

EXPERIMENTAL INVESTIGATION OF LAMINAR FLOW, POOL BOILING
HEAT TRANSFER, STABILITY, AND BUBBLE DYNAMICS IN NANOFLUIDS

by

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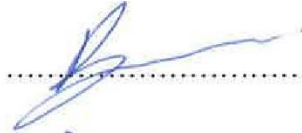
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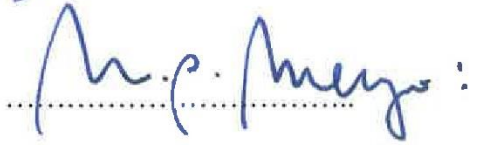
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ABSTRACT

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The advances in cooling microelectromechanical systems (MEMS) have led to extensive research efforts on flow and heat transfer in miniaturized channels during the last two decades. The need for high heat removal rates in many engineering devices used in microelectronics, energy industry, transportation, and optoelectronic industries such as plasmonics and thermophotovoltaic cells led to the development of smart engineering designs and/or efficient and high performance heat transfer fluids. An increase in heat transfer rate can be achieved by either optimizing the geometry and characteristic lengths or increasing heat transfer coefficients via enhancing thermophysical properties of fluids. Geometric optimization of engineering systems is limited to the production scale, when the system goes to miniaturization. Additionally, conventional coolants such as water, ethylene glycol, oil, refrigerants, and air result in limited heat transfer rates. Thus, suspending nanoparticles of average diameters of 1-100 nm with higher thermal conductivities in conventional liquid coolants was introduced which is called “nanofluids”. The term “nano” is related to nano-size particles, while the term “fluids” stands for base liquids which these nanoparticles are dispersed. Despite being one of the most studied subjects in heat transfer, many underlying mechanisms of nanofluids are still not fully elucidated due to its complex nature. In this thesis, pressure drop, single- and two-phase heat transfer of water-based nanofluids for different nanoparticles, concentrations, microtube hydraulic diameters, and heated lengths were investigated. Additionally, bubble dynamics in pool boiling on thin platinum wire, and flow in glass minichannels was studied. Finally, a novel method for increasing the stability of nanofluids was presented.

ÖZET

NANOAKIŞKAN'DA LAMİNAR AKIŞI, HAVUZ KAYNAMASI ISI TRANSFERİ, KARARLILIK, VE KABARCİK DİNAMİĞİNİN DENEYSEL İNCELENMESİ

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Mikroelektromekanik soğutma sistemlerindeki (MEMS) gelişmeler, son yirmi yılda minyatür kanallarda akış ve ısı transferi konusunda kapsamlı araştırmalara yol açmıştır. Mikroelektronik, enerji endüstrisi, ulaşım, plazmonikler ve termofotovoltaik hücreler gibi optoelektronik endüstrilerde kullanılan birçok mühendislik cihazında yüksek ısı giderme oranlarına duyulan ihtiyaç, akıllı mühendislik tasarımlarının ve/veya verimli ve yüksek performanslı ısı transfer akışkanlarının geliştirilmesine yol açmıştır. Isı transfer oranındaki bir artış, geometrinin ve karakteristik uzunlukların optimize edilmesiyle veya akışkanların termofiziksel özelliklerinin artırılması yoluyla, ısı transfer katsayılarının artırılması sağlanarak elde edilebilir. Mühendislik sistemlerinin geometrik optimizasyonu, sistem minyatür haline geldiğinde üretim ölçeğiyle sınırlıdır. Ek olarak, su, etilen glikol, yağ, soğutucu akışkanlar ve hava gibi geleneksel soğutucular, sınırlı ısı transfer hızlarına neden olur. Böylece, geleneksel sıvı soğutuculara daha yüksek termal iletkenlikleri ortalama çapı 1-100 nm olan nanopartiküllerin eklenmesi "nanoakışkan" olarak adlandırıldı. "Nano" terimi, nano boyutlu parçacıklarla ilgilidir; "akışkan" terimi ise, bu nanopartiküllerin dağılmış olduğu baz sıvılar anlamına gelmektedir. Isı transferinde en çok çalışılan konulardan biri olmasına rağmen, karmaşık yapısı nedeniyle nanoakışkanların altında yatan birçok mekanizma hala aydınlatılamamıştır. Bu çalışmada basınç düşüşü, farklı nanopartiküller, konsantrasyonlar, mikrotüp hidrolik çapları ve ısıtılmış uzunluklar için su bazlı nanoakışkanların tek- ve iki-fazlı ısı transferi araştırılmıştır. Buna ek olarak havuz kaynamasındaki ince platin tel üzerinde, ve cam minikanallarındaki akış içerisindeki kabarcık dinamiği incelenmiştir. Son olarak, nanoakışkanların stabilitesini arttırmak için yeni bir yöntem sunulmuştur.



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LIST OF SYMBOLS AND ABBREVIATIONS

Nomenclature

A	Area (m^2)
a	Advancing
ag	Asymmetrical growth
b	Bubble, Buoyancy
C_D	Drag coefficient (-)
C_p	Specific heat (kJ/kg.K)
c	Concentration, Cross section, Contact
cp	Contact pressure
D	Diameter (m)
e	Equilibrium
em	Evaporation momentum
ev	Evaporation
F	Force (N)
f	Fluid, Friction factor (-)
G	Mass flux ($\text{kg/m}^2.\text{s}$)
g	Gas, Gravitational acceleration (m/s^2)
h	Convective heat transfer coefficient ($\text{W/m}^2.\text{K}$)
h	Height (m), Hydraulic, Hydrodynamic
h_{fg}	Latent heat (J/kg)
HTC	Heat Transfer Coefficient ($\text{W/m}^2.\text{K}$)
I	Amperage (A)
i	Inertia, Interface
in	Inner
k	Thermal conductivity (W/m.K)
L	Length (m)
m	Mass (kg)
$\dot{m}$	Mass flow rate (kg/s)
Max	Maximum
Nu	Nusselt number
P	Electrical power (W)

Pr	Prantdl number
Q	Heat loss (W), Heat rate (W), Heat transfer rate (W)
q''	Heat flux (W/m ²)
$\dot{q}$	Volumetric heat generation rate (W/m ³)
R	Parameter (-), Resistance (°C/W.m ²)
Re	Reynolds number (-)
r	Radius (m)
T	Temperature (K)
t	Thickness (m), Time (s)
U	Uncertainty (%)
u	Flow velocity (m/s)
u,U	Velocity (m/s)
V	Voltage (V), Volume (m ³)
W	Width (m)
X	Uncertainty variable
x	Location (m)

Greek Symbols

ΔP	Pressure drop (Pa)
δ	Film thickness (m)
θ	Contact angle (kJ/kg)
μ	Viscosity (Pa.s)
ρ	Density (kg/m ³)
σ	Surface tension (N/m)
ϕ	Volume fraction ratio (%)

Subscripts

1-4	Thermocouple number
amb	Ambient
b	Boiling
bf	Base fluid
c	Contact, Cross section
dev	Developing
el	Electrical
ent	Entry

f	Fluid
h	Heated
i	Inner
in	Inlet
Ma	Marangoni
nf	Nanofluid
np	Nanoparticle
o	Outer
pl	Plane
qs	Quasi-steady
r	Receding, Reference
S	Sauter (diameter)
s	Shear, Sliding, Surface
sat	Saturation
ss	Stainless steel
sur	Surface
th	Thermocouple
theo	Theoretical
tot	Total
w	Wall

Superscripts

*	Dimensionless variable
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CHAPTER 1

1. INTRODUCTION AND OBJECTIVES

1.1. Introduction

The importance of cooling microelectromechanical systems (MEMS) has led to extensive research efforts on flow and heat transfer in miniaturized components and microchannels within the last two decades starting with the study of Tuckerman and Pease [1]. In general, to remove heat fluxes from a smaller area, air and conventional liquid coolants such as water, oil and refrigerant are used as the working fluids. Due to better thermophysical properties of liquids and higher attained heat transfer coefficients of liquids in comparison with air [2], liquids are preferred. Besides, heat and fluid flow in microscale is important in many applications such as microelectronic cooling of electronic chips, micro heat exchangers, bioengineering, human genome project, automobile industry, power generation systems, heating, air-conditioning, and chemical engineering. There are many parameters, which should be included when investigating convective heat transfer in microchannels such as size effect, rarefaction effect, surface roughness, axial heat conduction, surface geometry, and used coolants [3].

The need for high heat transfer rates in many engineering devices used in microelectronics, energy industry, transportation, and optoelectronic industries such as plasmonics [4] and thermophotovoltaic cells [5] led to the development of smart engineering designs and/or efficient and high performance heat transfer fluids. An increase in heat transfer rate can be achieved by either optimizing the geometry and characteristic lengths or increasing heat transfer coefficients via enhancing thermophysical properties of fluids. Geometric optimization of engineering systems is limited to the production scale, when the system goes to miniaturization. Additionally, conventional coolants such as water, ethylene glycol, oil, refrigerants, and air result in

limited heat transfer rates. Heat transfer performance of many industrial processes can be improved further by using new approaches, materials and fluids to eventually help the energy efficiency and eventual economic and environmental benefits. As a result of growing research efforts, enhancements in the performance are viable with enhanced micro-and nano structured surfaces [6–9] or working fluids with different surfactants or polymeric solutions [10,11].

Another possible method for modifying working fluids is implemented through dispersions of metallic and/or non-metallic, commonly metal-oxide, nanoparticles with higher thermal conductivities and size range of 1-100 nm, in conventional base fluids such as water, ethylene glycol, engine oil, propylene glycol, refrigerants and/or proper mixture of above-mentioned fluids [12]. Nanofluids have attracted great attention due to their potential in heat transfer enhancement, deposition-based applications and drug delivery. The former term “nano” is related to nano-size particles, while the latter term “fluids” stands for base liquids such as water, engine oil, ethanol. Despite being one of the most studied subjects in heat transfer, underlying mechanisms and physical principles of nanofluid boiling are still not well-known due to its complicated nature. Resulting nanofluids as alternative working fluids could significantly enhance heat transfer and thus have attracted much attention of many researchers around the world. Such fluids have found applications in thermal systems such as solar energy systems [13], heat exchangers [14], domestic refrigerators [15], heat pipes [16–18], thermosyphons [19,20] along with many other applications [21,22].

Nanoparticles are of great interest because of their outstanding properties afforded by their small size and high surface to volume ratio [23]. Addition of metallic (Cu, Al, Fe, Au, and Ag) or dielectric nanoparticles (Al_2O_3 , CuO, Fe_3O_4 , TiO_2 , CNT, SiO_2 and SiC) to a base fluid increases thermophysical properties of fluids such as thermal conductivity and enhances overall heat transfer capability [24,25]. Compared to micro/mini-particle-liquid suspensions for heat transfer enhancement, nanofluids have better performance due to more heat transfer surface between solid and liquid phases, high dispersion stability, reduced pumping power, reduced particle clogging, and adjustable thermal properties and surface wettabilities by varying particle concentrations [21]. Nanofluids constitute a new category of fluids consisting of uniformly dispersed and suspended nanometer-size particles in a base fluid.

Previous investigations indicate that nanoparticles alter thermophysical properties of base fluids and affect heat transfer [26–30]. One of the most promising properties of nanofluids is elevated thermal conductivity due to the fact that the thermal conductivity of nanoparticles is greater than that of standard base fluids [28,31]. Generally, the thermal conductivity of employed nanoparticles is higher than that of base fluids, such as water, ethanol and engine oil, and underlying mechanisms for enhancement in thermal conductivity of nanofluids can be attributed to layering at the solid/liquid interface, ballistic phonon transport, Brownian motion of the nanoparticles at the molecular and nano-scale level [32,33], agglomeration/deposition of nanoparticles [34,35]. Recently, a number of studies have been performed on the viscosity of nanofluids [31,36]. Accordingly, the viscosity of nanofluids increases with mass fraction and decreases with temperature [37,38].

1.2. Literature Review

1.2.1. Single-Phase Heat Transfer

There is a large number of studies in the literature related to convective heat transfer characteristics of nanofluids in mini/microchannels. However, many results are still contradictory to each other. Ho et al. [39] experimentally studied forced convective cooling performance of a copper microchannel heat sink, which consists of 25 parallel rectangular microchannels of a length of 50 mm and a cross-section of $283\ \mu\text{m} \times 800\ \mu\text{m}$, using Al_2O_3 /water nanofluid as the coolant. Their results showed that nanofluids have higher heat transfer coefficients compared to the pure coolant. Hwang et al. [40] experimentally investigated pressure drop and convective heat transfer coefficient of water-based Al_2O_3 nanofluids flowing through a uniformly heated circular tube of an inner diameter of 1.812 mm and a length of 2500 mm in the fully developed laminar flow regime. They discussed various effects such as thermal conductivities under static and dynamic conditions, energy transfer by nanoparticle dispersion, nanoparticle migration due to viscosity gradient, non-uniform shear rate, Brownian diffusion, and thermophoresis on the enhancement of the convective heat transfer coefficients with nanofluids, and they proposed that the flattening of the velocity profile is a possible

mechanism of the convective heat transfer enhancement, which cannot be explained by an increase in the thermal conductivity of nanofluids alone.

Selvakumar and Suresh [41] used CuO nanoparticle and water mixture with volume concentrations of 0.1% and 0.2% to investigate the effect of flow rate on the interface temperature, convective heat transfer coefficient, and pressure drop. According to their results, higher convective heat transfer coefficients were obtained with higher flow rate and nanoparticle fraction ratios. Singh et al. [42] investigated the effects of Al_2O_3 nanoparticle based nanofluid volume fraction and change in particle diameter and base fluids in 130, 211 and 301 μm hydraulic diameter microchannels. They found that nanofluids do not behave as a single phase liquid in microchannel and a non-uniform distribution of particles is present due to shear induced migration, which can be the main reason behind differential behavior of nanofluids. Kalteh et al. [43] investigated convective heat transfer of Al_2O_3 -water nanofluid flows inside a wide rectangular microchannel heat sink of 94.3 mm length, 28.1 mm width, and 580 μm height under laminar flow conditions. They found that two-phase Eulerian-Eulerian method is in better agreement with the experimental results when compared to the single phase homogenous approach. Two-phase results showed that nanofluids can be considered as a homogeneous mixture, and velocity and temperature profiles for two phase results are flatter than the homogeneous single phase model. Esmaeilzadeh et al. [44] experimentally studied hydrodynamics and heat transfer characteristics of $\gamma\text{-Al}_2\text{O}_3$ nanoparticles inside a circular tube for laminar flows and constant heat flux boundary conditions. They observed enhanced heat transfer coefficients with nanofluids compared to the base fluid.

Liu and Yu [45] experimentally investigated single-phase forced convection of nanofluids in a circular minichannel of 1.09 mm inner diameter. They found that convective heat transfer is enhanced with nanofluids for laminar flows, while it is deteriorated in the transition and the early stage of fully developed turbulent regions. Then, it gradually recovers after the fully developed turbulence is intensified. Murshed et al. [46] experimentally investigated forced convective heat transfer of nanofluids flowing through a cylindrical minichannel under laminar flow and constant wall heat flux conditions. They claimed that both convective heat transfer coefficient and Nusselt number increase considerably with the nanoparticle volume fraction as well as with Reynolds number.

Kumaresan et al. [47] studied heat transfer characteristics of CNT based nanofluids in a tubular heat exchanger of various lengths for cooling/heating applications. They claimed that, in contrast to the conventional heat transfer concept, the value of Nusselt number for the nanofluids increases with the decrease in Reynolds number, as the MWCNT concentration increases. The heat transfer performance of CNT based nanofluids flowing through a tube was investigated by Ding et al. [38], and significant heat transfer enhancements were observed. Meyer et al. [48] experimentally investigated convective heat transfer enhancement of aqueous suspensions of multi-walled carbon nanotubes flowing through a straight horizontal tube for a Reynolds number range of 1000–8000. They claimed that heat transfer was enhanced when comparing the data using Reynolds–Nusselt profile as reference. However, when comparing the data at the same velocity, it was shown that heat transfer was not enhanced, and at the end, they concluded that the tested carbon nanotube based nanofluids are not an ideal fluid for heat transfer enhancement.

Recently, Haghighi et al. [49] studied convective single-phase heat transfer performance of Al_2O_3 , TiO_2 and CeO_2 nanoparticle based nanofluids in a 500 μm diameter, 30 cm long test section under laminar flow conditions, and concluded that nanofluids show no advantage over pure water for laminar flow at equal pumping power. More recently, heat transfer behavior of water-based alumina, titania and carbon nanotube nanofluids in a straight pipe with constant heat flux at the wall was investigated by Utomo et al. [50]. Their results indicated that nanofluids enhance heat transfer by no more than 10% at a fixed mass flow rate or velocity.

Salman et al. [51] numerically investigated convective heat transfer using ethylene glycol (EG) as the base fluid for four different nanoparticles, Al_2O_3 , CuO , SiO_2 , and ZnO with different volume fractions, 1% to 4%, and different particle diameters, 25 to 85 nm, in a circular microtube. In their study, under laminar flow and under uniform heat flux boundary conditions, heat transfer enhancement was reported with nanoparticles in comparison with pure EG. Furthermore, they reported an increase in Nusselt number with Reynolds number for all the configurations.

From the mentioned studies, it can be seen that possible heat transfer enhancement and/or deterioration nanofluids is still a heated debate particularly for laminar flows. Moreover, many studies only show heat transfer characteristics, but Dynamic Light Scattering (DLS), Scanning Electron Microscopy (SEM), and zeta potential

characterization of nanofluids are discussed less. In addition, although there are many studies on nanofluids in the literature, convective heat transfer studies on nanofluids have not been conducted for thermally developing flow conditions. Thus, the objective of the section of this thesis related to the single-phase heat transfer is to characterize the nanofluids via their solid and hydraulic diameters, to investigate zeta potential for stability characterization, and to analyze the effect of nanoparticles on single phase heat transfer characteristics in a horizontal smooth hypodermic circular microtubes with inner and outer diameters of 502 and 717 μm at different Reynolds numbers ($200 < \text{Re} < 2500$) under thermally developing flow conditions. Water based nanofluids with TiO_2 and Al_2O_3 nanoparticles were prepared at different weight fraction ratios, 0.01-3 wt.%, and prepared nanofluids were tested in a simple open loop experimental setup. Pressure drops, friction factors, and single-phase heat transfer coefficients of the nanofluids were compared to the each other.

1.2.2. Pool Boiling Bubble Dynamics

Despite being one of the most investigated topics in heat transfer studies by virtue of its potential of removing large amount of heat, many principles and mechanisms of boiling phenomenon remain elusive due to its complexity. Nucleate boiling has crucial importance in many fields such as manufacturing, power plants, nuclear reactors, refrigeration and chemical processing [52–54]. However, limited fundamental studies have been performed on bubbles dynamics and their effects on boiling heat transfer. On the other hand, alternative solutions were sought to augment heat transfer such as changing the surface by deposition of materials [28] or coating nano/micro porous structures [6], adding nanoparticles to the base fluids [55], using pin fins [56], jet impingement [57], taking the advantage of secondary or turbulent flows [58].

Based on the boiling curve [59], the first regime of boiling is the natural convection regime, where bubbles do not form, although the wall temperature and the temperature of the hot liquid layer (fluid in contact with the wall) are both above the saturation temperature. When the heat flux or surface temperature is further increased, bubbles emerge from nucleation sites, and nucleate boiling commences. At this stage, the driving mechanisms of bubble dynamics strongly depend on the fluid bulk temperature.

Subcooled boiling occurs when the liquid temperature (distant from the boiling surface) is lower than the saturation temperature. This can induce temperature gradients and thermocapillary stresses on bubbles [60]. In saturated boiling, due to the lack of temperature gradients on bubble interface, Marangoni effects however have not been observed. These effects however become one of the major heat transfer mechanisms in subcooled boiling [61,62]. In general, a bubble attached to a heated surface is subject to evaporation from its base and condensation from its cap (if the bubble cap is exposed to the subcooled region). However, Marangoni convection originates from the non-condensable gases inside the bubble, which nonuniformly accumulate near the condensation surface [62]. Non-condensable gases hinder the condensation by up to 90 percent leading to increased saturation pressure gradient and temperature gradient at bubble interface so that Marangoni convection is induced along the bubble interface [62–66].

Nanoparticles within nanofluids can alter the heated surface characteristics leading to deposition of a thin layer of nanoparticles and influence the single-phase, flow and pool boiling heat transfer. According to many experimental studies, upon deposition of nanoparticles, a porous layer deposits on the surface and leads to a decrease in nucleation sites density [67,68]. Besides, the deposited layer introduces an extra thermal resistance. The aforementioned factors are found to be leading reasons behind pool boiling heat transfer deterioration. On the other hand, nanoparticles deposition, which can result in a decrease in contact angle, can cause an improvement in surface wettability and subsequent pool boiling critical heat flux (CHF) enhancement [69,70]. Another possible mechanism of this enhancement can be attributed to nanoparticle pinning effect at the contact line region owing to structural disjoining pressure, which is generated by ordered layering of nanoparticles in the evaporation meniscus contact line [31,71]. In addition, disjoining pressure enhances spreading and changes the force balance near contact line [72]. Furthermore, it is known that the deposition of nanoparticles and nanoparticle thin film coating affect surface roughness [67], wettability [68,73] as well as receding and advancing contact angles [73].

Unlike saturated boiling, the presence of temperature gradients along the bubble interface in subcooled boiling leads to Marangoni convection, which is a major heat transfer mechanism in this regime. Following natural convection regime, bubbles arise from the nucleation sites along with the augmentation of the heat flux. A bubble, which

is attached to a heated surface and exposed to subcooled fluid from its cap, is subjected to condensation from its cap and evaporation from its base. However, in the case, where non-condensable gases are present inside the bubble, they accumulate near the condensation surface, inhibit condensation leading a temperature gradient at the bubble interface and hence induce a Marangoni convection around the bubble [62].

Both enhancement and deterioration in boiling heat transfer with nanoparticles were reported due to the change in nucleation site density and surface roughness/morphology [55,69,70], while enhancement of critical heat flux (CHF) was achieved due to modified surface wettability, structural disjoining pressure and contact angle pinning [69,70]. Disjoining pressure, which has three components; namely molecular, electrostatic and structural, is one of the most important surface forces near the three-phase contact region and depends on the thickness of the thin film [74]. The structural disjoining phenomenon arises from ordered layering of nanoparticles inside the confined geometry of the wedge film forming between liquid drop or bubble and solid surface [75]. The reason for the pinning behavior is possibly structural disjoining pressure, which changes the force balance at the contact line.

On the other hand, dispersed nanoparticles in nanofluids change the surface characteristics due to deposition of nanoparticles on the surface and generate a thin layer, which affects boiling heat transfer. Deposition of nanoparticles may decrease the nucleation site density [67,68] and may introduce an extra thermal resistance via affecting the surface roughness [67], wettability [68,73], and receding and advancing contact angles [73]. The mechanisms of wetting in nanofluids are not yet fully understood because of additional particle-particle, particle-solid and particle-fluid interactions in comparison to pure fluids. Furthermore, both nanoparticle concentration and size of nanoparticles influence the contact angle [76]. A possible reason for the change in contact angle is due to solid-like layer ordering of nanoparticles in the contact line region [77,78]. Images related to bubble dynamics on wires in subcooled pool boiling were reported in the literature as sliding/sweeping of bubbles [61,79–86], leaping [64,87,88], oscillations [89,90], collision [79–81,83,85], bubble top-jet flow [84,91], circling [82,92]. However, the best knowledge of the authors, only few comprehensive and exhaustive studies have been performed so far [93,94]. The common point of these pioneering studies is the effect of the Marangoni convection generated by temperature gradients at the interface on heat transfer. Interactions among the bubbles and the nucleation sites as well as non-

condensable gases still remain an open question. Some of the interesting events observed in de-ionized (DI) water based pool boiling experiments related to bubble dynamics are sliding [66,79–81,83,93,94], oscillation [89,90,93], collision [66,79,80,93,94], leaping [64,87,88,93,94] circling [82,92] accompanied with bubble top jet flow [12,13,14,15]. However, only few investigations have been conducted with nanofluids [95]. The common point of these pioneering studies is that Marangoni convection plays a major role in subcooled pool boiling and augments heat transfer. However, more studies on the investigation of bubble-bubble interactions and nucleation site-bubble interactions are crucial to fully understand the Marangoni convection and its contribution to heat transfer.

The goal of the section of this thesis related to pool boiling bubble dynamics is to provide detailed information about bubble dynamics on a platinum wire and give insight about bubble-bubble interactions and nucleation site-bubble interactions as well as the role of nanoparticles in this bubble dynamics at different applied heat fluxes and bulk temperatures (between 30-50°C) using pure water, and TiO₂ nanoparticles/water nanofluids with two weight fractions (0.002% and 0.005%). Migration, collision, coalescence, leaping, oscillation and dancing of bubbles were presented and fundamentally discussed, with emphasis on physical mechanisms. Besides, the aim of this section is to investigate and explain the stick and oscillatory behavior in bubble dynamics, which has not been recorded in the literature on pure liquids, on a platinum wire with the use of nanoparticles at two different mass fractions and with nanoparticle deposition. Four cases, namely pure deionized water on both plain and TiO₂ nanoparticle deposited wires, TiO₂ nanoparticle/water nanofluids at two mass fractions of 0.002% and 0.005% on a plain wire, are compared by recording the bubble behavior images via a high speed camera.

1.2.3. Pool Boiling Heat Transfer

The physics and underlying mechanisms of boiling heat transfer remain not fully understood due to its intricate nature (despite the need for augmentation of the boiling heat transfer performance in the worldwide applications). Despite the wide range of boiling heat transfer studies found in the literature, bubble dynamics and its associated role in heat transfer, has not sufficiently considered, particularly in small scale. Transition from macro to micro scale by miniaturization offers new insights in boiling and stresses

the significance of comprehending the bubble dynamics in small scale emerges in conjunction with increasing attention to the development of high-tech miniature devices including micro components such as micro electro mechanical systems (MEMS), electronics cooling and micro energy components.

The effective parameters in nanofluid pool boiling heat transfer are nanoparticle material, average diameter, volume/mass fraction, heater's material, surface roughness, type, shape and base fluid. Different nanoparticles, such as Al_2O_3 [96–100], TiO_2 [101–103], SiO_2 [104–106], ZnO [105], ZrO_2 [107], Cu [96,108], CuO [109,110] and nanotubes such as carbon nanotubes (CNT) [111–114], have been widely used. Besides, Newtonian [115] and non-Newtonian [116] types of nanofluids were employed in experimental and numerical pool boiling heat transfer studies. There are also comprehensive reviews on nucleate pool boiling heat transfer and pool boiling critical heat flux using different nanoparticles, which reported both enhancement and deterioration depending on different important parameters such as heater size and shape, nanoparticle average diameter and materials [117], as well as nanofluid fraction ratio, roughness of the heated surface, size of cavity and nucleation sites [118].

Recently, Das and Bhaumik [119] have analytically developed a correlation to predict heat flux of pool boiling on a mechanically polished stainless steel plate for two different nanofluids, namely gamma- Al_2O_3 and TiO_2 nanoparticles dispersed in water, by considering temperature, volume fraction and nanoparticle parameters. This correlation was based on thermophysical properties of nanofluids such as heat capacity, density, viscosity and thermal conductivity calculated with well-known correlations. Brownian motion of nanoparticles, cluster/particle agglomeration and development of the liquid layer over the plate surface were neglected in their study, and their correlation was validated against the experimental data existing in the literature.

In another study, Naphon and Thongjirakul [120] experimentally investigated the effect of nanofluid concentration and pressure on pool boiling heat transfer on a cylindrical heater at different pressures of 50, 100 and 150 kPa. Nanofluids were prepared by dispersing TiO_2 nanoparticles in two different base fluids, R141b refrigerant and ethylene glycol, with volume fractions of between 0.01% and 0.075%. They concluded that both volume fraction and pressure played an important role in pool boiling heat transfer. The increase in volume fraction for nanofluids with both base fluids caused a deterioration in heat transfer compared to the pure base fluid.

In the study of Suriyawong and Wongwises [102], pool boiling heat transfer of TiO₂ nanoparticles dispersed in water was investigated for volume fractions of 0.00005, 0.0001, 0.0005, 0.005 and 0.01%. Copper and aluminum circular heaters with roughness of 0.2 and 4 μm were utilized as heaters. Enhancement in heat transfer was reported to only occur for nanofluids with the volume fraction of 0.0001 vol.% for the copper heater. For aluminum heater, deterioration in heat transfer was observed for all the concentrations. The decrease in roughness also had a positive effect on heat transfer. The effects of TiO₂ nanoparticles, which had an average diameter of 21 nm and were added to R141b refrigerant with volume concentrations of 0.01, 0.03 and 0.05%, and pressure (in the range of 200-500 kPa) on pool boiling heat transfer on a cylindrical copper heater, were examined by Trisaksri and Wongwises [103]. They found that an increase in volume fraction resulted in degradation in pool boiling heat transfer compared to the pure refrigerant, particularly at higher heat fluxes.

Sarafraz and Hormozi [110] experimentally investigated the effect of surfactants such as sodium dodecyl sulfate, SDS, sodium dodecylbenzene sulfonate, SDBS and Triton X-100 in CuO nanoparticles/water nanofluids on nucleate pool boiling heat transfer. Their results reported deterioration in pool boiling heat transfer in the absence of surfactants, whereas the addition of surfactants led to heat transfer enhancements. The mass fraction of copper oxide nanoparticles was in the range of 0.1% and 0.4%. The highest enhancement was obtained for the highest mass fraction ratio (i.e. 0.4%) and SDS as a surfactant (up to nearly 70%). The authors also have some other studies on different nanoparticles such as Al₂O₃ dispersed in ethylene glycol [98] or in water [121], carbon nanotube (CNT)/water nanofluids [114] and nanofluids with ZrO₂ nanoparticles in a mixture of water and ethylene glycol [107].

Hegde et al. [109] conducted experiments on pool boiling critical heat flux (CHF) of nanofluids with CuO nanoparticles at volume fractions from 0.01% to 0.5%. For all the concentrations, critical heat flux enhancements were above 60%, and the highest enhancement (about 130%) was pertinent to the optimum volume fraction of 0.2%.

Despite the fact that a large number of studies have focused on pool boiling heat transfer of nanofluids, there still exists a strong motivation to focus on this subject, particularly on nanofluids with CuO nanoparticles, on which only few investigations can be found, and on the effect of nanoparticle stability on pool boiling. In this regard, pool boiling heat transfer of TiO₂ nanoparticles/water and CuO nanoparticles/water nanofluids

was experimentally explored at different volume fractions over an aluminum flat plate heater. Visualization was carried out as well to observe the boiling images and to link it to the heat transfer performance. For CuO nanoparticles/water nanofluids, where the stability of nanoparticles was rather less, the surfaces were tested after the nanofluid experiments with the pure base fluid to shed light on heat transfer enhancement with CuO nanoparticles. As a result, the results were bolstered with both boiling images and repeated tests with the pure base fluid in the section of this thesis related to pool boiling heat transfer.

