

Numerical investigations on flow boiling heat transfer of ammonia water binary solution (NH₃/H₂O) in a horizontal microchannel

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29 **1 Introduction**

30 As the electronic components keep going smaller and their power densities continuously shoot higher,
31 flow boiling in microchannels has been widely recognized as one of the more promising and efficient
32 cooling methods due to its advantages such as large heat transfer area to volume ratio, small temperature
33 variation on heated surface and high heat transfer performance with small amount of required coolant
34 mass flux [1]. Zeotropic mixtures have been considered as alternative refrigerants replacing pure fluids
35 in certain applications to improve the overall energy efficiency of power-generation and refrigeration
36 systems. Especially in phase change-related thermal applications, the heat transfer irreversibility in heat
37 exchangers could be reduced significantly due to the temperature glide in non-isothermal phase change
38 processes of zeotropic mixtures at a constant pressure. Wang et al. [2] experimentally investigated the
39 zeotropic mixture effect on the low-temperature solar Rankine cycle performance and found that the
40 overall cycle efficiency of R245fa/R152a (0.7/0.3 by mass) mixture was 45.5% higher compared with
41 that of pure R245fa. Zheng et al. [3] also discovered that the R161/R600a (0.25/0.75 by mass) mixture
42 could enhance the system efficiency of a solar energy-powered refrigeration cycle by 39.6% and 54.7%
43 comparing with pure R600a and R161, respectively. Furthermore, the thermophysical properties of
44 zeotropic mixtures at a given pressure (e.g. saturation temperature) could be flexibly adjusted by tuning
45 the inlet concentration of the more volatile component. Thus, they are advantageous in multi-phase
46 related thermal managements where there are restrict limits for the maximum device/system
47 temperatures ensuring sustainable operations, such as in high power electronics industries. For example,
48 water cannot be boiled at 50 °C and 1 bar pressure but NH₃/H₂O mixture can if an appropriate NH₃
49 concentration is chosen. Marcinichen et al. [4] pointed out that the temperature of microprocessor chips
50 should be kept below 85 °C with small temperature nonuniformity in order to achieve satisfactory
51 calculating performance while maintaining high levels of reliability and safety. Leão et al. [5] conducted
52 experimental investigations on flow boiling heat transfer of R32/R125/R134a mixture (23/25/52 by
53 weight) at saturation temperature of 25 °C in multi-channel rectangular heat sink for thermal
54 management of high-power density electronic components, and found that the maximally achieved
55 average heat transfer coefficient (HTC) could be up to 30 kW/(m²·K).

56 Accordingly, experimental studies on flow boiling heat transfer of zeotropic mixtures in mini/micro-
57 channels have been conducted in literature. Guo et al. [6] investigated flow boiling heat transfer
58 performance of R134a/R245fa (0.82/0.18 by mass) mixture in a horizontal tube with inner diameter of
59 3 mm and found that the binary mixture had less pressure drop and higher HTC than that of pure R245fa.
60 Dang et al. [7, 8] carried out experimental studies on flow boiling heat transfer characteristics of
61 R134a/R245fa mixture in a single rectangular microchannel (1mm×1mm) and a seven-parallel
62 segmented microchannel (2mm×1mm) at constant saturated temperatures of 18.5 and 26 °C. They
63 discovered that the flow boiling HTC of zeotropic mixtures in mini-channels were typically lower than
64 the original pure fluids in most conditions, but the mixtures could delay surface dry-outs at high heat
65 fluxes and also increase the critical heat flux (CHF) values significantly compared with pure fluids.
66 Azzolin et al. [9] and In et al. [10] noticed that flow boiling HTCs of R1234ze(E)/R32 mixture and
67 R123/R134a mixtures in microchannels with diameters of 0.96 mm and 0.19 mm were smaller than

68 those of corresponding pure fluids at most experimental conditions, respectively. Results showed that
69 the zeotropic mixture had higher CHF_s than pure R134a though smaller HTC_s at most cases. Five flow
70 pattern regimes including the bubbly, confined bubbly, slug, churn-annular and annular flow were
71 observed and the transitions among different regimes for binary mixtures were delayed comparing with
72 pure fluids. It was suggested that the hysteresis, directly influenced by the inlet concentration of the
73 more volatile component, considerably affects the overall flow boiling heat transfer performance.

74 Besides experimental efforts, computational fluid dynamic (CFD) simulation has also been adopted as
75 an effective approach for describing multiphase flow heat transfer in microchannels since it could
76 provide visualized transient spatial and temporal distributions (e.g. temperature and flow patterns) in
77 complex flow and heat transfer processes, which cannot be accomplished through experiments. However,
78 most of those relevant numerical studies in literature [11] have been mainly focused on multiphase flow
79 and heat transfer processes of pure fluids such as water and other pure refrigerants. It is clear that,
80 comparing to pure fluids, multiphase flow behaviors of miscible zeotropic mixtures are much more
81 complex. Especially, the liquid/gas interface of miscible fluids under phase transitions changes
82 depending on many factors (e.g. the varying concentration of the more volatile component in the bulk
83 fluid) and difficult to be captured by regular mathematical descriptions. It is commonly known that the
84 volume of fluid (VOF) method is capable of simulating immiscible mixture by tracking separated
85 volume fraction of fluids [12]. However, in order to simulate flow boiling heat transfer of binary
86 mixtures, the VOF model has been modified with reasonable assumptions in literature. For example, by
87 coupling the effective diffusion model of liquid mixture with VOF method, Banerjee [13] investigated
88 mass and heat transfer process between ethanol/isooctane mixture and air in a 2D macro-scale
89 countercurrent stratified flow domain in ANSYS Fluent. Using the same methods, Zhang et al. [14]
90 studied flow boiling heat transfer of isobutene/pentane mixture in a 2D countercurrent flow domain with
91 inner diameter of 4 mm and outer diameter of 10 mm. Additionally, considering the vapor and liquid
92 solutions as uniform mixtures, Lima et al. [15] studied the steady heat and mass transfer of NH₃/H₂O
93 mixture flowing in a 2D macro-scale plate absorber using ANSYS CFX software.

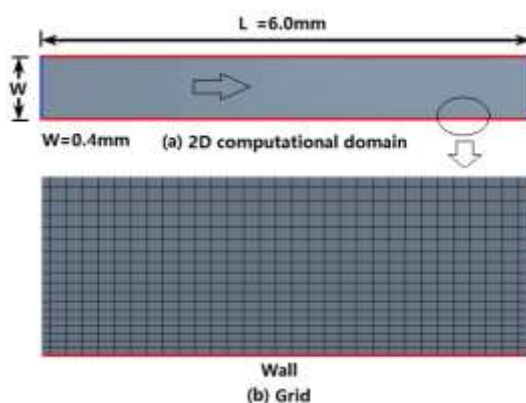
94 NH₃/H₂O mixture has been a useful working fluid in large industrial refrigeration systems for decades.
95 One advantage of NH₃/H₂O mixture is its high latent heat of evaporation. The latent heat of evaporation
96 of NH₃/H₂O, R134a/R245fa, and R123/R134a mixture (0.2/0.8 by mass) at saturated temperature of
97 323.15 K are 1689 kJ/kg, 167 kJ/kg and 155.5 kJ/kg, respectively (calculated by REFPROP). Kærn et
98 al. [16] systematically evaluated existing flow boiling heat transfer correlations for the macro-scale
99 NH₃/H₂O mixture heat exchanger design, including two NH₃/H₂O flow boiling correlations and three
100 flow boiling heat transfer correlations modified based on pool boiling correlations. Khir et al. [17]
101 investigated flow boiling heat transfer of NH₃/H₂O mixture in a vertical tube with inner diameter of 6
102 mm and validated Mishra's model [18] with a new set of correlations. Furthermore, there have also been
103 experimental studies specifically on flow boiling heat transfer of NH₃/H₂O mixture in mini-scale
104 channels. Taboas et al. [19, 20] studied flow boiling heat transfer of NH₃/H₂O mixture in a vertical plate
105 heat exchanger with hydraulic diameter of 4 mm. It was revealed that the flow boiling HTC of NH₃/H₂O
106 mixture was highly dependent on mass flux but negligibly affected by heat flux and pressure at high

107 vapor quality from 0.1 to 0.22. Arima et al. [21, 22] looked into flow boiling heat transfer of $\text{NH}_3/\text{H}_2\text{O}$
108 mixture in a vertical plate evaporator with a gap size of 2 mm. It was found that the local HTC's increased
109 with the increasing of mass flux but decreased with heat flux at certain experimental conditions. Bor et
110 al. [23] showed that the HTC of $\text{NH}_3/\text{H}_2\text{O}$ in a single channel annulus (hydraulic diameter of 0.4 mm
111 and length of 0.8 m) was increased with inlet vapor quality, mass flux and heat flux. However, it can be
112 noticed that most of those experimental studies were conducted in heat exchangers with large tube sizes
113 and those related empirical correlations will not be applicable in predicting flow boiling heat transfer
114 performance of $\text{NH}_3/\text{H}_2\text{O}$ in microchannels due to the differences in bubble dynamics and flow patterns
115 [24]. In addition, few numerical studies of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling heat transfer in microchannels are
116 available in literature. Hence, it is of great significance to conduct exclusive numerical studies towards
117 flow boiling heat transfer characteristics of $\text{NH}_3/\text{H}_2\text{O}$ mixture in microchannels.

