor volume fractions, there are higher interphase interactions, and consequently, the resisting force against fluid flow increases. Accordingly, an increase in fluid flow pressure drop is expected for high heat fluxes. The influence of heat flux and mass flow rate on the fluid flow pressure drop will be discussed quantitatively in the “Pure fluid flow boiling results” subsection.

3.3. Thermal performance of two-phase flow in comparison with single-phase flow

Before presenting flow boiling results, the heat transfer characteristics of single-phase flow is compared with two-phase flow boiling. The HFE-7700 dielectric fluid is used for the single-phase flow with a heat flux of 210 W/cm². The saturation temperature of this fluid is 167 °C in atmospheric pressure. Therefore, in the same conditions as those of HFE-7100 fluid, the HFE-7700 would maintain a single-phase flow. The single-phase flow of HFE-7100 with a heat flux of 120 W/cm² is also studied.

Wall temperature and bulk temperature distribution along the microchannel length are displayed in Fig. 5. In addition to the single-phase flow results, two-phase flow boiling results of HFE-7100 with a heat flux of 210 W/cm² is plotted. The mass flow rate is 7.5 g/s for all three cases. The wall and bulk temperatures of HFE-7700 flow are higher than in other cases. The onset of nucleate boiling distance from the inlet surface \( x_{ONB} \) is indicated in the flow boiling graph. It is observed that the wall temperature of the flow boiling case has different characteristics in comparison with the single-phase flows. The wall temperature distribution is approximately uniform after the onset of nucleate boiling point in the condition of two-phase flow boiling. The temperature of the fluid is fixed to the saturation temperature during boiling and this can baricade the excessive temperature increase in the heat sink solid. Additionally, bubble motion near the microchannel surface can disturb the thermal boundary layer. As a result, the fluid and the solid domain exchange heat at a high rate, and the wall maintains a uniform temperature distribution. Consequently, the flow boiling can be used to maintain a uniform and low temperature in a high heat flux device.

The local heat transfer coefficient is calculated using the approach presented by Lee and Mudawar [14], which is presented in Eq. (13). This approach is based on the difference between the local wall temperature and local fluid bulk temperature. The lower temperature difference between the wall and fluid bulk corresponds to a higher heat transfer coefficient. Local heat transfer coefficients along the microchannel length for single-phase and two-phase flows with 7.5 g/s mass flow rate are presented in Fig. 6. In the single-phase flows, because of thermal boundary layer growth, the local heat transfer coefficient decreases along the microchannel length. Nevertheless, in the two-phase boiling flow, the heat transfer coefficient increases after the onset of nucleate boiling point. This increase is because of the reduction in local temperature difference between wall and fluid bulk caused by the movement of bubbles near the microchannel walls. The onset of nucleate boiling point is indicated in the graph. Therefore, the start of boiling in the microchannel can enhance the heat transfer coefficient.

The modified curve, yellow line, in Fig. 6 represents the local heat transfer coefficient of HFE-7700 plus the offset of HFE-7700 and single-phase HFE-7100 heat transfer coefficients at the starting point of the

<table>
<thead>
<tr>
<th>( q' ) [W/cm²]</th>
<th>( T_B ) [°C]</th>
<th>Error [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Exp [21]</td>
<td>CFD</td>
</tr>
<tr>
<td>213.69</td>
<td>62.40</td>
<td>62.85</td>
</tr>
<tr>
<td>286.97</td>
<td>79.01</td>
<td>75.91</td>
</tr>
<tr>
<td>363.53</td>
<td>85.65</td>
<td>82.69</td>
</tr>
<tr>
<td>439.64</td>
<td>90.01</td>
<td>90.10</td>
</tr>
</tbody>
</table>

Table 5

Microchannel average base temperature presented in the experimental data [21] and derived from the current CFD study for subcooled flow boiling of HFE-7100 with a mass flow rate of 5 g/s.
3.4. Pure fluid flow boiling results

In boiling flow simulations, the mass flow rate is 5 g/s and 7.5 g/s. Heat fluxes are in the range of 210 W/cm² to 440 W/cm². These conditions correspond to the boiling numbers in the range of $2.3 \times 10^{-3}$ to $7.1 \times 10^{-3}$. The distribution of wall temperature along the microchannel length is shown in Fig. 7. The Onset of Nucleate Boiling (ONB) points are indicated by black plus signs and a black dashed line. So, the fluid flow before this line is a single-phase flow. As it is displayed, the slope of the wall temperature graph decreases after a small distance from the ONB point. At the point where the slope of graphs changes, the bubbles have a size that can affect the thermal boundary layer. As a result, at this point, the subcooled liquid contacts with the solid surface and reduces its temperature. Therefore, a decrease in wall temperature is observed along the microchannel length.