1.2.4. Bubble Dynamics in Minichannels

Despite many studies in the literature, physical mechanisms of boiling heat transfer have not been yet fully understood due to the complex nature. Among the two-phase heat transfer mechanisms and phase change phenomena, flow boiling is distinguished by its high heat dissipation characteristics. Indeed, the superiority of flow boiling systems over the single-phase heat transfer systems arises from the latent heat of phase change, in the development of miniaturized systems, understanding the underlying mechanisms as well as the effects of forces on bubble dynamics in small scales has cardinal importance on designing and developing novel high performance systems.

In flow boiling, the events of nucleation, growth, sliding and lift-off or departure of the bubbles occur. Each of these events should be investigated separately by taking the channel type (circular or rectangular channels) and orientation (vertical, horizontal and inclined) into account. Furthermore, bubble type and shape should be analyzed based on the acting forces, since small, confined and elongated bubbles exhibit different dynamics. Bubble nucleation can be categorized as quasi-homogeneous or of heterogeneous nucleation [122]. Once nucleation initiates, bubble undergoes a growth process starting with an inertia controlled step at the inception of the growth and continuing with a diffusion controlled step [123]. Some researchers focused on investigating the significant effects on bubble nucleation and growth [124–133]. Additionally, bubble contact diameter on the heated surface is another important factor in bubble dynamics and analysis of acting forces. An increase in the contact diameter causes an increase in the bubble departure diameter as a result of the change in the forces in both horizontal and vertical directions [126,128]. Bubble detachment which has a remarkable impact on heat

transfer, has become an appealing subject [125,126,128,130,131,134,135]. Ivey [136] presented different regions based on acting forces on the bubble, in the light of the experimental data and correlating the diameter and the frequency of the bubble: a) the hydrodynamic region, where the buoyancy and the drag forces are the major forces, b) transition region, where the surface tension, buoyancy and drag forces are comparable, and c) thermo-hydrodynamic region, in which thermal conditions during bubble growth are dominant.

Bubble sliding is one of the parameters, which enhances boiling heat transfer. The effect of bubble sliding process on heat transfer was analyzed by measuring the single bubble speed [137]. Kusuda et al. [138] proposed a theoretical model for predicting the heat transfer enhancement based on the temperature change, which occurred while bubble passed through a vertical rectangular channel. Later on, different studies on the prediction of heat transfer enhancement in a tube [139], on vertically up-flows and down-flows [140] and single vapor bubble on a downward facing heater surface [141,142] were performed. Different channels (circular and rectangular channels [125,137,143] or vertical wall [128]) and orientation (vertical [125,126,129,130,132,134,135], inclined, and horizontal [124,127,133,137,144]) were also considered in the literature. Furthermore, force analysis on a small bubble [125,126,129,132,137], a confined bubble [124,127,129,132,133] and an elongated bubble [127,129,132,144] is also present. Force analysis on liquid-vapor interfaces including major forces and scale effects on heat transfer in microchannels [145] as well as on bubbles nucleated from the nucleation site on horizontal, vertical and inclined plates [146] was performed in the literature. Klausner et al. [146] did a force balance for a growing bubble in a horizontal Pyrex glass square channel and proposed a model for predicting bubble detachment diameter. Zeng et al. [147,148] improved this by determining the inclination angle on dynamic basis, which was not required as an input. Later on, Mei and Klausner [149] focused on shear force acting on spherical bubbles. Following that, Klausner et al. [143] implied that the mean fluid velocity over the bubble determined the departure and lift-off diameter due to controlling the drag and wall superheat beneath the bubble.

Adding nano-size metallic and/or non-metallic particles with mean diameters of 1-100 nm, to conventional base fluids such as water, ethylene glycol, engine oil, refrigerants and/or proper mixtures of the above fluids, was introduced as nanofluids by Choi and Eastman [12], this approach and altered thermophysical properties of liquids.

Accordingly, nanoparticles might be enhanced heat transfer in both single-phase or boiling heat transfer. Different nanoparticles were utilized in flow boiling experiments, such as Al_2O_3 nanoparticles in water [150–153], CuO nanoparticles in R-113 [154,155], CuO and SiO_2 nanoparticles in R-134a [156] and Cu nanoparticles in water [157].

In the section of this thesis related to bubble dynamics in minichannels, a force analysis was performed on different size of bubbles, namely small, confined and elongated ones on plane surfaces and surfaces with TiO_2 nanoparticle deposition using various working fluids, namely pure water, pure ethanol, water-based and ethanol-based nanofluids with TiO_2 nanoparticles. An air bubble was injected inside a rectangular glass minichannel, which was electrically heated with tantalum a film heater. The bubble movement and temperature profiles along the channel were captured using high-speed and infrared (IR) cameras, respectively.

1.2.5. Subcooled Flow Boiling Heat Transfer

Flow boiling in small channels is becoming more and more popular in applications such as electronics cooling, fuel cell evaporators [158] and annular heat exchangers [159]. Based on the experimental studies in microchannels, four heat transfer mechanisms were reported for flow boiling: 1) nucleate boiling, where the heat transfer coefficient mainly depends on heat flux, 2) convective boiling, where the heat transfer coefficient is dependent on mass flux and vapor quality but not much on heat flux, 3) combined nucleate and convective boiling and 4) thin film evaporation of a thin liquid film, where elongated bubbles with a thin liquid film are observed [160].

Surface enhancement is one of the methods for enhancing heat transfer in flow boiling [161], which has led to the use of micro- and nano-structured surfaces [6]. As an alternative, the use of nanoparticles in conventional fluids is viable for enhancing both single-phase and flow boiling heat transfer in macro and micro scale. Various types of metallic and non-metallic nanoparticles were utilized in experimental studies [162] for enhancing thermal conductivity [163], yet nano-sized metal powders are dangerous to handle due to flammability. Thus, metal oxides can be regarded as suitable candidates for nanoparticles [164].

Various types of nanofluids were used in convective boiling heat transfer experiments, such as Al_2O_3 nanoparticles in water [150–153], CuO nanoparticle based R-113 refrigerant [154,155], CuO and SiO_2 nanoparticle based R-134a refrigerant [156] and Cu nanoparticles in water [157]. However, deposition and agglomeration of nanoparticles in boiling experiments might cause serious problems, which negatively affect cooling performances and might also cause clogging in microsystems leading to device failures. Subcooled flow boiling conditions could serve as a remedy for this problem since boiling incepts at bulk fluid temperatures lower than the saturation temperature [165], and thus, less evaporation effects inside the bulk fluid are expected, which may result in less deposition and agglomeration of nanoparticles and may increase stability.

While flow boiling studies mostly focused on saturated flow boiling [166,167] and on subcooled flow boiling [168,169], the literature on nanofluid flow boiling is rather scarce. Ahn et al. [153] made comparisons between pure water and alumina nanoparticle based nanofluids, with a volumetric fraction of 0.01vol.% and an average nanoparticle diameter of 47 nm. They concluded that flow boiling CHF could be increased by up to 40% at a nanofluid flow velocity of 4 m/s compared to pure water.

There are few experimental studies in the literature related to subcooled flow boiling in mini scale tubes, annuli and channels. Abedini and his coworkers performed their subcooled flow boiling experiments in a circular tube made of stainless steel having an inner diameter of 10 mm, a wall thickness of 1 mm and a test section length of 1 m. They utilized various nanoparticles, such as TiO_2 , Al_2O_3 , CuO, and ZnO [170–172] nanoparticles dispersed in water, with different nanoparticle volume concentrations of 0.01%, 0.01%, 0.5%, and 2.5% in horizontal and vertical tubes. According to their experimental results, mentioned nanofluids deteriorated subcooled boiling heat transfer at low concentrations, while increasing the concentration deteriorated heat transfer even more.

Kim et al. [151] performed their experiments in a test section made of stainless steel with an outer diameter 6.35 mm, a thickness of 0.41 mm, and a length of 100 mm, which was set up vertically. They used alumina, zinc oxide and diamond nanoparticles in water with volume fraction ratios less than 0.1%. They concluded that the heat transfer coefficients of nanofluids under subcooled flow boiling conditions were close to those of the base fluid (within $\pm 20\%$).

Few studies were present for vertical [173,174] or horizontal [175] annuli, too. Sarafriz and Hormozi [173] investigated subcooled flow boiling of CuO nanoparticle based nanofluids in a vertical annulus. According to their experimental results, CuO/water nanofluids with high mass fraction ratios caused a deterioration in heat transfer. Rana et al. [175] and Zangeneh et al. [174] used ZnO nanoparticles with different synthesis methods in their subcooled boiling experiments on vertical and horizontal annuli, respectively. Generally, they observed heat transfer enhancement with the mentioned nanoparticles.

Another type of heating, namely hot spot heating, which is rather different than our heating type, namely joule heating, was also studied for subcooled boiling studies with nanofluids. For example, Setoodeh et al. [176] and Soleimani et al. [177] employed this kind of heating in a horizontal tube and channel, respectively, for Al₂O₃/water nanofluids with different concentrations. In both studies, nanofluids enhanced subcooled flow boiling heat transfer for an optimum surface in roughness point of view.

There are also some comprehensive reviews on convective and pool boiling heat transfer enhancement by using nanofluids [178], single-phase and boiling heat transfer characteristics of nanofluids [179], pressure drop of nanofluids flows [180] and effect of nanoparticles deposition on boiling phenomenon [28]. In our previous study [181], single phase heat transfer characteristics of Al₂O₃ and TiO₂ nanoparticle based nanofluids in microtubes of an inner diameter of ~500 μ m was investigated. At low Reynolds numbers, convective heat transfer coefficient enhancement was negligible. For Reynolds numbers more than 1500, the effect of nanoparticles on heat transfer enhancement became visible.

The aim of the section of this thesis related to subcooled heat transfer is to provide insight into subcooled flow boiling of alumina nanoparticle based nanofluids, a popular, rather stable and widely used nanofluid type, in micro scale studies. The literature related to subcooled flow boiling in micro scale is already scarce for pure working fluids, while there are only few studies on subcooled flow boiling of nanofluids even in conventional scale. In this section, it is intended to address this lack of literature about micro scale subcooled flow boiling of nanofluids and to provide information about nanofluid stability under these conditions as well. Moreover, deposition and agglomeration of nanoparticles will less likely occur in subcooled flow boiling due to less fluid evaporation effects so that nanofluids could be repeatedly utilized along with a potentially better heat transfer performance. In the present section, experiments were performed on subcooled flow

boiling heat transfer by using mixture of spherical Al_2O_3 nanoparticles, which have an average diameter of 20 nm (with 0.05 wt.%, 0.2 wt.%, 0.5 wt.%, 1 wt.% and 1.5 wt.% mass fractions), and de-ionized water as the base fluid. The effects of mass flux and mass fraction on local heat transfer coefficients were discussed and heat transfer coefficients of nanofluids were compared to those of pure base fluid. The results on the stability of suspended nanoparticles and the deposition of nanoparticles were also presented along with their effects on boiling heat transfer.

1.2.6. Stability of Nanofluids

Nanofluids are considered to be the next-generation heat transfer fluids as they offer exciting possibilities to enhance heat transfer performance compared to pure liquids. As mentioned previously, conventional fluids such as water, ethylene glycol and oil have limited thermal properties. Increasing the thermophysical properties of classical fluid coolants by adding metallic or non-metallic solid nanoparticles (NPs), with an average size of 1-100 nm, to conventional fluids for better heat transfer performance was first introduced by Choi [12]. There are many advantages of nanofluids over classical fluids,[12] such as high specific surface area, high dispersion stability with predominant Brownian motion of nanoparticles, reduced pumping power to reach the same heat transfer intensification, reduced particle clogging, adjustable properties (thermal conductivity and wettability, by using different fraction ratios). It was observed that random movement (Brownian motion) of the nanoparticles [33,182] and thermophoresis [26,183] augment the energy transport process significantly when there is no bulk fluid motion. However, their limited durability and reusability constitute a challenge for their use in commercial product applications such as thermal management, drug delivery, and refrigeration. A gradual agglomeration of nanoparticles in nanofluids occurs with time and is accelerated by heating and boiling/evaporation of nanofluids due to the desorption and evaporation of the coating from the particle surface and the increase in volume fraction of the nanofluid and with their repetitive use in confined spaces [184–187]. As a result, both the stability and reusability of nanofluids decrease with time thereby shortening their shelf life. Prior art of the accordant subject involves the use of surfactants to avoid agglomeration and sedimentation of nanoparticles in nanofluids. Coatings such as the bilayer lauric acid coating generate a steric repulsion among particles, which keeps

them dispersed in the base fluid and contribute to the stability of nanofluids [188]. The forces responsible for particle-particle interactions are well known [189]. However, bubble-nanoparticle interactions are not clearly understood.

The cavitation phenomenon has a significant effect on jet flows in flow restrictive elements. The cavitation bubbles improve the efficiency of the microfluidic systems [190], enhance the combustion process [191] and increase the fragmentation ratio of the targeted areas [192] by generating dispersed and discretized spray structures. The pressure reduction at the element leads to cavitation bubbles with different sizes and distributions [193–195] depending on the fluid flow characteristics. Pioneering studies on hydrodynamic cavitation in micro scale revealed different characteristics of cavitating flow in the micro scale compared to cavitating flows in the macro scale [196–200]. The cavitation phenomenon also has a significant effect on the cone angle of the spray, penetration and spray breakup [201]. The sudden pressure drop and subsequent generation of cavitation bubbles result in better spray atomization and efficient spray primary and secondary breakup in the micro flow restrictive elements [202].

In the section of this thesis related to introducing a novel method for increasing the stability of the nanofluids, according to the importance of nanofluids in heat transfer applications, the stability and reusability of aggregated water-based Al_2O_3 nanofluids is investigated and an approach for eliminating the aggregation issue is proposed. Experiments were performed with a mixture of spherical 20 nm mean diameter nanoparticles in water as a base fluid. Three sets of scattering measurement results were obtained for the following conditions; prepared nanofluid, a heated sample representative of heat-related laboratory experiments samples [181,203] or industrial applications [13,14,16,204] and after the experiment with different pressures. By analyzing the dynamic light scattering measurement results, it can be concluded that by using spray jet and cavitation, producing bubbles and taking advantage of released energy from the collapse of bubbles at different pressures, agglomerated nanoparticles mixed in a base fluid can be deagglomerated and stabilized for near future use in laboratory experiments and finally utilizing them in industrial applications.

1.3. Research Objectives

The main goal of this study is to investigate the effect of nanoparticles on heat transfer coefficient and bubble dynamics. In order to achieve this goal:

- Single- and two-phase flow boiling heat transfer experiments were performed in the same experimental setup with different nanoparticles and concentrations.
- Pool boiling experimental setup was designed and effect of two different nanoparticles with various weight fraction ratios were investigated.
- Visualization of the bubbles and their behaviors in the presence of nanoparticles in both pool boiling on thin wire and in minichannel made of glass were performed and comprehensively were compared with bubble dynamics with pure base fluid.
- A novel method for increasing the stability and reusability of nanofluids was presented and feasibility experiments for a sample nanoparticle were performed.

In Chapters 2 and 3, details of employed experimental procedures and results will be given as:

- Convective heat transfer (Sections 2.1 and 3.1),
- Pool boiling bubble dynamics (Sections 2.2 and 3.2),
- Pool boiling heat transfer (Sections 2.3 and 3.3),
- Flow in minichannels bubble dynamics (Sections 2.4 and 3.4),
- Subcooled flow boiling heat transfer (Sections 2.1 and 3.5),
- Nanofluids preparation, characterization, and properties (Section 2.5), and
- Novel method for increasing the stability of nanofluids (Sections 2.6 and 3.6).

Finally, general conclusions will be given in Chapter 4 as well as contributions of this thesis in the literature (Section 4.2). The main parameters used in this thesis discussed in Chapter 3 can be summarized in Table 1.1.

Table 1.1: The summary of the main parameters employed in this thesis.

Section	Study	Nanoparticle NP	Diameter (nm)	Base Fluid	Fraction ratio (wt.%)
3.1	Single-phase Heat transfer	Al_2O_3	20	Water	0.01, 0.1, 0.5, 1, 2, 3
		TiO_2	10-30	Water	0.01, 0.1, 0.5, 1, 2
3.2	Pool boiling Bubble dynamics	TiO_2	10-30	Water	0.002, 0.005
3.3	Pool boiling Heat transfer	TiO_2	20-50	Water	0.001, 0.01, 0.05,
		CuO	10-20		0.075, 0.2 (for both)
3.4	Flow Bubble dynamics	TiO_2	10-30	Water	0.005
				Ethanol	0.005
3.5	Subcooled boiling Heat transfer	Al_2O_3	20	Water	0.05, 0.2, 0.5, 1, 1.5
3.6	Stability	Al_2O_3	20	Water	0.05

CHAPTER 2

2. EXPERIMENTAL PROCEDURE AND DATA REDUCTION

2.1. Convective and Subcooled Flow Boiling Heat Transfer

2.1.1. Single- and Two-Phase Experimental Setup

The schematic of the experimental setup (Figure 2.1(a) and Figure 2.1(c)) and test section (Figure 2.1(b) and Figure 2.1(d)) for convective and subcooled heat transfer experiments are displayed. It consists of a syringe pump (Cole-Parmer 200 Touch Screen Series, USA), a hypodermic stainless steel tube (Small Parts Inc., USA) as the test section with 502 μm and 717 μm inner and outer diameters, respectively, one or two pressure gauges (Omega Engineering Inc., USA) for measuring inlet and outlet pressures of the test section, a DC power supply (Xantrex Technology Inc., USA) for generating the desired heat flux with the integration of two alligator clips on the microtube (joule heating of the test section), 6 and 4 K-type thermocouples respectively for convective and subcooled studies (Omega Engineering Inc., USA), whose locations are displayed in Figure 2.1(b) and Figure 2.1(d) for measuring the outer surface wall temperatures of the microtube. Additionally, schematic of the tube and fluid in the test section region can be seen in Figure 2.1(e).

The experimental setup is an open loop system. The fluid was guided to a reservoir. Packing glands (Conax Technologies, USA), including gland body and sealant, were used for sealing the test section at the inlet. The inlet flow rate was controlled via the controller of the syringe pump. During experiments flow rate, voltage, current, wall temperatures, inlet and outlet pressures data were obtained under steady state conditions and were reduced to obtain mass flux, Reynolds number, pressure drop, friction factor, single- and two-phase heat transfer coefficients. The locations of thermocouples for

different experimental heated length cases are listed in Table 2.1, where x_{th} is the thermocouple location with respect to the beginning of the heated section and x_{th}^* is dimensionless position of thermocouple, the ratio of x_{th} to total heated length, L_h .

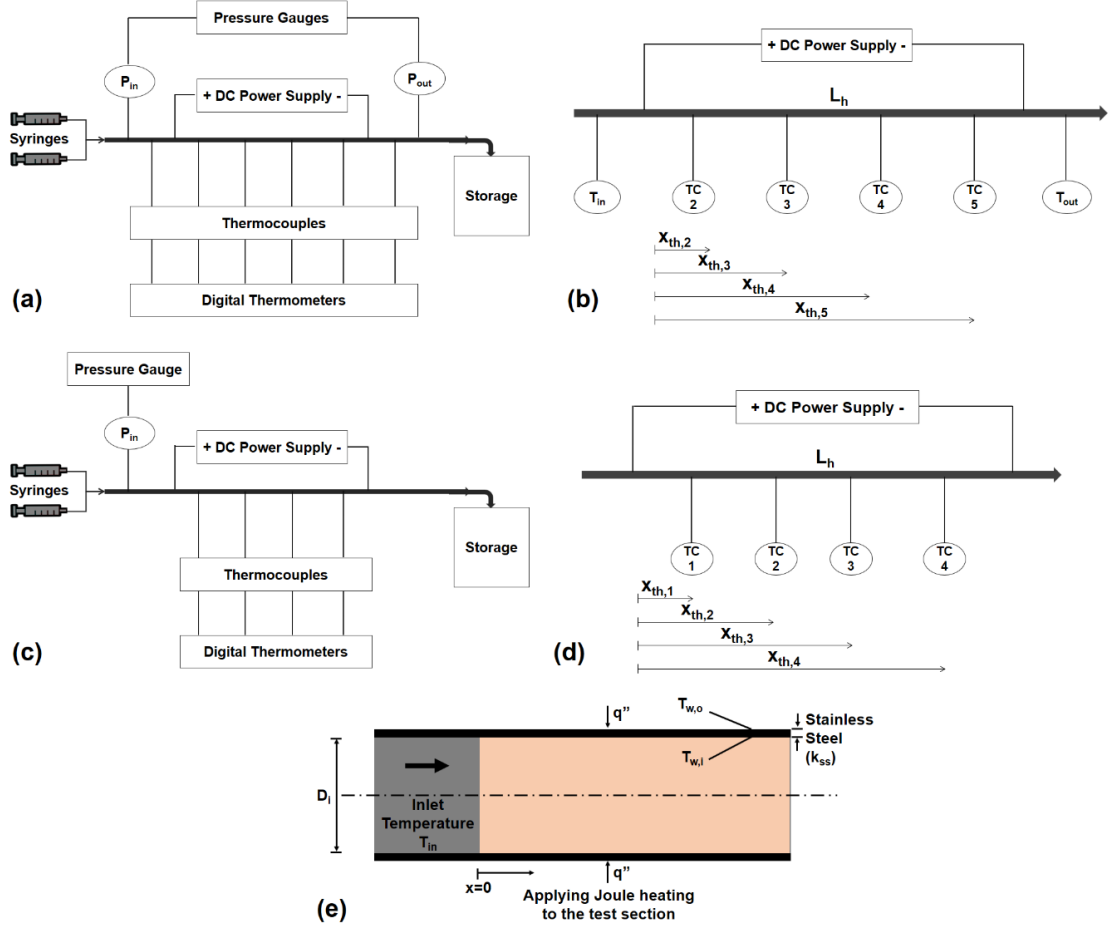


Figure 2.1: Schematic of: a) single-phase, c) subcooled flow boiling experimental setup, and test section of: b) single-phase, d) subcooled flow boiling. e) Sketch of the tube and fluid domain.

Table 2.1: Thermocouple locations on the test section for single-phase (SP) and two-phase (TP) heat transfers.

Case	$L_h [m]$	$x_{th,1} [m]$	$x_{th,2} [m]$	$x_{th,3} [m]$	$x_{th,4} [m]$	$x_{th,5} [m]$
SP	0.12	-	0.015	0.045	0.075	0.0105
TP1	0.07	0.015	0.03	0.045	0.065	-
TP2	0.12	0.02	0.05	0.08	0.115	-
Case	$L_h [m]$	$x_{th,1}^* [-]$	$x_{th,2}^* [-]$	$x_{th,3}^* [-]$	$x_{th,4}^* [-]$	$x_{th,5}^* [-]$
SP	0.12	-	0.125	0.375	0.625	0.875
TP1	0.07	0.21	0.43	0.64	0.93	-
TP2	0.12	0.17	0.42	0.67	0.96	-

2.1.2. Single- and Two-Phase Heat Transfer Data Reduction

Temperature and pressure data obtained from thermocouples and pressure sensors are used for deducing friction factor, single- and two-phase heat transfer coefficients, and Nusselt number.

Reynolds number is expressed as:

$$\text{Re}_{nf} = \frac{GD_i}{\mu_{nf}} \quad (2.1)$$

where D_i is the microtube inner diameter, μ_{nf} is the nanofluid viscosity, and G is the mass flux defined as:

$$G = \frac{\dot{m}}{A_c} \quad (2.2)$$

where $\dot{m}$ is the mass flow rate and A_c is the microtube cross-sectional area which is calculated as:

$$A_c = \frac{\pi D_i^2}{4} \quad (2.3)$$

The hydrodynamic performance of the system can be characterized by friction factor. Friction factor of the system is calculated as:

$$f = \frac{2\Delta P D_i}{L_{tot} \rho_{nf} u^2} \quad (2.4)$$

where f is friction factor, ΔP is pressure drop, D_i is microtube inlet diameter, L_{tot} is microtube total length, and u is flow velocity.

Assuming that the length of the tube is much larger than its thickness, 1D steady-state radial heat conduction analysis with constant properties and uniform volumetric heat generation is considered. Accordingly, the local inner surface temperature of the microchannel, $T_{w,i}$, is expressed in terms of the measured local outer surface temperature, $T_{w,o}$, as:

$$T_{w,i} = T_{w,o} + \frac{\dot{q}_{net}}{4k_{ss}} \left(\frac{D_o^2}{4} - \frac{D_i^2}{4} \right) - \frac{\dot{q}_{net}}{2k_{ss}} \frac{D_o^2}{4} \ln \left(\frac{D_o}{D_i} \right) \quad (2.5)$$

where $\dot{q}_{net}$ is volumetric net heat flux, k_{ss} is stainless steel thermal conductivity, and D_o is microtube outer diameter.

The volumetric heat generation rate is obtained using the net power and volume of heated stainless steel tube as:

$$\dot{q}_{net} = \frac{P_{el} - Q_{loss}}{\pi \left(\frac{D_o^2}{4} - \frac{D_i^2}{4} \right) L_h} \quad (2.6)$$

where P_{el} is the electrical power, Q_{loss} is the heat loss and L_h is the heated length.

The heat losses included in the calculations were determined from natural convective heat transfer analysis of air surrounding the microtube:

$$Q_{loss} = h_{air} A_s (T_{w,avg} - T_{amb}) \quad (2.7)$$

where A_s is the microtube outer surface area, $T_{w,avg}$ and T_{amb} are the average of wall temperatures and ambient temperature, respectively, and h_{air} is the natural convective heat transfer coefficient of the air, which can be calculated as [205]:

$$h_{air} = \frac{Nu_{air} k_{air}}{D_o} = \left(0.6 + \frac{0.387 Ra_{air}^{1/6}}{\left[1 + \left(\frac{0.599}{Pr_{air}} \right)^{9/16} \right]^{8/27}} \right)^2 \cdot \frac{k_{air}}{D_o} \quad (2.8)$$

where k_{air} is the thermal conductivity of the air, D_o is the microtube outer diameter, Pr_{air} is the Prantdl number of the air and Ra_{air} is the Rayleigh number defined as:

$$Ra_{air} = \frac{\beta g (T_{w,avg} - T_{amb}) D_i^3}{\alpha_{air}^2 Pr_{air}} \quad (2.9)$$

where β is the thermal expansion coefficient of the air, g is the gravitational acceleration, α_{air} is the thermal diffusivity of air.

Fluid temperature is calculated by using the following energy balance equation as:

$$T_{f,i} = T_{in} + \frac{P_{el} - Q_{loss}}{\dot{m} C_{p,nf}} \frac{x_{th}}{L_h} \quad (2.10)$$

where $T_{f,i}$ and T_{in} are fluid and inlet temperatures, respectively, P_{el} is electrical power, Q_{loss} is heat loss, $\dot{m}$ is mass flow rate, x_{th} is location of thermocouple, and x_{th} / L_h is the

dimensionless position. Since Reynolds numbers in this study were rather high, axial conduction effects were expected not to be significant.

With the wall temperature at the stainless steel tube and the fluid temperature, net heat power, and heated surface area, the single phase local heat transfer coefficient of nanofluid is calculated as:

$$h = \frac{P_{el} - Q_{loss}}{A_{sur,i} (T_{w,i} - T_{f,i})} \quad (2.11)$$

where h is heat transfer coefficient. Heated surface area, $A_{sur,i}$, is expressed as:

$$A_{sur,i} = \pi \cdot D_i \cdot L_h \quad (2.12)$$

For single-phase average heat transfer coefficient, $T_{w,i}$ in Eq. (2.11) is taken as the average inner wall temperature expressed in Eq. (2.5), and $T_{f,i}$ is taken as the average fluid temperature obtained from exit and inlet fluid temperatures. Thus mean average temperature difference $T_{w,i} - T_{f,i}$ was used for average heat transfer coefficient calculations [13]:

$$T_{w,i} - T_{f,i} = \frac{1}{4} (T_{w,2} + T_{w,3} + T_{w,4} + T_{w,5}) - \frac{1}{2} (T_{in} + T_{out}) \quad (2.13)$$

Finally, Nusselt number can be found as:

$$Nu = \frac{hD_i}{k_{nf}} \quad (2.14)$$

Under subcooled boiling conditions ($T_{w,i} > T_{sat}$), the obtained heat transfer coefficient corresponds to the two-phase heat transfer coefficient, while it is the single phase heat transfer coefficient under single phase conditions ($T_{w,i} < T_{sat}$).

The uncertainties listed in Table 2.2 and Table 2.3 for single- and two-phase heat transfer, respectively, were provided by the manufacturer's specification sheets and/or obtained using the propagation of uncertainty method included in the article of Kline and McClintock [46]. Accordingly, if an experimental parameter is expressed as a function of different variables:

$$R = R(x_1, x_2, \dots, x_n) \quad (2.15)$$

the uncertainty in the parameter is calculated as:

$$U_R = \left[\left(\frac{\partial R}{\partial x_1} U_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} U_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} U_n \right)^2 \right]^{1/2} \quad (2.16)$$

Table 2.2: Uncertainties in experimental parameters for single-phase heat transfer.

Uncertainty Parameter	Error
Electrical power (P_{el})	$\pm 0.32\%$
Flow rate	$\pm 0.4\%$
Friction factor (f)	$\pm 1.4\%$
Inner diameter (D_i)	$\pm 2\mu\text{m}$
Nusselt number (Nu)	$\pm 3.5\%$
Outer diameter (D_o)	$\pm 1\mu\text{m}$
Pressure drop (ΔP)	$\pm 0.3\%$
Fluid Temperature ($T_{f,i}$)	$\pm 0.1^\circ\text{C}$
Heat transfer coefficient (h)	$\pm 3.2\%$

Table 2.3: Uncertainties in experimental parameters for two-phase heat transfer.

Uncertainty Parameter	Uncertainty
Electrical power (P_{el})	$\pm 0.76\%$
Flow rate	$\pm 0.35\%$
Friction factor (f)	$\pm 1.22\%$
Pressure (P)	$\pm 0.25\%$
Heat transfer coefficient (h)	$\pm 3.29\%$

2.1.3. Nanofluids Properties

Thermophysical properties of nanofluids such as density, viscosity, and heat capacity were calculated from the existing expressions, which include the properties of base fluid and nanoparticles, as follows:

Nanofluid density [207]:

$$\rho_{nf} = (1 - \varphi)\rho_{bf} + \varphi\rho_{np} \quad (2.17)$$

where φ is the weight fraction and ρ_{nf} , ρ_{bf} and ρ_{np} are the nanofluid, base fluid and nanoparticle densities, respectively. (According to the data obtained from NIST [208], the density of pure gamma-alumina phase is $\rho_{np} = 3.965\text{g.cm}^{-3}$.)