118 As discussed, $\text{NH}_3/\text{H}_2\text{O}$ (latent heat ~ 1689 kJ/kg) flow boiling in microchannels is a potential effective option
119 for advanced thermal management of electronics cooling with maximum operating temperature of 85°C ,
120 which is a temperature lower than saturated temperature of water at atmospheric pressure. Therefore, in
121 present work, numerical simulations were carried out to investigate the effects of inlet mass flux, inlet NH_3
122 concentration (by mole) and heating wall temperature on the overall and local heat transfer performance of
123 $\text{NH}_3/\text{H}_2\text{O}$ mixture in a single horizontal microchannel (0.4 mm width and 6 mm length) at constant wall
124 temperature boundary condition. The effective thermophysical and transport properties of $\text{NH}_3/\text{H}_2\text{O}$ mixture
125 as well as modified phase change models (Lee model) were incorporated into VOF model (two fluid model)
126 through UDFs in ANSYS Fluent.

127 **2 Mathematical models**

128 This study simulated the flow boiling heat transfer performance of $\text{NH}_3/\text{H}_2\text{O}$ mixture in a 2-dimensional
129 rectangular microchannel domain (0.4 mm width and 6.0 mm length) shown in Fig. 1(a). The non-
130 uniform quadrilateral mesh was adopted for the entire computational domain displayed in Fig. 1(b),
131 which was gradually refined towards the heating walls for capturing the small-size nucleating bubbles
132 within the viscous boundary layers.



134 **Fig. 1** Schematic of numerical simulation construction

135 The VOF model coupled with Lee method has been widely adopted to predict immiscible multiphase

136 flow heat and mass transfer in mini/microchannels because of its robustness, time-saving and especially
137 accuracy for mass conservation [11, 25]. In the present study, a transient VOF-explicit method and
138 modified Lee model have been employed to track the liquid-gas interfaces of NH₃/H₂O mixture in flow
139 boiling and characterize the mass and heat transfer associated with phase change. The NH₃/H₂O mixture
140 at both liquid and gas state were assumed as a homogeneous saturated working fluid. In addition, it should
141 be pointed out that the diffusion equations in the liquid and in the vapor phase are not directly considered in
142 the present numerical simulation due to computational cost reasons and the lack of such appropriate diffusion
143 equations, though remedies have been taken to better fit the numerical model for simulating the flow boiling
144 of NH₃/H₂O mixture, including the use of effective thermophysical and transport properties of the mixture
145 and modified Lee model in VOF model. The thermodynamic properties of NH₃/ H₂O mixture (e.g.
146 enthalpy) at vapor-liquid equilibrium were determined using correlations from Patek and Klomfar [26].
147 Other thermophysical properties of NH₃/H₂O mixture, including critical temperature and pressure,
148 specific thermal capacity, thermal conductivity, dynamic viscosity, surface tension and density, were
149 obtained by formulations proposed by Conde [27]. Furthermore, in the original Lee model, the mass
150 transfer at liquid/gas interface is driven by the deviation of interfacial temperature from the saturation
151 temperature of pure fluid, which is, apparently, not suitable for zeotropic mixture like NH₃/H₂O with
152 more than one fluid components. It is generally accepted that the phase change phenomena of zeotropic
153 NH₃/H₂O mixture are governed by the difference between local transient and saturated NH₃
154 concentration (a function of temperature and pressure) at the liquid/gas interface. Accordingly, a set of
155 effective thermophysical properties of NH₃/H₂O mixture and modified Lee model were integrated in the
156 simulation by UDF in ANSYS Fluent to fully describe the unique flow boiling behaviors of zeotropic
157 NH₃/H₂O mixture.

158 Theoretically, when nucleation starts, a laminar single-phase flow will be disrupted due to the
159 interactions between small bubbles and their neighboring liquids [28]. The realizable k- ϵ model was
160 adopted due to its superior performance for complex flow and strong heat transfer, which has been seen
161 and validated in previous investigations on flow boiling heat transfer in mini/microchannels from other
162 researchers [29, 30]. The PISO algorithm was chosen for pressure-velocity coupling, the second-order
163 upwind discretization for momentum and energy equations, as well as PRESTO and Geo-Reconstruct
164 discretization for pressure and volume fraction interpolation, respectively. The variable time step was
165 controlled by the Global Courant number up to 0.25 and the absolute residuals of the continuity equation
166 was set to $1e^{-4}$.

167 *2.1 VOF Method*

168 In present work, the VOF method has been used to track the liquid-vapor interfaces in NH₃/H₂O mixture
169 flow boiling by solving Navier-Stokes mass, momentum and energy conservation equations. The gravity
170 effect was ignored for the 2-D computational domain. Also, Revellin et al. [31] found that gravity had
171 little impact on flow boiling in microchannels when the channel size was less than 0.5 mm mainly due
172 to the increased surface tension effect.

173 The continuity equation for each phase:

174
$$\frac{\partial}{\partial t}(\alpha_q \rho_q) + \nabla \cdot (\alpha_q \rho_q \vec{v}_q) = S_{m,pq} \quad (1)$$

175
$$\sum_{q=1}^n \alpha_q = 1 \quad (2)$$

176 where, α_q , ρ_q and \vec{v}_q are the volume fraction, density and velocity of the q^{th} fluid in a grid, respectively.
 177 The sum of volume fractions equals to unity. $S_{m,pq}$ is the mass source term (kg/s) from the p^{th} phase to
 178 the q^{th} phase, which could be calculated by phase change model.

179 In this simulation, there are two phases including the liquid (p) and gas (q) phases in NH₃/H₂O mixture.

180 The momentum and energy equations:

181
$$\frac{\partial}{\partial t}(\rho \vec{v}) + \nabla \cdot (\rho \vec{v} \vec{v}) = -\nabla p + \nabla \cdot [\mu(\nabla \vec{v} + \nabla \vec{v}^T)] + \vec{F}_{surf} \quad (3)$$

182
$$\frac{\partial}{\partial t}(\rho E) + \nabla \cdot [\vec{v}(\rho E + p)] = \nabla \cdot (k_{eff} \nabla T) + S_h \quad (4)$$

183 where, density ρ , dynamic viscosity μ and the effective thermal conductivity k_{eff} are all volume-averaged
 184 variables, while energy E and temperature T are mass-averaged values in a grid cell. S_h is the energy
 185 source term generated corresponding to the phase change process.

186
$$\rho = \sum_{q=1}^n \alpha_q \rho_q \quad (5)$$

187
$$E = \sum_{q=1}^n \alpha_q \rho_q E_q \Big/ \sum_{q=1}^n \alpha_q \rho_q \quad (6)$$

188 where, E_q is energy for each phase based on the specific heat of the q^{th} phase and the shared temperature
 189 value of two phases.

190 Surface tension force \vec{F}_{surf} is considered as a source term in momentum equation, calculated by means of
 191 the continuum surface force method (CSF) as follows [32]:

192
$$\vec{F}_{surf} = \sigma \frac{\rho \kappa \nabla \alpha}{(\rho_p + \rho_q)/2} \quad (7)$$

193 where, κ is the local interface curvature and σ is the surface tension coefficient.

