NUMERICAL SIMULATION AND PARAMETRIC STUDY OF BUBBLE GROWTH DURING WATER AND NANOFLOWD FLOW BOILING IN A MICROCHANNEL

A thesis submitted in partial fulfillment of the requirements
for the degree of Master of Science
in Mechanical Engineering

By

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May 2015
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Dr. Robert Ryan  Date

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Dr. Abhijit Mukherjee, Chair  Date

California State University, Northridge
Dedication

This work is dedicated to my family, especially my father Dr. Ahmad Khalighi and my mother Parvin Tahlidost who continuously and generously supported me during my entire education. I like to thank my dearest, Nazilla Forootan who always encouraged me to continue my education and efforts. I wanted to thank my advisor, Dr. Abhijit Mukherjee and my committee members, Dr. Sidney Schwartz and Dr. Robert Ryan for their support.
# Table of Contents

Signature Page ii
Dedication iii
List of Figures vi
Nomenclature viii
Abstract x

## Chapter 1: Introduction

- Microchannels and Micro Scale Flow Boiling 1
- Flow Boiling Regimes in Microchannels 1
- Bubble Dynamics in Microchannels 3
- Addition of Nanoparticles 8

## Chapter 2: Objectives

9

## Chapter 3: Numerical Model

11

- Computational Domain 11
- Governing Equations 12
- Initial Conditions 14
- Boundary Conditions 14
- Flow Regime and Reynolds Number 15
- Nanofluid Properties 15
- Particle Deposition 16
- Wall Superheat 16
- Experimental Validation 17

## Chapter 4: Results and Discussion

18

- Effect of Reynolds Number 19
- Effect of Wall Superheat 22
Effect of Bubble Contact Angles 26
Effect of Thermal Properties Change by Using Nanofluids 30
Instability of Bubble Interface 32
Heat Transfer from Walls 36
Chapter 5: Conclusion 44
References 45
List of Figures

Fig 1.1 Boiling flow regimes [11].
Fig 1.2 Forces acting on a vapor bubble in a conventional flow boiling condition.
Fig 1.3 Static vapor slugs inside horizontal channels [1].
Fig 1.4 Microlayer under a growing bubble.
Fig 1.5 Elongated vapor bubbles or bubble slugs inside a microchannel [19].
Fig 3.1 Computational domain.
Fig 4.1 Bubble shapes.
Fig 4.2 Bubble growth rate (\(\theta=40-30\), \(\Delta T=10 \, ^\circ C\)).
Fig 4.3 Top view of the channel. Bubble at the lowest Reynolds number reaches the side walls earlier (\(\theta=40-30\), \(\Delta T=10 \, ^\circ C\)).
Fig 4.4 Uniform velocity vectors (\(\theta=40-30\), \(\Delta T=10 \, ^\circ C\)).
Fig 4.5 Temperature contours (\(\theta=40-30\), \(\Delta T=10 \, ^\circ C\)).
Fig 4.6 Bubble growth rate. Case of \(\Delta T=10 \, ^\circ C\) grows linearly to reach the domain exit which is not fully shown here (\(\theta=40-30\), Re=100).
Fig 4.7 Top view of the bubble growing at different wall superheats (\(\theta=40-30\), Re=100).
Fig 4.8 Uniform velocity vectors. Bubbles have the same volumes (\(\theta=40-30\), Re=100).
Fig 4.9 Fig 3.9 Bubble growth rate (Re=100, \(\Delta T=10 \, ^\circ C\)).
Fig 4.10 Temperature contours (Re=100, \(\Delta T=10 \, ^\circ C\)).
Fig 4.11 Velocity vectors (Re=100, \(\Delta T=10 \, ^\circ C\)).
Fig 4.12 Temperature contours and velocity vectors (Re=100, \(\Delta T=10 \, ^\circ C\)).
Fig 4.13 Bubble growth rate (\(\Delta T=10 \, ^\circ C\), Re=100, \(\theta=40-30\)).
Fig 4.14 Velocity vectors (\(\Delta T=10 \, ^\circ C\), Re=100, \(\theta=40-30\)).
Fig 4.15 Temperature contours (\(\Delta T=10 \, ^\circ C\), Re=100, \(\theta=40-30\)).
Fig 4.16 Distorted bubble interface (\(\Delta T=15 \, ^\circ C\), Re=250, \(\theta=20-10\)).
Fig 4.17 Temperature contours and uniform velocity vectors (ΔT=15 °C, Re=250, θ=20-10).

Fig 4.18 Temperature contours and uniform velocity vectors at different Reynolds number (ΔT=15 °C, θ=20-10).

Fig 4.19 Heat transfer at the top wall (θ=40-30, ΔT=10 °C).

Fig 4.20 Heat transfer at the side wall (θ=40-30, ΔT=10 °C).

Fig 4.21 Heat transfer at the bottom wall (θ=40-30, ΔT=10 °C).

Fig 4.22 Heat transfer at the top wall (θ=40-30, Re=100).

Fig 4.23 Heat transfer at the side wall (θ=40-30, Re=100).

Fig 4.24 Heat transfer at the bottom wall (θ=40-30, Re=100).

Fig 4.25 Heat transfer at the top wall (Re=100, ΔT=10 °C).

Fig 4.26 Heat transfer at the side wall (Re=100, ΔT=10 °C).

Fig 4.27 Heat transfer at the bottom wall (Re=100, ΔT=10 °C).
# Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
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<tbody>
<tr>
<td>$A$</td>
<td>wall area</td>
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<tr>
<td>$C_p$</td>
<td>specific heat at constant pressure</td>
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<tr>
<td>$D$</td>
<td>characteristic dimension</td>
</tr>
<tr>
<td>$d$</td>
<td>grid spacing</td>
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<tr>
<td>$F_i$</td>
<td>inertia force per unit length</td>
</tr>
<tr>
<td>$F_M$</td>
<td>momentum change force of evaporation</td>
</tr>
<tr>
<td>$F_S$</td>
<td>surface tension force per unit length</td>
</tr>
<tr>
<td>$F_v$</td>
<td>viscous force per unit length</td>
</tr>
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<td>$F_g$</td>
<td>gravitational force per unit length</td>
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**Greek symbols**

<table>
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<th>Symbol</th>
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<tr>
<td>$\beta_T$</td>
<td>coefficient of thermal expansion</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>interfacial curvature</td>
</tr>
<tr>
<td>$\mu$</td>
<td>dynamic viscosity</td>
</tr>
<tr>
<td>$\nu$</td>
<td>kinematic viscosity</td>
</tr>
<tr>
<td>$\rho$</td>
<td>density</td>
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\begin{itemize}
  \item $\sigma$ surface tension
  \item $\tau$ time period
  \item $\phi$ level set function
  \item $\varphi$ contact angle
  \item $\theta$ advancing and receding contact angles set
  \item $\delta$ microlayer thickness
\end{itemize}

Subscripts
\begin{itemize}
  \item $evp$ evaporation
  \item $in$ inlet
  \item $L, l$ liquid
  \item $sat$ saturation
  \item $v$ vapor
  \item $w$ wall
  \item $x$ $\partial/\partial x$
  \item $y$ $\partial/\partial y$
  \item $z$ $\partial/\partial z$
\end{itemize}

Superscripts
\begin{itemize}
  \item $*$ non-dimensional quantity
  \item $\rightarrow$ vector quantity
\end{itemize}
NUMERICAL SIMULATION AND PARAMETRIC STUDY OF BUBBLE GROWTH DURING WATER AND NANOFLUID FLOW BOILING IN A MICROCHANNEL

By

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Master of Science in Mechanical Engineering