Nanofluid viscosity [209]:

$$\mu_{nf} = \mu_{bf}(1 + 2.5\phi) \quad (2.18)$$

where μ_{nf} and μ_{bf} are the nanofluid and base fluid viscosities, respectively.

Nanofluid specific heat capacity [210]:

$$C_{p,nf} = (1 - \phi)C_{p,bf} + \phi C_{p,np} \quad (2.19)$$

where $C_{p,nf}$, $C_{p,bf}$ and $C_{p,np}$ are the specific heat capacities of nanofluid, base fluid and nanoparticles, respectively. (The gamma-alumina nanoparticles specific heat capacity is taken as $C_{p,np} = 0.795 \text{ kJ/kg.K}$ [211].)

Nanofluid thermal conductivity [212]:

$$k_{nf} = \frac{2k_{bf} + k_{np} + 2\phi(k_{np} - k_{bf})}{2k_{bf} + k_{np} - \phi(k_{np} - k_{bf})} k_{bf} \quad (2.20)$$

where k_{nf} , k_{bf} and k_{np} are thermal conductivity of nanofluid, base fluid, and nanoparticles, respectively.

2.2. Pool Boiling Bubble Dynamics

The experimental tests related to pool boiling bubble dynamics were conducted using a pool boiling facility, which consists of a boiling chamber, a platinum wire, a cartridge heater, two thermocouples, a DC power system and a high speed visualization system (Figure 2.2). The boiling chamber includes a cylindrical pressure vessel of stainless steel with transparent glass windows on front and back and a valve at the top for charging the system with working fluid. A platinum wire having a diameter and length of 250 μm and 80 mm, respectively, is placed into the chamber. A cylindrical cartridge heater attached to a thermostatic control system as well as a thermocouple at the top left of the chamber is embedded at the bottom right of the chamber for preheating the working fluid. Another thermocouple is connected to a thermometer to check the fluid bulk temperature. Heating for the wire is provided using a DC power supply connected to the ends of the wire. A high-speed camera (Phantom V4) with a frame rate of 1000 fps and resolution of 512×512 pixels is utilized to record boiling images.

The boiling chamber is filled with nanofluid to a level about 2.5 cm above the wire and is heated up by setting the thermostatic controller to the desired fluid bulk

temperature. Bulk temperatures are chosen as 30, 35, 40, 45 and 50 °C in this study. By altering the voltage and current, heat flux is applied to the wire. The backlight is adjusted for visualization and the focus was on the bubbles with the use of the high speed camera and appropriate lenses. Image post processing is done utilizing both PCC 2.7 and ImageJ software. Bubble average speeds are measured with the PCC 2.7 software based on the distance covered by a bubble within 30 ms. All the experiments are conducted at atmospheric pressure.

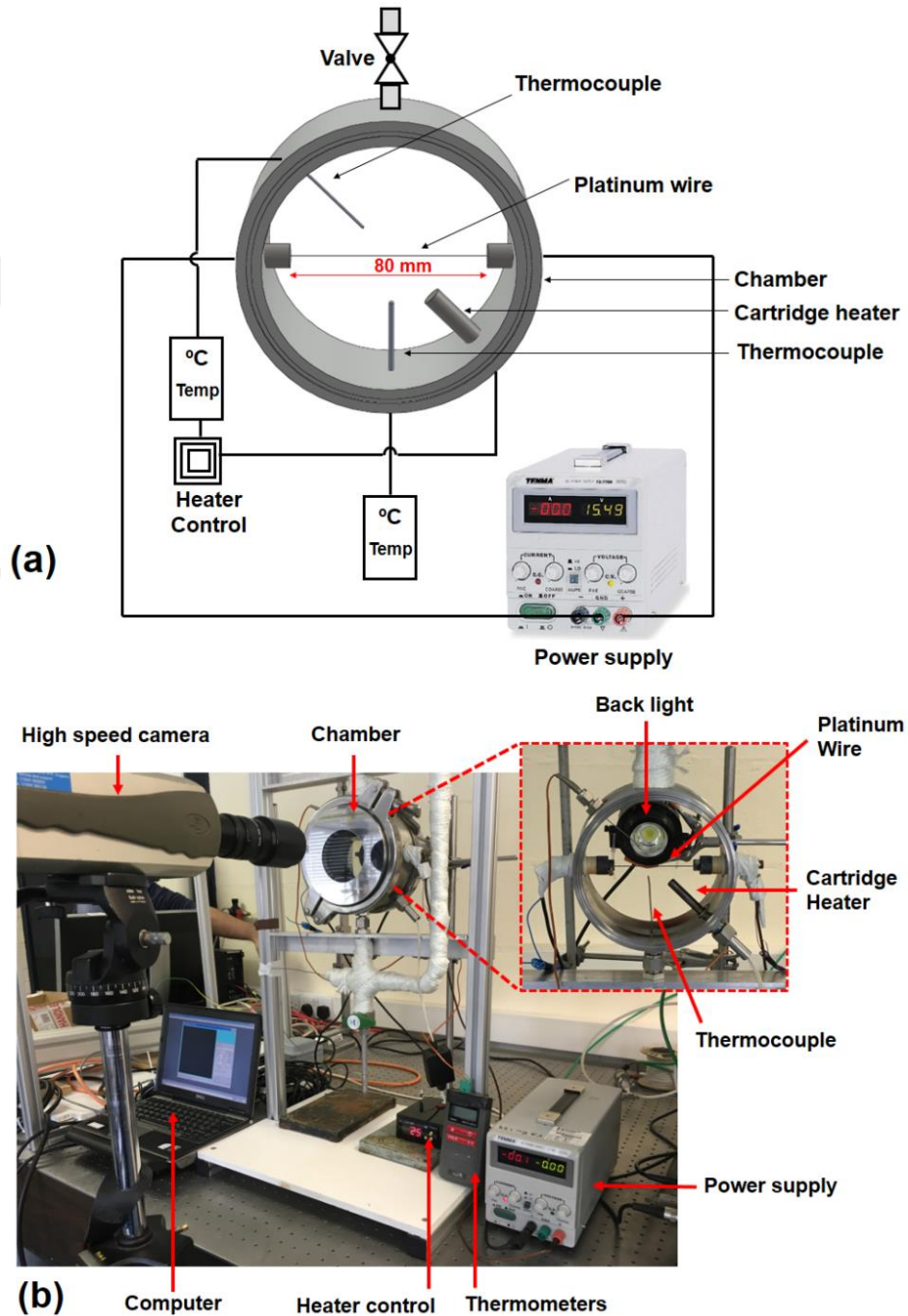


Figure 2.2: a) Schematic, and b) an image of the pool boiling bubble dynamics setup.

2.3. Pool Boiling Heat Transfer

2.3.1. Pool Boiling Heat Transfer Experimental Setup

The pool boiling experimental setup is schematically displayed in Figure 2.3. It has a glass pool of 6 mm thickness having dimensions of $50 \times 50 \times 50 \text{ mm}^3$, which was sandwiched with an aluminum plate heater with dimensions of $62 \times 62 \times 25 \text{ mm}^3$ through two holder plates. The heater has four holes at the bottom part of the heater (2 cm below its surface), where the cartridge heaters (Istanbul Resistance Co., Turkey) were press-fit and high quality conductive grease was used for filling the air gap around the heater. The gasket sealing elements were used between the glass pool and the holder plates to prevent leakage and heat dissipation. The heater plate has also three small holes for inserting K-type thermocouples (Omega Engineering Inc., USA) in order to measure the surface temperature at the close proximity of the heater surface (0.8 mm below the surface), which is in contact with the working fluid. A T-type thermocouple (Omega Engineering Inc., USA) was also inserted to the fluid to measure the saturation temperature of the nanofluids. A reflux condenser (IsoLab GmbH, Germany), with inner and outer glass tubes diameters of 22 and 40 mm, respectively, was connected to the pool in order to keep the nanofluids at nearly same amount through condensing escaped vapor from the pool. The amount of water and nanofluids measured before and after the experiments confirmed the proper functionality of the condenser. The condenser was open to atmosphere to maintain atmospheric conditions for the experiments and was connected to a tap to change the water inside for proper condensation operation. The cartridge heaters provided heat to the heater plate through a power supply. The readings from the thermocouples and the power supply were used to calculate heat transfer coefficients (HTC) corresponding to nanofluid samples. The heat flux was supplied by connecting the cartridge heaters to a DC power supply and tuning the voltage and current values. Experiments on nanofluid samples were carried out for several times to make sure about the repeatability of experiments. The average values were taken, while the deviations from the average values were about $\pm 3.1\%$. The experiments were performed at heat fluxes below the critical heat flux under saturated conditions.

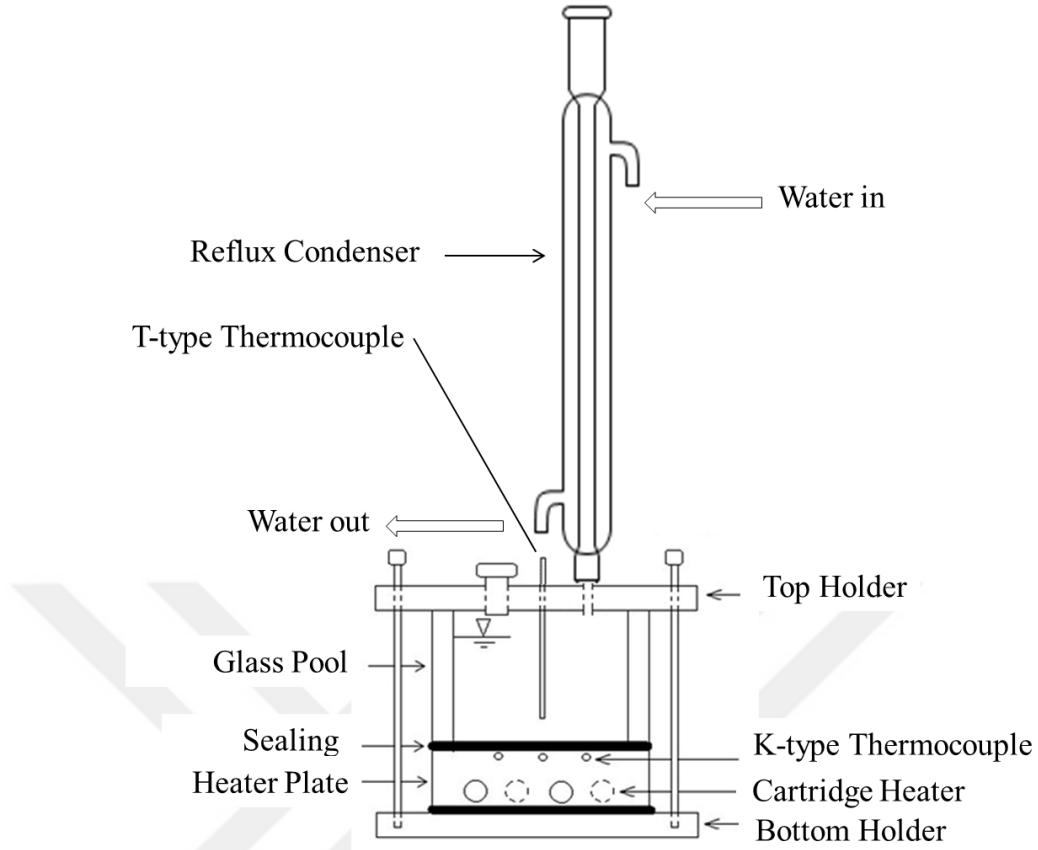


Figure 2.3: Schematic of the pool boiling heat transfer experimental setup.

2.3.2. Pool Boiling Heat Transfer Data Reduction

The recorded temperatures from the bulk and wall surface, voltage and current values of the DC power supply were used for calculating pool boiling heat transfer coefficients at different heat fluxes.

The net heat flux, q'' , is obtained as:

$$q'' = \frac{VI - Q_{loss}}{A} \quad (2.21)$$

where V and I are voltage and current readings of the power supply, respectively, A is heated surface area, Q_{loss} is heat loss obtained from natural convection analysis, which was between 2.5% and 5% relative the electrical power.

Nucleate pool boiling heat transfer coefficient is calculated as:

$$h = \frac{q''}{T_s - T_{sat}} \quad (2.22)$$

where T_{sat} is fluid saturation temperature and T_s is the surface temperature deduced from the average value from readings of the thermocouples, T_{th} , and thermal contact resistance, R_c :

$$T_s = T_{th} - q'' \cdot R_c \quad (2.23)$$

In the light of the uncertainty propagation method [213], the overall uncertainty analysis of the experimental result $U = U(X_1, X_2, \dots, X_N)$ is given as:

$$\delta U = \left(\sum_{i=1}^N \left(\frac{\partial U}{\partial X_i} \delta X_i \right)^2 \right)^{1/2} \quad (2.24)$$

where X_i is variable and δX_i is its corresponding uncertainty (given by the manufacturer's specification sheets of used equipment in the case of equipment use). Accordingly, the average uncertainties in electrical power and heat transfer coefficients were found to be $\pm 1.1\%$ and $\pm 2.7\%$, respectively.

2.4. Flow in Minichannels Bubble Dynamics

The experimental setup consists of a horizontal tantalum coated borosilicate glass rectangular minichannel, a high-speed camera, an infrared (IR) camera, a syringe pump, two integrated thermocouples and a DC power supply. The channel walls were deposited with a tantalum film heater with thickness of a couple of ten nm via Magneto-sputtering. The thickness of this film was optimized by balancing the transparency and application of heat flux. The glass channel which was horizontally positioned as shown in Figure 2.4, and has a width of $W_{in} = 4$ mm, height of $h_{in} = 2$ mm, and a wall thickness of $t_{wall} = 0.4$ mm. The length of the heated section is fixed to 7 cm. The working fluid was pumped through the minichannel at desired flow rates via a syringe pump. The inlet and outlet fluid temperatures were measured with K-type thermocouples, which were placed at the inlet and outlet of the minichannel. A high speed camera with a frame rate of 200 fps was installed above the minichannel along with the IR camera

(ThermoVision® 900). However, owing to the lack of enough space for both high speed and IR cameras, the IR camera was tilted to a few degrees relative to its vertical position in order to fix the mapped area of the outlet of the channel heated section. It is worth mentioning that while the whole channel heated length (7 cm) was within the visualization limits of the high speed camera, the IR camera covered 4 cm of the channel heated length (Figure 2.4) due to its limited focusing distance. Fiber optic back light was positioned under the test section for supporting visualization. A stainless steel needle with a diameter of 125 μm was inserted through the hole drilled of the connection to the channel. The hole was insulated by Dow Corning 795 silicon building sealant via curing for 24 hours. A single air bubble was introduced to the channel by utilizing the syringe, which was manually controlled.

The mass flux range was chosen to be between 2.1 $\text{g/m}^2.\text{s}$ to 20.8 $\text{g/m}^2.\text{s}$ for water and water-based nanofluids and between 1.6 $\text{kg/m}^2.\text{s}$ to 16.5 $\text{kg/m}^2.\text{s}$ for ethanol and ethanol-based nanofluids. The applied heat flux changed from 0 to 20 kW/m^2 . All the experiments were performed at the ambient inlet temperature and atmospheric pressure.

From boiling point of view, when nanofluids are utilized, nanoparticle deposition on channel walls becomes one of the leading parameters. Hence, in order to investigate the deposition effect on single air bubble dynamics, inner walls of a minichannel were deposited with TiO_2 nanoparticles via boiling the TiO_2 /water nanofluid at the mass fraction ratio of 0.005 wt.% inside the channel. Working fluids were chosen as pure DI water, pure ethanol, TiO_2 /DI water nanofluid with a fraction ratio of 0.005 wt.% and TiO_2 /ethanol nanofluid with a fraction ratio of 0.005 wt.%. The average uncertainties in mass flow rate, temperature, and bubble size are $\pm 0.4\%$, $\pm 0.1^\circ\text{C}$, and $\pm 10 \mu\text{m}$, respectively, according to the manufacturer's specifications.

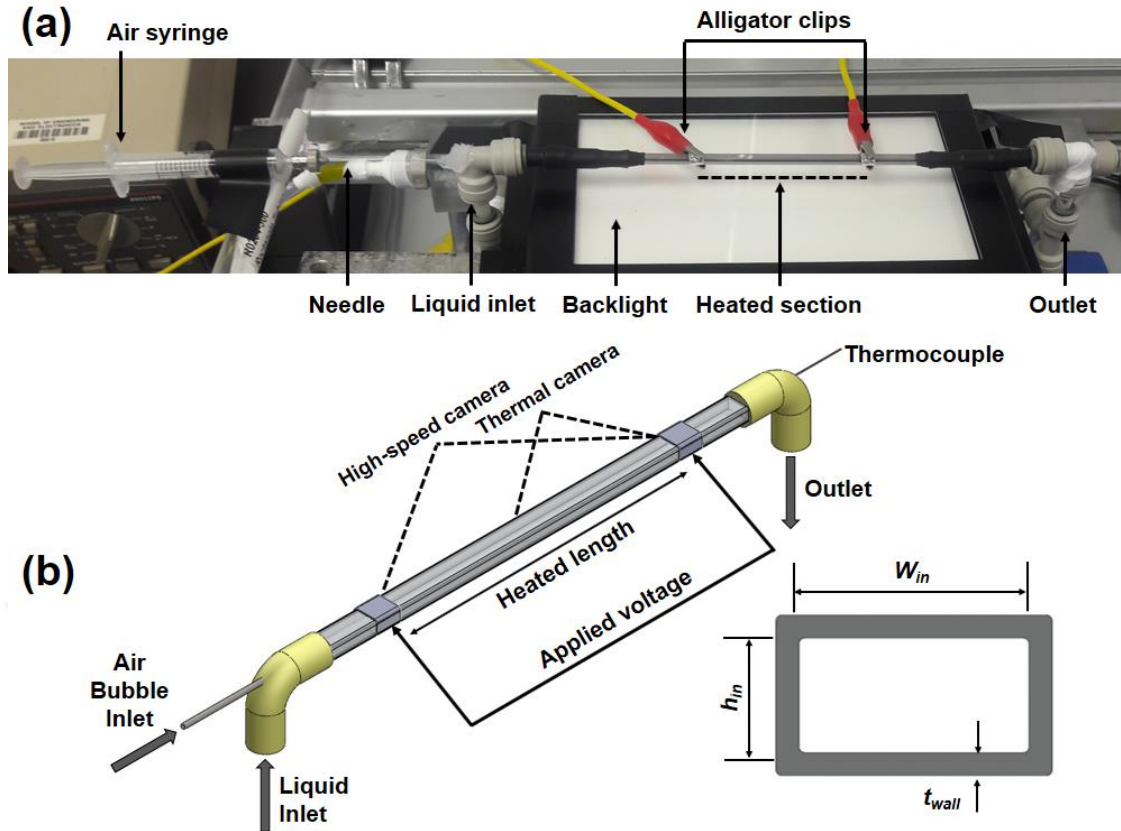


Figure 2.4: a) Photo of the real experimental setup and b) schematic diagram of the test section related to the bubble dynamics in minichannels.

2.5. Nanofluids Preparation, Characterization and Properties

2.5.1 Single- and Two-Phase Heat Transfer

In single-phase section of this study, spherical TiO_2 (rutile-Titania) and $\gamma\text{-Al}_2\text{O}_3$ (γ -Alumina) (Ionic Liquids Technologies, IoLiTec GmbH, Germany) nanoparticles were used, and deionized water was used as a base fluid. Nanofluids with different weight fraction ratios were prepared by performing first minimum half an hour sonication, and then having magnetic stirring conducted for at least for one hour. The stability and thermophysical properties of the nanofluid directly depend on the size, morphology, type, and physical properties of the nanoparticles being used. Ensuring the stability of the prepared nanofluid is vital for retaining the improved thermal properties of the nanofluids. Figure 2.5 shows the prepared nanofluids at different mass fraction ratios for single-phase heat transfer experiments. Prepared titania- and alumina-nanoparticles/water nanofluids

were coded as TiO₂ (I, II, III, IV, V, VI) and Al₂O₃ (I, II, III, IV, V, VI) for 0.01, 0.1, 0.5, 1, 2, and 3% mass fractions, respectively, as shown in Table 2.4.

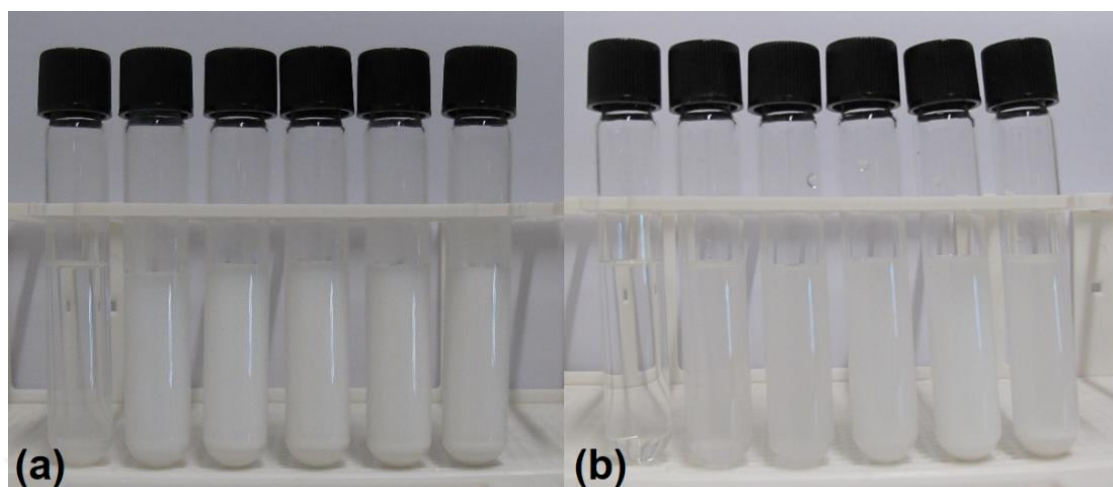


Figure 2.5: a) TiO₂-, and b) Al₂O₃-nanoparticles/water nanofluids, for 0.01, 0.1, 0.5, 1, 2, and 3%, weight fraction ratios each.

Table 2.4: Types, codes, and weight fraction ratios of prepared nanofluids.

NF Code	wt. %	NF Code	wt. %
TiO ₂ (I)	0.01	Al ₂ O ₃ (I)	0.01
TiO ₂ (II)	0.1	Al ₂ O ₃ (II)	0.1
TiO ₂ (III)	0.5	Al ₂ O ₃ (III)	0.5
TiO ₂ (IV)	1	Al ₂ O ₃ (IV)	1
TiO ₂ (V)	2	Al ₂ O ₃ (V)	2
TiO ₂ (VI)	3		

Figure 2.6 shows the SEM images of the TiO₂ and Al₂O₃ nanoparticles. As understood from SEM images, particles solid diameters are around 10-30 nm. All SEM images of nanoparticles indicate a spherical shape of primary particles.

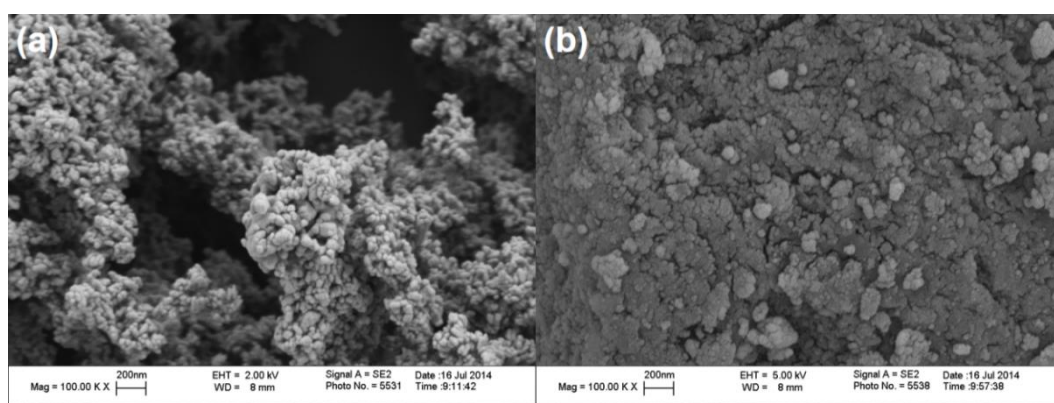


Figure 2.6: SEM images of: (a) TiO₂ nanoparticles, and (b) Al₂O₃ nanoparticles.

In addition to particle solid diameters, TiO_2 and Al_2O_3 nanofluids hydrodynamic diameters were also characterized via DLS measurements (Malvern ZS Zetasizer, Malvern Instruments Ltd., UK). It is understood from the DLS measurements that hydraulic diameters vary between 225-250 nm. Dry nanoparticle is on average 20 nm in diameter, and the average hydrodynamic particle size is around 10 times bigger than dry phase, namely around 225-250 nm. Zeta potential was used to characterize the stability of prepared nanofluids. Nanofluids having an absolute value of zeta potential more than 60 mV are considered as highly stable, while nanofluids having absolute zeta potential values down to 30 mV are physically stable, and nanofluids having absolute zeta potential values less than 5 mV are not considered as stable [214]. Zeta potentials of nanofluids are shown in Figure 2.7. The same samples were measured for 3 times to control the accuracy of the results. It can be stated from the figures that zeta potentials of the samples are almost identical for 3 measurements, and prepared nanofluids are mainly stable. Prepared nanofluids were also observed just after the experiments, and it was seen that they preserved their stability. Two absolute zeta potential profiles for Al_2O_3 and TiO_2 nanoparticle based nanofluids at a weight fraction of 0.5 wt.%, with peak values of 35 mV and 42 mV are shown as samples in Figure 2.7, respectively.

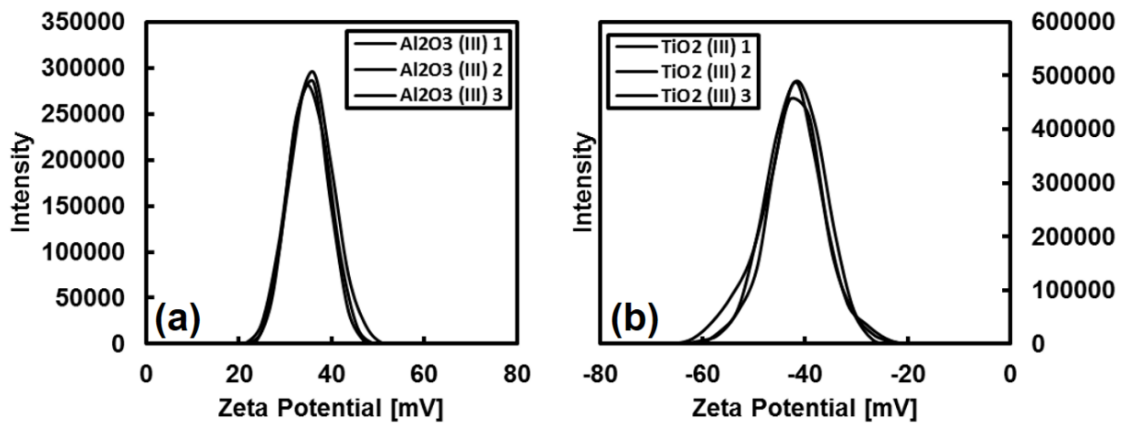


Figure 2.7: Zeta potential measurements of: a) 0.5 wt.% Al_2O_3 , and b) 0.5 wt.% TiO_2 . (3 times measurements were done for each figure.)

Another Scanning Electron Microscopy (SEM) Secondary Electron (SE) image of Al_2O_3 nanoparticles can be seen in Figure 2.8. Accordingly, particle solid diameters are around 10-30 nm, and the particles have a spherical shape. Besides the particle solid diameter, hydrodynamic diameter of Al_2O_3 nanoparticles in the base fluid was measured using the Dynamic Light Scattering (DLS) apparatus (Malvern ZS Zetasizer, Malvern

Instruments Ltd., UK). Figure 2.8(a) shows typical DLS measurement results, which were performed for three times for samples with a mass fraction of 0.5 wt.%. According to the DLS measurements results, the peak size or distribution size of alumina nanoparticles in mixture is nearly 250 nm. The same method was used for other mass fractions, namely 0.05 wt.%, 0.2 wt.%, 1 wt.% and 1.5 wt.%, and average of measurements for each concentration can be seen in Figure 2.8(b). Distribution sizes for all of the nanofluids are approximately the same.

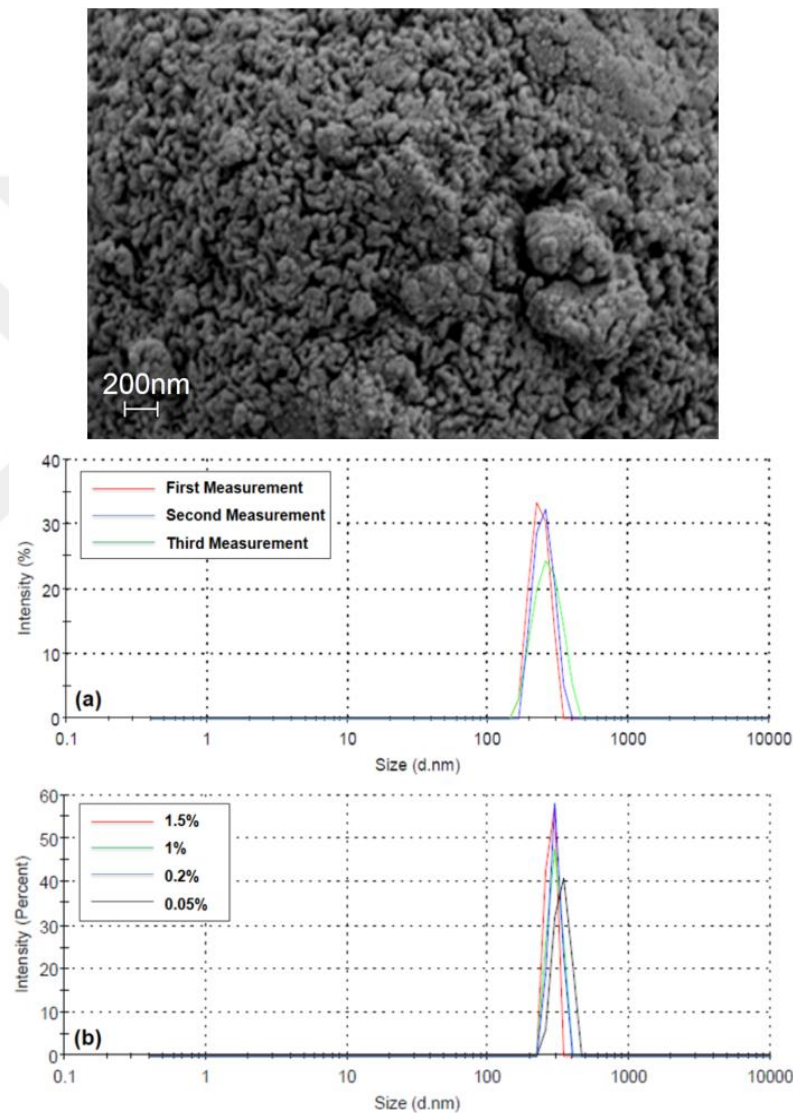


Figure 2.8: SEM-SE image of Al_2O_3 nanoparticles (upper). DLS measurement results of Al_2O_3 /water nanofluid samples: a) three measurements at the mass fraction of 0.5 wt.%, and b) average of three measurements at other fraction ratios.