194 2.2 Phase change Model

195 Lee model is a simplified version of Schrage model, considering the quasi-thermo-equilibrium phase
 196 change existing at constant pressure condition and driven by the deviation of local temperature from the
 197 saturated temperature of the pure fluid [33]. As aforementioned, the evaporation of NH₃/H₂O mixture is
 198 a non-isothermal phase change process. Hence, the phase change phenomenon of NH₃/H₂O mixture is
 199 considered to be triggered by the difference between local transient and saturated NH₃ concentration of

200 the liquid mixture (a function of local temperature and pressure) in each interfacial grid cell. Furthermore,
 201 the mass transfer rate at the liquid/gas interface should be proportional to the NH₃ concentration
 202 deviation. Therefore, the Lee model could be modified as follows:

$$203 \quad \begin{cases} S_{m,pq} = -S_{m,qp} = -r_i \alpha_q \rho_q \left| \frac{x - x_{sat}}{x_{sat}} \right|, & x < x_{sat} \\ S_{m,pq} = -S_{m,qp} = +r_i \alpha_p \rho_p \left| \frac{x - x_{sat}}{x_{sat}} \right|, & x > x_{sat} \end{cases} \quad (8)$$

$$204 \quad S_h = S_{m,pq} \cdot q_{LH} \quad (9)$$

205 where, q_{LH} is the latent heat of evaporation, x and x_{sat} are the local transient NH₃ concentration and the
 206 corresponding saturated concentration, the relaxation factor r_i is an empirical coefficient with the unit
 207 of s⁻¹, which is determined by factors such as mesh size, operational conditions and geometric parameters of
 208 heat sinks in specific cases [34, 35]. The value could be ranged from 0.1 to 1e⁷ s⁻¹ for the least saturation
 209 temperature deviation in literature [11]. It was also pointed out that a small r_i value might cause the
 210 temperature in interfacial cells deviating from the saturation temperature, and a large value might
 211 increase the difficulty for the convergence of governing equations. In this study, 100 was selected for
 212 flow boiling heat and mass transfer simulation of NH₃/H₂O mixture, referring to previous related
 213 research [36, 37, 38, 39].

214 2.3 Initial and Boundary conditions

215 Considering the length of the computational domain is only 6 mm and the transient multiphase flow
 216 passes the channel in merely several milliseconds, the temperature of heating walls was set as a constant
 217 for this small dimensional and rapid process [13, 14, 40]. A velocity-inlet and pressure-outlet conditions
 218 were used in present simulation. To ensure fully developed flows at the channel outlet, the back-flow
 219 temperature was adjusted equal to the mass-averaged temperature of forward flow at the precious-step
 220 time (calculated by UDF) as follows:

$$221 \quad T_{backflow} = \frac{\sum_{i=1}^m (\rho_i u_{xi} A_i \cdot T_i)}{\sum_{i=1}^m (\rho_i u_{xi} A_i)}; u_{xi} > 0 \quad (10)$$

222 where, $T_{backflow}$ is the mass-averaged back-flow temperature and ρ_i , u_{xi} , A_i , T_i are the density, velocity in
 223 the x direction, area and temperature of mixture at the i_{th} grid cell of the outlet boundary, respectively.

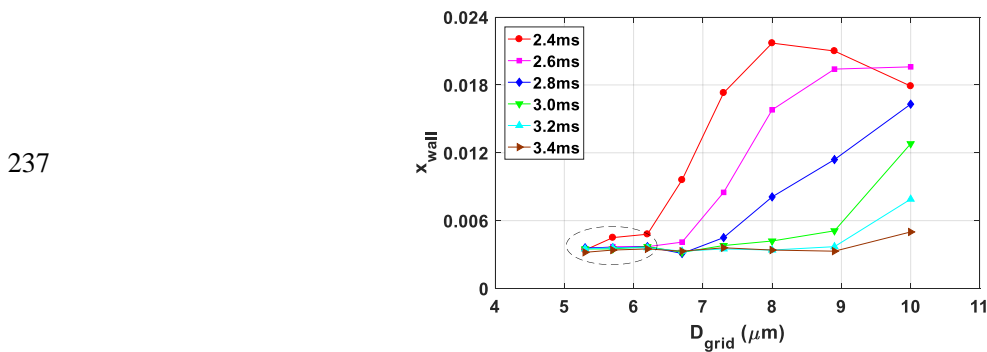
224 Since the NH₃/H₂O mixture will be entering the channel at subcooling state, a steady single-phase flow
 225 field was simulated first and then the convergent results with residual of 10⁻⁵ were adopted as the initial
 226 condition for later flow boiling heat transfer simulation [41].

227 2.4 Mesh independence test and validation

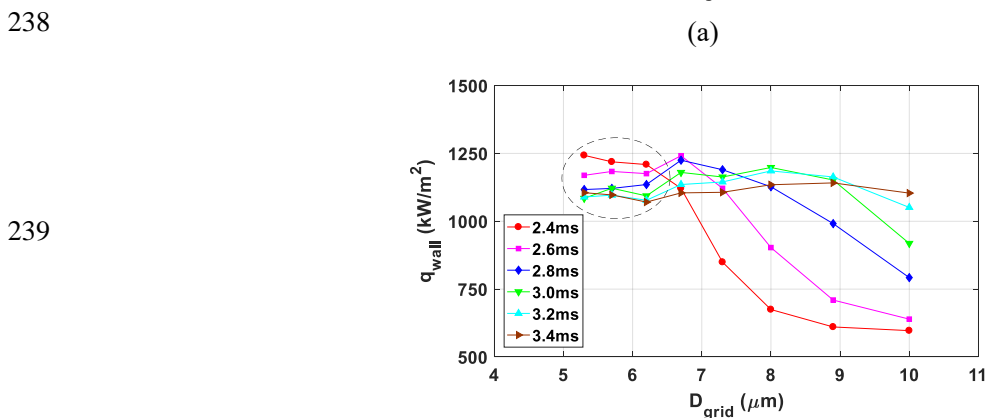
228 A mesh independence test was conducted in the computational domain, shown in Figure 1. Eight mesh

229 sizes were evaluated at mass flux of 46 kg/(m²·s), NH₃ concentration of 30% and heating wall
 230 temperature of 50 °C.

231 The values of area-averaged vapor fraction on heating surfaces (i.e. x_{wall}) and heat flux at heating walls
 232 (i.e. q_{wall}) are plotted against varying grid size in Fig. 2 to demonstrate the grid size effect on the
 233 numerical results. It can be observed from the figure that when the average grid size was decreased to
 234 and below 6.2 μm, the variations of vapor fractions and heat fluxes among cases with different grid sizes
 235 were less than 3%. Therefore, an average grid size of 6.2 μm satisfactorily fulfilled the rule of mesh
 236 independence and was selected for the simulations in present study.



(a)



(b)

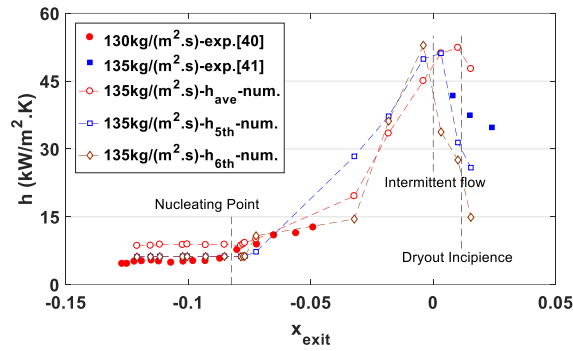
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241 **Fig. 2** Grid independent tests based on the vapor fraction (a) and heat flux (b) at the heating walls

242 Since few experimental studies of NH₃/H₂O mixture flow boiling in microchannels are available in
 243 literature and H₂O could be considered as a special NH₃/H₂O mixture with zero NH₃ concentration, the
 244 numerical construction in this study (e.g. VOF- original Lee model) was validated with experimental
 245 data of subcooled ($T_{in}=30$ °C) water flow boiling heat transfer in copper rectangular microchannels with
 246 similar hydraulic diameters [42, 43]. The contact angle of copper surface was chosen as 86° [44].

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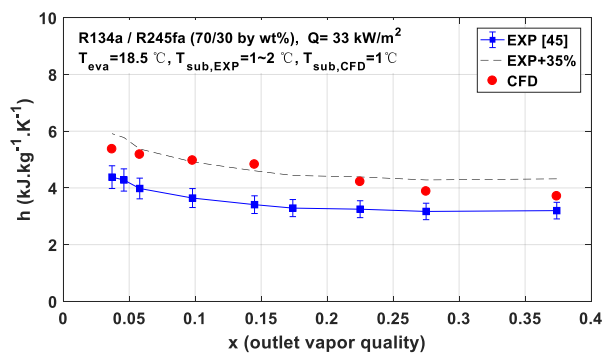


248 **Fig. 3** Comparisons of HTC results among numerical and referenced experimental cases

249 The experimental results of overall HTC (h) versus exit vapor quality (x_{exit}) at various operating
 250 conditions as well as the HTC values calculated using the numerical set-up in this study were compared
 251 and are demonstrated in Fig. 3. The negative x_{exit} values were related to inlet subcooling conditions [45,
 252 46]. In Fig. 3, the solid data points denote experimental results and the connected hollow-dots are for
 253 numerical results. h_{ave} is the average HTC of the whole computational domain, h_{5th} and h_{6th} are the
 254 average HTC values at channel axial locations of 4-5 mm and 5-6 mm, respectively. Fig. 3 indicates that
 255 the numerical results obtained based on the numerical system construction of this study are in good
 256 agreements with the experimental data within relative errors of $\pm 25\%$.