For heat flux higher than 290 W/cm², the wall temperature increases in the second half of the microchannel. This increase is because high volume fraction bubbles are in contact with the microchannel surface. However, the slope of temperature increment in two-phase regions is lower than the slope in a single-phase region.

As it was indicated, the ONB point is important in subcooled flow boiling, due to the change in heat transfer characteristics after this point. At this point, the mass transfer from the liquid phase to the vapor phase starts in small active nuclei. The distance of the ONB point from the microchannel inlet ($x_{ONB}$) as a function of heat flux is displayed in Fig. 8. This distance is decreased with an increase in heat flux and it is expected to reach zero for very high heat fluxes. For the higher mass flow rates, the nucleate boiling begins in greater distance from the microchannel inlet. The degree of subcooling affects the onset of nucleate boiling significantly; however, its effect is not investigated in this study.

Variation of HFE-7100 liquid phase volume fraction along the microchannel length is displayed for both mass flow rates in Fig. 9. This volume fraction is averaged in the bulk of the fluid domain. The liquid volume fraction is unity near the microchannel inlet. It is observed that the liquid volume fraction decreases for higher heat fluxes and lower mass flow rates. The oscillations in volume fraction are due to the high volume fraction of small bubbles in the fluid bulk. Because of the high subcooling of fluid bulk, the mass is transferring with a low rate to the vapor phase in low heat fluxes. Consequently, as it was also observed in Fig. 4, the 210 W/cm² and 290 W/cm² heat fluxes have similar liquid volume fraction in the second half of the microchannel (Fig. 10).

The local heat transfer coefficient is inversely related to the local difference between the microchannel surface temperature and the fluid bulk temperature. Therefore, a mechanism that decreases this local temperature difference can enhance the local heat transfer coefficient. Local heat transfer coefficient distribution for all working conditions are displayed in Fig. 9. The black plus signs and dashed line indicate the ONB point along the microchannel length. It is observed that the heat transfer coefficient has a downward trend in the microchannel entry region up to the ONB point. This region corresponds to a single-phase flow. As it is demonstrated in Fig. 6, in a single-phase flow, the local heat transfer coefficient along the microchannel length decreases because of thermal boundary layer growth. After the start of nucleate boiling, the heat transfer coefficient increases. This increase is because of disturbance in the thermal boundary layer caused by bubble movements. Hence, the local heat transfer coefficient graph has a minimum located after the ONB point. Abedini et al. [38] also reported a minimum in local heat transfer coefficient distribution of a boiling flow.

The pressure drop through the microchannel is expected to be high because of its small dimensions. The difference between inlet and outlet pressures as a function of heat flux is plotted for both mass flow rates in Fig. 11. Obviously, the higher mass flow rate has a higher pressure drop. As it is observed, pressure drop increases for high heat fluxes with a constant mass flow rate. For high heat fluxes, the fluid mass transfers...
from the liquid phase to the vapor phase with a high rate, and the mean volume fraction of the vapor phase increases. Therefore, as it was indicated in Fig. 4, larger bubbles exist in higher heat fluxes. In addition to the viscous interactions of liquid and vapor phases with microchannel surfaces, the viscous interaction between the liquid phase and the vapor phase exists in high heat fluxes. Consequently, the pressure drop in high heat fluxes is greater than the lower ones. In the mass flow rate of 5.0 g/s, the pressure drop along the microchannel length for the highest heat flux is increased by 13.9% in comparison with the lowest heat flux. The maximum pressure drop for operating conditions of this study is 85 kPa, which can be handled using small commercial cooling pumps with a water head pressure of 10 m.

3.5. Nanofluid flow boiling results

The mean microchannel wall temperature as a function of heat flux for all nanofluid concentrations and the mass flow rate of 5.0 g/s are displayed in Fig. 12. Nanofluid increases the thermal conductivity of the base fluid, and it transfers heat with a higher rate than the base fluid. As a result, the wall temperature decreases with an increase in nanofluid concentration.

Local heat transfer coefficient distribution for the minimum heat flux (210 W/cm²) and the maximum heat flux (440 W/cm²) derived from nanofluid flow boiling is displayed in Fig. 13. The results of this figure are obtained from a single time step, and they are not time-averaged. It is observed that adding nanoparticles to the HFE-7100 enhances the local heat transfer of flow boiling. In heat flux of 210 W/cm², the first half of the microchannel has a higher heat transfer enhancement than the second half, which its maximum value is equivalent to a 7% enhancement in local heat transfer coefficient for the 7.5 g/s mass flow rate. For this heat flux, the local heat transfer enhancement percentage decreases to 2% after the start of nucleate boiling. This asymmetry may be caused by the fact that nanoparticles affect the liquid phase thermophysical properties. Therefore, for the low heat flux of 210 W/cm², the heat transfer enhancement in the single-phase flow region is higher than the enhancement in the two-phase flow region. As it was explained earlier, the border of these two regions can be determined by the minimum in the local heat transfer coefficient distribution.