2.5.2. Pool Boiling Bubble Dynamics

In nanofluid preparation of pool boiling bubble dynamics section, spherical rutile TiO_2 nanoparticles were purchased in powder form (Ionic Liquids Technologies, io-li-tec GmbH, Germany) and have an average diameter of 10-30 nm, they were mixed with DI water. To achieve the desired weight fraction of the nanofluids (0.002 and 0.005 wt.% in this study), nanoparticles with the required mass were dispersed in a specific amount of DI water. DI water was degassed for one hour prior to the nanofluid preparation. Accordingly, the liquid was added in the degassing glass chamber and was heated by the heating coil. In order to avoid any agglomeration and to provide a homogenous dispersion, a two-step procedure was implemented. In the first step, the mixture of degassed DI water and nanoparticles was exposed to sonication (Fisher Scientific Ltd., cat. no. FB15047) for one hour. In the second step, a magnetic stirrer was used to mix the nanoparticles in degassed DI water for another hour. Additionally, one of the options of the sonicator is “degas” mode, which was also used during the first step of nanofluid preparation. No additives such as surfactants were used in this study.

To analysis nanoparticles size distribution, scanning electron microscopy (SEM) image of the nanoparticles was taken before dispersing in water, Figure 2.9(a). Furthermore, the hydrodynamic diameter of the dispersed nanoparticles was measured with a dynamic light scattering (DLS) equipment (Nanosizer 90PLUS/BI-MASS, Multi angle particle sizing option, Brookhaven Instruments Corporation, USA), Figure 2.9(b). The hydrodynamic diameter of nanoparticles is typically bigger than the solid diameter due to existence of nano-layer around the solid particles and some agglomeration due to the employment of the DLS technique. The average of the measurements (out of three measurements) is plotted in Figure 2.9(b). The peaks of the DLS measurements are at 214 nm and 117 nm for nanofluids with weight fraction ratios of 0.002% and 0.005%, respectively.

The experiments related to pool boiling bubble dynamics were conducted at atmospheric pressure by: i) filling the pool boiling chamber from the upper part with working fluids, namely pure deionized water or two different nanofluids, up to a level of 2.5 cm above the wire, ii) adjusting the bulk temperature to the desired temperature starting from 30°C with intervals of 5°C up to 50°C, iii) applying heat flux to the platinum wire by adjusting the voltage and amperage values of the DC power supply, and iv) recording the videos utilizing a Phantom V4 high speed camera with the resolution of

512×512 and frame rate of 1000 fps. Afterwards, the captured images were analyzed by using the PCC 2.7 software. The figures displaying the results were prepared by the ImageJ software.

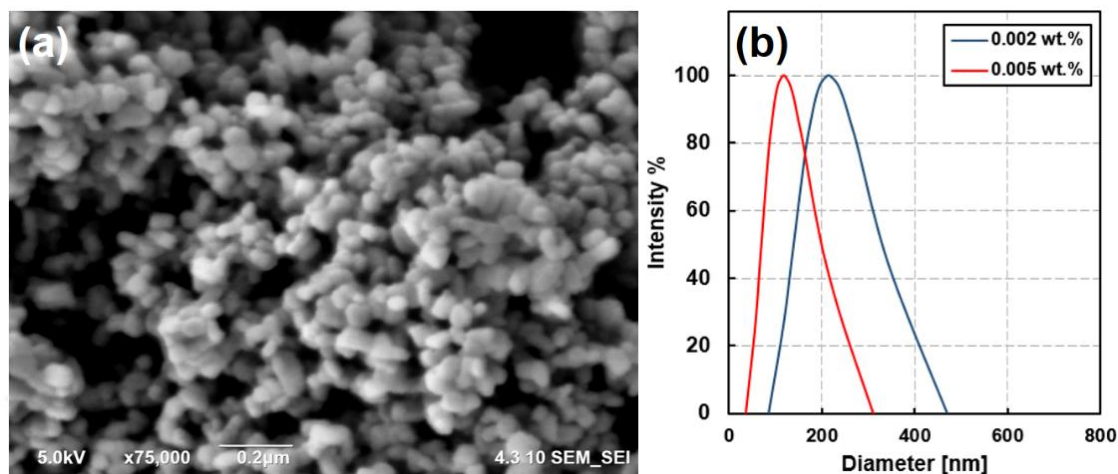


Figure 2.9: a) Scanning electron microscopy (SEM) image of TiO₂ nanoparticles, and b) dynamic light scattering (DLS) measurements of TiO₂/water nanofluids with mass fraction ratios of 0.002% and 0.005%.

Scanning electron microscopy image energy-dispersive X-ray spectroscopy (EDX) measurements of the used platinum wire before and after the experiments (TiO₂ nanoparticles deposition) are shown in Figure 2.10(a) and Figure 2.10(b), respectively. Additionally, energy-dispersive X-ray spectroscopy (EDX) measurements of platinum wire (before and after the experiments) are provided in Table 2.5.

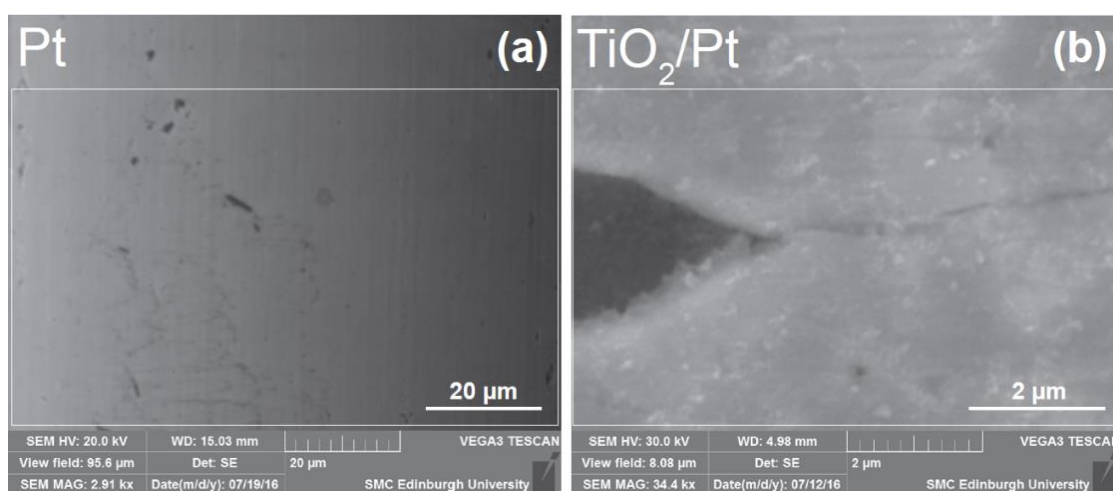


Figure 2.10: Scanning electron microscopy (SEM) images of platinum wire: (a) before, and (b) after experiments (TiO₂ nanoparticle deposition on the wire).

Table 2.5: Energy-dispersive X-ray spectroscopy (EDX) measurements of platinum wire with TiO₂ nanoparticle deposition after the experiments.

Sample area	Pt (wt.%)	Ti (wt.%)	O (wt.%)
#1	22.9	6.1	71.7
#2	49.1	1.5	49.1
#3	23.0	3.8	73.2
#4	54.7	2.4	42.8
#5	6.2	4.0	89.8
#6	24.8	3.6	71.6
Average	30.1	3.6	66.3

2.5.3. Pool Boiling Heat Transfer

In the study related to pool boiling heat transfer on plate heater, spherical TiO₂ (Titania) and CuO nanoparticles with average diameters of 20-50 nm and 10-20 nm, respectively, (both Ionic Liquids Technologies, io-li-tec GmbH, Germany) nanoparticles were dispersed in distilled water. The mass fractions, namely 0.001%, 0.01%, 0.05%, 0.075% and 0.2 %, were considered. The mass fractions were common mass fractions, which were widely used in boiling studies in the literature [102,103,109,110,120,215,216]. Typically, the mass fractions used in boiling studies are much lower than those used in single-phase studies [50,170,217–219]. It was not possible for the authors to have a mass fraction lower than the smallest one in the manuscript. Rather high mass fractions (>1%) were not considered, since nanoparticle deposition rigorously occurs at high mass fractions upon boiling. Nanofluids were prepared by adding the required amount of nanoparticles to distilled water, as base fluid (for obtaining above-mentioned mass fractions), performing sonication (Bandelin Sonorex, Germany) and magnetic stirring (JEIOtech, Korea). In this study, both of sonication and magnetic stirring took an hour (for each) in the preparation of all the nanofluids, which was found sufficient for preventing agglomeration and was also in parallel lines with nanofluid studies [220–226].

Typical Scanning Electron Microscopy (SEM) images of TiO₂ and CuO nanoparticles are displayed in Figure 2.11(a) and Figure 2.11(b). Accordingly, TiO₂ nanoparticle solid diameters are around 20-50 nm and CuO nanoparticles diameters are around 10-20 nm, and the particles have spherical shape.

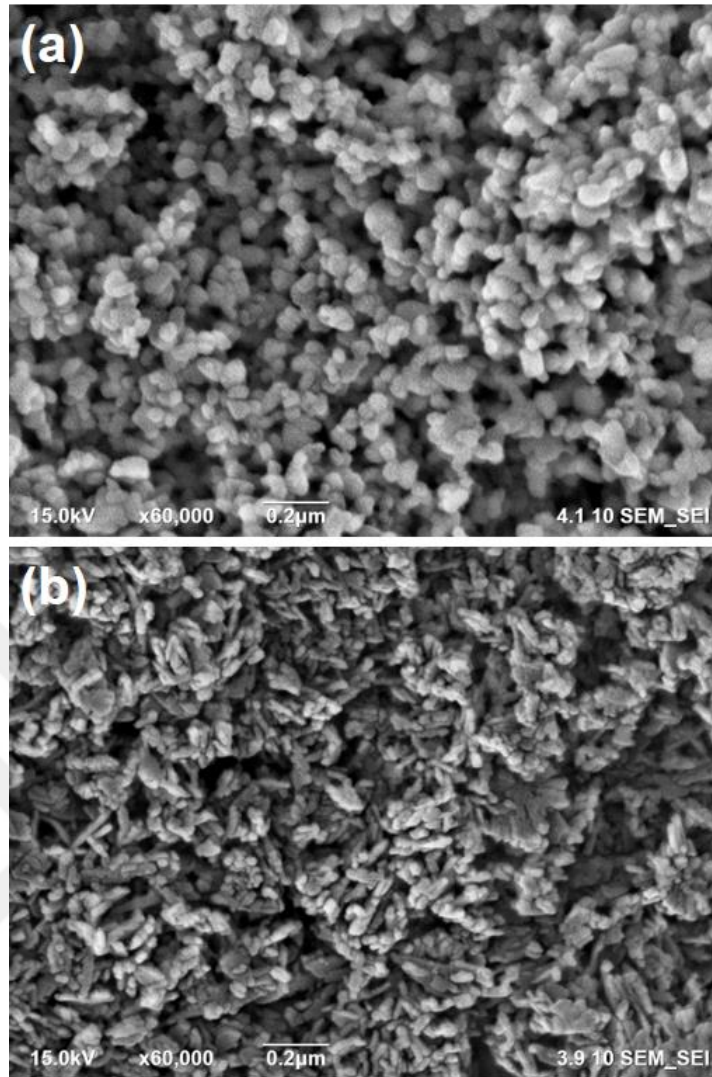


Figure 2.11: SEM image of: (a) TiO_2 nanoparticles and (b) CuO nanoparticles.

Additionally, Transmission Electron Microscopy (TEM) images of TiO_2 and CuO nanoparticles are shown in Figure 2.12(a) and Figure 2.12(b), respectively. According to the images, the same range of particle size can be measured.

The hydrodynamic diameters of TiO_2 and CuO nanoparticles in the base fluid were also measured using the Dynamic Light Scattering (DLS) apparatus (Malvern ZS Zetasizer, Malvern Instruments Ltd., UK), as shown in Figure 2.13(a) and Figure 2.13(b), respectively. This was repeated for three times for each sample, and the average values were taken for each set of measurements. According to the DLS measured results, peak sizes or distribution sizes of TiO_2 and CuO nanoparticles in nanofluids are approximately 340 ± 40 nm and 350 ± 50 nm, respectively. It can be seen that nanoparticles sit well-dispersed in the aqueous medium.

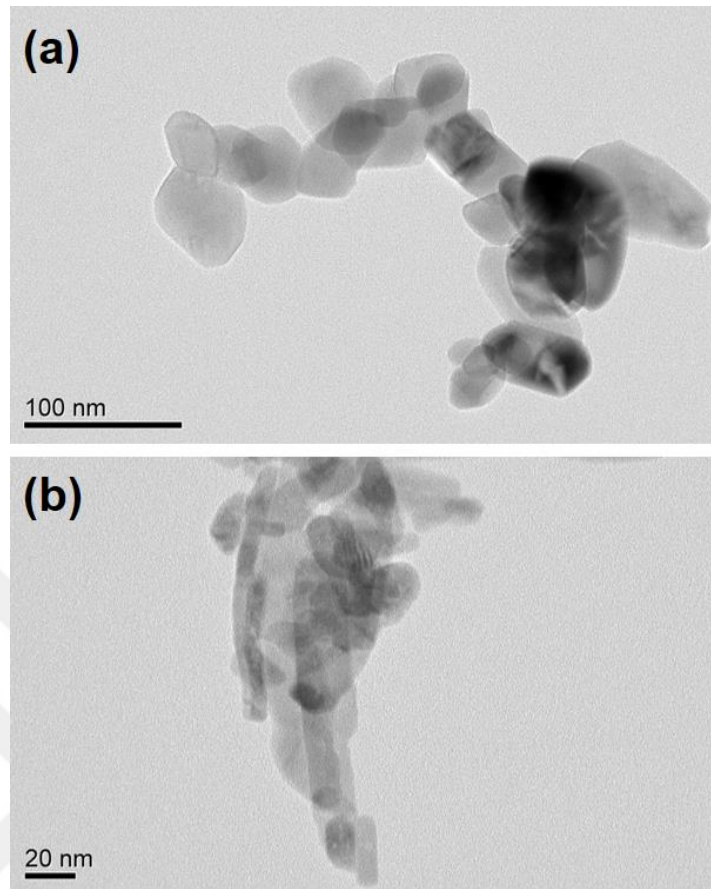


Figure 2.12: TEM image of: a) TiO₂ nanoparticles and b) CuO nanoparticles.

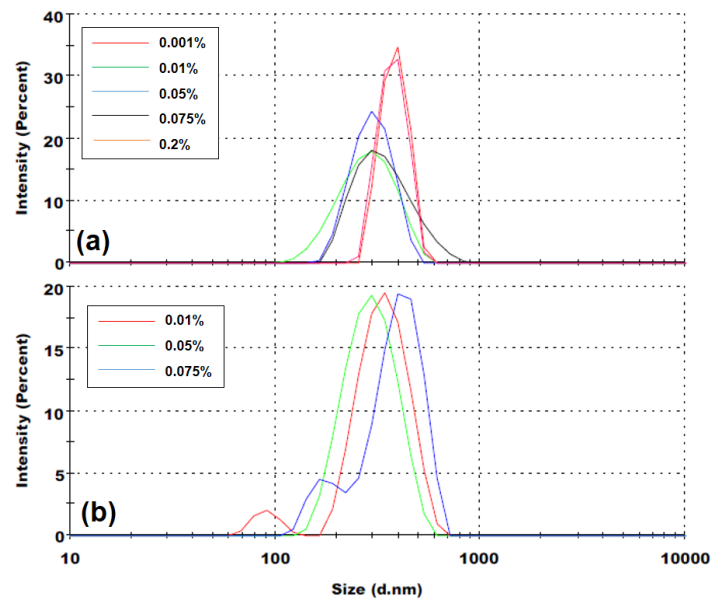


Figure 2.13: DLS measurement results of prepared: a) TiO₂ and b) CuO, nanoparticles/water nanofluids for different mass fractions.

2.5.4. Flow in Minichannels Bubble Dynamics

In the study related to the flow of nanofluids in minichannels bubble dynamics, spherical rutile TiO_2 nanoparticles (Ionic Liquids Technologies, io-li-tec GmbH, Germany) with an average diameter of 10-30 nm were utilized. Scanning electron microscopy (SEM) image of the nanoparticles before dispersion in the water is shown in Figure 2.14(a). Nanofluids were prepared by adding required mass of nanoparticles for achieving the mass fraction ratio of 0.005 wt.%, to both deionized water (DI water) and ethanol as the base fluids. No surfactant was used for studying the effect of nanoparticles to avoid any side effect of surfactant in bubble dynamics. Sonication (Fisher Scientific Ltd., cat. no. FB15047) was done for at least half an hour for properly dispersing the nanoparticles. Following this step, a magnetic stirrer was utilized for half an hour for a better dispersion. The hydrodynamic diameter of the nanoparticles in nanofluids is another parameter, which shows whether the nanoparticles are well-dispersed in the base fluid or not. Dynamic light scattering (DLS) measurements for both ethanol based and DI-water based nanofluids were performed by the Nanosizer equipment (90PLUS/BI-MASS, Multi angle particle sizing option, Brookhaven Instruments Corporation, USA), Figure 2.14(b). Three measurements were made for each nanofluid, and the averages of the measurements are plotted in Figure 2.14(b). The peaks of the DLS measurements are located at 151 nm and 117 nm for DI-water based nanofluids and ethanol-based nanofluids, respectively.

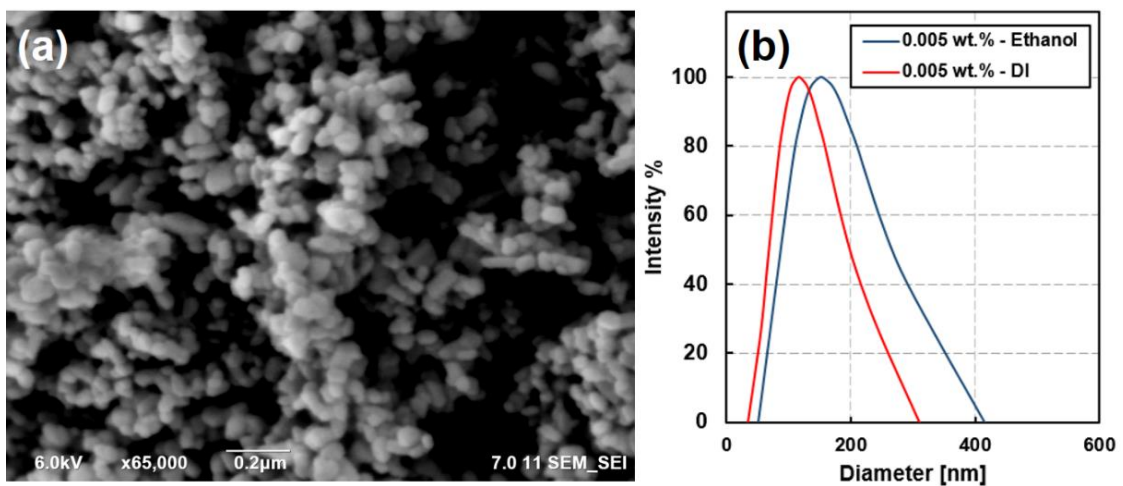


Figure 2.14: a) Scanning electron microscopy (SEM) secondary electron (SE) image of TiO_2 nanoparticles, and b) dynamic light scattering (DLS) measurement of nanofluids with different base fluids.

2.6. Novel Method for Increasing the Stability of Nanofluids

The schematic of the proposed technique, which is based on a recent patent application by the authors [227], is shown in Figure 2.15. The nanofluid having nanoparticle clusters is guided to a short micro flow restrictive element such as an orifice or a venturi. This flow restrictive element induces a sudden pressure drop due to low pressure; small bubbles from the flow restrictive element emerge due to cavitation. The emerging bubbles interact with the agglomerated structure of nanoparticles and decrease its size through hitting and/or shock waves and liquid jets, which are generated by their collapse. As a result, the resulting average nanoparticle size decreases at the exit of the flow restrictive element, and the stability of nanofluids is promoted.

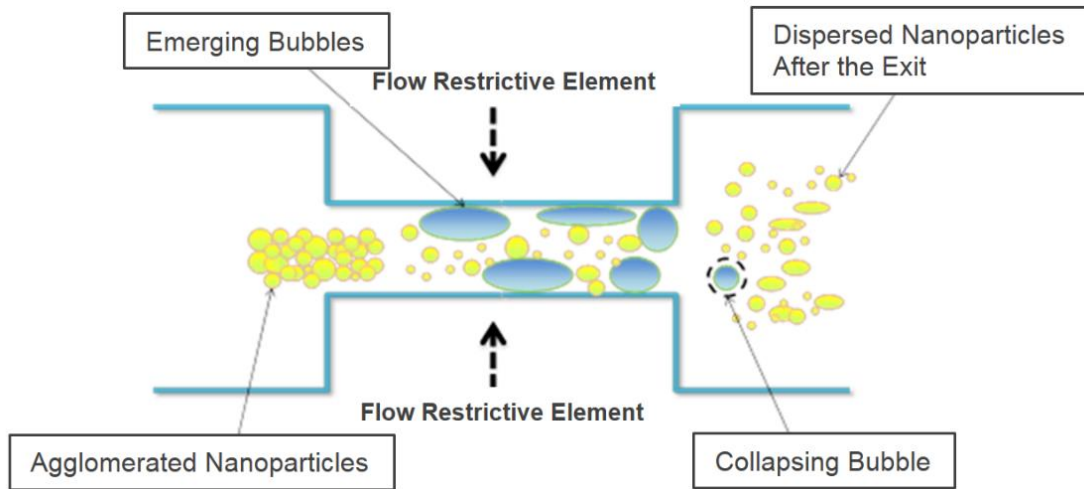


Figure 2.15: Schematic of the proposed technique for increasing the stability of the nanofluids.

The current approach is based on cavitating jet flows through a micro orifice with an inner diameter of 152 μm . For this purpose, the average solid diameter and mean hydrodynamic diameter of the employed Al_2O_3 nanoparticles were measured using the Scanning Electron Microscopy (SEM), Transmission Electron Microscopy (TEM), and Dynamic Light Scattering (DLS, Zetasizer Nano ZS, Malvern) techniques, respectively. The SEM and TEM-Bright Field (BF) images of the initially used Al_2O_3 nanoparticles are displayed in Figure 2.16 and Figure 2.17.

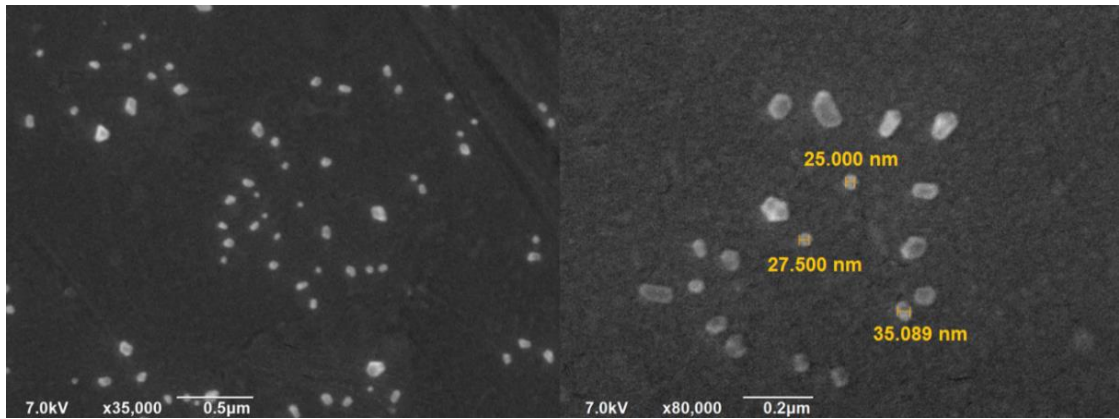


Figure 2.16: SEM images of initially used Al_2O_3 nanoparticles.

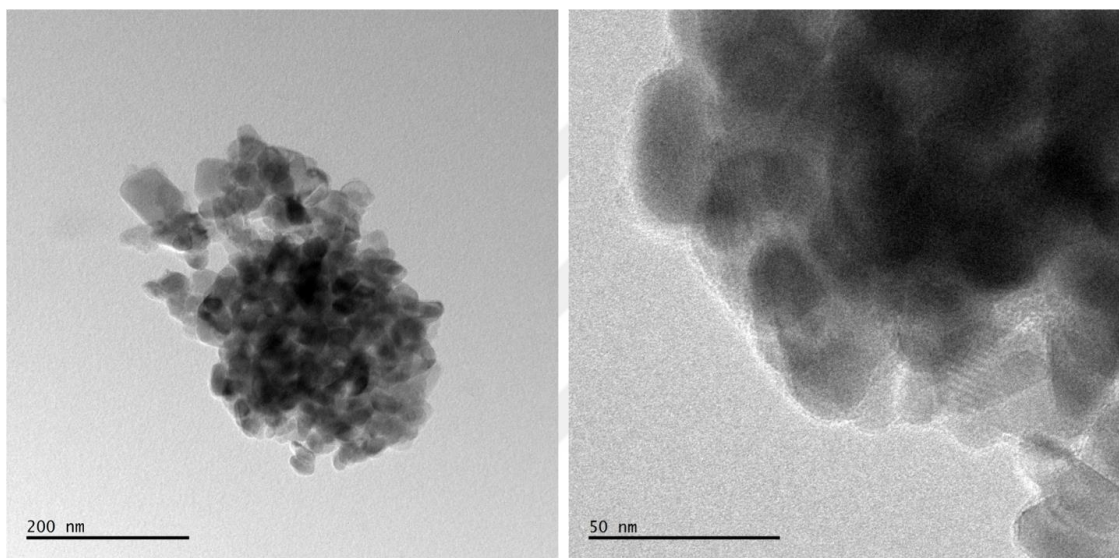


Figure 2.17: TEM-BF images of initially used Al_2O_3 nanoparticles.

In order to implement microscopy techniques, the nanofluid samples must be dropped on substrates and dried with air before taking the microscopy images. Due to the possible agglomeration of the nanoparticles during this drying process, it is not possible to image the real distribution of the particles in the base fluid using both SEM and TEM (Transmission Electron Microscopy) techniques and parametric effects cannot be captured. The DLS analysis, on the other hand, enables monitoring of the particle sizes in the base fluid without any processing. The DLS technique is employed for measuring the hydrodynamic diameter of the suspended nanoparticles in the base fluid with the use of Zetasizer Nano ZS (Malvern). This system consists of four main components: a laser serving as a light source to illuminate the sample particles, a detector to measure the intensity of the scattered light, an attenuator to regulate the laser intensity, and a correlator and Zetasizer software to analyze the measured data [220]. It should be noted that the

hydrodynamic size measured in a base fluid with the DLS technique is different from the particle (solid particle size) size measured by the SEM/TEM techniques. In the applications with nanofluids, the hydrodynamic size is considered as an important parameter and is typically significantly larger than the solid particle size [25,228]. In addition, no surfactant was used in this study for facilitating nanoparticle agglomeration.

A schematic of the experimental setup and micro orifice configuration, and spray structure are shown in Figure 2.18. The experimental setup consists of a high pressure pure nitrogen tank (Linde Gas, Gebze, Kocaeli), a liquid container (Swagelok, Erbusco BS, Italy), pressure sensors (Omega, USA.), fine control valves (Swagelok) at different locations, a micro filter (Swagelok), a turbinemeter (Omega, USA), fittings (Swagelok), and microchannel (Small Parts, USA.) connected to the setup with appropriate fittings. The tank was used as a container for deionized (DI) water, while the high pressure nitrogen tank was connected to this liquid container in order to maintain high input pressure and propel the fluid to the micro orifice. The filter was employed to prevent the flow of any particle larger than 15 μm into the system. Several pressure sensors were located at different parts of the experimental setup to measure the pressure.

The experimental setup was equipped with several valves to start/stop the flow. Cavitating flows were generated with a sudden pressure drop across a 4.5 mm short microchannel with an inner diameter of 152 μm . The experimental conditions, channel geometry and dimensions were selected based on our previous studies [229,230] for ensuring cavitation bubbles at the exit to validate the proposed method. According to the results provided in our previous study [229], there are few cavitation bubbles downstream of the channel at an upstream pressure of 10 bars when long channels are employed. More generated cavitation bubbles can extend to the outlet of the microchannel at higher pressures. For a short microchannel, the possibility of occurrence of bubble collapse downstream of the microchannel is higher compared to a long microchannel so that the energy released from the collapse of the cavitation bubbles can be further exploited in the proposed technique, and erosion due to the collapse of bubbles inside the channel can be avoided. As a result, the microchannel length was kept as short as possible.