In recent years, heat dissipation in micro-electronic devices has become a significant limitation for electronics manufacturers. The amount of heat generation per unit area increases as electronic devices become smaller. Micro heat exchangers have become one of the most effective cooling solutions due to their ability to remove a significant amount of heat from small electronic devices. Utilization of two phase flow in these heat exchangers provides an advantage of removing more heat than single phase flow. Microchannel is the most important part of the micro heat exchangers but it is also the most challenging part due to complications of boiling phenomenon in microchannels. Investigating the bubble dynamics inside these channels helps to better understand the bubble behavior to improve current designs of micro heat exchangers. In this study, a numerical model has been used to perform a parametric study of the bubble dynamics inside a microchannel. Effect of Reynolds number, wall superheat, and bubble contact angles has been studied. Effect of using nanofluids as a homogenous mixture to improve heat transfer has been studied, too. The results show a significant change in bubble
dynamics by changing the wall superheat and bubble contact angles. On the other hand, changing the Reynolds number and nanofluid concentration were found to be less effective in altering the bubble dynamics.
Chapter 1: Introduction

Microchannels and Micro Scale Flow Boiling

Advances in computer chips and microprocessors introduced a new limitation to electronic systems which is thermal management. Dissipation of heat from highly compacted power consumer devices such as CPUs is being performed by large heat sinks which can dissipate up to 250 W/cm² [2]. In the near future, higher rates of heat dissipation up to 1000 W/cm² will be required in electronics [3]. Micro heat exchangers have shown a great potential of heat dissipation in a constrained space [4]. In single phase flow of liquid, the micro heat exchanger can remove heat up to 10,000 W/m² from the surface while this amount can be increased up to 100,000 W/m² by using two phase flow [5]. While the heat dissipating potential of the heat exchangers consisting of microchannels seems promising, the boiling phenomenon in these channels must be well understood in order to make the technology available and reliable for commercial applications.

Microchannels have hydraulic diameters ranging from 10 to 200 µm compared to minichannels with hydraulic diameters ranging from 200 µm to 3 mm. Macro scale or conventional channels have hydraulic diameters greater than 3 mm. The small scale of microchannels results in different flow boiling behavior than conventional channels [1, 6]. Flow boiling in these microchannels is composed of complex behavior that has been subjected to numerous research studies in the past decade. The small scale experimental setups restrict the high resolution data recording during the experiment. In addition to that, boiling in microchannels is a rapid and transitional phenomenon which makes the data recording even more limited. Hence, devices such as high speed cameras along with appropriate lenses, small and highly accurate thermocouples and image processing software are required to record data from a small test section. These requirements resulted in a limited number of reliable experimental studies with a limited range of tested variables until now. According to a suggestion by Kandlikar and Mukherjee [7], appropriate and validated numerical models can help in understanding details of boiling in microchannels. Details such as temperature profiles inside the channel, bubble dynamics, and the effect of different parameters on heat transfer can be investigated with relatively lower cost by a numerical simulation. Here it will be informative to review some of the most important findings in published experimental studies of microchannel flow boiling.

Flow Boiling Regimes in Microchannels

In general, five flow regimes are identified in a microchannel during boiling. Bubbly, slug, churn, wispy-annular, and annular flows are observed in the literature as shown in Fig 1.1 [6, 8-11]. A post-dryout flow regime is also reported that follows the annular
flow. This regime which is also called the inverted-annular flow emerges when the critical wall heat flux has been reached [11]. As mentioned by Ribatski and Bigonha [1], unlike the conventional channels, the stratified flow does not appear inside the microchannels. Lack of stratification is due to the importance of the capillary (surface tension) force over the buoyancy force. Consequently, orientation of the microchannel is not an important parameter in a microchannel flow regime study [6, 8].

Fig 1.1 Flow boiling regimes [11].

In a study by Garimella and Harirchian [11], it was shown that the transition between each flow regime is a function of wall heat flux. Also, increase in mass flux increases the required heat flux for flow transition. They observed that higher mass flux causes the
bubbles to grow slower and become more elongated in the bubbly flow regime. They also performed a series of experiments on channels with different width size ranging from 100 to 5850 µm but all with the same depth of 400 µm. They reported that flow patterns in 100 µm- to 250 µm-wide microchannels are similar. They observed no bubbly flow in microchannels smaller than 250 µm-wide while they observed this regime in larger channels. While flow boiling regimes are different in microchannels, a single vapor bubble in a microchannel also acts differently compared to a bubble during flow boiling in a conventional channel as discussed in the following section.

**Bubble Dynamics in Microchannels**

In flow boiling inside a conventional channel, forces acting on the liquid-vapor interface of the bubble consist of forces caused by momentum change due to evaporation, flow inertia force, surface tension force, shear force, and gravitational force. Surface tension and forces due to evaporation tend to retain the bubble attached to the wall, while the flow inertia and gravitational force tend to detach the bubble from the surface. Fig 1.2 schematically shows the forces acting on a bubble in a horizontal channel in flow boiling.

![Fig 1.2 Forces acting on a vapor bubble in a conventional flow boiling condition.](image)

As the evaporation takes place, the vapor phase leaves the interface of the bubble at higher velocities due to the difference between liquid and vapor phase densities. Change in momentum of the fluid particles results in forces acting on the interface. The magnitude of momentum change is the highest near the heating surface where
evaporation rate is the largest. Kadlikar [6] analyzed the forces acting on a vapor bubble in flow boiling. In case of saturated boiling for which wall heat flux results in an evaporation mass flux of \( \frac{q}{h_{fg}} \), force due to momentum change can be described per unit length as:

\[
F_M = \frac{qD}{h_{fg}} \frac{q}{h_{fg}} \frac{1}{\rho_v} = \left( \frac{q}{h_{fg}} \right)^2 \frac{D}{\rho_v}
\]  

(1.1)

where \( D \) is the characteristic length. The characteristic length can be defined based on the bubble diameter or channel diameter.

The inertia force acting on the bubble due to flow can be described as:

\[
F_I = \rho_l V^2 D = \frac{\rho G D^2}{\rho_l}
\]  

(1.2)

The force due to surface tension is given by:

\[
F_S = \sigma \cos \theta
\]  

(1.3)

where \( \theta \) is the bubble contact angle that can be the advancing or receding contact angle depending on the direction of the liquid motion.

Shear or viscous force per unit length is given by [12]:

\[
F_v \sim \frac{\sigma \mu V D^2}{D} = \mu V = \frac{\mu G}{\rho}
\]  

(1.4)

The gravitational force acting on the bubble can be described as follows:

\[
F_g = (\rho_l - \rho_v) g D^2
\]  

(1.5)

According to the above correlation, the magnitude of the gravitational force is dependent on the square of the characteristic dimension. In microchannels, the characteristic length is in microns; hence, it results in a quite small magnitude of the gravitational force compared to the other forces. This small gravitational force has been seen in experimental studies. The effect of gravity on bubble slugs is experimentally shown by Ribatski and Bigonha [1]. They observed that by increasing the microchannel size from 400 \( \mu \)m to 2000 \( \mu \)m, shape of the static vapor slugs inside the channel changes as shown in Fig 1.3. This indicates that the effect of gravitational force is negligible in 400 \( \mu \)m or smaller microchannels.

Analysis of boiling heat transfer in microchannels is also complicated due to the effect of channel size not only on flow regimes but on a single bubble, too. Once the nucleation is initiated in a microchannel, a bubble grows rapidly due to release of the accumulated heat in superheated liquid surrounding the bubble [6]. As the channel size
decreases, the amount of superheated liquid inside the channel increases. Hence, more superheated liquid is present around the recently nucleated bubble which results in a faster bubble growth rate. This is different from the bubble growth mechanism in conventional subcooled nucleate boiling in which most of the superheated liquid is present under the bubble and not around it. It is a good idea to review what happens to a bubble in a subcooled boiling condition.