257 To further validate the VOF/modified LEE model in this study, additional numerical simulations were
 258 conducted based on a model of flow boiling heat transfer of R134a/R245fa mixture (70/30 by wt.%) in
 259 a 2D microchannel. The numerical conditions duplicated the experimental conditions in [47] such as the
 260 constant heat flux of 33 kW/m^2 and the evaporation temperature of 18.5°C . The thermodynamic &
 261 transport properties of R134a /R245fa binary mixture were obtained by using NIST REFPROP 9.1 [48].
 262 The results of HTC and flow pattern comparisons between experimental and numerical cases are
 263 demonstrated in Fig. 4 and Fig. 5.

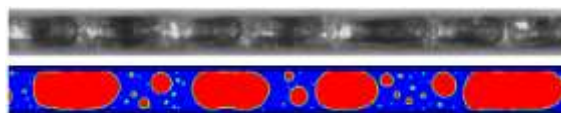
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265 **Fig.4** HTC result comparisons among numerical and referenced experimental cases of R134a/R245fa
 266 mixture (70/30 by wt.%) in the microchannel

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268



269

(a) Confined flow

270



271

(b) Slug flow

272

Fig.5 Comparisons of corresponding confined bubble and slug flow patterns of R134a/R245fa mixture (70/30 by wt.%) between the experiment and numerical simulation

273

274

As shown in Fig.4, the simulation results based on VOF - modified Lee model could well predict the tendency of HTC values as a function of channel outlet vapor quality under the constant heat flux conditions and the deviations of individual HTC values between experiment (blue points) and numerical simulation (red points) are within satisfactory expectations (35%). As the results indicated, the simulated values of HTC were over-predicted in comparison with the experimental data. One possible reason could be that mass diffusion equations were not considered in the present numerical model, which has been agreed as one of the main reasons that causes the heat transfer degradation of zeotropic mixture compared to pure fluids [47]. Ammonia water, as a zeotropic mixture, associates with more than two species (i.e. $\text{NH}_3 \cdot \text{H}_2\text{O}$, NH_4^+ , NH_2^- , OH^- and H_3O^+). Thus, the mass diffusion model exclusively for $\text{NH}_3/\text{H}_2\text{O}$ mixture can be rarely found in literature. Furthermore, the computational cost would be too overwhelming even though such mass diffusion model exists. Alternatively, the mass transfer phenomena of different components in the $\text{NH}_3/\text{H}_2\text{O}$ mixture has been taken into account by the use of effective thermophysical and transport properties of $\text{NH}_3/\text{H}_2\text{O}$ mixture and modified phase change models (i.e. modified Lee model embedded in VOF model).

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Further illustrated in Fig.5, the simulated flow patterns (e.g. confined bubble in Fig.5 (a) and slug flow in Fig.5 (b)) also match well with the corresponding experimentally visualized flow patterns of R134a/R245fa mixture (70/30 by wt.%) flow boiling in microchannel. Hence, the numerical system construction in this study (i.e. VOF - modified Lee model) has been validated and therefore can be properly used for the following investigations of $\text{NH}_3/\text{H}_2\text{O}$ mixture flow boiling characteristics in the microchannel under the condition that there is almost no experimental data of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling in microchannels.

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3 Results and discussion

294

In flow boiling research community, it is generally accepted that there are two main mechanisms (convection and nucleate boiling) for flow boiling heat transfer in microchannels. And the mass flux (dominates convection) and heat flux (related to wall superheat) are the most important factors affecting the flow boiling heat transfer performance [43, 51]. Furthermore, the inlet concentration of the more volatile component of a zeotropic mixture is one of the most distinctive parameters that differentiates flow boiling of zeotropic mixtures from pure fluids [7-10]. Therefore, as the numerical model was successfully validated in section 2.4, the effects of mass flux, inlet NH_3 concentration and heating surface temperature on flow boiling heat transfer of $\text{NH}_3/\text{H}_2\text{O}$ mixture in microchannels were comprehensively investigated and compared under a constant wall temperature boundary condition. During the simulations, the heating wall temperature was set below 85 °C to comply with industrial standard of the maximum functional temperature of common micro-scale electronics [4, 49].

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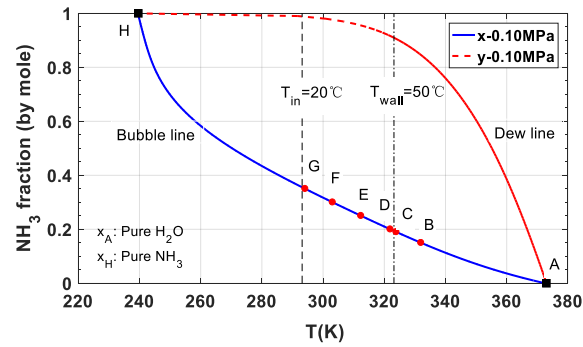
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305 Accordingly, the relevant simulation conditions in this study are detailed in Table 1. Fig. 6 shows the
 306 phase diagram of NH₃/H₂O by mole fraction and temperature at 0.1 MPa, where the left line (blue) and
 307 the right line (red) is the saturated liquid and vapor line of NH₃, respectively. Based on the phase diagram,
 308 the mole fraction of NH₃ was selected from 0.2 to 0.36 to ensure the NH₃/H₂O mixture was at subcooled
 309 condition ($T_{inlet}=20\text{ }^{\circ}\text{C}$) before entering the microchannel and underwent flow boiling under a wall
 310 temperature of 50 °C.

311 **Table 1** Simulation conditions on flow boiling of NH₃/H₂O mixture

Parameter	Range
Mass flux (kg/(m ² ·s))	46~552
NH ₃ concentration (by mole)	0.15~0.35
Heating wall temperature (°C)	20.5~70
Inlet fluid temperature (°C)	20
Operational pressure (MPa)	0.1

312



313

314

Fig. 6 Phase diagram of NH₃/H₂O mixture at 0.1 MPa

315 In addition, the microchannel heating surfaces were deliberately treated as superhydrophilic surfaces
 316 (zero contact angle) in this study featuring optimum flow boiling performance. In literature, it has been
 317 pointed out that superhydrophilic surface could considerably enhance flow boiling HTC and CHF in
 318 microchannels at high heat flux conditions mainly due to the uniform thin liquid film distribution on heating
 319 surface and the consequent delay to partial dryout [50, 51].

320 Both overall and local heat transfer performance of NH₃/H₂O mixture flowing through the microchannel
 321 were investigated. As mentioned earlier in Section 2, the transient arithmetic model was adopted in this
 322 simulation. However, the overall and local HTC results were obtained when the system reached dynamic
 323 equilibrium.

324 The overall HTC was calculated by

$$325 \quad h_{overall} = \sum_{i=1}^n \left[\frac{(q_{L,i} + q_{R,i})/2}{T_{wall} - (T_{in} + T_{out,i})/2} \right] / n \quad (11)$$

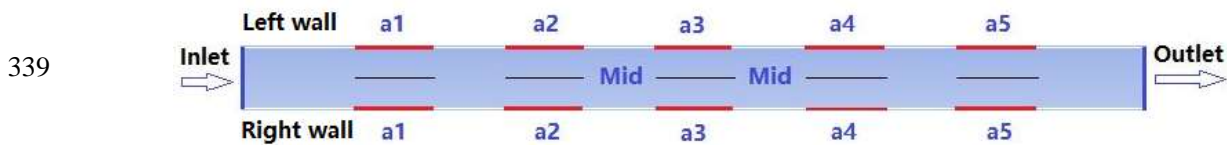
326 where, $h_{overall}$ is the overall heat transfer coefficient (kW/(m²·K)), q_L and q_R is the area-averaged heat flux
 327 (kW/m²) at either the left and right heating wall of the 2-D microchannel, as shown in Fig. 7. T_{wall} , T_{in}

328 and T_{out} are the wall temperature (K), the inlet and outlet fluid temperature (K), respectively. The index
 329 “ i ” denotes an individual transient point in time after the dynamic equilibrium of flow boiling. A total
 330 number of “ n ” individual time points were considered for evaluating the time-average HTC within the
 331 equilibrium state.