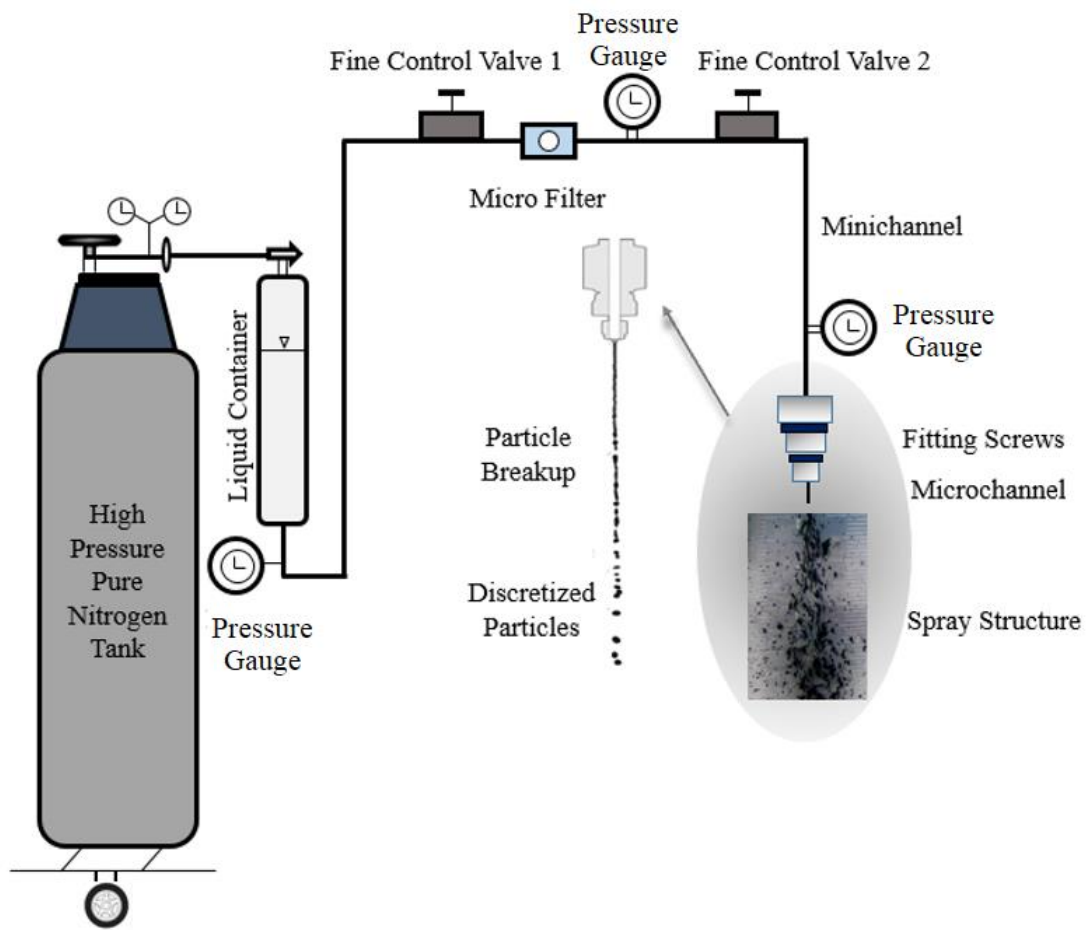


Figure 2.18: Schematic of the experimental setup, the microchannel configuration and the spray structure at the outlet.

The experiments were conducted by applying different inlet pressures. Inlet pressures varied between 10 and 90 bars, while the outlet pressure was fixed at 1 atm. The volumetric flow rate was increased with the change in inlet pressure, which affects the flow at the outlet of the micro orifice. The upstream pressure was measured at the outlet of the container. The flow passes through the system of tubes with an inner diameter of 3 mm and another pressure gauge records the pressure downstream of the microfilter to measure the pressure loss. Two fine valves control the flow. Finally, the pressure prior to the micro orifice (inlet pressure) was measured using another pressure gauge. Heated nanofluid samples (up to 99.8°C), where agglomeration of nanoparticles and disturbance in their stability took place leading to larger hydrodynamic diameters, were guided to the micro orifice. The size of the nanoparticles was measured after passing the fluid through the micro orifice to monitor any change in the stability.

CHAPTER 3

3. RESULTS AND DISCUSSION

3.1. Single-Phase Convective Heat Transfer

First, the experimental setup was validated by conducting the experiments with deionized water to determine the friction factor and heat transfer coefficients. It was observed that experimental friction factors for deionized water matched well with theoretical predictions, as can be seen at Figure 3.1(a). For theoretical friction factor coefficient predictions, existing conventional friction factor expression recommended for fully developed laminar flows in circular tubes ($f = 64 / \text{Re}$) is used. Figure 3.1(b) shows a comparison between experimental heat transfer coefficient and theoretical heat transfer coefficient. Theoretical heat transfer coefficients were calculated by using the following equations [231,232]:

$$h_{\text{theo}} = \frac{Nu_{\text{dev}} \cdot k_{\text{bf}}}{D_i} \quad (3.1)$$

$$Nu_{\text{dev}} = 0.517(f \cdot \text{Re})^{1/3} (x^*)^{-1/3} \quad (3.2)$$

$$x^* = \frac{x_{\text{dev}}}{D_i \cdot \text{Re} \cdot \text{Pr}} \quad (3.3)$$

Again, the experimental heat transfer coefficients matched well with theoretical heat transfer coefficients and are predicted within $\pm 15\%$. The correlation recommended for thermally developing flows is employed for the comparison since the thermal entry length varies from 7.3 to 31.9 cm in this study for the pure working fluid, while the heating length is kept as 12 cm. Thermally developing lengths are estimated ($x_{\text{ent}} = 0.05 \text{Re} \cdot \text{Pr} \cdot D_i$) and are shown in Table 3.1. Since the measured surface temperatures increased linearly along the microtube, the thermal conductivity of stainless

steel is rather high (>20 W/mK), and the thickness of the microtube is not large compared to the inner diameter of the microtube, the constant heat flux case is more appropriate for the experimental conditions in this study, and the correlation recommended for the constant heat flux case was employed.

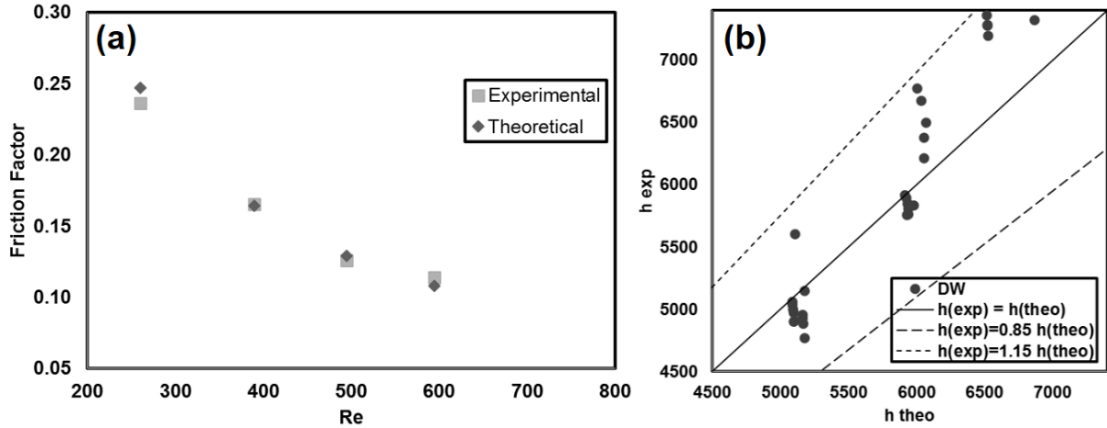


Figure 3.1: Comparison of experimental and theoretical: a) friction factor, and b) heat transfer coefficients for deionized water.

Table 3.1: Thermally developing length of nanofluids at different Reynolds numbers.

Re	531±14	1070±25	1665±15	1940±24	2519±35
Developing length (cm)	7.3	15.7	21.3	24.8	31.9

After validating the experimental setup, the experiments were performed using the nanofluids under different operating conditions, and friction factors, single-phase experimental and theoretical heat transfer coefficients of the nanofluids are compared to each other. Figure 3.2 shows experimental friction factor coefficient for all the nanofluids. Thermophysical properties of the nanofluid were taken into account in the data reduction. As can be seen, the results match well with theoretical friction factor predictions ($f = 64/Re$) for a wide variety of Reynolds numbers. It can be also observed that the introduction of nanoparticles and the mass fraction do not have any significant effect on friction factor. However, the impact of agglomerated nanoparticles should still be considered further. Although the stability was preserved after the experiments, a gradual agglomeration of nanoparticles is expected for their repetitive use in the setup and at elevated temperatures.

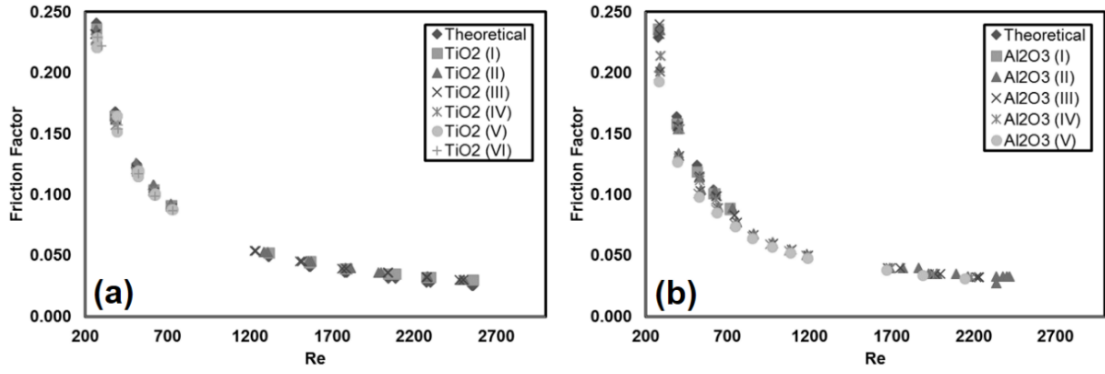


Figure 3.2: Experimental and theoretical friction factor for water-based nanofluids with TiO_2 and Al_2O_3 nanoparticles at different fraction ratios in different Reynolds numbers.

Figure 3.3 shows the average heat transfer coefficients of nanofluids with TiO_2 and Al_2O_3 nanoparticles up to Reynolds number of 1200. Experimental results reveal that measured heat transfer coefficients for nanofluids with TiO_2 and Al_2O_3 nanoparticles are almost the same, as seen in Figure 3.3. Thus, it is understood that mass fraction of nanofluids does not have any considerable effect on heat transfer coefficient and no considerable enhancement exists for this region. This finding is in agreement with the literature [49]. This might be due the lack of mixing of nanoparticles with respect to the base fluids under laminar flow conditions.

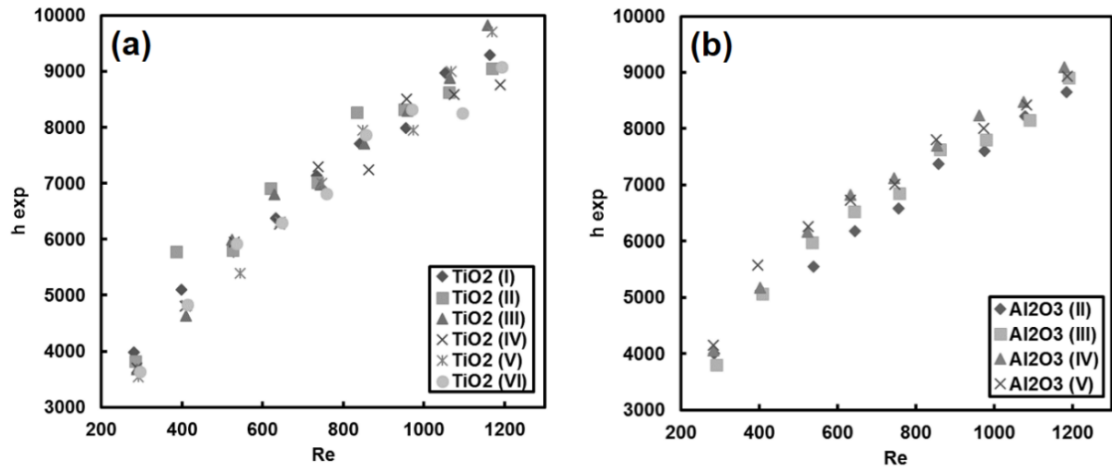


Figure 3.3: Experimental heat transfer coefficient of nanofluids with: a) rutile-titania, and b) γ -alumina nanoparticles.

Figure 3.4 shows average experimental heat transfer coefficients and Nusselt numbers of nanofluids with Al_2O_3 nanoparticles for $\text{Re} > 1500$. As seen from Figure 3.4(a), maximum heat transfer coefficients in this figure increase up to 25% with mass

fraction ratio. This increasing trend is due to convective heat transfer enhancement with nanoparticle motion under transition regime and pending turbulent flow conditions. Under these conditions, enhancing effects due to nanoparticle motion and subsequent additional mixing are present leading to enhanced heat transfer. As understood from Figure 3.4(b), with increasing the mass fraction ratio of nanoparticles, Reynolds number is decreasing due to increasing viscosities at higher mass fraction ratios.

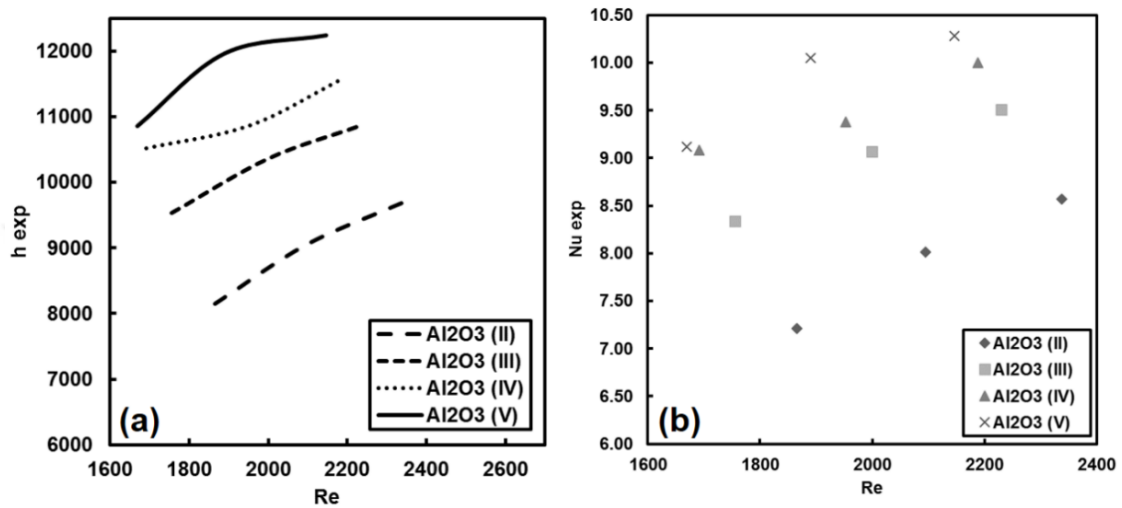


Figure 3.4: a) Heat transfer coefficients, and b) Nusselt number of nanofluids with Al₂O₃ nanoparticles.

Figure 3.5 shows experimental local heat transfer coefficients of nanofluids with TiO₂ nanoparticles for Reynolds numbers of 289, 636, 849, and 1070. Local heat transfer coefficients for nanofluids with TiO₂ nanoparticles have almost a constant profile for different mass fractions beyond a certain distance, which implies that fully developed conditions were reached beyond a certain distance from the inlet. Similar trend is observed for nanofluids with Al₂O₃ nanoparticles at Reynolds numbers of 403, 751, 846, and 1082 as shown in Figure 3.6. Local heat transfer coefficient profiles show that thermal entrance length is rather short at low Reynolds numbers and heat transfer coefficient becomes constant beyond this entry length.

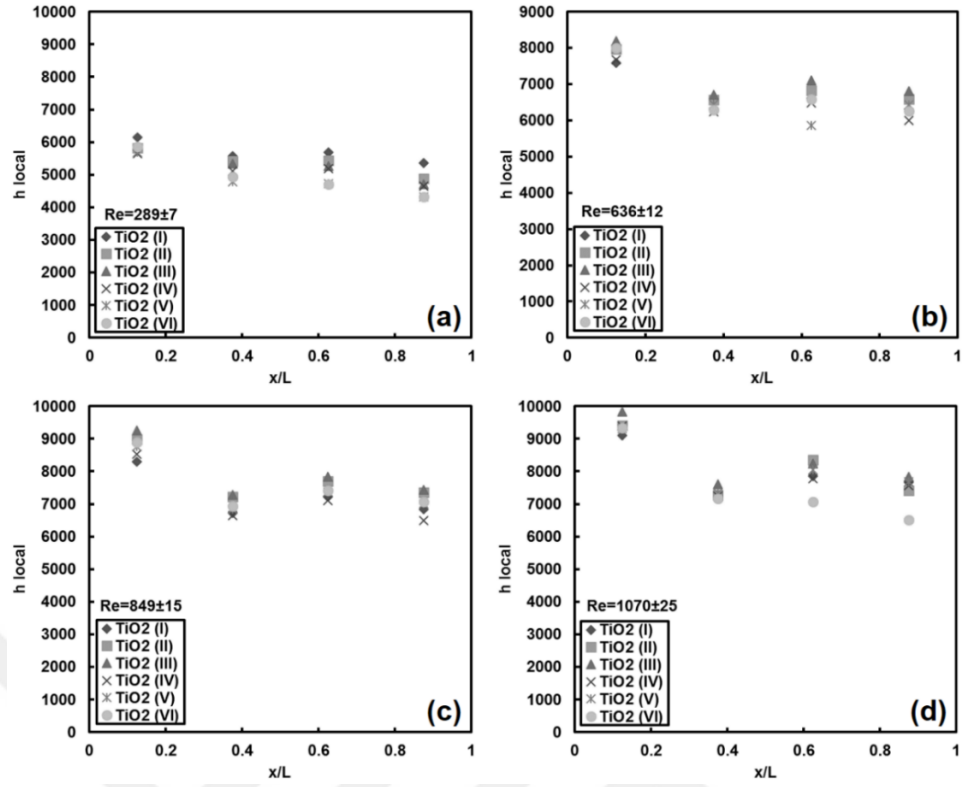


Figure 3.5: Local heat transfer coefficients of nanofluids with TiO_2 nanoparticles at lower Reynolds numbers.

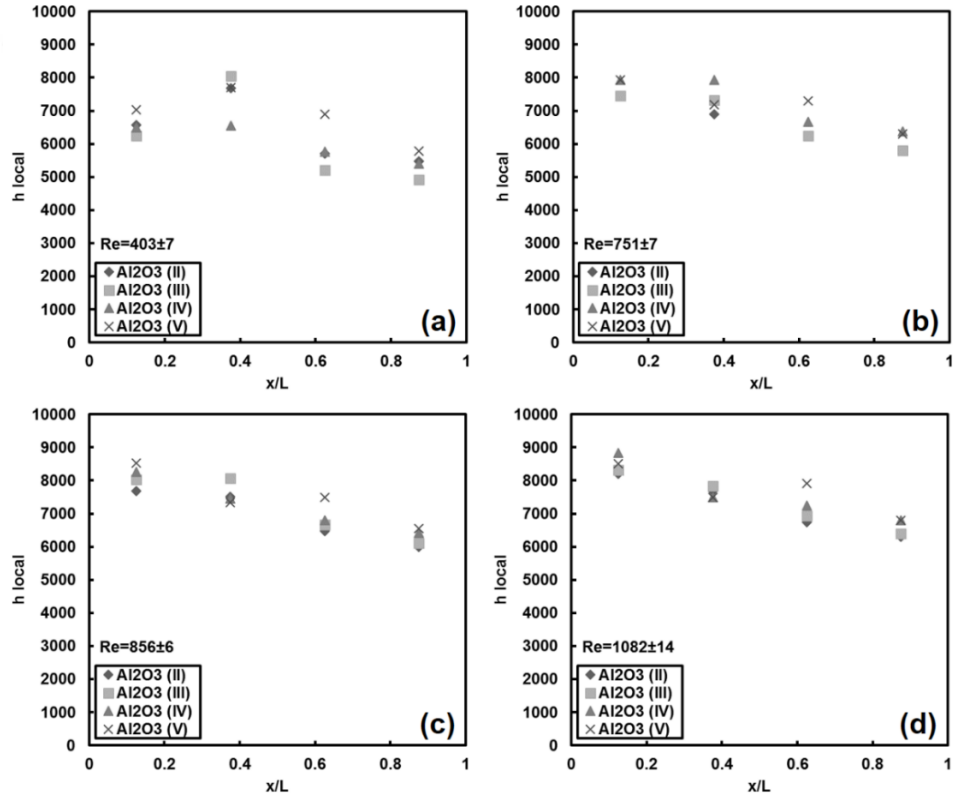


Figure 3.6: Local heat transfer coefficients of nanofluids with Al_2O_3 nanoparticles at lower Reynolds numbers.

Nanofluids with TiO_2 nanoparticles display a similar trend as shown in Figure 3.7 at Reynolds numbers of 1789 and 2036. At higher Reynolds numbers, thermally developing length becomes longer. Under these conditions, heat transfer coefficient continues to decrease along the axial distance as shown in Figure 3.7 leading to a less flat profile. Figure 3.8 shows experimental local heat transfer coefficients of nanofluids with Al_2O_3 nanoparticles for Reynolds numbers of 1682, 1933, and 2190. A decreasing trend in heat transfer with axial distance coefficient is present. Moreover, it can be observed that local heat transfer coefficients increase at higher Reynolds numbers.

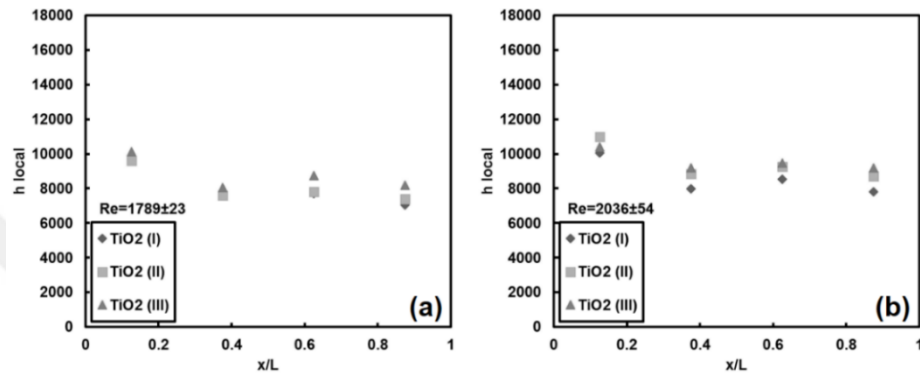


Figure 3.7: Local heat transfer coefficients of nanofluids with TiO_2 nanoparticles at higher Reynolds numbers

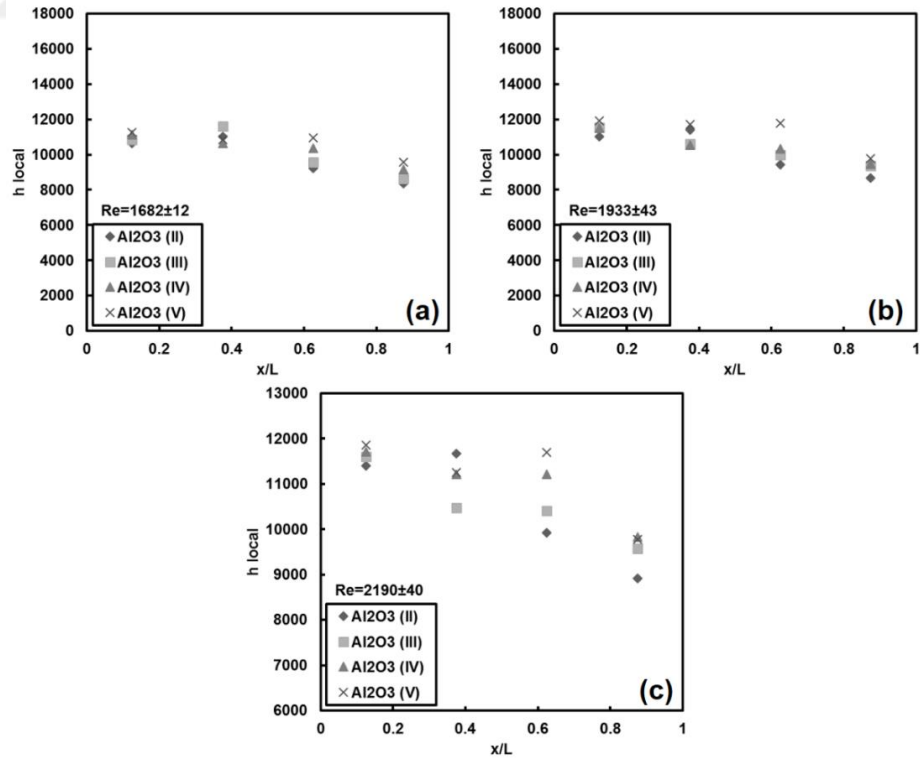


Figure 3.8: Local heat transfer coefficients of nanofluids with Al_2O_3 nanoparticles at higher Reynolds numbers

Figure 3.9 displays average experimental heat transfer coefficients of nanofluids with TiO_2 and Al_2O_3 nanoparticles with theoretical predictions. Theoretical predictions were obtained using nanofluids' thermophysical properties in similar lines to the pure working fluid. It can be seen that experimental heat transfer coefficients at different mass fractions of nanofluids can be predicted within $\pm 15\%$, and the existing theory for convective heat transfer analysis could be implemented for nanofluids with appropriate use of their thermophysical properties for thermally developing flows.

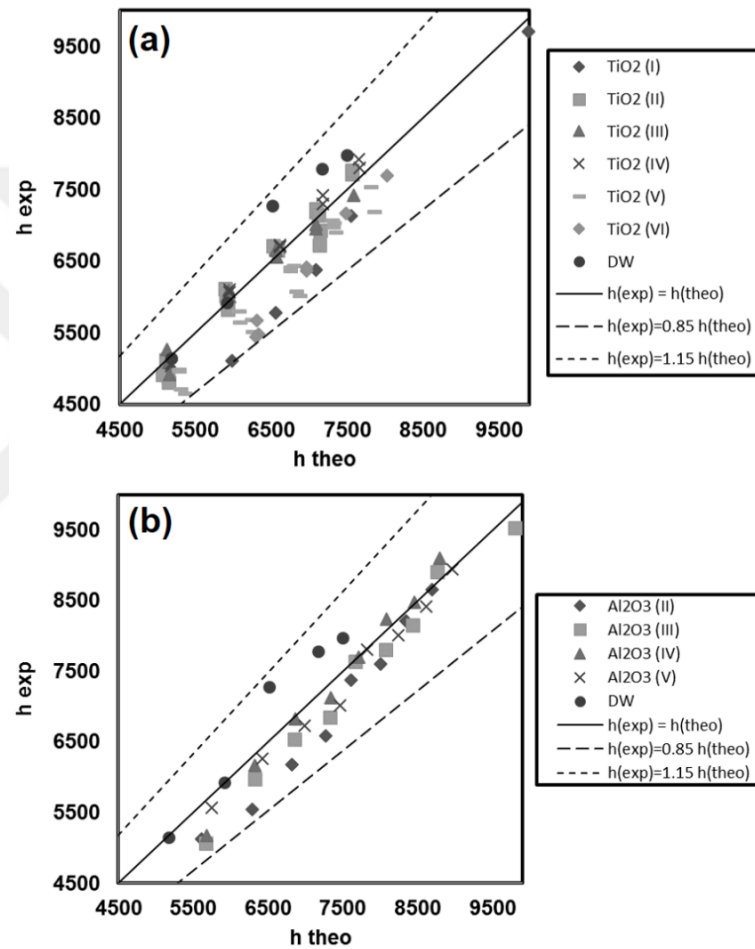


Figure 3.9: Experimental versus theoretical average heat transfer coefficients for nanofluids with: a) titania, and b) alumina nanoparticles.

3.2. Pool Boiling Bubble Dynamics

3.2.1. Bubble Dynamics for Pure Water

3.2.1.1. Bubble growth at ONB point

Applying a heat flux on the wire in contact with the liquid resulted in the formation of a hot liquid layer surrounding the wire. The liquid layer becomes less dense than the bulk fluid and rises with the driving force of buoyancy. The adjacent cooler fluid then surrounds the wire. This results in natural convection and constitutes the heat transfer mechanism preceding the onset of nucleate boiling regime (ONB). Boiling inception is associated with nucleation of bubbles. As long as the vapor pressure is higher than the ambient pressure, a vapor bubble forms and grows in the liquid. As far as bubble formation is concerned, nucleation, growth, departure and waiting time emerge as important parameters.

The energy transferred to the liquid molecules disturbs the stable state condition and leads to the formation of a molecule clusters (embryo). Upon reaching a critical size, these clusters are called nuclei [233]. Nucleation can be of quasi-homogeneous or of heterogeneous nature [122]. If the heated solid material has microscopic defects and crevices on its surface, vaporization begins in these locations. Gases or vapor within the defects require lower energy (to overcome the free energy barrier) to form a nucleus. This type of nucleation is known as heterogeneous nucleation [234,235]. On the other hand, if the molecule clusters completely form within the superheated liquid, it is called homogeneous nucleation [236]. While a bubble grows hemispherically, a thin layer of liquid (microlayer) between the bubble base and the heated wall is trapped contributing to the bubble growth [237]. The pressure inside the bubble drops along with temperature at the bubble interface where the enthalpy of the superheated layer is converted into latent heat at the bubble interface. As a result, the interfacial temperature decreases and heat diffuses to the interface.

Bubble growth has two stages, inertially controlled growth and thermally controlled growth. The inertially controlled growth is very fast and occurs in the initial stage of the growth process. It is only limited by the momentum that is transferred to the surrounding liquid for its displacement. Thermally controlled growth occurs when heat is conducted from liquid to the liquid-vapor interface and is limited by the heat conducted

to the interface. The growth rate of this stage is much slower than the inertially controlled one.

According to bubble growth rate measurements obtained from the experiments, as the bubble diameter increases, the growth rate decreases as can be seen in Figure 3.10 (a portion of bubble growth process, the whole growth process took approximately 58 sec). The growth rate between t s and $t+10$ s, between $t+10$ s and $t+30$ s, and between $t+30$ s and $t+45$ s are $17 \mu\text{m/s}$, $8 \mu\text{m/s}$, and $4 \mu\text{m/s}$, respectively.

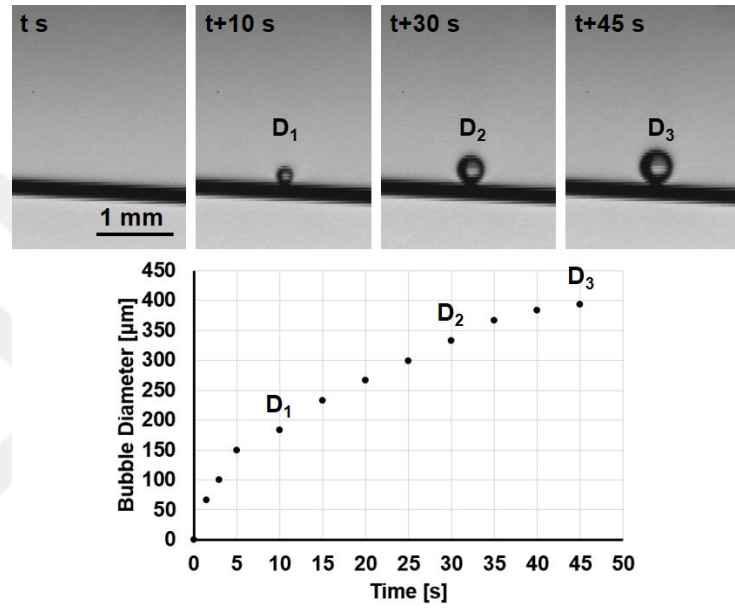


Figure 3.10: Bubble growth process for a bubble from a nucleation site at ONB and the corresponding diameter as a function of time. ($T_f=30^\circ\text{C}$, $q''=508 \text{ kW/m}^2$)

A bubble can depart horizontally by migrating along the wire or vertically by rising through the bulk fluid when it reaches a critical size dependent on the applied heat flux. In subcooled nucleate boiling, vertical detachment from the nucleation site is prominent at low heat fluxes (ONB process), ($q''=508 \text{ kW/m}^2$), owing to the fact that higher critical sizes increase the buoyancy force. Moreover, inertia, hydrodynamic drag and thermocapillary forces should be taken into account for the growth rate. Inertia and hydrodynamic drag forces are negligible when the bubble grows slowly and lower temperature gradients cannot produce adequate thermocapillary forces to resist the lifting of the bubble. On the other hand, increasing the heat flux results in horizontal bubble departure, where thermocapillary force is more dominant to hold the bubble on the wire. The bubble migration process will be discussed later. The thermal boundary layer is

ruptured by the bubble departure and redeveloped to allow a new bubble nucleation. The time between the bubble departure and a new bubble nucleation is called the waiting time. The hot liquid layer is only regenerated during the waiting time by transient conduction.