![Fig 1.3 Static vapor slugs inside horizontal channels [1].](image)

In subcooled macro scale nucleate boiling, vapor generation has been found to occur by the evaporation from a very thin liquid layer, the microlayer, which forms between the vapor bubble and hot surface as shown in Fig 1.4 [13]. The average microlayer thickness $\delta$ can be determined in terms of the kinematic viscosity of the fluid $\nu$ and the bubble life time $t_b$ through the following correlation [14]:

$$\delta = \frac{8}{7} (3\nu t_b)^{0.5}$$  \hspace{1cm} (1.6)

In case of water nucleate boiling where $t_b = 0.04 \text{ s}$, microlayer thickness is 200 $\mu$m. This indicates the fact that the thin layer of liquid between the vapor bubble and the wall in microchannels, if it exists, must be significantly thinner than the microlayer in macro scale boiling. However, contribution of the microlayer in evaporation in microchannels with hydraulic diameters less than 250 $\mu$m is unlikely to be important in case of saturated flow boiling as discussed by Kandlikar [6]. Instead, the bubble growth is highly influenced by the surrounding superheated liquid as mentioned earlier.
It must be noted that the microlayer discussed above is different from the thin layer which forms when the vapor bubble slugs move along the microchannel. These vapor slugs as shown in Fig 1.5 slide over a very thin liquid layer that separates the bubble from the channel walls. This thin layer has a significant contribution to evaporation and heat transfer inside the microchannel as investigated by Thome and Jacobi [15]. Though the microlayer in slug flow has been subject to many of the recent studies, there is no experimental research work on microlayer evaporation during a single bubble growth after its nucleation inside the microchannel.

![Microlayer under a growing bubble](image)

**Fig 1.4 Microlayer under a growing bubble.**

There are also very few experimental studies in the literature that trace a single bubble during its growth from early stages until filling the microchannel cross section. Some of the first recorded bubble growth data was published by Pan et al. [16] in which the growing vapor bubbles in a trapezoidal microchannel were filmed using a high speed camera. They observed that bubbles grow linearly in time. They compared their observation with the Rayleigh equation for bubble growth in a homogenous and superheated bulk liquid. Their experimental data showed a growth rate of $0.237 \pm 0.008 \mu m/ms$ while the Rayleigh equation predicted $4890 \mu m/ms$ of growth rate in bubble diameter which is significantly larger. They also compared the results with the Cooper model [17] which models the growth of the bubble radius. It is based on the evaporation of the microlayer underneath the growing bubble. This model represents the growth of the bubble radius as a function of the square root of time which is inconsistent with the authors’ observation of linear bubble growth. The authors stated that the microlayer may not be present in the microchannels due to this inconsistency.
In another study, Walton et al. [18] measured and plotted the bubble growth in time inside multiple square microchannels with hydraulic diameter of 194 µm. Bubbles were reported to grow nonlinearly and have three different stages of growth. The first stage is when the bubble is relatively small in the channel and grows rapidly due to evaporation from the surrounding superheated liquid. The second stage begins when the bubble comes in contact with the liquid at upper layers which has a lower temperature. In this stage the bubble grows slower due to lower evaporation rate on its interface. As the bubble continues to grow, it reaches the superheated liquid present near the side walls and its growth rate increases again. This growth rate is highly nonlinear.

Mukherjee and Edel [2] also measured the growth rate of a bubble inside a single microchannel at Reynolds numbers of 100 and 200. The bubble grew linearly in time at a Reynolds number of 100 while it showed a nonlinear growth rate at the Reynolds number of 200. This growth rate of the bubble is higher than the other results mentioned above. Due to the inconsistency of the reported bubble growth rates inside microchannels, it seems necessary to investigate the bubble dynamics in microchannels with appropriate numerical models to look at the effect of different parameters on the bubble growth rate.

As it is evident, one of the main goals of studying flow boiling in microchannels is to find the best way to increase the rate of heat transfer. Changing the fluid or enhancing the thermal and physical properties of the current working fluids can be options to reach the desired heat transfer characteristic. In the next section, one of these enhancements will be discussed.
Addition of Nanoparticles

Direct application of nanoparticles to working fluids inside microchannel is a common practice to increase the convection heat transfer rate. A variety of nanofluid mixtures with different particle concentrations have been studied for single phase flows. It has been well known that nanoparticles can increase the single phase convection heat transfer coefficient [20-22]. Nanoparticles mainly enhance thermal properties of the base fluid which results in an increase in single phase convective heat transfer coefficient.

However, nanoparticles have a secondary role during nanofluid boiling. Nanoparticles deposit on the heated surface during boiling due to evaporation of the base fluid. Much of the research conducted in the past years regarding microchannel and nanofluid flow boiling has shown similar effects from nanoparticle deposition and enhancements to the critical heat flux (CHF). Kim et al. [23] experimentally saw the effects of nanoparticle deposition on a surface during pool boiling. During boiling, nanoparticles had been deposited on the heated surface and this layer had significantly improved the surface wettability, which was shown by a reduction in the contact angle between the fluid and surface. It was also observed that surface texture changes due to particle deposition. Vafaei and Wen [24] reported that nanoparticles can modify bubble dynamics significantly by modifying the contact angle through both nanoparticle suspension and deposition. The deposition of nanoparticles was found to decrease the contact angle for a given volume concentrations of nanoparticles.

More recently, Kim et al. [25] experimentally studied the effects of different nanofluid mixtures including Al$_2$O$_3$ and water, ZnO and water, and diamond and water. The heat transfer coefficient increased with mass flux rate for both pure water and nanofluid; however, the addition of nanoparticles to the base fluid also saw an increase in the CHF. According to this dual effect of nanoparticles during boiling, it seems informative to numerically study the effect of thermal properties enhancement and contact angle change due to presence of nanoparticles. This will be defined as one of the objectives of this study in the next chapter.
Chapter 2: Objectives

A large ratio of vapor bubble volume to channel hydraulic diameter in microchannels results in larger interactions between the bubble and channel walls compared to flow boiling in mini or macro channels. Backward flow caused by rapid bubble growth or flow regulation in multiple microchannels are examples of such interactions that can be seen only in microchannel flow boiling [26]. Moreover, changes in dynamics of a single bubble may result in a different flow boiling regime and eventually a different heat transfer rate in the microchannel. Hence, understanding the dynamics of a single bubble in microchannel is important not only to avoid undesired conditions such as surface dryout and backward flow, but also to know the key parameters in a microchannel flow boiling. In case of constant fluid properties and fixed microchannel geometry, parameters that can affect the bubble dynamics according to the forces discussed in Chapter 1 would be the liquid mass flux or Reynolds number as its non-dimensional form, applied heat to the channel and bubble contact angle. In case of boiling in microchannels smaller than 250 µm, the gravity effect is negligible and the bubbly flow inside the microchannel which is shown in Fig 1.1 (a) never appears. Instead, each bubble grows to the size of the microchannel and fills the entire channel cross-section. This study will focus on the early stages of a single bubble growth when the bubble volume is not large enough to fill the microchannel cross-section. The growing bubble will be studied after its nucleation along a channel length three times larger than the channel width.