332 The local heat transfer coefficient was calculated by

$$333 \quad h_{local,j} = \sum_{i=1,j}^n \left[\frac{(q_{l,j,i} + q_{r,j,i})/2}{T_{wall} - T_{fluid,j,i}} \right] / n \quad (12)$$

334 where, h_{local} is the local heat transfer coefficient (kW/(m²·K)), q_l and q_r is the local area-averaged wall
 335 heat flux (kW/m²) at either the left and right heating wall. The index “ j ” indicates the individual locations
 336 for where the local HTC being investigated, shown as “a1...a5” in Fig. 7. T_{fluid} is the local mass-averaged
 337 fluid temperature (K) corresponding to the individual locations “a1...a5”. The length of each local area
 338 “ a_j ” is 0.5 mm.



340 **Fig. 7** Locations in microchannel domain for obtaining local parametric values

341 3.1 The effect of mass flux

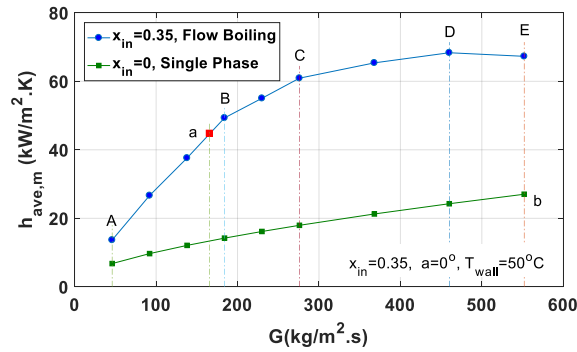
342 The mass flux effect on subcooled flow boiling heat transfer of NH₃/H₂O mixture ($x_{in}=0.35$) at a constant
 343 heating wall temperature of 50 °C was investigated and compared with single-phase convective heat
 344 transfer of H₂O ($x_{in}=0$).

345 3.1.1 The overall heat transfer performance

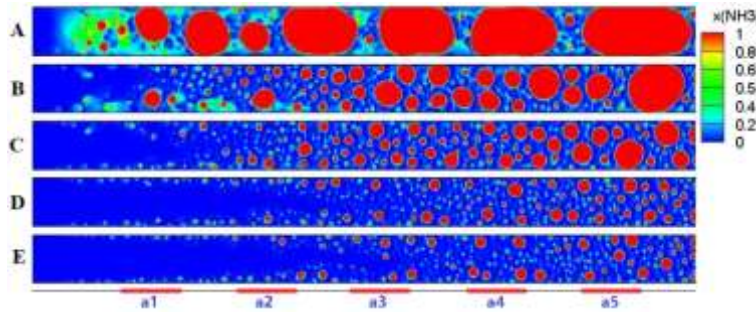
346 The overall HTC of single-phase water convection and flow boiling of NH₃/H₂O are plotted against
 347 mass flux in Fig. 8.

348 As shown in Fig. 8(a), while the HTC of single-phase flow (green) is linearly correlated with varying
 349 mass flux, the HTC of NH₃/H₂O flow boiling (blue) firstly increases linearly with mass flux (A-B),
 350 followed by gradual decrease in growth rate (B-D) and eventually tended to be nearly constant with the
 351 further increase of mass flux (D-E). The corresponding flow patterns at A-E are illustrated in Fig. 8(b)
 352 as A, B are slug flow and D, E are bubbly flow. As the results show, the HTC vs. mass flux tendency of
 353 NH₃/H₂O is similar with the well-known flow boiling behavior, which features that HTC in nucleate
 354 boiling dominant region is depended upon heat flux but far less sensitive to mass flux and vapor quality,
 355 while it is dependent with mass flux and vapor quality but independent with heat flux in the convective
 356 boiling dominant region. Moreover, the nucleate boiling region is normally associated with low vapor
 357 quality that favor relatively small bubbles from nucleation, whereas the convective boiling region
 358 features high vapor quality flow pattern with nucleation inhibited [43, 52].

359 Moreover, it can be observed from Fig. 8(a) that the HTC values of NH₃/H₂O flow boiling are greater
 360 than those of single-phase water convection for all mass fluxes at 50 °C wall temperature. This means
 361 for a thermal application that has to operate below 50 °C, NH₃/H₂O (two phase) is a better choice than
 362 water (cannot boil at 50 °C at 0.1 MPa) in terms of heat transfer performance.



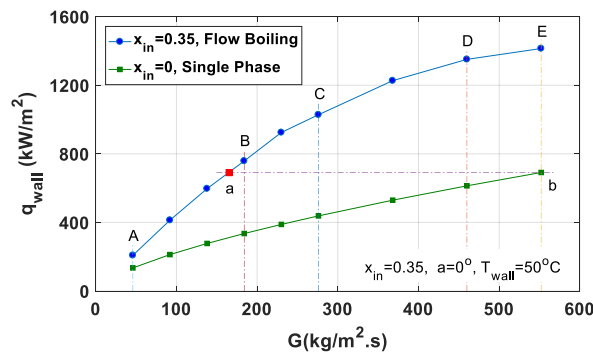
(a)



(b)

363
364
365
366
367 **Fig. 8** The mass flux effect on overall HTC (a) and vapor fraction distribution (b) of NH₃/H₂O flow
 368 boiling in microchannel

369 To more completely evaluate the flow boiling heat transfer performance of NH₃/H₂O, in addition to the
 370 overall HTC results, the values of overall heat flux (average heat flux over the length of microchannel
 371 under constant wall temperature) are plotted as a function of mass flux in Fig. 9 to demonstrate the heat
 372 dissipation capability of NH₃/H₂O flow boiling in the microchannel.



373
374 **Fig. 9** The mass flux effect on overall heat flux of NH₃/H₂O in microchannel

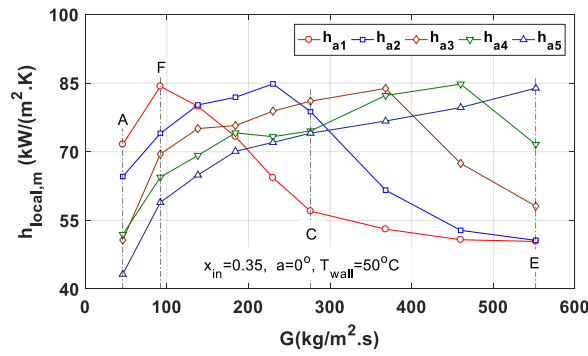
375 As the figure shows, for the same mass flux of 552 kg/(m².s) at case point “E” and “b” in Fig. 9, the

376 overall heat flux of $\text{NH}_3/\text{H}_2\text{O}$ mixture flow boiling in a microchannel could reach up to 1.41 MW/m^2 ,
377 which is 2.05 times the value of the single-phase H_2O convection (0.69 MW/m^2). Furthermore, for the
378 same heat dissipation rate of 0.69 MW/m^2 at case point “a” and “b” in Fig. 9, to keep a constant device
379 temperature (e.g. $50 \text{ }^\circ\text{C}$), the required mass flux of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling is only $166.2 \text{ kg}/(\text{m}^2\cdot\text{s})$, which
380 is 30% the requirement of single-phase H_2O convection and therefore means less required pumping
381 power supply. Hence, flow boiling heat transfer of $\text{NH}_3/\text{H}_2\text{O}$ mixture in microchannels at least shows
382 promises as an alternative thermal management method for high power density electronics though there
383 are problems to be studied and solved in the future.