3.2.1.2. Lateral migration

A bubble departs from the nucleation site and migrates along the wire from nucleation site to another bubble or end of the wire. Usually, when a moving bubble encounters another bubble, it changes its direction in the opposite direction and continues its translational motion at relatively lower heat fluxes. Increasing the heat flux results in an increase in the bubble velocity, Figure 3.11. This behavior is often seen in the experiments until a certain heat flux (approximately 1400 kW/m²).

A bubble migration and collision with another bubble is illustrated in Figure 3.12 at a water bulk temperature of 30°C and heat flux of $q''=682$ kW/m². As can be seen, while a moving bubble with a constant velocity, 32 mm/s, is approaching a stationary bubble, it decelerates gradually down to the velocity of 23 mm/s in 40 ms ($t+90$ ms). In the next 10 ms, the deceleration becomes greater so that within 10 ms the velocity suddenly drops down to 8 mm/s ($t+100$ ms), and finally it stops at a close location to a stationary bubble, while its size decreases. After stopping for 2 ms, it reverses its direction at $t+113$ ms. In the reverse direction, it accelerates to the velocity of 27 mm/s ($t+135$ ms). After 45 ms, its velocity tends to a constant value of 30 mm/s ($t+180$ ms), and it continues to migrate at this constant velocity. It should be mentioned that this process does not comprise the inception of the bubble and its acceleration to a constant velocity, which is illustrated as supplementary material in Figure 3.13.

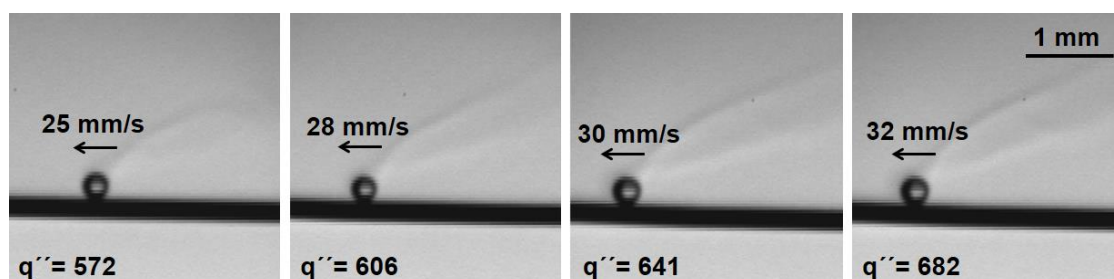


Figure 3.11: Lateral migration acceleration of bubbles with approximately the same size on the wire at water bulk temperature of 30°C and at different applied heat fluxes [kW/m²]. (Tail of the bubbles presents the direction of migration.)

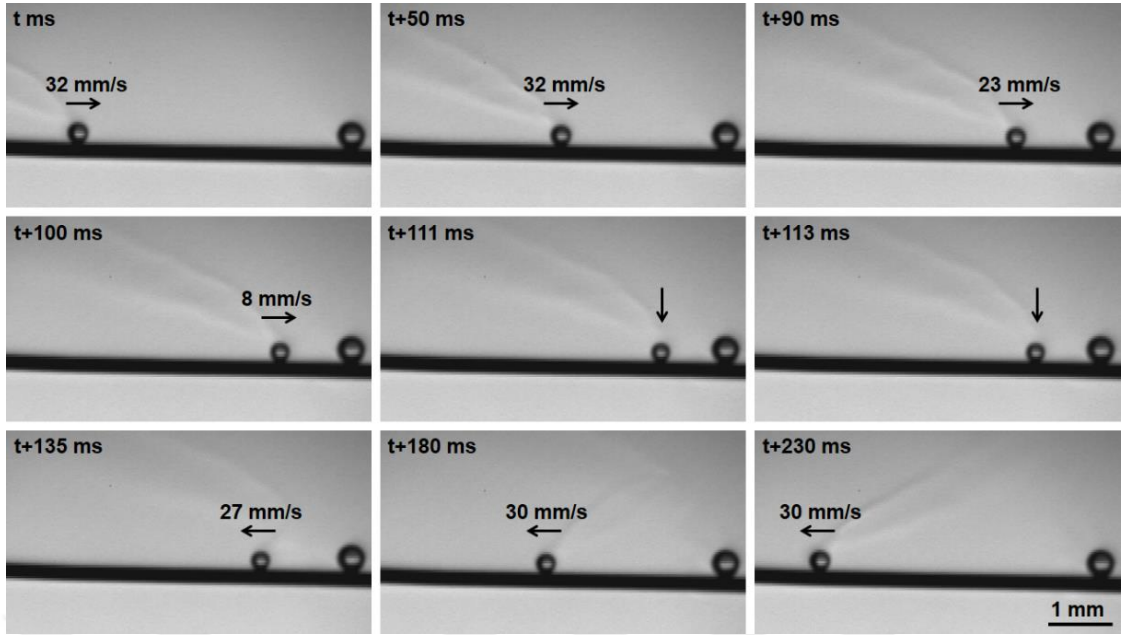


Figure 3.12: Lateral migration of a moving bubble on the wire in pure water. Deceleration, stopping time and acceleration are presented. ($T_f=30^\circ\text{C}$, $q''=682 \text{ kW/m}^2$)

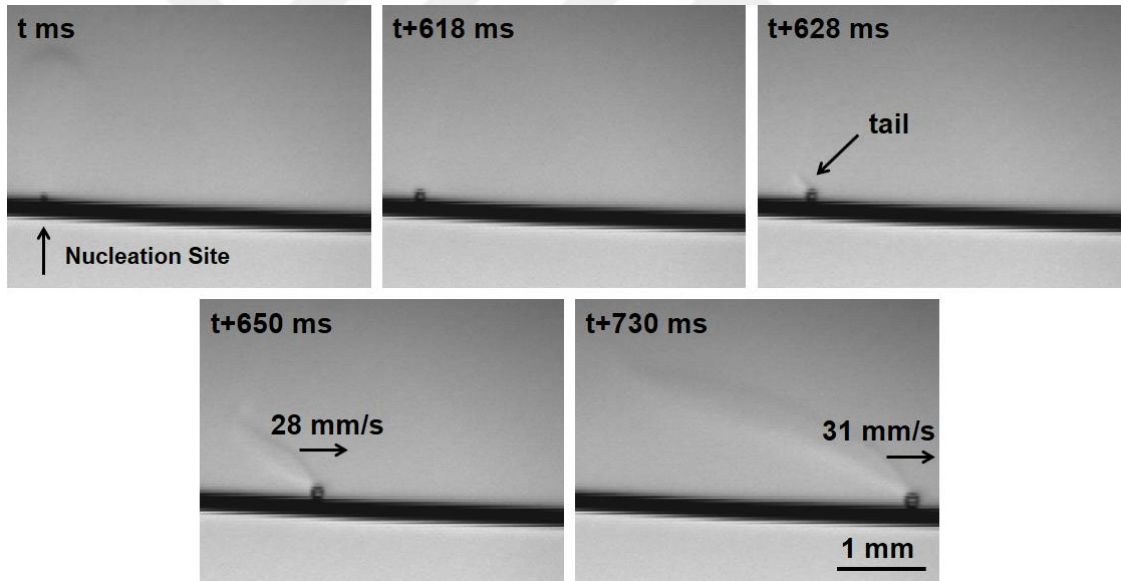


Figure 3.13: Lateral migration acceleration of a new bubble from the nucleation site and its reaching to a constant speed on the wire. Bubble after nucleation and reaching to a certain diameter at the bulk temperature and applied heat flux starts to migrate along the wire and after a certain time (here 10 ms) tail is observed. ($T_f=30^\circ\text{C}$, $q''=641 \text{ kW/m}^2$)

At the nucleation site, a bubble grows due to quickly absorbing the heat from the microlayer underneath the bubble, and the bubble is accelerated driven by Marangoni effect. The Marangoni effect arises from surface tension gradient at the interface between liquid and vapor phases. The surface tension depends on either concentration or

temperature. The Marangoni effect can be referred to as solutocapillary effect, when concentration determines the surface tension, and as thermocapillary effect, when the surface tension varies with temperature. Considering a force balance in the vertical direction on a stationary or moving bubble on a wire, the buoyancy force acts on the bubble in upward direction (Figure 3.14). The gravity force is not sufficient to balance the buoyancy force. Along the bubble interface, evaporation happens at the bubble interface on the hot liquid layer (between the bubble base and the wire), while condensation occurs at the bubble interface on the surrounded bulk liquid. The temperature gradient at the bubble interface between the hot liquid layer and the bubble base gives rise to the thermocapillary flow near and above the bubble. The thermocapillary effect always generates convection from higher to lower temperature along the interface. The corresponding flow is called top jet flow and has a tail-like appearance. It is speculated that Marangoni convection would not be strong enough to move the bubble unless non-condensable gases exist inside the bubbles owing to the fact that the presence of non-condensable gases degrades condensation heat transfer [62,86]. A lower condensation heat transfer coefficient disturbs the uniformity of the surface temperature distribution resulting in stronger Marangoni convection [64].

A high temperature region drags the thermal boundary layer into the cold region, and thus, the liquid is pumped from the hot region to the cold region, and the bubble jet begins to develop. Since surface tension depends on the local temperature gradient, the Marangoni flow exerts a thrust force (thermocapillary force) on the bubble, and the vertical component of the thermocapillary force prevents bubble rising under the influence of buoyancy force [94].

The horizontal component of the aforementioned thermocapillary thrust force plays a major role in the bubble lateral migration. As the bubble moves on the wire, an asymmetrical temperature distribution develops around the bubble [88]. Since the bubble consistently absorbs heat from the hot liquid layer, the wire surface temperature hence the hot liquid layer temperature changes as the temperatures behind the bubble will be lower than the temperatures in front of this latter. Therefore, the bubble front surface is warmer than its back. The horizontal component of the thermocapillary force pushes the bubble along the bubble interface in the same direction. The temperature difference between the wire and the bulk fluid temperatures in front of the bubble is higher than that

behind the bubble, hence, the Marangoni flow is more intense in front of the bubble leading to a pressure gradient, which further pushes the bubble in the same direction [82]. In addition to the horizontal component of the thermocapillary force, a drag force acts on the moving bubble in the opposite direction. When the bubble is accelerated, this indicates that the Marangoni force is greater than the drag force. As the moving bubble continues to grow by absorbing heat from the wire the velocity of the bubble increases, the drag force also increases. On the other hand, the temperature gradient decreases as the bubble approaches the stationary bubble. As a result, the thermocapillary force decreases, since the stationary bubble acts as a cold source. At some point, the thermocapillary force and the drag force balance each other leading to bubble constant velocity. As the bubble moves further, the drag force becomes greater than the thermocapillary force causing a decrease in the bubble speed. When the bubble encounters the stationary one, it stops. At this moment, the temperatures on front of the bubble are lower than those on the backside. Thus, the bubble migration shifts to the opposite direction.

As mentioned before, the moving bubble becomes smaller upon stopping. This effect is more obvious at higher applied heat fluxes ($q''=739 \text{ kW/m}^2$), Figure 3.14. A sudden deceleration of the moving bubble may change the non-condensable gas accumulation area from the condensing surface to the evaporating surface and enhances condensation, leading to a small temperature gradient on the bubble surface with the absence of Marangoni convection near the bubble, within a very short period of time. When the bubble stops, the remaining tail rises, and the newly formed tail becomes visible after a short period of time. This does not happen immediately after the start of bubble migration in the reverse direction.

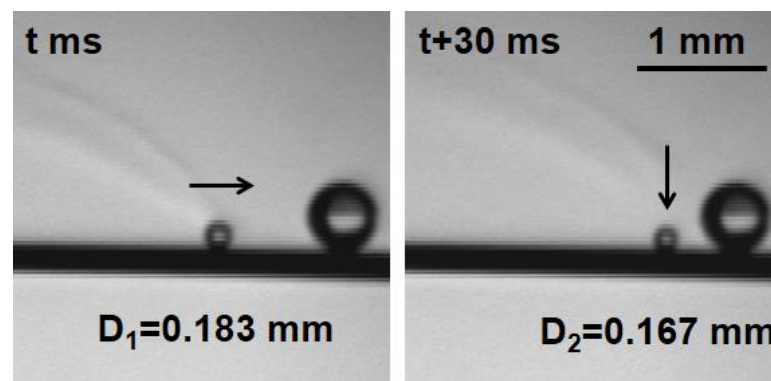


Figure 3.14: Shrinkage of a bubble after stopping near a stationary one. By assuming the bubble as a sphere and calculating the volume before and after shrinkage, it can be concluded that the volume of the bubble shrinks with a rate of approximately 24%. ($T_f=30^\circ\text{C}$, $q''=739 \text{ kW/m}^2$)

Another type of bubble-bubble interaction regarding bubble migration and pre-jet flow can be seen in Figure 3.15. At higher applied heat fluxes ($q''=641 \text{ kW/m}^2$), when a moving bubble approaches a newly nucleated bubble, which has not completed the growth process and has not started with its migration, there are two possibilities. As can be seen, a new bubble nucleates from the nucleation site, and after reaching a certain diameter at $t+620 \text{ ms}$ after nucleation, it starts to move without interfering with other bubbles. In the meanwhile, a new bubble emerges and continues growing. Here, an important event assists in determining whether this newly nucleated bubble will migrate away from the moving bubble, which is approaching it (Case 1) or will merge with it (Case 2). At $t+900 \text{ ms}$, a pre-jet initiates from the top of the bubble, which is demonstrated with dashed lines and rises above the bubble with a pre-jet velocity of 15 mm/s . While this bubble is growing, a moving bubble approaches ($t+1054 \text{ ms}$), and because of the pre-jet, the moving bubble pushes the nucleating bubble away probably because of perturbation of the symmetrical temperature distribution around the nucleating stationary bubble, which generates a thermocapillary force on the stationary bubble to overcome the contact line force. Thus, its growth process continues away from the nucleation site [85]. On the other hand, the initially moving bubble reverses its direction after stopping for some time. The three sub-figures starting from $t+1302 \text{ ms}$ are related to Case 2. First, the inception of the moving bubble occurs. Then, it approaches a new bubble, which is nucleated from the same nucleation site, and merges with this stationary bubble, because the pre-jet event does exist before the interaction. The pre-jet may affect and decrease the radial temperature distribution around the nucleating stationary bubble at the nucleation site. Meanwhile, before the pre-jet, the bubble is smaller and is exposed to the hotter fluid.

According to systematic observations, the bubble detachment is seen due to buoyancy. However, particularly at very high heat fluxes, bubble motion underneath the wire is observed at lower bubble velocities (presumably because of gravity), Figure 3.16, because the heater boundary layer is smaller than that on top of the wire resulting in weaker Marangoni convection [66,86].

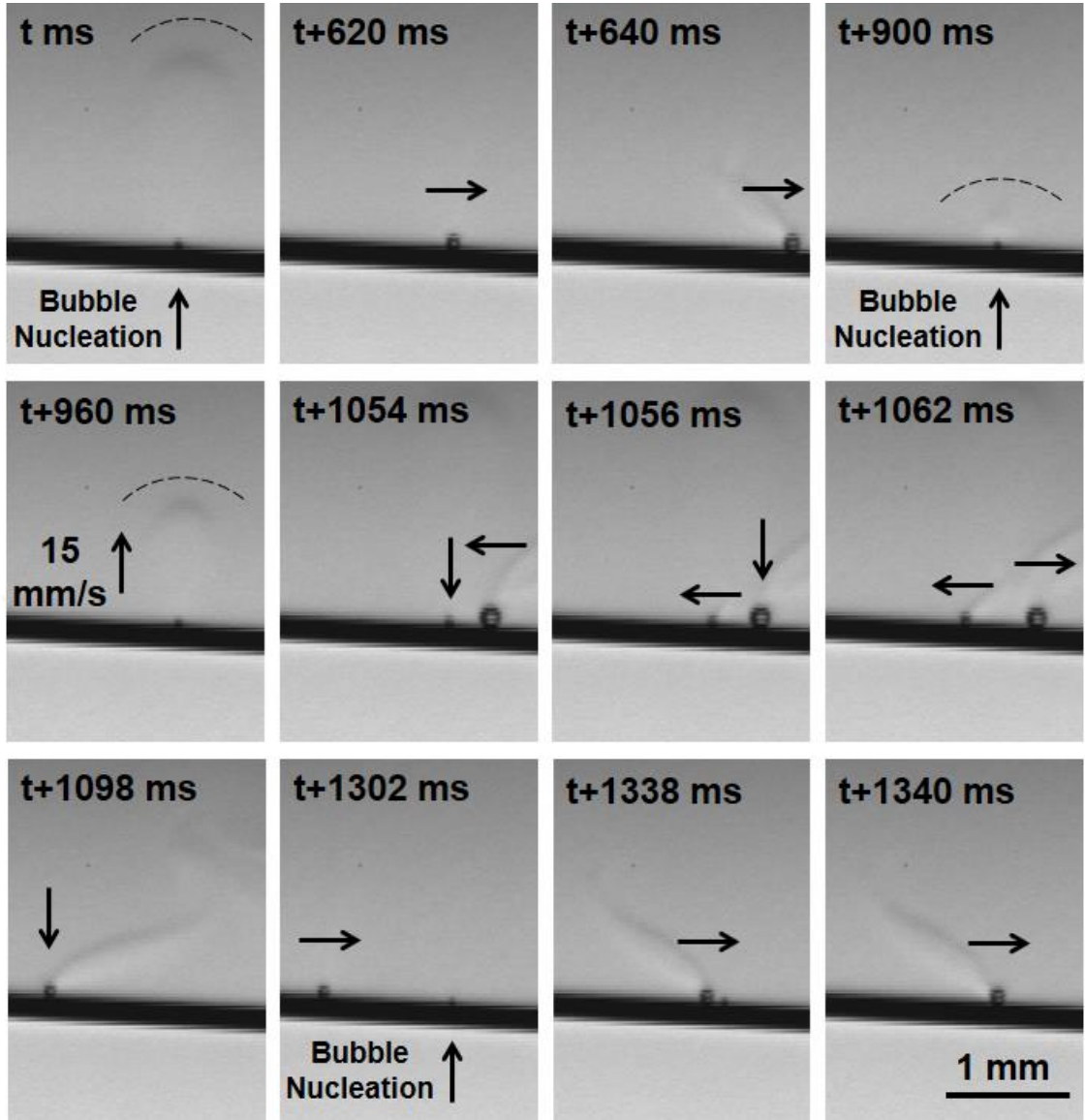


Figure 3.15: Pre-jet event for a new bubble. ($T_f=30^\circ\text{C}$, $q''=641 \text{ kW/m}^2$)

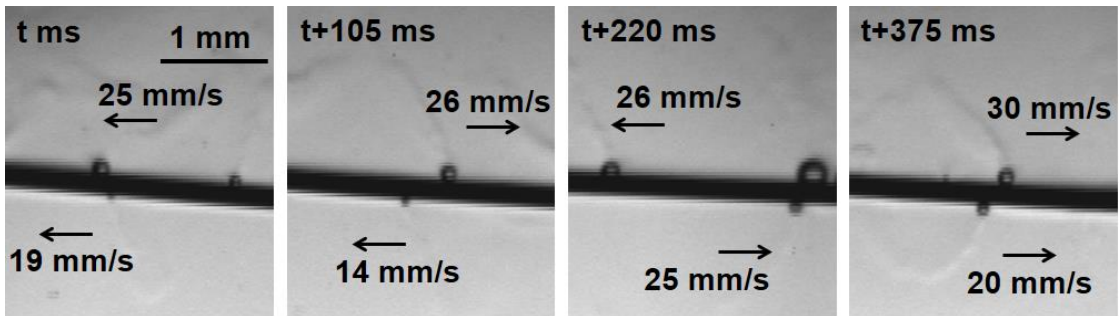


Figure 3.16: Comparison between lateral migration speed of the bubble on the wire and that under the wire. Generally, speeds and mean speed under the wire are less those on the wire. ($T_f=30^\circ\text{C}$, $q''=1396 \text{ kW/m}^2$)

3.1.2.3. Coalescence

Bubble coalescence usually occurs when the distance between bubbles is smaller than the sum of the radii of the bubbles [93]. Based on the experimental observations, different types of coalescence take place: two bubbles moving towards each other (Case 1), Figure 3.17(a), two bubbles moving in the same direction (Case 2), Figure 3.17(b), a bubble approaching a stationary bubble, which is formed at the nucleation site (Case 3), Figure 3.17(c). Cases 1 and 3 are commonly observed at moderate heat fluxes and high heat fluxes ($640 \text{ kW/m}^2 < q'' < 1400 \text{ kW/m}^2$).

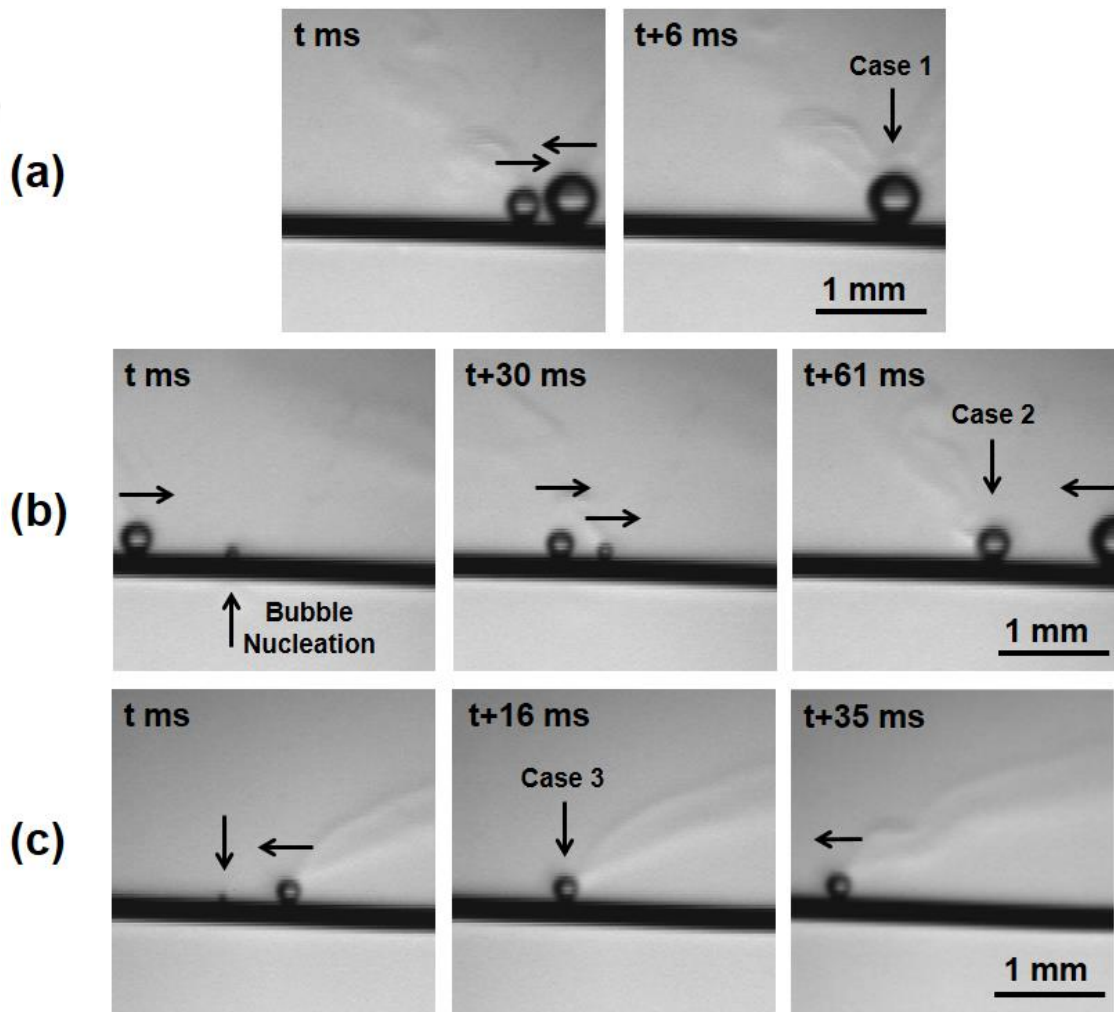


Figure 3.17: Coalescence when: a) two bubbles move towards each other (Case 1) ($T_f=30^\circ\text{C}$, $q''=1036 \text{ kW/m}^2$), b) two bubbles move in the same direction (Case 2) ($T_f=30^\circ\text{C}$, $q''=1036 \text{ kW/m}^2$), and c) a bubble approaches to a new bubble (Case 3). ($T_f=30^\circ\text{C}$, $q''=707 \text{ kW/m}^2$).

Figure 3.18 illustrates bubbles merging process with respect to different bubble sizes and velocities at the water bulk temperature of 30°C and heat flux of $q''=998 \text{ kW/m}^2$. Two bubbles approach each other with a velocity of 28 mm/s. Then, bubbles merge and the newly formed bubble (bubble III) having two tails stops for a while ($t+10 \text{ ms}$). It should be noted that the radius of bubble III is smaller than the sum of the radii of bubbles I and II. After merging, bubble III starts to migrate to the right direction. It can be observed that when the velocities of the moving bubbles are the same, the merged bubble motion is in the same direction as the moving bubble with a larger diameter. Bubble III then continues with its motion and encounters a smaller moving bubble (bubble IV) ($t+114 \text{ ms}$). At this moment, the velocities of bubble III and bubble IV are 21 mm/s and 29 mm/s, respectively. After the coalescence ($t+122 \text{ ms}$), the newly formed bubble (bubble V) migrates in the same direction as the bubble IV, which has a smaller diameter and but a larger velocity ($t+222 \text{ ms}$). Clearly, when the big bubble has a higher velocity with respect to the smaller bubble, the merged bubble migrates in the direction of the larger velocity and diameter ($t+222 \text{ ms} - t+277 \text{ ms}$).

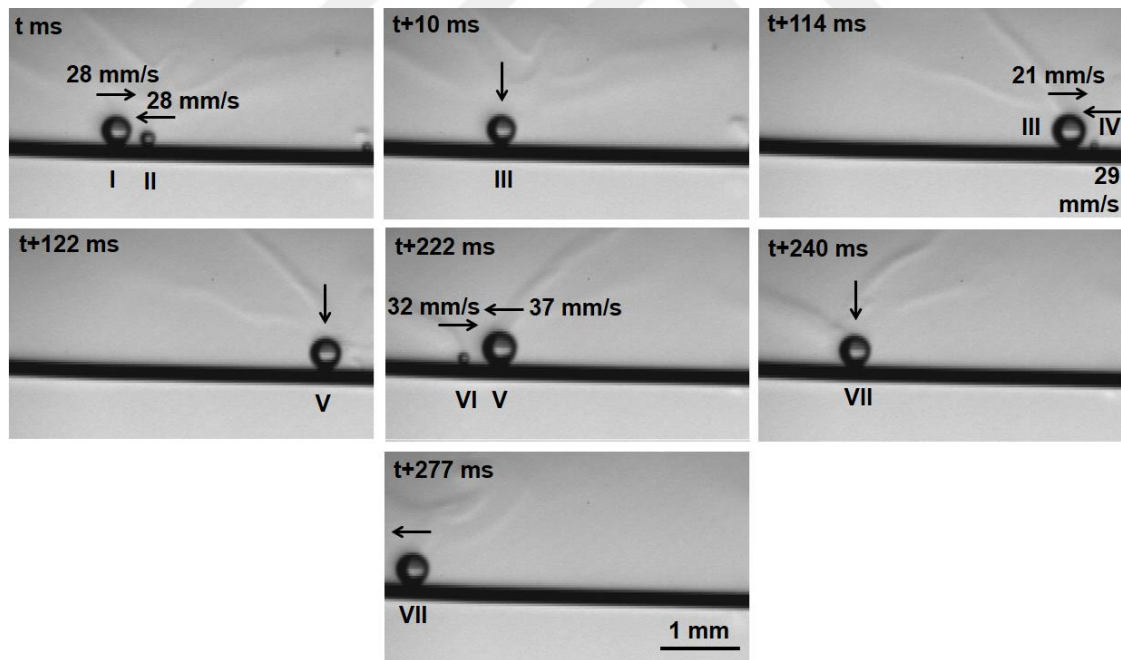


Figure 3.18: Merging bubbles and movement in different directions at a high heat flux. ($T_f=30^\circ\text{C}$, $q''=998 \text{ kW/m}^2$)

For most of the most fluids, surface tension diminishes with increasing temperature. The elevation in the wire temperature gives rise decrease in surface tension

for hot liquid on the wire. The temperature gradient along bubbles interface will be greater at higher heat fluxes, this increases the thermocapillary force driving the bubbles through the high temperature region, where the surface tension is lower. Subsequently, the greater thermocapillary force causes an increase in the velocity of the bubbles. On the other hand, the viscosity decreases with increasing temperature therefore, the drag force in higher heat fluxes is weaker compared to that in lower heat fluxes.

In the case of lower heat fluxes, considering when bubbles are at the closest position, where the moving bubble stops, the combined effect of inertia and thermocapillary forces is not enough for bubbles to overcome the capillary threshold for coalescence. Bubbles as the hottest spots generate a radial temperature profile in liquid around them [122,238,239]. Beyond a certain threshold, thermocapillary force drives the moving bubble into the other bubble. The bubbles have a certain surface energy and always tend to minimize it. The tendency to minimize surface free energy leads the bubbles to form spherical or spherical-like shapes due to the minimum surface area to volume rate of such shapes. The total surface free energy of two bubbles will be the minimum when they merge into one single bubble by shrinking the surface area. The radius of the merged bubble is expected to be smaller than the sum of the two moving bubble radii. Interestingly, based on the observations, the merged bubble becomes even smaller within a very short time period, Figure 3.19.