During flow boiling, magnitude of the channel wall superheat degree is a key parameter in heat transfer from the wall. As discussed in Chapter 1, a large amount of superheated liquid is present around the recently nucleated bubble causing rapid growth of the bubbles inside the microchannel. A higher rate of heat transfer from the channel walls results in presence of more superheated liquid around the bubble which increases the bubble growth rate. Accordingly, microchannel wall superheat is likely to have a larger effect on bubble dynamics compared to Reynolds number and bubble contact angles. This study investigates and compares the effect of Reynolds number, wall superheat and bubble contact angles on the dynamics of a single bubble after its nucleation inside a microchannel. Typical values of these parameters in microchannel flow boiling have been selected and the effect of each one will be discussed in detail and the parameters which have the most significant effect on bubble dynamics under the defined conditions will be recognized. Bubble interface experiences deformation due to combined effect of these parameters. The condition which results in deformation of the bubble interface will also be discussed in this study.

Moreover, the effect of using nanofluids on bubble dynamics will be investigated as well. As discussed earlier, nanofluids have dual effects on flow boiling. According to the experimental studies mentioned before, even diluted nanofluids significantly affect the rate of heat transfer in channels during boiling. It seems that the nanoparticle deposition
is responsible for this change and not the base fluid thermal properties enhancement. In this study, the effect of thermal properties change caused by nanoparticle addition will be compared to the effect of surface enhancement caused by nanoparticle deposition on a single bubble during boiling by means of a three-dimensional numerical model. By using this model, we are able to closely look at the bubble growing inside the channel and find the main cause of its faster growth rate when a nanofluid is the working fluid. The mixture used in this study is alumina-water nanofluid which is a typical nanofluid used in microchannels due to its superior properties compared to other nanofluid mixtures.
Chapter 3: Numerical Model

Mukherjee and Dhir [27] developed a three dimensional numerical model which employed the level set function method in order to study lateral merger of vapor bubbles during nucleate boiling. Later, Mukherjee and Kandlikar [7] developed a complete model of a growing bubble inside a microchannel. This numerical model has served as a baseline of the present study of bubble dynamics in a microchannel.

Computational Domain

The computational domain is shown in Fig 3.1. Cartesian coordinates have been used with a domain of 3.96×0.99×0.99, non-dimensional units in size, and the grid is uniform. The liquid leaves the domain at x=3.96. The nucleation cavity is placed on the bottom plane or x-z plane at y=0 and at the center of the channel cross-section with the same distance from the side walls (z=0). The non-dimensional unit of the channel length is divided into 80 cells to optimize the computational time and minimize the computational error.

![Fig 3.1 Computational domain.](image1)

The complete incompressible Navier-Stokes equations are solved using the SIMPLER method [28]. The continuity equation is turned into an equation for the pressure correction and a pressure field is derived from the known velocity field. After each iteration, the velocities are corrected using velocity-correction formulas. The
computations proceed to convergence via a series of continuity satisfying velocity fields. The algebraic equations are solved using a TDMA (Tri-Diagonal Matrix Algorithm) algorithm. The speed of convergence of this line-by-line technique is increased more by supplementing it with the block-correction procedure [29]. The multi-grid technique is employed to solve the pressure equations.

Sussman et al. [30] developed a level set approach where the interface was captured implicitly as the zero level set of a smooth function. The level set function was typically a smooth function, denoted as $\phi$. This formulation eliminated the problems of adding/subtracting points to a moving grid and automatically took care of merging and breaking of the interface. Furthermore, the level set formulation generalized easily to three dimensions. The present analysis is done using the mentioned level set technique.

The liquid vapor interface is identified as the zero level set of a smooth distance function $\phi$. The level set function $\phi$ is positive outside the vapor bubble and negative inside the vapor bubble. The interface is located by solving the level set equation. A fifth order WENO (Weighted, Essentially Non-Oscillatory) scheme is used for left sided and right sided discretization of $\phi$ [31]. While $\phi$ is primarily a distance function, it will not remain so after solving the level set equation. Maintaining $\phi$ as a distance function is necessary for providing the interface with a width fixed in time. This is achieved by re-initialization of $\phi$. A modification of Godunov’s method is used to determine the upwind directions. The re-initialization equation is solved in fictitious time after each fully complete time step. With $\Delta \tau = \frac{h}{2u_0}$, ten $\tau$ steps are taken with a third order TVD (Total Variation Diminishing) Runge Kutta method.

**Governing Equations**

**Momentum equation:**

$$\rho \left( \frac{\partial \vec{u}}{\partial t} + \vec{u} \cdot \nabla \vec{u} \right) = -\nabla p + \rho \vec{g} - \rho \beta_T (T - T_{sat}) \vec{g} - \sigma \kappa \nabla H + \nabla \cdot \mu \nabla \vec{u} + \nabla \cdot \mu \nabla \vec{u}$$

$$\text{for } \phi > 0$$

$$\nabla \cdot \mu \nabla \vec{u}$$

(3.1)

**Energy equation:**

$$\rho C_p \left( \frac{\partial T}{\partial t} + \vec{u} \cdot \nabla T \right) = \nabla \cdot k \nabla T \text{ for } \phi > 0,$$

$$T = T_{sat} \text{ for } \phi \leq 0$$

(3.2)

**Continuity equation:**
\[ \nabla \cdot \vec{u} = \frac{\dot{m}}{\rho^2} \cdot \nabla \rho \quad (3.3) \]

The curvature of the interface:

\[ \kappa(\phi) = \nabla \cdot \left( \frac{\nabla \phi}{|\nabla \phi|} \right) \quad (3.4) \]

The mass flux of liquid evaporating at the surface:

\[ \dot{m} = \frac{k_i \nabla T}{h_{fg}} \quad (3.5) \]

The vapor velocity at the interface due to evaporation:

\[ \vec{u}_{evp} = \frac{\dot{m}}{\rho_v} = \frac{k_i \nabla T}{\rho_v h_{fg}} \quad (3.6) \]

To prevent instabilities at the interface, the density and viscosity are defined as:

\[ \rho = \rho_v + (\rho_l - \rho_v)H \quad (3.7) \]
\[ \mu = \mu_v + (\mu_l - \mu_v)H \quad (3.8) \]

\( H \) is the Heaviside function given by:

\[ H = \begin{cases} 1 & \text{if } \phi \geq +1.5d \\ 0 & \text{if } \phi \leq -1.5d \end{cases} \quad (3.9) \]

\[ H = 0.5 + \frac{\phi}{3d} + \frac{\sin \frac{2\pi \phi}{3d}}{2\pi d} \quad \text{if } |\phi| \leq 1.5d \]

where \( d \) is the grid spacing.