The overall heat transfer coefficient of flow boiling with nanofluids as a function of heat flux is displayed in Fig. 14 (a). In all heat fluxes, the heat transfer coefficients are enhanced by the addition of nanoparticles to the base fluid. Therefore, nanofluid might be a solution to increase the overall heat transfer coefficient of flow boiling. In Fig. 14 (b), the overall Nusselt number is plotted against the heat flux for all nanofluid concentrations. The trend of the Nusselt number graphs is the same as the heat transfer coefficient graphs. In a mass flow rate of 7.5 g/s, also similar overall results are observed.

The best way to evaluate the heat transfer enhancement caused by nanofluids is to calculate the nanofluid overall heat transfer coefficient enhancement percentage with respect to the base fluid overall heat transfer coefficient for a given heat flux. This percentage is called nanofluid heat transfer enhancement percentage, and its variation with heat flux is displayed in Fig. 15. As it is expected, the heat transfer enhancement increases for higher nanofluid concentrations. The average enhancement percentage of all nanofluid concentrations for mass flow rates of 5.0 g/s and 7.5 g/s are 1.9% and 1.6%, respectively. For the volumetric nanofluid concentration of 4%, the nanofluid heat transfer enhancement is 3% on the average for both mass flow rates.
4. Conclusion

Highly subcooled flow boiling of a pure fluid and nanofluids in a microchannel is studied using three-dimensional computational fluid dynamics simulations. In this study, the dielectric HFE-7100 fluid is considered as the base fluid. The equivalent thermophysical properties of the alumina nanofluids are calculated using correlations. The operating conditions of current simulations correspond to Reynolds numbers in the range of 530 to 2000 and boiling numbers in the range of $2.3 \times 10^{-3}$ to $7.1 \times 10^{-3}$. The numerical model of flow boiling in a microchannel heat sink is validated against experimental data and the maximum numerical error for microchannel average base temperature and overall heat transfer coefficient are below 4% and 6%, respectively. In two-phase flow simulations, the VOF multiphase model is employed. The computational domain of this research is a microchannel of a microchannel heat sink. Therefore, all results derived for the single microchannel can be generalized to the microchannel heat sink. The independency of heat transfer coefficient results from the computational cell sizing is confirmed. Results of this study can be summarized as below:

- The thermal performance of the single-phase flow and two-phase
boiling flow were compared in the same operating conditions. The microchannel wall temperature in flow boiling was much lower than the wall temperature in single-phase flow.

- It was observed that the upward trend of the wall temperature along the microchannel length decreased after the onset of nucleate boiling point. While boiling fluid temperature does not exceed the saturation temperature, and this may have a positive effect on the wall temperature distribution.
- The local heat transfer coefficient trend was descending because of the thermal boundary layer growth in the single-phase flow region. After the start of nucleate boiling, owing to thermal boundary layer disturbance caused by bubble movements near the microchannel surface, the local heat transfer coefficient increased. Therefore, the ONB point plays a significant role in two-phase flow heat transfer enhancement.
- The rise in mass transfer from the liquid phase to the vapor phase in high heat fluxes increased the interphase viscous interactions between two phases. Consequently, in high heat fluxes, the resistance force against the fluid flow increased in a constant mass flow rate.
- In nanofluid flow boiling, the average microchannel wall temperature slightly decreased with an increase in nanofluid concentrations. This wall temperature reduction delayed the onset of nucleate boiling.
- The application of nanofluids with 4% volumetric concentration enhanced the overall heat transfer coefficient by up to 3%.
- It is concluded that the heat transfer enhancement obtained from nanofluid usage is not remarkable with respect to the heat transfer enhancement obtained from the two-phase flow application.

In the future researches, it is suggested to:

1. Investigate different nanoparticle materials with different nanoparticle effective diameters.
2. Investigate the effects of the microchannel aspect ratio.
3. Develop a proper scheme to consider the effects of nanoparticle deposition in simulations.
4. Study the nanofluid flow boiling using the simulation of nanoparticles’ motions.

CRediT authorship contribution statement

Ali Soleimani: Writing - original draft, Conceptualization, Methodology, Software, Investigation. Amirmohammad Sattari:...
Fig. 13. Local heat transfer coefficient distribution along the microchannel length for 210 W/cm² and 440 W/cm² and mass flow rate of a) 5.0 g/s and b) 7.5 g/s.

Fig. 14. a) overall heat transfer coefficient and b) Nusselt number of nanofluids flow boiling as a function of heat flux for a mass flow rate of 5.0 g/s.

Fig. 15. The variation of nanofluid heat transfer enhancement percentage with heat flux for a) 5.0 g/s and b) 7.5 g/s.