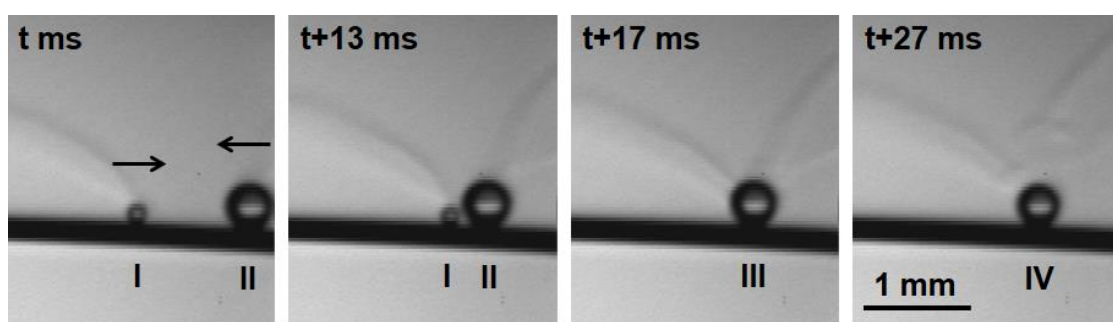


Figure 3.19: Shrinkage of merging bubbles (I and II) just after merging (bubble III) ($t+17$ ms) (down to 41%) and after a while (bubble IV) ($t+27$ ms) (down to 53%) by assuming a bubble as a sphere. ($D_1 = 0.450$ mm, $D_2 = 0.165$ mm, $D_3 = 0.430$ mm, $D_4 = 0.400$ mm) ($T_f=30^\circ\text{C}$, $q''=998$ kW/m²)

The coalescence process may also trigger the condensation of merged bubble due to its impact, which results in alteration in the accumulation area of the non-condensable gases. The change in the tails of the bubbles provides insights to the analysis of the

coalescence in detail as can be seen in Figure 3.20. When the two moving bubbles merge, the tails of the initial moving bubbles are still present. However, it is not clear whether the tails are still attached to the bubble or just traces of previous ones. After coalescence, within a very short period of time, the tails rise rectilinearly while having the highest lifting velocity right above the merged bubble. The inelastic collision and coalescence impact might be the reason for both condensation of the merged bubble and the maximum tail rising velocity. As mentioned before, considering a moving bubble, Marangoni flow would not be strong enough to move the bubble unless non-condensable gases are present to reduce the condensation effects. A small amount of non-condensable gases could reduce the condensation heat transfer coefficient by more than 90% [62,64]. However, they probably have little effect on evaporation coefficient due to the tendency of non-condensable gases to accumulate at the condensing surface [66].

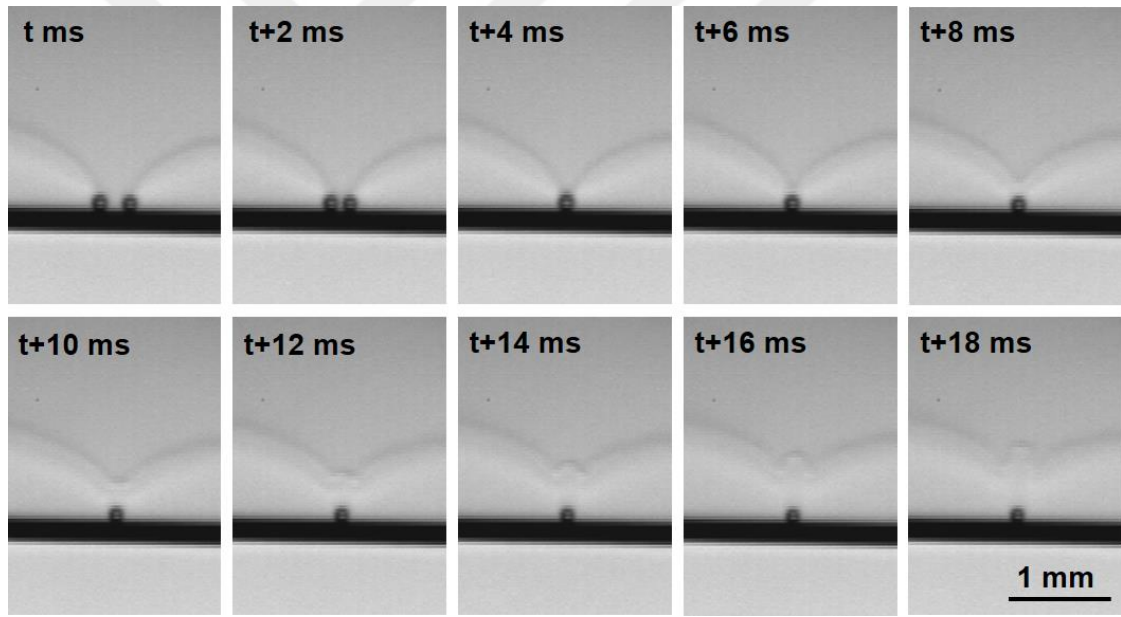


Figure 3.20: Tails of bubbles during merging. Bubbles after merging hesitate for a couple of ms and during this hesitation period, the tails of initially moving bubbles resemble to a bird's wings. The tail above the merged bubble rises with a higher speed due to inertia force due to merging while a shape change in vertical direction. ($T_f=30^\circ\text{C}$, $q''=869 \text{ kW/m}^2$)

The impact of the inelastic collision and coalescence may affect the non-condensable gases accumulation area and makes the bubbles evaporate less or/and condense more within a very short period of time. Meanwhile, the Marangoni flow near the merged bubble is not as intense as near a moving bubble since condensation is strong

enough to decrease the temperature gradient on the bubble surface. Therefore, two remaining tails rise right after the coalescence. In addition to that, because of the inelastic collision impact, the merged bubble deforms and tends to bounce fast. As a result of this tendency, the maximum lifting velocity (49 mm/s) of the remaining tails is observed right above the bubble cap.

3.1.2.4. Leaping

At high heat fluxes ($q'' > 1000 \text{ kW/m}^2$), not only horizontal but also vertical bubble motion is observed. The bubble departs from the wire and changes its size until it reaches the maximum rise height then returns to the wire again, all of which occurs periodically. The physics of the leaping process is still not well understood. Two types of leaping processes can be observed: leaping on the nucleation jet flow without motion in horizontal direction (Case 1), Figure 3.21, leaping with lateral migration (Case 2), Figure 3.22. Case 1 is usually observed at fixed nucleation sites with a bubble which has a diameter much smaller than the wire diameter. Figure 3.21 presents Case 1 at the bulk temperature of 30°C and heat flux of $q'' = 1036 \text{ kW/m}^2$. Following the events of a bubble explosion at the nucleation site ($t = 0 \text{ ms}$) and the nucleation jet flow ($t + 1 \text{ ms}$), a small bubble appears on the wire ($t + 11 \text{ ms}$). The bubble suddenly jumps away from the wire in the vertical direction along with a decrease in its size ($t + 13 \text{ ms}$) and then returns to the wire with an increase in its size ($t + 17 \text{ ms}$). Again, the bubble hops from the wire ($t + 19 \text{ ms}$) and returns back ($t + 24 \text{ ms}$). This cyclic process ends with bubble migration or a transition in behavior to hopping with sliding.

The occurrence of Case 1 depends on the nucleation jet velocity and is usually observed after the eruption at the nucleation site. The bubble lifts off from the wire with the aid of the jet impact and then condenses. However, even at the highest bubble position, large temperature gradients are still present to generate a thermocapillary force on the bubble acting on liquid/vapor interface and thus the bubble is repelled back to the wire. Furthermore, non-condensable gases in the bubble may facilitate the bubble reattachment process by increasing the temperature gradients [64].

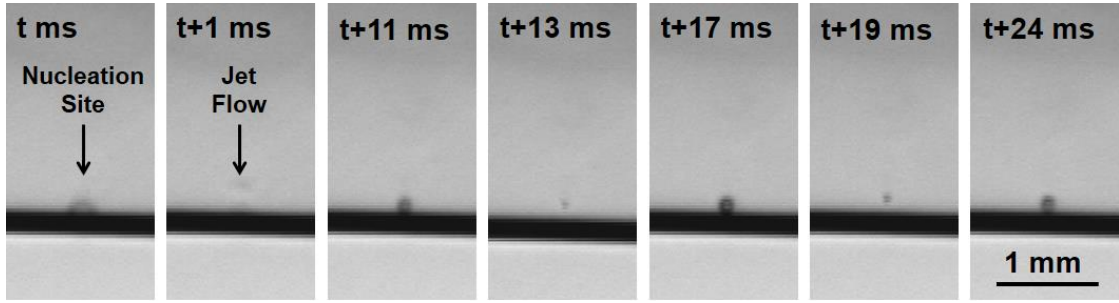


Figure 3.21: Leaping on the nucleation jet flow without moving in vertical direction (Case 1). ($T_f=30^\circ\text{C}$, $q''=1036 \text{ kW/m}^2$)

The typical leaping with sliding process (Case 2) is illustrated in Figure 3.22 at the bulk temperature of 30°C and heat flux of $q''=1132 \text{ kW/m}^2$. A bubble with a diameter of 0.500 mm , which is attached to the wire has a tail different from a typical sliding bubble tail, can be seen. This bubble then moves slightly in both horizontal and vertical directions along with a decrease in size ($t+5 \text{ ms}$). It rises up to the height of 0.050 mm from the wire. After that, it reattaches to the wire and moves on the wire along with an increase in its size up to approximately the same as the initial size ($t+8 \text{ ms}$). A second leaping event pursues the first one ($t+10 \text{ ms} - t+16 \text{ ms}$). This bubble leaping with a sliding process continues until the bubble encounters another bubble, Figure 3.23, or in some cases nucleation jets, Figure 3.24.

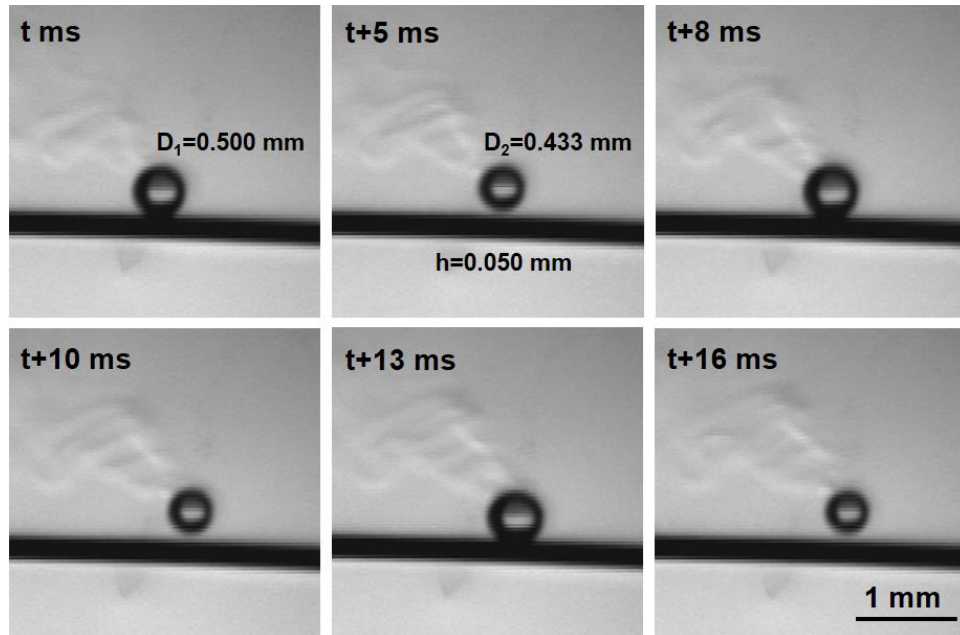


Figure 3.22: Leaping big bubble on the wire in pure water (Case 2). ($T_f=30^\circ\text{C}$ and $q''=1132 \text{ kW/m}^2$)

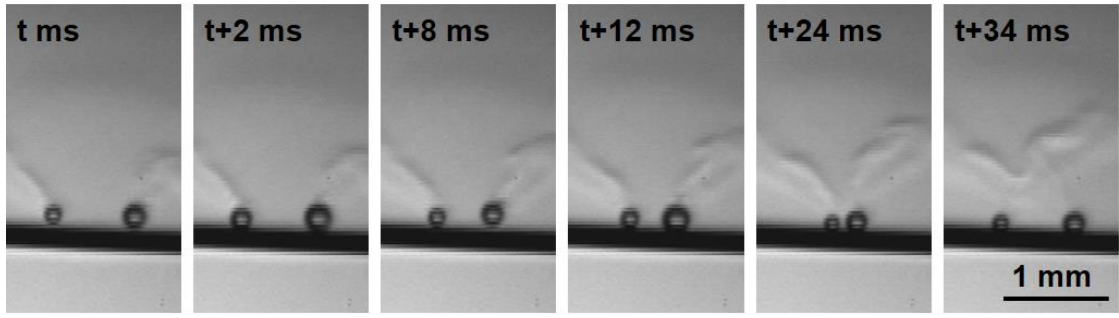


Figure 3.23: Collision of two leaping bubbles. The bubbles continue their lateral migration without leaping. ($T_f=30^\circ\text{C}$, $q''=1314 \text{ kW/m}^2$)

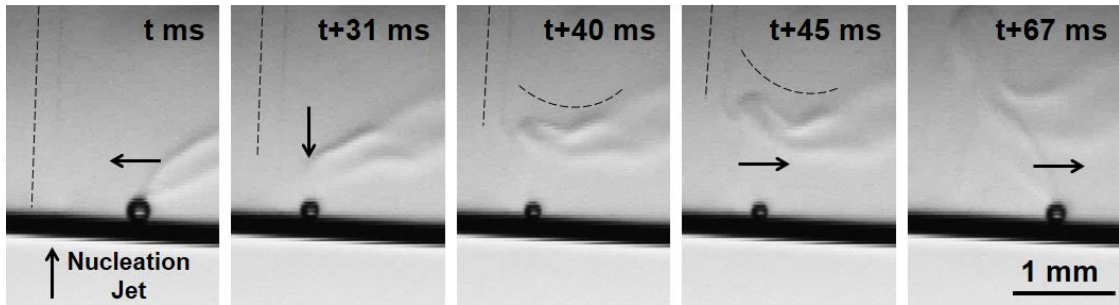


Figure 3.24: A leaping bubble approaches to a nucleation jet, it decelerates, stops ($t+31 \text{ ms}$) for 9 ms (its tail is rising by means of nucleation jet inertia force), shrinks in size and starts to migrate against its previous direction. ($T_f=30^\circ\text{C}$, $q''=1223 \text{ kW/m}^2$)

The observations indicate that both cases depend on the jet flows at the nucleation sites. Not only buoyancy force but also the jet impact can initiate Case 2. Under some circumstances, when the bubble approaches the nucleation site, Figure 3.25, Case 2 initiates. However, it is worth mentioning that a latent nucleation site may be present in the pathway, which might be recognized by using a 3D visualization technique, while Case 2 generally starts when a moving bubble passes the nucleation site, Figure 3.26, or a bubble, which completes its growth process at the nucleation site, starts sliding with leaping, Figure 3.27.

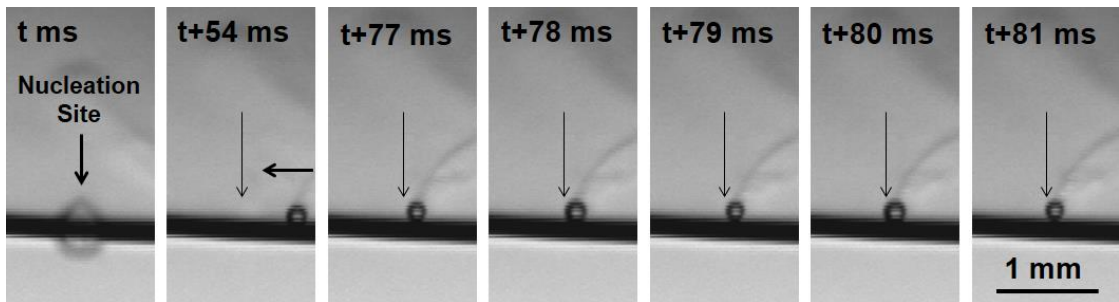


Figure 3.25: Leaping on the nucleation jet flow with movement in horizontal direction. ($T_f=30^\circ\text{C}$, $q''=1314 \text{ kW/m}^2$)

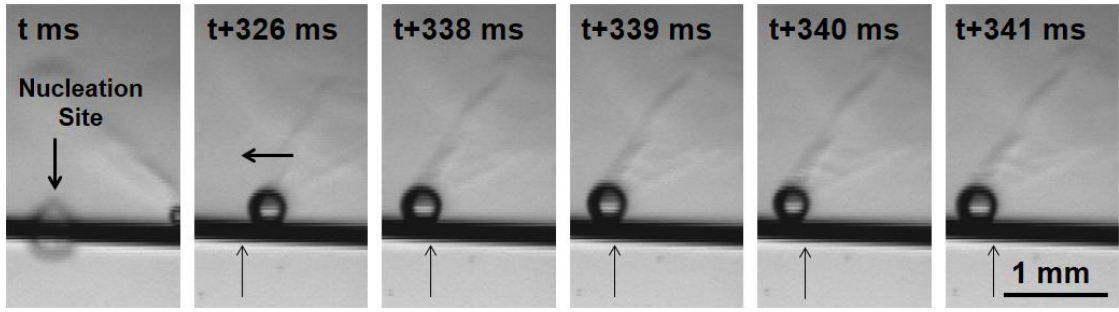


Figure 3.26: Leaping bubble after passing above the nucleation site. ($T_f=30^\circ\text{C}$, $q''=1314\text{kW/m}^2$)

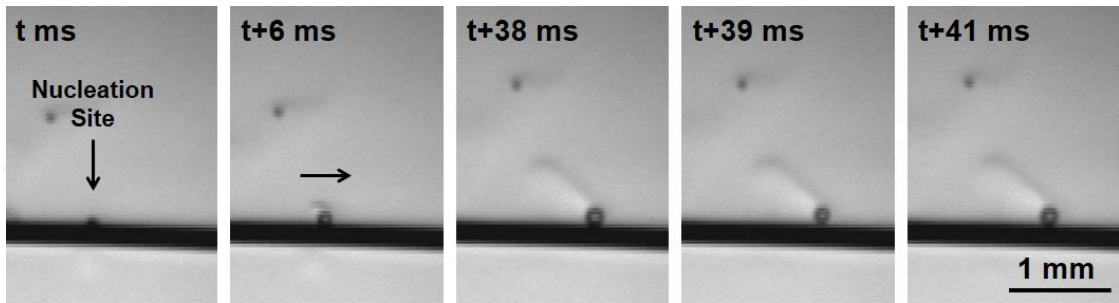


Figure 3.27: Leaping bubble after nucleation from the nucleation site and migration. ($T_f=30^\circ\text{C}$, $q''=1036\text{ kW/m}^2$)

The jet flow on the nucleation site feeds the bubble passing on the nucleation site by adding vapor to it. The bubble suddenly gets bigger at this position. At the same time, continuous jet flow provides a lift force for the bubble. A combined effect of the buoyancy force and the lift force from the jet flow causes the moving bubble to rise. When the bubble departs from the wire, it becomes smaller and the thermocapillary thrust force (now larger than the lifting effects) repels the bubble back to the wire. Again, the effect of the accumulation position of the non-condensable gases may also be a prominent factor. When the amount of non-condensable gases is small or moderate and their accumulation area is not around the bubble cap, where the bubble is exposed to cooler liquid, the temperature distribution on the bubble surface becomes more uniform, which results in a weaker thermocapillary thrust force [62]. Additionally, phase change heat transfer deteriorates with the presence of non-condensable gases in the bubble [64]. When two leaping bubbles encounter each other, they usually merge, Figure 3.28, and the merged bubble shrinks while it stops leaping. Then, the merged bubble starts to migrate without leaping. However, under some conditions, when two leaping bubbles approach each other, they start shrinking before the collision. They continue shrinking while

approaching each other. After the collision, both of them reverse their directions, Figure 3.23.

Regarding bubble detachment, it occurs when a bubble reaches a critical size, where buoyancy overcomes capillary forces. According to the observations, not only buoyancy force but also lifting forces, which arise from the nucleation jets, Figure 3.29, and coalescence impact, Figure 3.30, can initiate the bubble detachment, whereas the thermocapillary thrust force acts as the major resistive force.

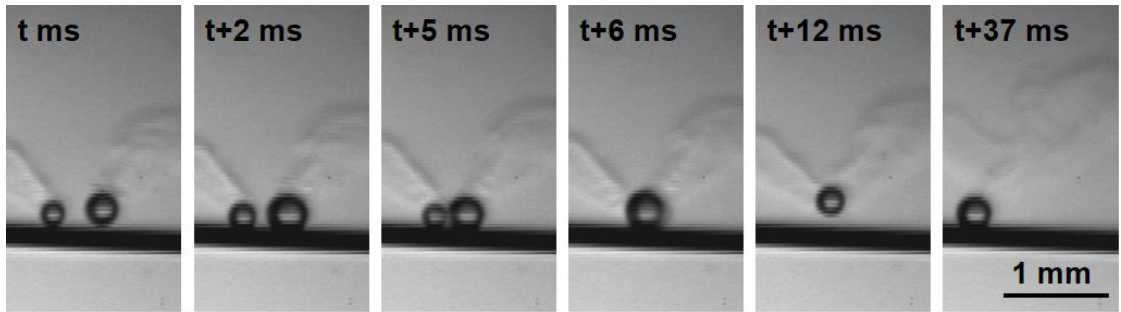


Figure 3.28: Merging two leaping bubbles. The merged bubble continues its lateral migration without leaping. ($T_f=30^\circ\text{C}$, $q''=1314 \text{ kW/m}^2$)

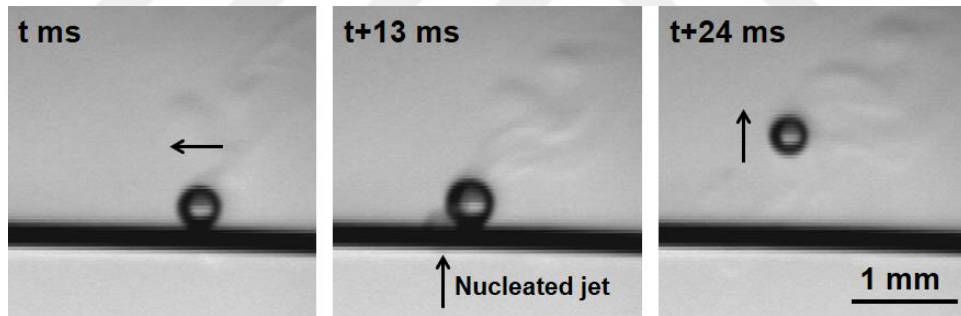


Figure 3.29: Detachment of a bubble due to nucleated jet from a nucleation site. ($T_f=30^\circ\text{C}$ and $q''=1113 \text{ kW/m}^2$)

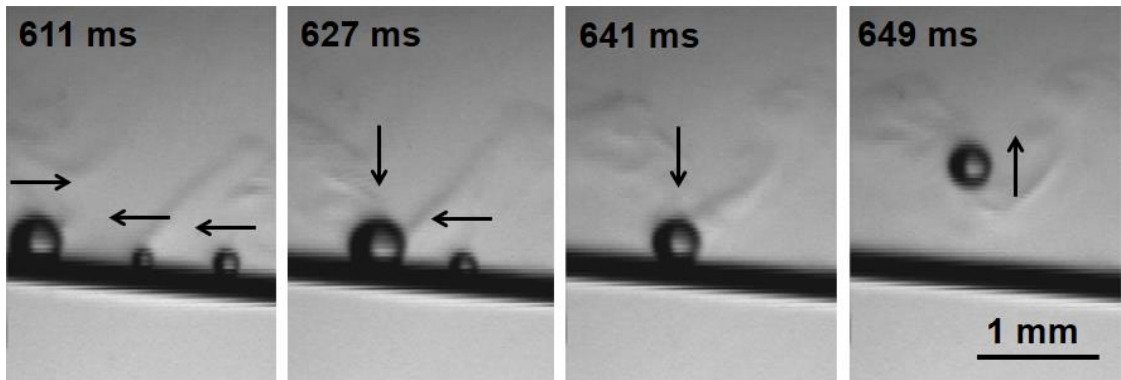


Figure 3.30: Merging small bubbles and detachment of the merged big bubble in pure water. ($T_f=30^\circ\text{C}$, $q''=1396 \text{ kW/m}^2$)

3.2.2. TiO₂ Nanoparticles/Water Nanofluid with Mass Fraction of 0.002 wt. %

3.2.2.1. Lateral migration

Upon inception, bubbles grow up to a critical size at nucleation sites. Collision is usually observed at relatively lower heat fluxes in NF-I. After motion, depending on the existence of another bubble in the vicinity or a cold source (wall) at the end of the wire, bubbles migrate towards another bubble along the wire and experience acceleration, constant speed and deceleration, respectively, under the influence of the forces, which will be discussed later on. When the moving bubble encounters another bubble, it reverses its direction and continues its migration in the other direction, along with acceleration. An increase in heat flux results in an increase in bubble velocity. A bubble migration is demonstrated in Figure 3.31 for NF-I bulk temperature of 40°C and heat flux of $q''=423$ kW/m². A bubble approaches with a constant velocity of 26 mm/s (t ms) to a stationary bubble, decelerates until it stops ($t+232$ ms), reverses its direction, accelerates up to a velocity of 24 mm/s and carries on moving.

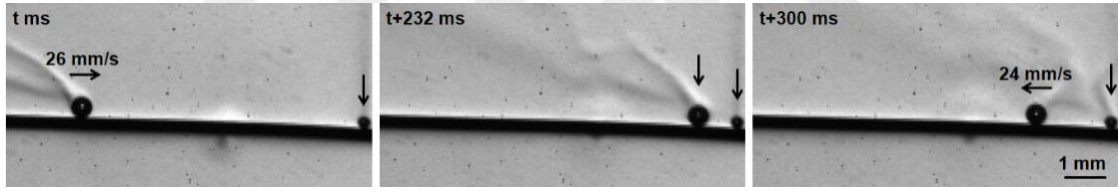


Figure 3.31: Bubble lateral migration and encounter. ($\phi=0.002$ wt.%, $T_f=40^\circ\text{C}$, $q''=423\text{kW/m}^2$)

A newly formed bubble is subjected to heat from its base and grows through evaporation. Unlike saturated boiling, when a growing bubble reaches to a certain size, at which the bubble cap is exposed to cooler liquid, it tends to shrink in subcooled boiling. When condensation from the bubble cap is hindered, convection, which is called thermocapillary convection occurs from a higher to lower temperature zone along the interface. Marangoni convection originates from surface tension gradient at the interface between liquid and vapor phases, which depends on either concentration or temperature. The temperature gradient between the fluid surrounding bubble base (hot liquid layer) and cap causes thermocapillary convection, and concentration gradients between the surrounding nanofluid of bubble base and top generates solutocapillary effect. Marangoni convection is a combination of thermocapillary and solutocapillary effects. The resulting

event is called top jet flow and has a tail-like appearance on the bubble. It is speculated that the presence of non-condensable gases inside bubbles, which decreases condensation heat transfer coefficient and disturbs the uniformity of surface temperature distribution, gives rise to Marangoni convection, which has an important role in heat transfer [62]. Although this phenomenon depends only on the temperature in pure liquids (thermocapillary), both of the concentration and temperature affect Marangoni convection in nanofluids [95].

The magnitude of each force depends on experimental conditions in the system. Buoyancy force acts in an upward direction, while both gravity and the vertical component of Marangoni forces act in a downward direction. Microgravity studies showed that gravity force had a little effect on bubbles compared to Marangoni force [240–243]. It would be more accurate to label this as Marangoni thrust force, instead of Marangoni force since it is believed that Marangoni convection exerts a thrust force on the bubble. The vertical component of Marangoni force prevents the bubble from rising under the influence of buoyancy [66,79,80,83,86,91,93].

Regarding the horizontal components of the forces acting on bubbles, the horizontal component of Marangoni force acts on the bubble to promote migration. On the other hand, drag friction forces act on the bubble in the opposite direction. Furthermore, the contact line force acts at the interface, tangent to the surface around the bubble and changes its magnitude with a change in advancing and receding contact angles in the case of a moving bubble due to differences in these contact angles. The horizontal component of Marangoni force plays a major role in the migration of bubbles on the wire. In addition, as the bubble moves, the temperature in front of the bubble will be higher than temperature behind the bubble. Therefore, Marangoni convection is anticipated to be more vigorous in front of the bubble thus generating a pressure gradient [83].

Drag force acts on the moving bubble in the opposite direction of the horizontal component of Marangoni force. In the acceleration process, Marangoni force is greater than the drag force owing to the reason that temperature gradient in front of the bubble is higher than that behind the bubble, thereby generating a stronger Marangoni force. The drag force increases with bubble velocity and with bubble diameter during the growth process. Meanwhile, as the bubble approaches another stationary bubble, the temperature gradient decreases resulting in a decrease in Marangoni force, since the stationary bubble acts as a cold source. When Marangoni and drag forces balance each other, the velocity

of the bubble becomes constant. When the moving bubble encounters the stationary bubble, temperatures in front of the bubble becomes lower than those behind the bubble. As a result, the bubble stops. After that, the bubble moves to the opposite direction. With the addition of nanoparticles to pure water, additional forces come into play.

Nanoparticle deposition is expected to be more pronounced at higher concentrations in this study. Thus, the friction force in NF-II should be greater than NF-I resulting in an increase of sticking bubbles on the wire. Deposition of TiO_2 nanoparticles on the heated surface, increases the wettability, hence increasing the contact angle. Therefore, in addition to drag force, contact line force and friction force are additional resistive forces in the case of nanofluids as working fluids. Furthermore, increasing nanofluid concentration promotes these resistive forces.

The porous layer due to nanoparticle deposition depends on concentration, heat flux and surface temperature [28]. It is known that increasing the concentration leads to an increase in surface wettability, the possibility of filling the cavities with liquid, and subsequent decrease in the number of nucleation sites. Due to collection of nanoparticles in a confined wedge, structural disjoining pressure is introduced as a mechanism along with the pinning effect in the contact line region [31,74,244].

Our observations indicate that two different scenarios for bubble interactions exist; a bubble moving towards a stationary bubble (Case 1), Figure 3.31, and two bubbles initially moving in the same direction (Case 2), Figure 3.32. It is worth mentioning that Case 2 is often observed when the leading bubble is smaller than the bubble following leading it. These cases were also observed in other studies, where the working fluid was water [79,80,85,93,245]. As can be seen in Figure 6, at t ms, a stationary bubble (bubble I) with a velocity of 35 mm/s approaching a nucleation site where the jet flow is evident. Then, the stationary bubble encounters with a newly formed bubble (bubble II) at the nucleation site ($t+24$ ms), bubble II moves away from bubble I, because bubble I perturbs the uniform temperature distribution around bubble II and generates Marangoni force on bubble II enough to overcome the contact line force, while bubble I continues its motion ($t+31$ ms). Bubble II continue growing while moving, since it absorbs heat from the wire and hot liquid layer, hence the temperature of the wire becomes lower. The decrease in the wire temperature has two effects on bubbles: the speed of bubble II is reduced, while the speed of bubble I speed increases. When bubble II is far enough away from bubble I, it stops. On the other hand, bubble I continues with its motion at the same time ($t+47$ ms).