Since the vapor is assumed to remain at saturation temperature, the thermal conductivity is given by:

\[ k = k_i H^{-1} \quad (3.10) \]

The level set equation is solved as:

\[ \frac{\partial \phi}{\partial t} + (\vec{u} + \vec{u}_{evp}) \cdot \nabla \phi = 0 \quad (3.11) \]

After every time step, the level-set function, \( \phi \) is reinitialized as:

\[ \frac{\partial \phi}{\partial t} = S(\phi_0)(1 - |\nabla \phi|)u_0 \quad (3.12) \]
\( \phi(x, 0) = \phi_0(x) \)

\( S \) is the sign function which is calculated as:

\[
S(\phi_0) = \frac{\phi_0}{\sqrt{\phi_0^2 + d^2}}
\]  

(3.13)

The governing equations are made non-dimensional by the aid of time scale and length scale. The length scale is width/height of the channel. The non-dimensional temperature is defined as:

\[
T^* = \frac{T - T_{sat}}{T_w - T_{sat}}
\]  

(3.14)

The Nusselt number \((Nu)\) is calculated based on the area-averaged heat transfer coefficient \((\bar{h})\) at the wall given by

\[
\bar{h} = \frac{1}{A} \int_0^A h dA
\]  

(3.15)

where \(A\) is the wall area and \(h\) is obtained from:

\[
h = \frac{-k_l \partial T}{\partial y}_{\text{wall}} \text{ for horizontal walls,}
\]  

(3.16a)

and

\[
h = \frac{-k_l \partial T}{\partial z}_{\text{wall}} \text{ for vertical walls.}
\]  

(3.16b)

The wall Nusselt number is defined as:

\[
Nu = \frac{\bar{h}l_0}{k_l}
\]  

(3.17)

**Initial Conditions**

The bubble is placed at \(x=0.99, y=0\) and \(z=0\), with \(0.1l_0\) radius in the domain. All velocities at the initial grid points are set to zero. The liquid inlet temperature is set to \(0.25T_w\) and the wall temperature is set to values that will be discussed later. The vapor inside the bubble is set to the saturation temperature of 100 °C. The initial liquid temperature inside the domain is set equal to the inlet liquid temperature. The contact angle at the walls is set as specified later in the text.

**Boundary Conditions**

The boundary conditions are as follows:
At the inlet (x=0):

\[ u = u_0; \quad v = w = 0; \quad T = T_{in}; \quad \phi_x = 0 \]  \hspace{1cm} (3.18)

Constant inlet flow velocity has been specified in the numerical calculations. In parallel microchannel heat exchangers uniform inlet flow velocity is necessary to maintain stable operating conditions, which can be achieved using flow restrictions at the inlet [32].

At the outlet (x=3.96)

\[ u_x = v_x = w_x = T_x = 0; \quad \phi_x = 0 \]  \hspace{1cm} (3.19)

At the plane of symmetry (z=0)

\[ u_z = v_z = w_z = T_z = 0; \quad \phi_z = 0 \]  \hspace{1cm} (3.20)

At the walls (y= 0, y=0.99)

\[ u = v = w = 0; \quad T = T_w; \quad \phi_y = -\cos \varphi \]  \hspace{1cm} (3.21)

where \( \varphi \) is the contact angle.

At the wall (z=0.495)

\[ u = v = w = 0; \quad T = T_w; \quad \phi_z = -\cos \varphi \]  \hspace{1cm} (3.22)

Flow Regime and Reynolds Number

In this study, boiling of water and a nanofluid in a square cross-section microchannel is considered. The cross-sectional dimension of the square microchannel is chosen as 226x226 \( \mu \text{m} \). The nanofluid is a mixture of water, as the base fluid, and \( \text{Al}_2\text{O}_3 \) as the metallic nanoparticles introduced to the base fluid. Due to the very small size of the particles, it may be satisfactory to assume that the mixture can be modeled as a homogeneous fluid which is suggested by Roy et al. [33]. The flow regime of the nanofluid in the microchannel is laminar and Reynolds numbers of 50, 100, and 250 have been selected which are typical of microchannel flows.

Nanofluid Properties

A mixture of water as the base fluid and alumina oxide as nano additives is used to investigate the effect of nanofluids with different volumetric concentrations (vol\%) on bubble dynamics. Water-alumina oxide is one of the most common nanofluids being used in nanofluid heat transfer studies due to its superior thermal properties as well as its great mixture stability. All of the thermal-physical properties of the water-\( \text{Al}_2\text{O}_3 \) nanofluid are calculated at 100 °C for 1 vol\% and 2 vol\%. Viscosity of the nanofluid is calculated
based on the model and experimental results for water-Al₂O₃ nanofluid presented by Bose et al. [34], and the calculated viscosity compared to another model suggested by Vajjha [35] to ensure the calculated value is consistent. Thermal conductivity of the water-Al₂O₃ is calculated based on the data and model presented by Bose et al. [34]. Density of the nanofluid is calculated based on the theoretical equation suggested by Pak and Cho [36]. Specific heat of the nanofluid is predicted based on the correlation developed by Vajjha and Das [37] for water-Al₂O₃ nanofluid. Table 3.1 shows the properties of pure water and water-Al₂O₃ nanofluid with 1 vol% and 2 vol% concentration. It must be noted that the physical nanoparticles are not modeled in this study and the mixture is assumed to be homogenous.

<table>
<thead>
<tr>
<th>Volumetric Concentration</th>
<th>( k ), W/m K</th>
<th>( C_p ), kJ/kg K</th>
<th>( \mu \times 10^{-3} ), kg/m s</th>
<th>( \rho ), kg/m³</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pure Water</td>
<td>0.680</td>
<td>4.23</td>
<td>0.279</td>
<td>958</td>
</tr>
<tr>
<td>1 vol%</td>
<td>0.699</td>
<td>4.13</td>
<td>0.312</td>
<td>987</td>
</tr>
<tr>
<td>2 vol%</td>
<td>0.718</td>
<td>4.04</td>
<td>0.355</td>
<td>1016</td>
</tr>
</tbody>
</table>

Table 3.1 Pure water and water-Al₂O₃ nanofluid properties at 100 °C

Particle Deposition

During boiling of the nanofluids, nanoparticles deposit on the boiling surface. This deposition changes the surface roughness and surface wettability which was reported by Kim et al. [25]. The change in surface characteristic alters the contact angle of the bubbles on the surface. Kim et al. also showed that the contact angle of a static droplet on a metallic surface decreases from 80 to 23 degrees after the surface was used to boil the water-Al₂O₃ nanofluid. In the current numerical study, three sets of 60-50, 40-30, and 20-10 degrees as advancing and receding contact angles have been chosen. These three sets of contact angles were selected to reflect the different surface wettability caused by the nanoparticle deposition. Contact angles are assumed to be constant during growth of the bubble.

Wall Superheat

Recording the local wall superheat in microchannels is almost an impossible task due to the small scale of the test section and rapid nature of the boiling in microchannels. Most of the experimental studies have reported the average wall superheat during bubble growth inside the microchannel. Also, axial variation and fluctuation of the wall temperature is high due to instability of the flow and blockage of the channel by grown vapor bubbles which has been shown by Edel and Mukherjee [38]. Edel and Mukherjee [38] and Walton et al. [18] reported wall superheat up to 120 °C during flow boiling inside a microchannel. In this study, wall superheat remains constant during growth of
the bubble. Wall superheats of 5 °C, 10 °C, 15 °C, and 20 °C have been selected in order to study their effect on bubble dynamics inside the channel. Wall superheats of 2 °C and 3 °C were also selected in order to compare the bubble growth rate with the experimental data reported by Edel and Mukherjee [2].

Experimental Validation

Edel and Mukherjee [2] experimentally recorded bubble growth rates ranging from 140 μm/ms to 280 μm/ms with an average wall superheat of 102 °C and a liquid inlet temperature of 80 °C at a Reynolds number of 100. It was stated that the observed bubble grows linearly in time at Reynolds number of 100 while it does not grow linearly at Reynolds of 200. It should be noticed that temperature variation along the tested channel was reported to be high and the local wall superheat where the bubbles had nucleated is unknown. The local wall superheat at the bubbles location could be any temperature between 100 °C to 120 °C.

Lack of data on local wall superheat and liquid superheat surrounding the growing bubble makes it difficult to validate numerical simulation results quantitatively with experimental data. However, it is possible to compare the range of bubble growth rates reported in the literature with the numerical results. Table 3.2 compares different reported bubble growth rates with two simulated cases.