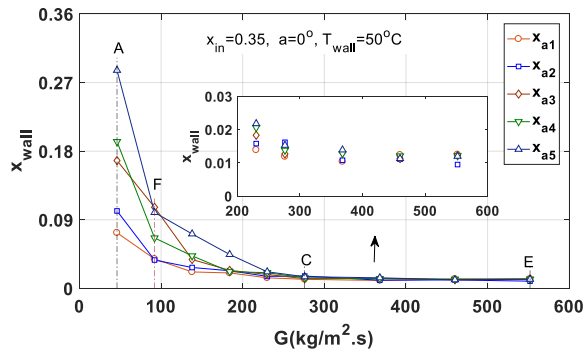
384 *3.1.2 The local heat transfer performance*

385 The local HTC and wall vapor fraction results of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling at various locations, shown as
386 “a1-a5” in Fig. 7, within the microchannel are plotted in Fig. 10. As it can be noticed from Fig. 10(a),
387 the HTC at “a1” that is close to the microchannel inlet, firstly increased from “A” to “F” and then
388 decreased from “F” to “E” against mass flux. At low mass flux (A-F), increasing mass flux positively
389 affected the local flow boiling heat transfer performance at “a1” by taking vapor bubbles away from the
390 surface at a higher rate thereby benefiting local bubble growth. At high mass flux (F-E), local flow
391 boiling at the inlet location “a1” was suppressed because there was not enough time for the fluid to
392 interact with the hot surface and be fully boiled due to the high flow rate. Therefore, there was an
393 optimum mass flux point at “F” where the two effects were best balanced. In addition, it can be observed
394 from Fig. 10(a) that there existed a same maximum local HTC value for all locations (a1-a5) around 85
395 $\text{kW}/(\text{m}^2\cdot\text{K})$ though occurring at different optimum mass fluxes. This shows the existence of a same upper
396 limit for the effect of mutually benefiting nucleation and convection on local flow boiling performance,
397 mainly governed by NH_3 concentration (i.e. the concentration of the more volatile component) under
398 the constant heating wall temperature, regardless the location in microchannel.

399 However, the local flow boiling behavior near the microchannel outlet should be different from that at
400 the inlet and that’s why the HTC trends are different in Fig. 10(a) at different channel locations. This
401 can be explained that the fluid just barely started to boil at the inlet and flow boiling was fully developed
402 at the outlet as the fluid already went through the whole length of the heating surface. Another important
403 situation to be noticed was that the NH_3 concentration in the bulk fluid, as the more volatile component,
404 decreased along the channel (i.e. from “a1” to “a5”). Hence, at “a5” that is close to the microchannel
405 outlet, the flow pattern was fully developed slug flow at low mass flux and fully developed bubbly flow
406 at high mass flux under constant wall temperature condition (Fig. 8(b)) and more importantly there were
407 limited amount of NH_3 left in the bulk fluid to be boiled. Accordingly, as shown in Fig. 10(a), the HTC
408 value at “a5” grew with mass flux. As boiling was fully developed and little NH_3 left in the bulk fluid,
409 the heat transfer performance was mainly dependent on the convective part and mass flux, which can be
410 further supported by the near-zero wall vapor fraction of $\text{NH}_3/\text{H}_2\text{O}$, as shown in Fig. 10(b).



(a)



(b)

Fig. 10 The mass flux effect on local HTC (a) and vapor fraction at heating surfaces (b) of NH₃/H₂O flow boiling in microchannel

3.2 The effect of inlet NH₃ concentration

The concentration of the more volatile component, related to both thermal and mass diffusion, is the most essential factor governing the phase change heat transfer of binary mixtures. Therefore, the inlet concentration of NH₃ has to be carefully investigated especially that in this study the heated wall temperature was only above the saturation temperature of NH₃ not H₂O throughout the entire microchannel. Also, the degree of subcooling at a certain pressure is determined by the inlet concentration of NH₃, which will affect the temporal and spatial distributions of NH₃/H₂O flow boiling along the channel axis. Hence, the effect of inlet NH₃ concentration, including one single phase case (with $x_{in}=0.19$) and four subcooled flow boiling cases (with $x_{in}=0.2, 0.25, 0.3$ and 0.35), on heat transfer performance has been discussed below under a constant heating temperature of 50 °C and operational pressure of 0.1 MPa while the saturated concentration of NH₃/H₂O mixture is 0.194 at the corresponding conditions.

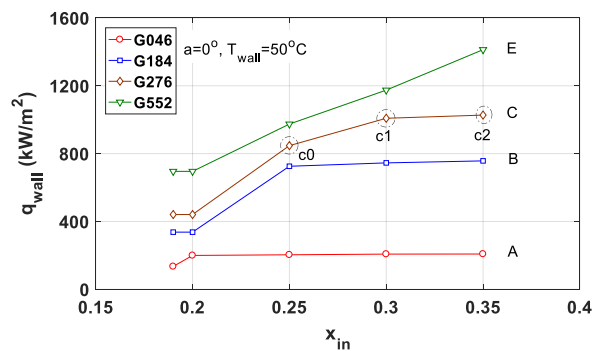
3.2.1 The overall heat transfer performance

Accordingly, the heat flux and HTC results of NH₃/H₂O mixture as a function of inlet NH₃ concentration are illustrated in Fig. 11. As it can be observed from Fig. 11(b), the average flow boiling HTCs at $G=276$ kg/(m².s) (brown line) were almost a constant when x_{in} of NH₃ < 0.2, which was so small that the effect of nucleate boiling was very limited. The overall HTCs were then increased with x_{in} because the effect of nucleate boiling came into play and got strengthened as x_{in} raised (e.g. c₀-c₁ in Fig. 11(b)) until the

435 flow boiling characteristics within the channel reached a steady state as x_{in} was continuously increased.
 436 The reason might be that the increase of NH_3 concentration would not always be beneficial to the overall
 437 flow boiling performance as the effective number of nucleation sites might be reduced by the
 438 incremented number of vapor bubbles sitting on the heating surface. Also, the adverse effects on heat
 439 transfer from surface tension gradient and NH_3 dilution and dissolution induced by the NH_3
 440 concentration gradient within the microchannel also have to be considered. Eventually, the HTC almost
 441 got to the same value between two different x_{in} conditions, shown as c_1 and c_2 in the figure.

442 Furthermore, as indicated in the figure, the trends of HTC against x_{in} are different among various flow
 443 rates. The most important finding is that the HTC reached to a steady value earlier at smaller values of
 444 x_{in} for lower mass flow rate cases, which can be deduced through the comparison between the HTC
 445 curves of $G=46 \text{ kg}/(\text{m}^2\cdot\text{s})$ (red line) and $G=276 \text{ kg}/(\text{m}^2\cdot\text{s})$ (brown line). Flow rate is another factor besides
 446 inlet NH_3 concentration which affect the timing of the onset of nucleate boiling as well as the
 447 development of flow boiling stages within the length of the channel. As mentioned above, a faster fluid
 448 velocity means the fluid would have less time interacting with the heating channel surface which closely
 449 connects with triggering the boiling phenomena while the fluid would interact with the heating surface
 450 more for a lower fluid velocity. Similarly, the heat flux values dissipated from the heating wall to
 451 NH_3/H_2O mixture converged to constant values as the inlet NH_3 concentration incremented for different
 452 mass flow rate cases, as shown in Fig. 11(a) (e.g. red, blue line). The figure reveals that there is a
 453 threshold of inlet NH_3 concentration to maintain a certain level of heat dissipation rate at a given mass
 454 flow rate, for example, $x_{in}=0.25$ for $G=184 \text{ kg}/(\text{m}^2\cdot\text{s})$ (blue line). In addition, at $x_{in}=0.25$, it can be noticed
 455 that the wall heat flux value increased with the mass flux (Fig. 11(a)) as discussed earlier while the HTC
 456 did not follow the same principle, such as, with HTC at $G=276 \text{ kg}/(\text{m}^2\cdot\text{s})$ greater than that at $G=552$
 457 $\text{kg}/(\text{m}^2\cdot\text{s})$. The possible reason for this was that fluid temperature within the microchannel had to be
 458 considered besides heat flux when calculating the HTC and NH_3/H_2O mixture is a zeotropic mixture
 459 with non-isothermal phase change process.