<table>
<thead>
<tr>
<th>Source</th>
<th>Re</th>
<th>$T_w$ [°C]</th>
<th>$T_l$ [°C]</th>
<th>$T_in$ [°C]</th>
<th>Growth rate type</th>
<th>Bubble growth rate [μm/ms]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lee. et al. [16]</td>
<td>25-70</td>
<td>98-157</td>
<td>103-124</td>
<td>N/A</td>
<td>Linear</td>
<td>0.24-95.3</td>
</tr>
<tr>
<td>Walton et al. [18]</td>
<td>43-125</td>
<td>N/A</td>
<td>N/A</td>
<td>71</td>
<td>Nonlinear</td>
<td>N/A</td>
</tr>
<tr>
<td>Edel and Mukherjee [2]</td>
<td>100</td>
<td>102</td>
<td>N/A</td>
<td>80</td>
<td>Linear</td>
<td>140-280</td>
</tr>
<tr>
<td>Edel and Mukherjee [2]</td>
<td>200</td>
<td>100.9-103.7</td>
<td>N/A</td>
<td>80</td>
<td>Nonlinear</td>
<td>N/A</td>
</tr>
<tr>
<td>Case 1</td>
<td>100</td>
<td>102</td>
<td>101.5</td>
<td>101.5</td>
<td>Linear</td>
<td>300</td>
</tr>
<tr>
<td>Case 2</td>
<td>100</td>
<td>103</td>
<td>100.75</td>
<td>100.75</td>
<td>Linear</td>
<td>223</td>
</tr>
</tbody>
</table>

Table 3.2 Comparison of experimental bubble growth rates with simulated cases

Bubble growth rate of Case 2 is inside the range reported by Edel and Mukherjee [2]. Though Case 2 has a higher wall superheat, it grows slower than Case 1 with wall superheat of 102 °C. This is due to the higher superheat of the surrounding liquid in Case 1.
Chapter 4: Results and Discussion

The bubble with the specified contact angles is nucleated at x=0.226 mm at the start of the numerical calculations. The bubble is placed at the first one-fourth of the channel near the channel inlet. The bubble grows due to evaporation of fluid at its interface. The shape of the bubble is spherical initially, however, as the bubble grows to the size of the channel, it starts to elongate in the direction of the flow. Fig 4.1 shows the growing bubble with contact angles of 40-30 inside the microchannel when wall temperature is 110 °C and Reynolds number is 100. The corresponding time to each frame is indicated on the top left corner of each frame in milliseconds. The bubble grows in both positive and negative x directions.

Fig 4.1 Bubble shapes.
Fig 4.2, Fig 4.6, Fig 4.9, and Fig 4.13 show the equivalent diameter of the bubble against time during growth of the bubble at different values of Reynolds number, wall superheat, nanofluid concentration and bubble contact angles, respectively. The equivalent diameter is the diameter of a sphere with the same volume as the bubble. The initial diameter of the bubble at 0 ms is 0.0452 mm since the initial bubble radius is set to 0.1\(l_0\).

Since the bubble has almost a linear growth rate, it is a good idea to compare the bubble growth rates with a base case. A vapor bubble with contact angles of 40-30, in pure water with Reynolds number of 100 and wall superheat of 10 °C, is taken as the base case. It is shown by a black solid line in each bubble growth rate plot.

**Effect of Reynolds number**

Fig 4.2 shows the effect of Reynolds number on bubble growth rate. Equivalent diameter is the diameter of a sphere that has the same volume as the bubble. Bubble growth rate increases by 13% as the Reynolds number drops from 250 to 50. At lower Reynolds numbers, the bubble grows nearly spherically before contacting the channel side walls. Higher Reynolds number or higher fluid velocity suppresses the bubble. Larger viscous forces caused by higher fluid velocity passing around the bubble results in more elongation of the bubble in the flow direction.

![Fig 4.2 Bubble growth rate (\(\theta=40-30\), \(\Delta T=10\) °C).](image)

19
At higher Reynolds number, inertia of the incoming liquid is larger compared to forces due to momentum change in evaporation. This results in sliding of the bubble inside the channel, which is consistent with the observation of sliding bubbles at higher Reynolds numbers reported by Edel and Mukherjee [2]. These effects can be seen in Fig. 4.3 which is a top view of the bubble at different Reynolds numbers. It also can be seen that the bubbles at Reynolds number of 50 and 100 come in contact the channel side walls before exiting the computational domain while the bubble with larger volume at Reynolds of 250 has not contacted the side wall.

Fig 4.3 Top view of the channel. Bubble at the lowest Reynolds number reaches the side walls earlier ($\theta=40-30$, $\Delta T=10$ °C).
Fig 4.4 represents the velocity vectors inside the channel for the three different Reynolds numbers. Vector magnitudes are uniform in this figure. Ability of the growing bubble to push the surrounding liquid toward the upstream and the upper wall reduces as the Reynolds number increases. This is due to higher inertia force of the incoming liquid that suppresses the bubble. Again, slide and elongation of the bubble is visible at Reynolds number of 250.

![Velocity vectors](image1)

**Fig 4.4 Uniform velocity vectors ($\theta=40-30$, $\Delta T=10^\circ C$).**

Fig 4.5 shows the temperature contours in the x-y plane and through the bubble. The thermal boundary layer at the top wall is pushed against the wall due to growth of the bubble. The thermal boundary layer is more squeezed when the Reynolds number is lower due to the rapid growth of the bubble.
Fig 4.5 Temperature contours ($\theta=40-30$, $\Delta T=10 \, ^\circ C$).

**Effect of Wall Superheat**

As mentioned before, wall superheats of 5 °C, 10 °C, 15 °C, and 20 °C have been selected in this study. Fig 4.6 shows the effect of wall superheat on the bubble growth rate inside the microchannel. The bubble grows up to 370% faster by increasing the wall superheat from 5 °C to 20 °C.

Fig 4.7 shows the top view of the bubbles inside the channel at different wall superheats. All bubbles have the same volumes. The bubble on the least superheated wall has contacted the side walls. It is elongated less than the other bubbles and has a more spherical shape.
At higher degrees of wall superheat, bubbles are not in contact with the side walls. Two mechanisms contribute to this phenomenon. The first mechanism comes from the fact that bubbles grow faster on more superheated walls. A fast growing bubble tends to push the surrounding liquid with a higher velocity in all directions. Consequently, liquid entrapped between the bubble and the side walls must also move fast. This rapid movement produces a very large viscous force that prevents the liquid from evacuating the gap between the bubble and the side wall. Therefore, bubbles grows more in the upstream and downstream directions as shown in Fig 4.8. The second mechanism is the force balance between the liquid inertia force and the momentum change force of evaporation. This balance causes the bubble to slide toward the channel exit at low wall superheats. Downstream movement of the bubble reduces the viscous forces acting on its interface due to a smaller velocity gradient between the interface and the surrounding liquid. Smaller viscous forces result in less elongation which allows the bubble to grow more spherically. Hence, the bubble can reach the side walls at a smaller volume.

Fig 4.6 Bubble growth rate. Case of $\Delta T=10$ °C grows linearly to reach the domain exit which is not fully shown here ($\theta=40-30$, Re=100).
Fig 4.7 Top view of the bubble growing at different wall superheats ($\theta$=40-30, Re=100).
Fig 4.8 Uniform velocity vectors. Bubbles have the same volumes ($\theta=40-30$, Re=100).
In Fig 4.7, the bubble on the 20 °C superheated wall shows signs of instability at its upstream meniscus which is facing the incoming liquid. Other cases have been run in order to study the bubble instability and will be discussed in Instability of Bubble Interface section.