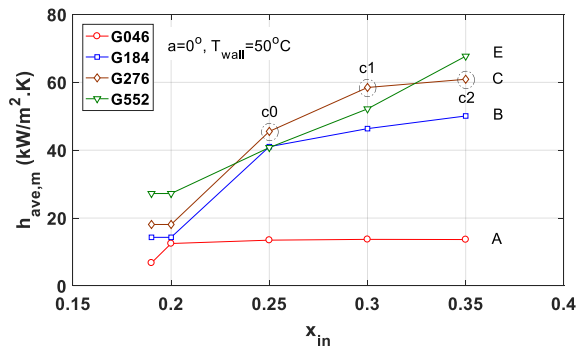
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(a)

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(b)

Fig. 11 The NH₃ concentration effect on overall heat flux (a) and HTC (b) of NH₃/H₂O flow boiling in microchannel

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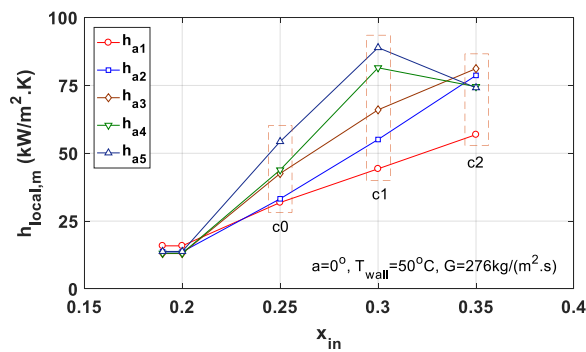
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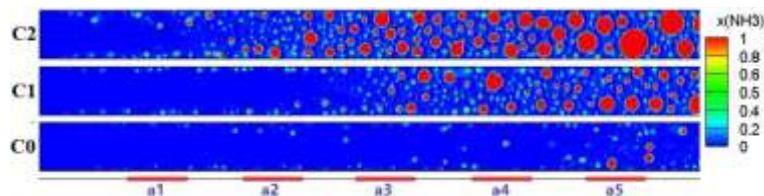
3.2.2 The local heat transfer performance

467 In addition, the effect of NH₃ inlet concentration on local flow boiling heat transfer characteristics was
 468 investigated. As illustrated in Fig. 12(a), at a fixed flow rate of 276 kg/(m²·s), the HTC at “a1” (red line)
 469 linearly grew as the inlet concentration of NH₃ increased while the HTC at “a5” (blue line) first
 470 incremented to a maximum value before its decline. The explanation of this local HTC behavior is
 471 similar to the effect of NH₃ inlet concentration on overall flow boiling heat transfer performance. It was
 472 at early stage of nucleate boiling at “a1” near the channel inlet so that the added NH₃ concentration
 473 promoted the local bubble formation and nucleate boiling. On the other hand, boiling had already been
 474 developed for a long distance through the channel before arriving at “a5” where excessive and large
 475 vapor bubbles on the heating surface started suppressing boiling heat transfer at high NH₃ concentration.
 476 This could be further supported by the local flow patterns, shown in Fig. 12(b).

477



(a)



(b)

Fig. 12 The NH₃ concentration effect on local HTC (a) and vapor fraction distribution (b) of NH₃/H₂O flow boiling in microchannel

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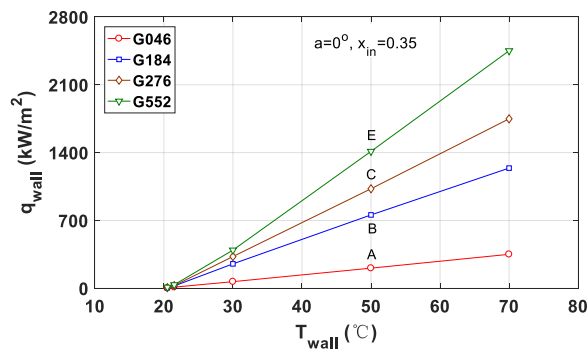
483 3.3 The effect of heating wall temperature

484 Previously, the mass flux and inlet NH₃ concentration effects on NH₃/H₂O mixture flow boiling heat
 485 transfer performance were investigated under the constant heating wall temperature of 50 °C. However,
 486 in practical applications related to this study, the functional temperature of most electronics would vary
 487 in a wider range that may be either lower or higher than 50 °C. Thus, the influence of heating wall
 488 temperature and mass flux on NH₃/H₂O mixture flow boiling heat transfer performance have been
 489 further discussed at a constant NH₃ inlet concentration of 0.35.

490 3.3.1 The overall heat transfer performance

491 In Fig. 13, the results of heat flux and HTC at different mass fluxes are plotted against heating wall
 492 temperature. As Fig. 13(a) shows, the overall heat flux of NH₃/H₂O mixture flow boiling heat transfer
 493 in the microchannel was linearly elevated as heating wall temperature increased due to the enhanced
 494 nucleate boiling. Also, the heat flux was larger at higher mass fluxes because of the enhancing effect of
 495 convection on the overall flow boiling heat transfer. Nevertheless, the trend of HTC, displayed in Fig.
 496 13(b), was different from that of heat flux, in which the HTC did not linearly increase with heating wall
 497 temperature. The HTC value of NH₃/H₂O mixture flow boiling heat transfer was inherently connected
 498 with the flow patterns within the microchannel which was governed by the balance among heating wall
 499 temperature, NH₃ concentration and mass flux. As the heating wall temperature went higher, the HTC
 500 first increased linearly in bubbly flow, then got flattened in slug flow and tended to reach a steady value
 501 before transforming to annular flow.

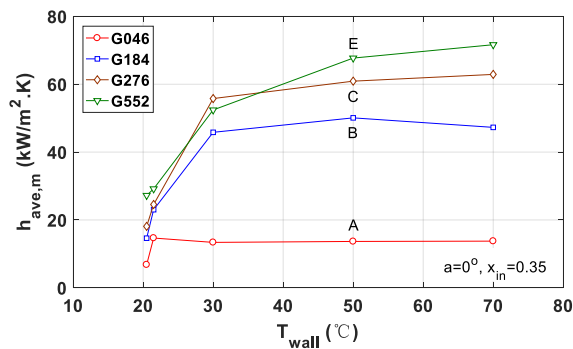
502



503

(a)

504



505

(b)

506 **Fig. 13** The heating temperature effect on overall heat flux (a) and HTC (b) of NH₃/H₂O flow boiling

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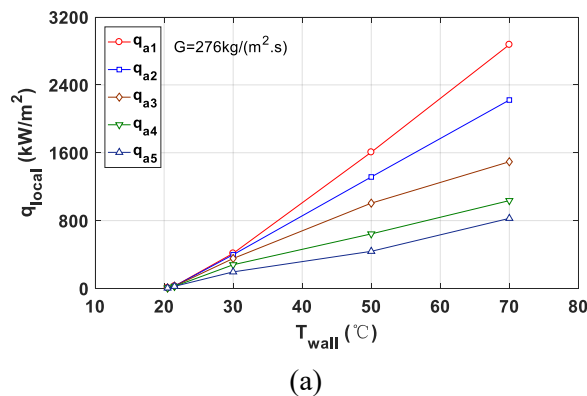
in microchannel

508 *3.3.2 The local heat transfer performance*

509 The local heat flux and HTC at 276 kg/(m²·s) are plotted based on different heating wall temperatures
 510 in Fig. 14. It can be observed from Fig. 14(a) that the local heat flux increased with heating wall
 511 temperature which is the driving force for NH₃/H₂O mixture flow boiling heat transfer. Whereas the
 512 slope of heat flux changing as a function of heating wall temperature was not the same for different
 513 locations at a1-a5. As shown in Fig. 14(a), the heat flux magnitude and the slope of heat flux vs. heating
 514 wall temperature decreased from a1 (red line) to a5 (blue line). This can be explained by that, as
 515 NH₃/H₂O flow boiling developed and NH₃ concentration in the bulk fluid decreased, it was nucleating
 516 boiling dominant bubbly flow at a1 near the microchannel inlet but approaching to convective dominant
 517 slug flow at a5 close to the outlet, as illustrated in Fig. 15(b). Furthermore, in Fig. 14(b), the HTC at
 518 T_{wall}=30 °C increased along the length of the microchannel (a1-a5) as the NH₃/H₂O mixture developed
 519 from subcooled flow boiling towards saturated flow boiling, as shown in Fig. 15(a). However, the HTC
 520 values at T_{wall}=50 °C and 70 °C were not considerably departed from each other except for HTC at
 521 location “a1” since the NH₃/H₂O mixture developed and reached to a certain stable flow boiling pattern
 522 faster at higher heating wall temperatures, as displayed in Fig. 15.