Effect of Bubble Contact Angles

Fig 4.9 shows the effect of change of contact angles change on the bubble growth rate. As the advancing and receding contact angles decrease from 60-50 to 40-30, the bubble grows faster. Further decrease of contact angles down to 20-10 results a significant increase in the bubble growth rate. The bubble with the contact angles of 20-10 grows up to 80% faster than the bubble with contact angles of 60-50.

![Fig 4.9 Bubble growth rate (Re=100, ΔT=10 °C).](image)

Fig 4.9 Bubble growth rate (Re=100, ΔT=10 °C).

Fig 4.10 represents the temperature contours of the bubbles with different contact angles inside the channel and at the same moment in time. The bubble with the contact angles of 20-10 has a larger volume. A closer look at the downstream portion of the bubble shows that a very thin liquid layer is formed between the bubble interface and the bottom wall. A large temperature gradient exists in this layer. This large gradient causes a high evaporation rate of the liquid which is shown in Fig 4.11. Large magnitude velocity
vectors are generated from the thin liquid layer formed under the bubble which represents a high evaporation rate.

In addition to the thin layer, there is a small amount of liquid entrapped under the bubble with contact angles of 20-10. This can be seen in Fig 4.10. Fig 4.11 shows large velocity vectors generated from the entrapped liquid. A magnified and sequential view of the entrapped liquid is given in Fig 4.12. The entrapped liquid evaporates with a high rate due to heat transfer from the bottom wall. This layer dries out at later stages of the bubble growth as shown in Fig 4.12. It was observed that the entrapped liquid evaporates faster as the wall superheat increases. The corresponding figures to those cases are not presented here.

Fi 4.10 Temperature contours (Re=100, ΔT=10 °C).

In addition to the thin layer, there is a small amount of liquid entrapped under the bubble with contact angles of 20-10. This can be seen in Fig 4.10. Fig 4.11 shows large velocity vectors generated from the entrapped liquid. A magnified and sequential view of the entrapped liquid is given in Fig 4.12. The entrapped liquid evaporates with a high rate due to heat transfer from the bottom wall. This layer dries out at later stages of the bubble growth as shown in Fig 4.12. It was observed that the entrapped liquid evaporates faster as the wall superheat increases. The corresponding figures to those cases are not presented here.
Formation of the thin liquid layer under the downstream portion of the bubble and evaporation of the entrapped liquid each makes a great contribution to evaporation and bubble growth. These two phenomena have not been observed in the bubbles with larger contact angles and it explains why the bubble with contact angles of 20-10 grows significantly faster than the other bubbles.

Fig 4.11 Velocity vectors (Re=100, ΔT=10 °C).
Fig 4.12 Temperature contours and velocity vectors (Re=100, ΔT=10 °C).
Effect of Thermal Properties Change by Using Nanofluids

Fig 4.13 shows the effect of nanofluid concentration on bubble growth rate. Bubble growth rate is up to 13% faster in the nanofluid with a higher concentration.

Al₂O₃ nanoparticles increase the thermal conductivity, viscosity and density but reduce the specific heat of the base fluid. Calculation of the Prandtl number based on properties in Table 2.1 shows 5% and 10% increase in Prandtl number, and also 3% and 4% increase in thermal diffusivity for nanofluid with 1 vol% and 2 vol% concentration, respectively. These changes in Prandtl number and thermal diffusivity represent enhancement in thermo-physical properties. This enhancement increases the rate of heat transfer from the walls.

On the other hand the viscosity of the nanofluid with 2 vol% is 27% higher than the pure water viscosity. Calculation of the Darcy friction factor shows that the friction factor of the nanofluid with concentration of 2 vol% concentration is 20% higher than that of pure water. This higher friction factor increases the pressure drop through the channel and requires higher pumping power.

Fig 4.14 shows the velocity vectors inside the channel. No significant change can be seen in the velocity pattern; however, the bubble has a larger volume in the nanofluid
with a higher concentration. Fig 4.15 shows the temperature contour through the bubble. There is also no significant change in the temperature contours.

Fig 4.14 Velocity vectors (ΔT=10 °C, Re=100, θ=40-30).

The bubble growth rate increase resulting from thermal properties enhancement is significantly less than the growth rate increase from contact angles reduction. This shows that the particle deposition plays a more important role in changing the bubble dynamics when concentration of nanofluid is less than 2 vol%. It should not be forgotten that higher concentrations of nanofluids in microchannels are not practical due to over deposition of the particles and blockage of the channel.
Instability of Bubble Interface

The wall superheats of 15 °C and 20 °C cause the bubble interface to experience instability. An example is represented in Fig 4.16 where the bubble experiences deformation at its upstream meniscus. This deformation appears when the bubble contact angles are 40-30 and 20-10. The bubble with contact angles of 60-50 experiences less deformation.
Fig 4.16 Distorted bubble interface ($\Delta T=15$ °C, $Re=250$, $\theta=20-10$).

Fig 4.17 shows the temperature contours along with the velocity vectors of the same bubble as shown in Fig 4.16. As the bubble grows, the thermal boundary layer at the upstream of the bubble rapidly becomes thicker due to the high wall superheat. This layer tends to diffuse in all directions but it is restricted by the bubble from the right side. On the other hand, the incoming fluid is pushing this layer from its left side. Hence, the only direction that this layer can diffuse would be upward. Since the bubble has low contact angles, a part of this layer is located under the bubble. Therefore, fast upward movement of this layer causes the bubble interface to deform as shown in Fig 4.17. In the case of 60-50 contact angles, the portion of the thermal boundary layer sitting under the bubble interface is small and its upward movement does not distort the interface as significantly as the cases with lower contact angles.
Fig 4.17 Temperature contours and uniform velocity vectors (ΔT=15 °C, Re=250, θ=20-10).

Reynolds number also plays a role in the instability of the bubble. The bubble experiences less interface deformation at the Reynolds of 50 and 100 while a significant deformation happens at the Reynolds of 250 as shown in Fig 4.18. At low Reynolds numbers, the thermal boundary layer under the bubble upstream meniscus is able to push the incoming liquid back and diffuse in the upstream direction. As the momentum of the incoming liquid increases, the thermal boundary layer moves upward instead of upstream which causes the bubble interface to deform.
Fig 4.18 Temperature contours and uniform velocity vectors at different Reynolds number (ΔT=15 °C, θ=20-10).

It is important to remember the fact that bubble behavior is also dependent on other parameters such as the channel size and heat conduction inside the solid part of the channel. By increasing the channel size, the gravitational force becomes important and bubbles may detach from the surface. Moreover, conduction heat transfer and fin effect of the side walls is important too. Temperature distribution in the microchannel can affect the bubble dynamics. Study of other influential parameters which are kept constant here will help to have a broader picture of flow boiling in microchannels.
Heat Transfer from Walls

Figs 4.19 - 4.27 compare the area averaged heat transfer from top, side and bottom walls as a function of time. The Nusselt number is obtained using equations 3.16 and 3.17. The heat transfer from the wall is very high initially as the liquid contacts the wall but it decreases as the thermal boundary layer develops over time. It can be seen that as the bubble grows and contacts the top and the side walls, the heat transfer drops due to poor heat transfer in the vapor phase.

Fig 4.19 - 4.21 show the effect of Reynolds number on the heat transfer from the top, side and bottom walls of the channel, respectively. In Fig 4.19, it can be seen that Reynolds number change has little effect on the heat transfer from the top wall. This is because the velocities associated with bubble growth are much larger compared to the incoming fluid velocities in the selected range of Reynolds number. Fig 4.19 also shows that at about 0.35 ms the top wall heat transfer rate starts to increase when Reynolds number is 50. This increase happens when the bubble grows and pushes back the thermal boundary of the top wall resulting in larger heat transfer. This trend starts earlier and faster for lower Reynolds number because the bubble grows faster and reaches the top wall earlier.

![Fig 4.19 Heat transfer at the top wall (θ=40-30, ΔT=10 °C).](image-url)
Fig 4.20 shows the effect of Reynolds number on the heat transfer at the side wall. The Nu number values are almost the same for all values of Reynolds number until 0.25 ms and then they start to deviate. It can be seen that after 0.25 ms the wall heat transfer rate decreases significantly for the cases of Reynolds of 50 and 100 but it starts to slightly increase and then decrease for the case of Reynolds of 250. This can be easily explained by the bubble shapes shown in Fig. 4.3. As discussed earlier, a bubble at higher Reynolds number contacts the side wall later and instead it pushes back the wall thermal boundary layer increasing the heat transfer rate. But when the Reynolds number is lower, the bubble grows more spherically and contacts the side wall earlier, decreasing the heat transfer from the wall due to poor convection heat transfer of through the vapor phase.

![Graph showing heat transfer at the side wall](image)

**Fig 4.20 Heat transfer at the side wall ($\theta=40-30$, $\Delta T=10 ^\circ C$).**

Fig 4.21 shows the effect of Reynolds number on the heat transfer at the bottom wall. Higher liquid velocities passing over the surface improve the heat transfer. Also, higher Reynolds number suppresses the bubble and causes slower bubble base expansion as discussed earlier. This allows the bottom wall to stay in contact with the liquid phase for a longer time. These two effects both improve the heat transfer from the bottom wall as shown in Fig 4.21.
Fig 4.21 Heat transfer at the bottom wall (θ=40-30, ΔT=10 °C).

Fig 4.22 shows the effect of wall superheat on the heat transfer at the top wall. The wall heat transfer decreases at the early stages of the bubble growth but as the bubble volume increases, the wall heat transfer rate increases too. This heat transfer increase is due to pressing of the already developed thermal layer toward the wall by the growing bubble. Therefore, the heat transfer improves as the bubble interface moves towards the top wall.

Fig 4.23 shows the effect of wall superheat on the heat transfer at the side wall. The heat transfer from the wall is improved as the wall superheat increases, due to faster bubble growth rate, but once the bubble contacts the wall the heat transfer rate starts to decrease drastically.

Fig 4.24 shows the effect of wall superheat on the heat transfer at the bottom wall. The value of the Nusselt number reduces faster as the wall superheat increases. At higher wall superheats bubble volume and the associated bubble base grow faster resulting in rapid reduction of the heat transfer.
Fig 4.25 shows the effect of bubble contact angles on the wall averaged Nusselt number at the top wall. The Nusselt number decreases as the thermal boundary layer thickens until the bubble growing on the opposite bottom wall prevents it from thickening further. As the bubble on the bottom wall elongates along the channel, it starts to push back the thermal boundary layer of the top wall, thus increasing the heat transfer. This increase in heat transfer starts at around 0.1 ms for the bubble with 20-10 degrees contact angles. As the contact angles of the bubble increase, this phenomenon happens later due to slower bubble growth rate as shown in Fig. 4.9.

Fig 4.26 shows the effect of bubble contact angles on the heat transfer at the side wall. The bubble with the lowest contact angles shows the highest heat transfer from the side wall due to rapid bubble growth. Also, the heat transfer at the side wall starts to decrease significantly as the vapor patch initiates for all the cases.
Fig. 4.27 shows the effect of bubble contact angles on the bottom wall heat transfer. Expansion of the bubble base area is dependent on the evaporation momentum forces and the surface tension forces. Decrease in contact angles increases the surface tension force causing the bubble base area to decrease. At 0.15 ms, decrease in contact angles from 60-50 to 40-30 degrees results in larger Nusselt number. Opposite to that effect, the Nusselt number reduces as the contact angles decreases from 40-30 to 20-10. In the case of 20-10 contact angles, the evaporation momentum forces are significantly larger than other cases due to formation of the thin liquid layer around the bubble base as discussed earlier. Large evaporation from this layer causes rapid expansion of the bubble base and decreases the heat transfer from the bottom wall.
These results show that the wall heat transfer in a single microchannel is mainly driven by the wall superheat and the bubble contact angles. It can be seen that growth of the bubble and expansion of the bubble base always result in heat transfer reduction at the bottom wall. At the side wall, the heat transfer increases as long as the bubble does not contact the wall. Once the vapor patch forms, the heat transfer from the side wall begins to drop. However, the heat transfer at the top wall improves as the growing bubble prevents thickening of the thermal boundary layer.

A change in the incoming liquid mass flux does not significantly affect the wall heat transfer. However, high mass flux suppresses the growing bubble and delays the vapor patch formation which improves the heat transfer from the walls. On the other hand, the velocities generated by the growing bubble are significantly larger than the incoming liquid velocity. The net effect shows no significant change in the heat transfer from the walls.
Increase in wall superheat results in faster bubble growth rate pushing back the thermal boundary layer toward the walls which causes an increase in heat transfer. However, a rapidly growing bubble forms a vapor patch earlier reducing the heat transfer and may lead to wall dry-out and CHF.

The bubble contact angles were found to be a significant parameter in changing the heat transfer. Low contact angles result in larger surface tension which keeps the bubble base small. This results in formation of a thin liquid layer around the bubble base that significantly improves the evaporation and increases the heat transfer. On the other hand, higher evaporation rate results in faster bubble base expansion which reduces the heat transfer. These two opposing effects result in the highest growth rate of the bubble with the contact angles of 20-10 degrees but also the highest heat transfer reduction rate.
Fig 4.26 Heat transfer at the side wall (Re=100, ΔT=10 °C).

Fig 4.27 Heat transfer at the bottom wall (Re=100, ΔT=10 °C).
Chapter 5: Conclusion

Understanding the bubble dynamics inside a microchannel is important in order to use flow boiling in microchannels as a practical solution for heat transfer. Collecting reliable data from experiments is a difficult task due to rapid nature of the flow boiling and the small scale of microchannels. Hence, numerical study is an appropriate tool to study the bubble dynamics and predict the bubble behavior during flow boiling in microchannels.

Bubble dynamics inside a microchannel has been numerically investigated in this study. The effect of Reynolds number, wall superheat, bubble contact angles and the use of nanofluids have been studied. The following trends have been found in this study:

1. Increase in Reynolds number causes the bubble to elongate more inside the channel for a specified bubble volume. Higher Reynolds number slightly decreases the bubble growth rate. In case of low wall superheats, large Reynolds number causes the bubble to slide along the channel. The wall heat transfer rate is found to slightly increase by increasing the Reynolds number.

2. Higher wall superheat significantly increases the bubble growth rate due to higher heat transfer from the walls. Bubbles have more elongated shapes at higher wall superheats for a specified bubble volume.

3. Low bubble contact angle causes the formation of a thin liquid layer between the bubble and the wall at downstream of the bubble which results in a significantly higher rate of evaporation. Low bubble contact angles result in higher heat transfer from the walls.

4. Higher nanofluid concentration slightly increases the bubble growth rate due to change in thermo-physical properties of the mixture.

5. The effect of bubble contact angle change on bubble dynamics is larger than the effect of thermal properties change for nanofluids. This indicates that the main contribution of nanofluids in boiling (at least in the selected range of vol% studied) is through the surface modification not through the thermal properties enhancement.

6. A bubble experiences instability at its interface when the wall superheat is 20 °C. This instability happens most when the bubble contact angle is 20-10 degrees and the Reynolds number is 250.
References