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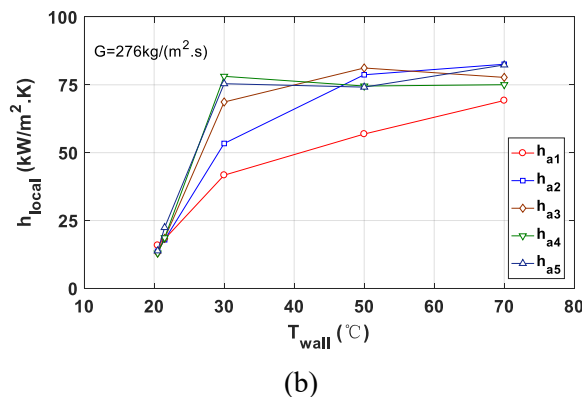


Fig. 14 The heating temperature effect on local heat flux (a) and HTC (b) at 276 kg/(m²·s) of NH₃/H₂O flow boiling in microchannel

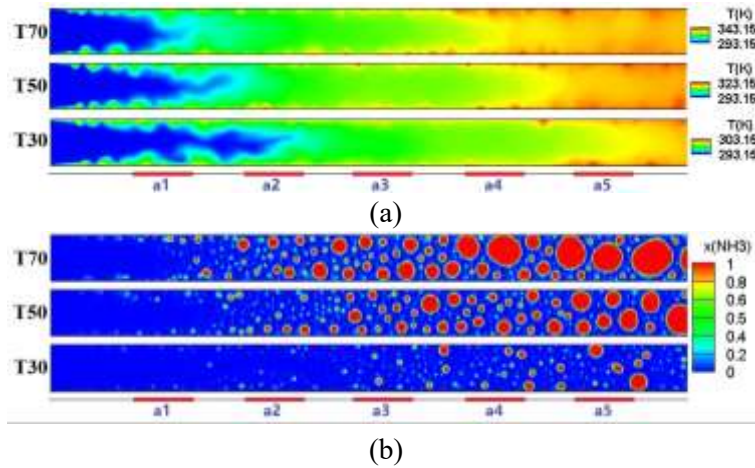
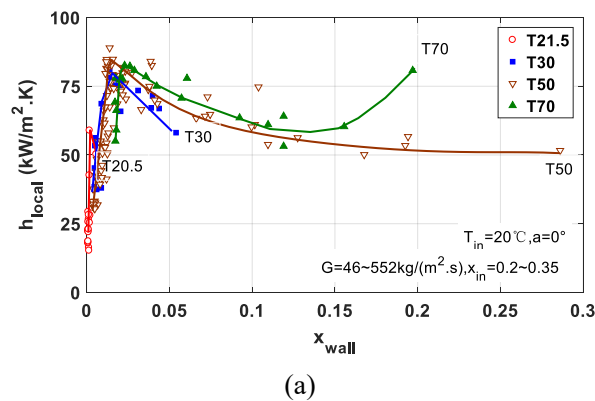


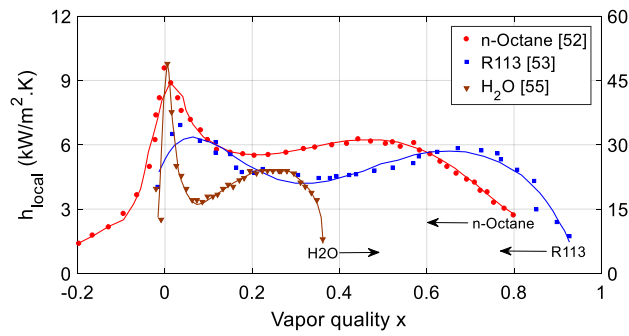
Fig. 15 The heating temperature effect on fluid temperature distribution (a) and vapor fraction distribution (b) of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling in microchannel

3.4 Overall performance evaluation of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling in the microchannel

The local HTC results of $\text{NH}_3/\text{H}_2\text{O}$ flow boiling under various conditions of mass flow rate, inlet NH_3 concentration and heating wall temperature are summarized in Fig. 16(a) as a function of local wall vapor fraction, which is difficult to be physically and accurately measured in flow boiling experiments. As indicated in the figure, the numerical results of local HTC vs x_{wall} , regardless mass flow rate, inlet NH_3 concentration and heating wall temperature, followed the classic experimental M-shape curve of flow boiling HTC against local vapor quality, as shown in Fig. 16(b) for different types of working fluids [52-55]. It can be seen in Fig. 16(a), within the microchannel, the local HTC of $\text{NH}_3/\text{H}_2\text{O}$ mixture linearly grew at low local wall vapor fraction (i.e. nucleation dominant region), declined in the middle (i.e. bubbly flow) after a maximum HTC value and kept decreasing with a slower rate at higher local wall vapor fractions (i.e. slug flow) until a sudden jump to a greater value (i.e. thin film evaporation flow) such as the “T70” curve. Furthermore, it can be concluded from the figure that flow boiling crisis and local dry-out were successfully prevented for $\text{NH}_3/\text{H}_2\text{O}$ mixture under all the experimental conditions investigated in this study (i.e. $T_{\text{wall}} < 70^\circ\text{C}$, $x_{\text{inlet}} < 0.35$, $G < 552 \text{ kg}/(\text{m}^2\cdot\text{s})$).



(a)



(b)

Fig. 16 Local HTC against vapor fraction on heating wall surface of NH₃/H₂O flow boiling in microchannel (a) present results (b) “M” shape curves in literature

4 Conclusions

In this study, numerical simulations were conducted to investigate the flow boiling heat transfer performance of NH₃/H₂O mixture in a single horizontal microchannel with 0.4 mm width and 6 mm length at various conditions. The effects of mass flux (46~552 kg/(m²·K)), inlet NH₃ concentration (0-35% by mole) and heating wall temperature (20.5~70 °C) on the overall and local heat transfer performance in the microchannel have been thoroughly evaluated. The main concluding remarks are as follows:

(1) Based on the numerical results, the flow boiling heat transfer performance of zeotropic NH₃/H₂O mixture in the microchannel was better than single-phase H₂O under a constant heating wall temperature of 50 °C. For the same mass flux of 552 kg/(m²·s), the heat dissipation rate of NH₃/H₂O mixture flow boiling could reach up to 1.41 MW/m², which was 2.05 times the value of H₂O single-phase convective cooling with 0.69 MW/m². While for achieving the same heat flux of 0.69 MW/m², the required mass flux of NH₃/H₂O flow boiling is 166.2 kg/(m²·s), which is 30% of the demanded H₂O flow for single-phase convective cooling.

(2) The numerical results showed that the NH₃/H₂O mixture flow boiling heat transfer in microchannel followed the general flow boiling characteristics except for the non-isothermal phase change feature of zeotropic NH₃/H₂O mixture (i.e. the saturation temperature of NH₃/H₂O mixture in the microchannel was a function of NH₃ concentration and pressure). The results also revealed that there was a threshold of inlet NH₃ concentration above which a steady level of heat dissipation rate was obtained at a given mass flow rate, that is, further increasing the inlet NH₃ concentration would no longer benefit the amount of heat being dissipated, for example, the threshold $x_{in}=0.25$ at $G=184$ kg/(m²·s).

(3) It was also indicated by the numerical simulations that the local HTC curve of NH₃/H₂O mixture flow boiling in the microchannel obeyed the general trend of the classic experimental M-shape curve of flow boiling HTC vs. local vapor quality. Furthermore, there were no local dry-outs throughout the microchannel length under all the simulation conditions in this study (i.e. $T_{wall} < 70$ °C, $x_{inlet} < 0.35$, $G <$

581 552 kg/(m²·s)). Therefore, it can be concluded that the zeotropic NH₃/H₂O mixture is good at preventing
582 local dry-outs and as a result it is a promising alternative coolant for maintaining a certain functional
583 temperature of high power density electronic devices.

584

585 **Declaration of conflicting interests**

586 The authors declare that there is no conflict of interest.

587

588 **Acknowledgement**

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590 Research Council (EPSRC) of the United Kingdom (Grant Nos. EP/N000714/1 and EP/N021142/1).

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602 **Nomenclature**

A	area (m ²)
E	energy (J/kg)
\bar{F}_{surf}	surface tension force (N/m ³)
h	Heat transfer coefficient (kW/(m ² ·K))
k_{eff}	effective thermal conductivity (W/(m·K))
p	pressure (Pa)
q_{LH}	Latent heat (J/kg)
q	Heat flux (kW/m ²)
r_i	relaxation factor during phase change process (m ⁻¹)
S_m	mass source term (kg/(m ³ ·s))
S_h	energy source term (J/(m ³ ·s))
T	temperature (K)
$T_{backflow}$	mass-averaged back-flow temperature (K)
u_x	velocity in the x direction at each grid cell (m/s)
\vec{v}	velocity (m/s)
x	NH ₃ concentration in liquid NH ₃ /H ₂ O mixture (by mol)
y	NH ₃ concentration in gaseous NH ₃ /H ₂ O mixture (by mol)

603 *Greek symbols*

α	volume fraction
κ	local interface curvature (m ⁻¹)
μ	dynamic viscosity (kg/(m·s))
ρ	density (kg/m ³)
σ	surface tension coefficient (N/m)

604 *Subscripts*

p, q	liquid and gas phases of fluid
sat	saturation
local	local heat transfer performance
overall	overall heat transfer performance
wall	heating wall

L, R	left and right side of heating wall
l, r	left and right side of heating wall
fluid	fluid parameter

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