After that, a second encounter of bubbles occurs at $t+72$ ms. However, this time, bubble I reverses its direction, whereas bubble II continues its motion.

Bubbles can migrate not only above (and along) the wire, but also under the wire as seen in Figure 3.33. However especially at low heat fluxes, bubbles usually migrate above the wire due to buoyancy. There is weaker Marangoni convection under the wire due to the fact that boundary layer under the wire is smaller than that on top of the wire. Increasing heat flux causes an increase in the number of nucleation sites and creates enough temperature gradients under the wire.

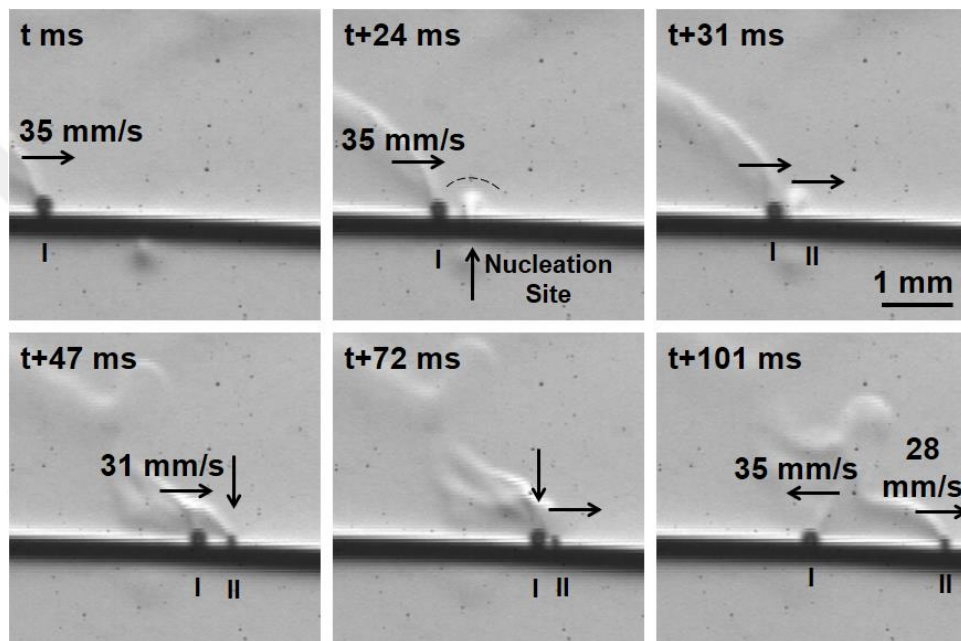


Figure 3.32: Encounter of moving bubble (I) with a newly nucleated bubble (II) after pre-jet event ($t+24$ ms), second encounter of the same bubbles ($t+72$ ms) and change in their directions ($t+101$ ms). ($\phi=0.002$ wt.%, $T_f=30^\circ\text{C}$, $q''=739$ kW/m²)

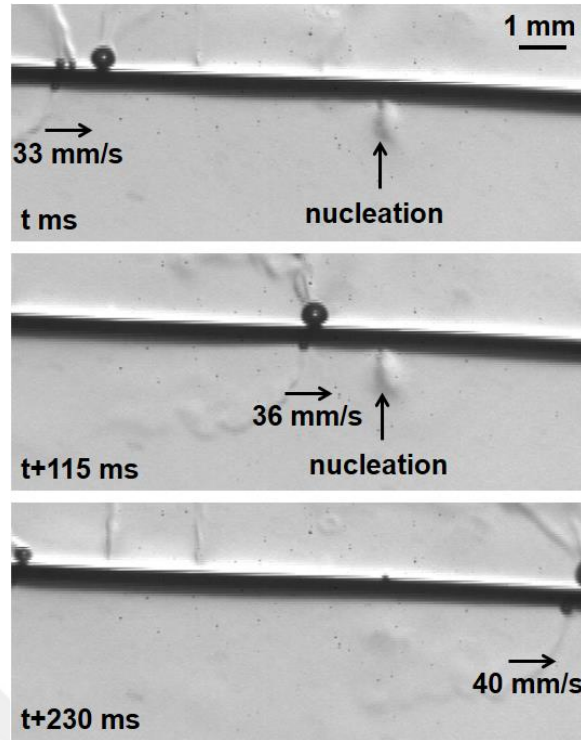


Figure 3.33: Nucleation and lateral migration of a bubble under the wire in the nanofluid. ($\phi=0.002$ wt.%, $T_f=35^\circ\text{C}$, $q''=1160$ kW/m²)

3.2.2.2. Bubble coalescence

Based on our observations, different types of coalescence take place: two bubbles moving towards each other (Case 1), Figure 3.34(a), a smaller bubble following a bigger one (Case 2), Figure 3.34(b), a bigger bubble following a smaller one (Case 3), Figure 3.34(c), two leaping bubbles moving towards each other, (Case 4), Figure 3.34(d), two leaping bubbles moving in the same direction (Case 5), Figure 3.34(e). The common feature of all the cases is that the size of the eventually formed bubble after the coalescence is smaller than the sum of the sizes of original bubbles, sometimes even smaller than the size of the originally bigger bubble, due to the mass transfer after coalescence in the form of flung small bubble or vapor pack.

Figure 3.34(a) shows the coalescence of two bubbles as well as a small remaining bubble after the coalescence in NF-I at a bulk temperature of 30°C and a heat flux of $q''=869$ kW/m². Two bubbles moving towards each other (t ms and $t+7$ ms). After coalescence of these two bubbles ($t+8$ ms), interestingly, a small bubble (bubble IV) is flung out of the newly formed bubble (bubble III). Then bubble IV continues growing ($t+9$ ms) at the same position, while bubble III moves away from bubble IV. This

observation indicates that not only momentum of initially moving bubbles but also remaining bubbles could determine the migration direction of the formed bubble since the remaining bubble can alter the wire temperature and guide the formed bubble. Similarly, this event can also be seen in Figure 3.34(b) and Figure 3.34(d). Nevertheless, sometimes instead of a bubble, a vapor pocket can be flung, as seen in Figure 3.34(c) and Figure 3.34(e).

The surface tension for liquids decreases as the temperature increases. At moderate heat fluxes, $q'' > 450 \text{ kW/m}^2$, the temperature gradient near the bubbles is greater resulting in an increase in Marangoni force, which drives the bubbles to high temperature regions with lower surface tension. Velocity and momentum of the bubbles increase with greater Marangoni force. Moreover, an increase in the temperature leads to a decrease in the velocity, subsequently, the drag force at moderate heat fluxes is lower. A radial temperature profile occurs around the bubbles where the bubble itself is the hottest [122,238,239].

Generally, bubbles adopt spherical or spherical-like shapes in order to minimize their surface free energy. When coalescence takes place, the surface free energy of the merged bubble is lower than the total free surface energy of the separate bubbles. In fact, adding nanoparticles changes surface tension of bubbles, thereby affecting the coalescence process. At relatively low heat fluxes, when the moving bubble encounters another bubble, it reverses its direction and continues its migration to the other direction, along with acceleration. In this case, the inertia of the moving bubble cannot overcome the reversal in Marangoni force: Thus, moving bubble reverses its direction [85]. In another case, a stationary bubble starts to move away from the approaching bubble due to the fact that the temperature distribution around the stationary bubble is now not symmetric leading to a Marangoni force, which overcomes the contact line force [85]. Increasing the heat flux typically results in an increase in the bubble speed. Hence, coalescence takes place instead of repulsion.

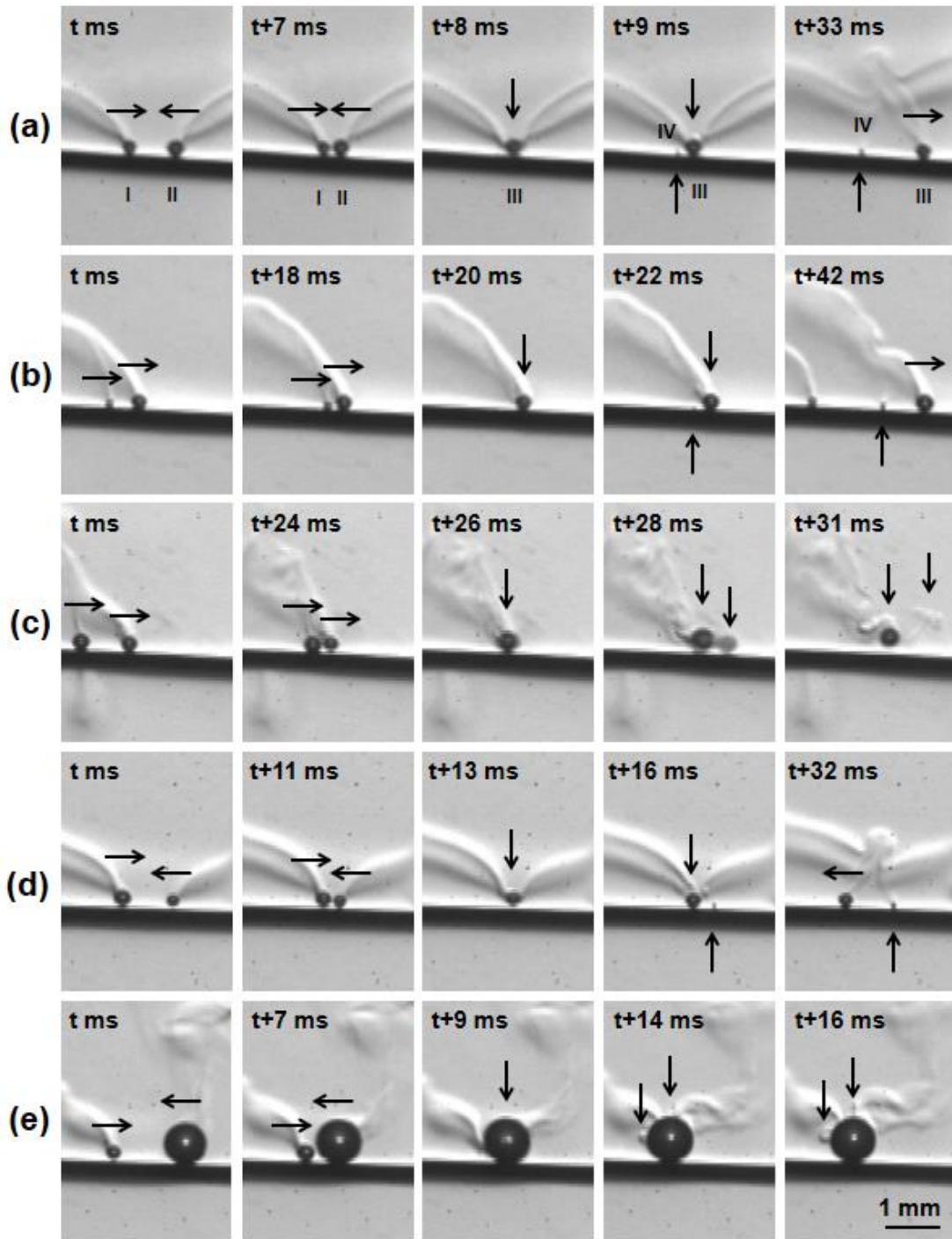


Figure 3.34: Cases corresponding to: (a) coalescence of two moving bubbles in the opposite direction ($T_f=30^\circ\text{C}$, $q''=869\text{ kW/m}^2$), (b) coalescence of two moving bubbles in the same direction ($T_f=40^\circ\text{C}$, $q''=1003\text{ kW/m}^2$), (d) coalescence of two leaping bubbles in the opposite direction ($T_f=35^\circ\text{C}$, $q''=888\text{ kW/m}^2$), and cases with jets after: (c) coalescence of two moving bubbles in the same direction ($T_f=40^\circ\text{C}$, $q''=1003\text{ kW/m}^2$), and (e) coalescence of two leaping bubbles in the same direction ($T_f=35^\circ\text{C}$, $q''=1029\text{ kW/m}^2$). ($\phi=0.002\text{ wt.}\%$)

3.2.2.3. Leaping

Our observations indicate that two types of leaping behavior of bubbles exist (at relatively high heat fluxes ($q'' > 900 \text{ kW/m}^2$)): vertical leaping (Case 1), Figure 3.35, vertical and horizontal leaping (Case 2), Figure 3.36. Typically, in Case 1, the bubble jumps away from the wire in the vertical direction and then returns to the wire, which occurs periodically. Behavior described in Case 1 is usually observed at nucleation sites with a bubble, which has a diameter much smaller than the wire diameter. Figure 3.35 presents Case 1 at NF-I bulk temperature of 35°C and heat flux of $q'' = 1029 \text{ kW/m}^2$. The bubble emerges at the nucleation site at $t \text{ ms}$ and suddenly jumps away from the wire in the vertical direction ($t+1 \text{ ms}$). When the bubble is exposed to the cooler fluid, its size diminishes due to condensation ($t+1 \text{ ms}$). Following this, the events of a second hopping ($t+11 \text{ ms}$) and returning back to the wire ($t+16 \text{ ms}$) can be observed. This periodic behavior terminates with bubble detachment or shifts to bubble migration.

Obviously, nucleation jet and its intensity affect the vertical leaping of bubbles. The lift off of the bubbles is initiated by nucleation jet flow. However, the difference between the surrounding fluid temperatures of the bubble base and its cap is still present, and thus, temperature gradient generates a great enough Marangoni force on the bubble to push it along the wire. It is expected that the contact angle of the bubble decreases with the introduction of nanoparticles to water at low concentration. Therefore, the vertical component of the contact line force slightly increases. In addition, non-condensable gases in the bubble may increase the temperature gradient around the bubble and this fact may play a major role in facilitating the bubble reattachment process.

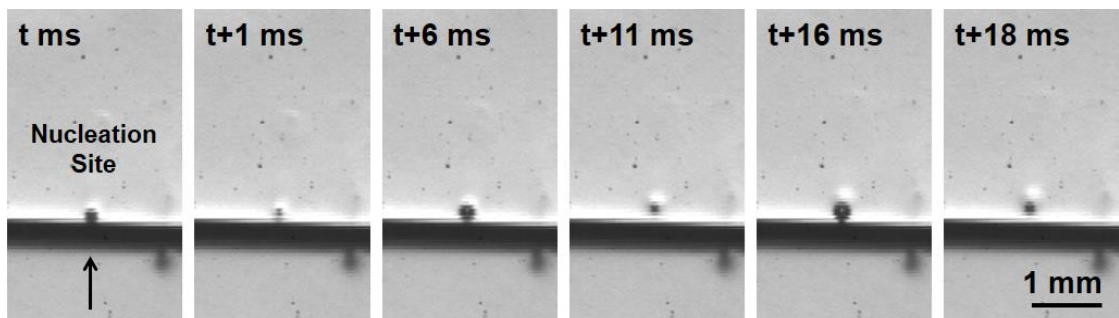


Figure 3.35: Leaping a bubble vertically without moving in horizontal direction.
($\phi = 0.002 \text{ wt.}\%$, $T_f = 35^\circ\text{C}$, $q'' = 1029 \text{ kW/m}^2$)

Figure 3.36 demonstrates Case 2 at a NF-I bulk temperature of 35°C and a heat flux of $q''=979 \text{ kW/m}^2$. The tail of a bubble, which is attached to the wire, with a size of 718 μm (Figure 3.36) is different from a typical tail of a sliding bubble, due to its continuous motion in both horizontal and vertical directions. During the horizontal motion, the bubble detaches from the wire up to a height of approximately 90 μm . There, its diameter decreases slightly down to 656 μm (t+2 ms). Thereafter, the bubble attaches back to the wire and its size increases up to approximately the initial diameter (t+4 ms). This behavior (leaping) continues periodically until the bubble stops leaping because of approaching a nucleation jet (Figure 3.37) or merging another bubble (Figure 3.36).

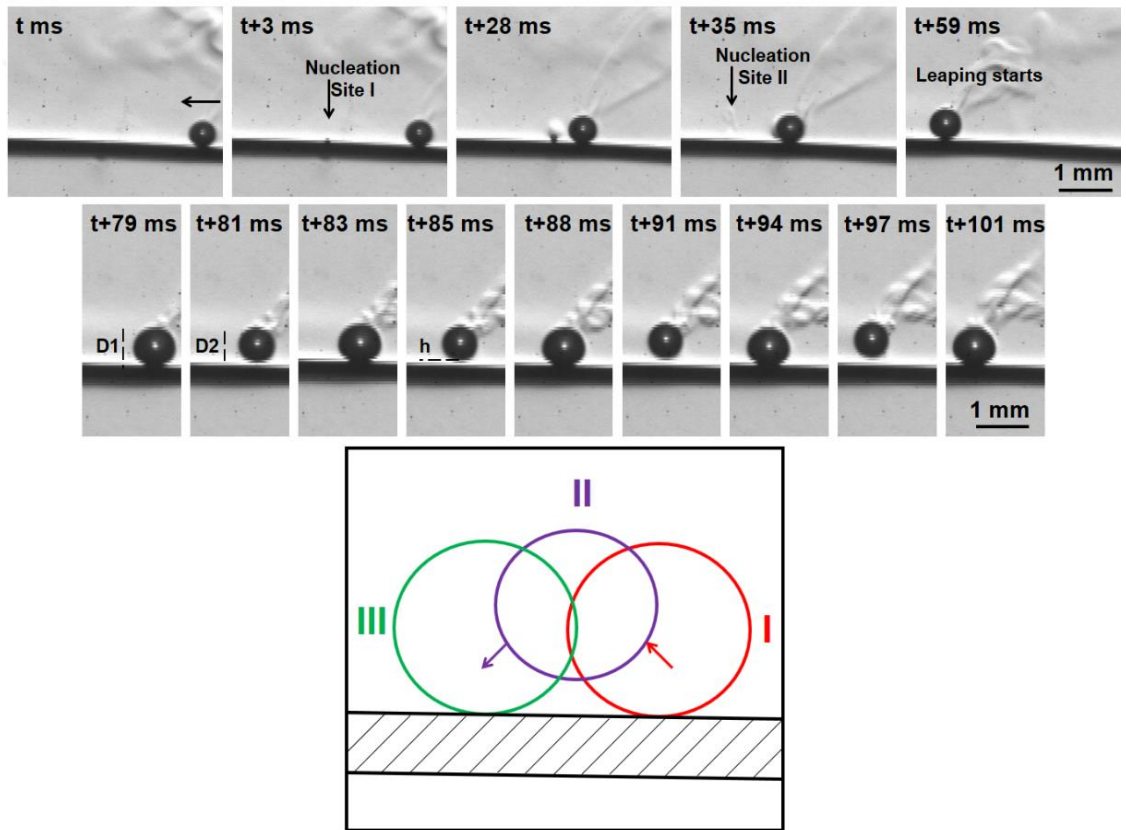


Figure 3.36: Leaping of a big bubble on the wire (above) due to existence of nucleation sites as initiator of leaping process, and schematic of the periodic leaping process at positions I, II and III (below). (Diameter difference during the leaping because of the effect of condensation) ($\phi=0.002 \text{ wt.}\%$, $T_f=35^\circ\text{C}$, $q''=979 \text{ kW/m}^2$)

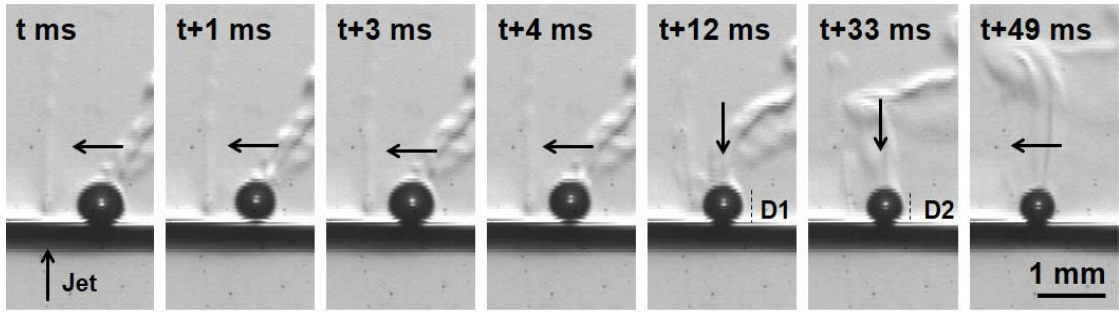


Figure 3.37: A leaping bubble eliminates the jet and its volume decreases down to 14%. ($\phi=0.002$ wt.%, $T_f=35^\circ\text{C}$, $q''=1029$ kW/m²)

Figure 3.38 shows the coalescence of two leaping bubbles in NF-I bulk temperature of 35°C and heat flux of $q''=1160$ kW/m². Two leaping bubbles approach each other (t ms and t+1 ms). When evaluating bubble size there is indication that the bigger bubble shrinks more when it reaches its maximum height in the leaping process just before the coalescence, probably due to the Marangoni convection of the other bubble (t+2 ms). When these two bubbles coalesce at t+5 ms, the newly formed bubble stops and shrinks until it reaches a size which is smaller than the initially leaping bigger bubble, at t+17 ms and then it starts to migrate without leaping.

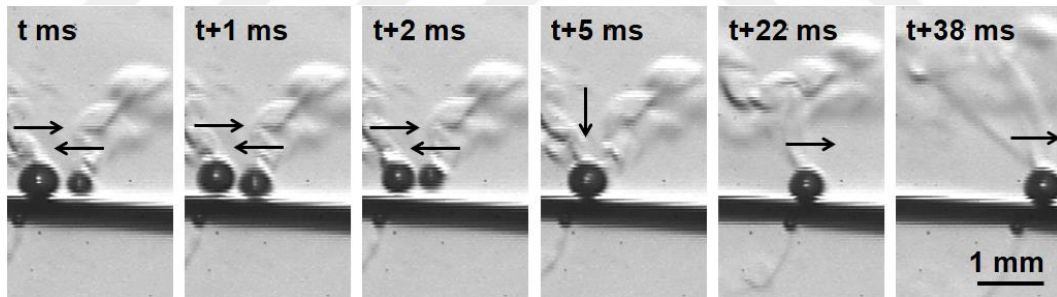


Figure 3.38: Merging of two leaping bubbles and the merged bubble without leaping on the wire in nanofluid. ($\phi=0.002$ wt.%, $T_f=35^\circ\text{C}$, $q''=1160$ kW/m²)

Both Case 1 and Case 2 are usually initiated by nucleation jet flows. Figure 3.39 displays the nucleation jet effect on bubble leaping in NF-I at a bulk temperature of 35°C and heat flux of $q''=1160$ kW/m². A bubble with a size of $345\text{ }\mu\text{m}$ approaches the nucleation jet at t ms and then it stops near the nucleation jet while its diameter decreases down to $326\text{ }\mu\text{m}$ (t+19 ms). After that, it passes the nucleation site while its diameter increases up to $460\text{ }\mu\text{m}$ and leaping starts (t+30 ms). A bubble gets bigger when passing over a nucleation site due to feeding with vapor from the nucleation site. With time, the jet flow applies a lift force. The combination of this force and buoyancy force raises the

bubble above the wire. The diameter of the bubble decreases above the wire due to condensation. Then Marangoni thrust force, which is now larger than the combination of lift and buoyancy forces, pushes the bubble back to the wire. Another important parameter in this process may be the accumulation location of non-condensable gases, which deteriorate phase change heat transfer [62]. Weaker Marangoni thrust force results in the more uniform temperature distribution on the bubble surface due to higher condensation heat transfer coefficient. Leaping can be observed for both mass fractions as well as for the pure base fluid. A major observation regarding the nanoparticle effect is the decrease in contact angle during leaping. The effect of the nucleation site on leaping bubble does not diminish instantly so that the bubble can jump for many times as seen in Figure 3.36.

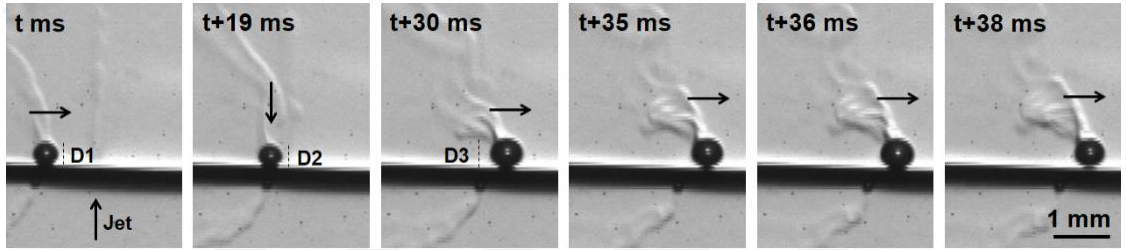


Figure 3.39: Decrease in diameter, leaping and increase in diameter of a bubble due to a nucleation jet. ($D_1=345 \mu\text{m}$, $D_2=326 \mu\text{m}$, $D_3=460 \mu\text{m}$) ($\phi=0.002 \text{ wt.}\%$, $T_f=35^\circ\text{C}$, $q''=1160 \text{ kW/m}^2$)

3.2.2.4. Nucleation jet-bubble interactions

Nucleation jet-bubble interactions have not been paid enough attention in the literature. However, not only inception of the leaping process, Figure 3.39, but also usually detachment of bubbles is affected by nucleation jets. Furthermore, some other interesting phenomena regarding nucleation jet-bubble interactions, such as bubble 3D motion near the nucleation jets, Figure 3.40, and bubble oscillation between two nucleation jets, Figure 3.41 can be observed.

As can be seen in Figure 3.40, in NF-I at a bulk temperature of 35°C and an applied heat flux of $q''=1003 \text{ kW/m}^2$, a bubble with a diameter of $345 \mu\text{m}$ approaches a nucleation site ($t+11 \text{ ms}$). After reaching the nucleation site it stops because the nucleation jet makes the wire cooler there, and its diameter decreases down to $307 \mu\text{m}$ ($t+26 \text{ ms}$). Interestingly, this bubble performs a 3D motion, turns and passes the nucleation site from the side of the wire (from $t+43 \text{ ms}$ to $t+54 \text{ ms}$).

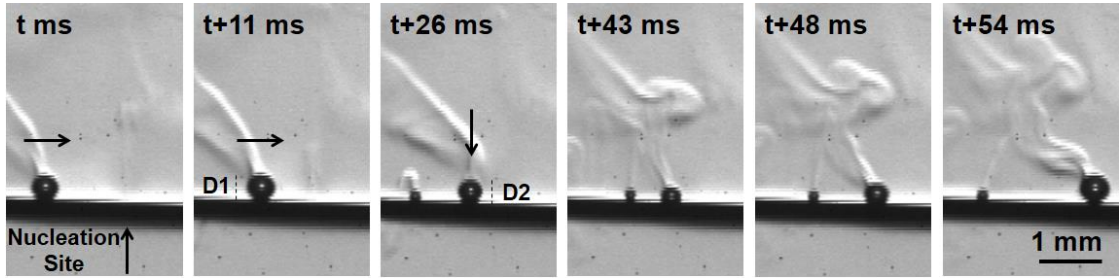


Figure 3.40: 3D turn of a bubble over a nucleation site and decrease in the bubble size. ($D_1=345 \mu\text{m}$, $D_2=307 \mu\text{m}$) ($\phi=0.002 \text{ wt.}\%$, $T_f=35^\circ\text{C}$, $q''=1003 \text{ kW/m}^2$)

In Figure 3.41, a bubble moves on the nucleation jet I at a velocity of 33 mm/s (t ms). When it encounters the nucleation jet I, it stops and its volume decreases (t+38 ms). Thereafter, it reverses its direction and moves to the nucleation jet II at a speed of 35 mm/s along with an increase in its volume (t+73 ms). Likewise, when it encounters the nucleation jet II, it stops and its volume decreases (t+101 ms). This bubble oscillation process continues until this bubble coalesces with another moving bubble.

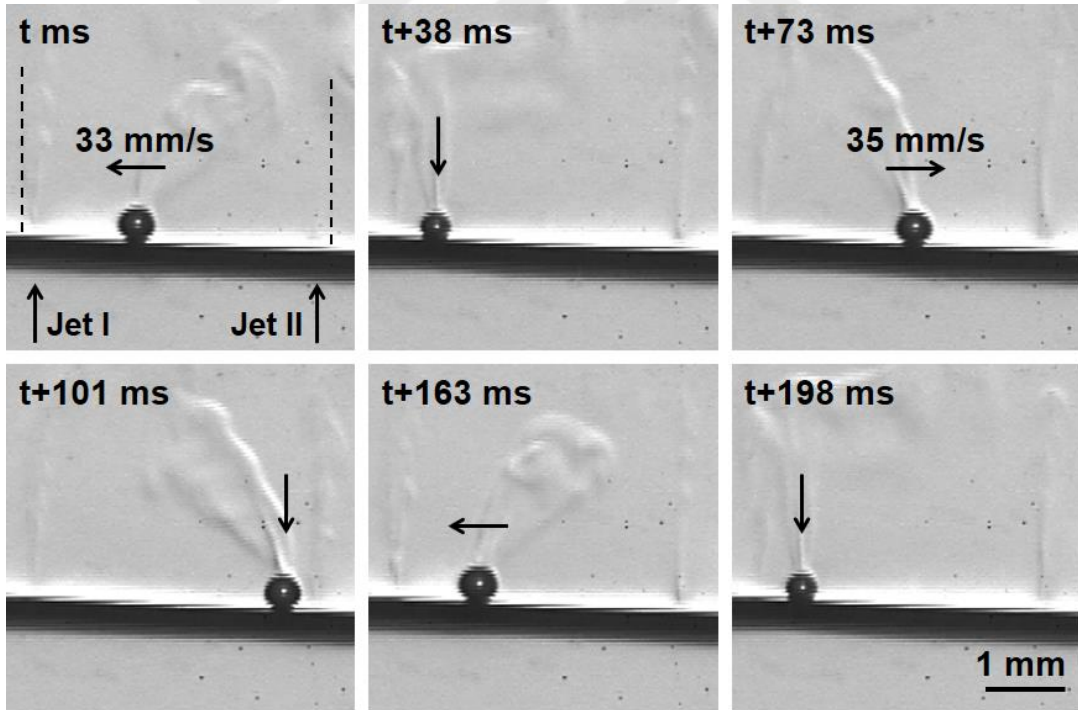


Figure 3.41: Oscillation of a bubble between two nucleation jets. ($\phi=0.002 \text{ wt.}\%$, $T_f=35^\circ\text{C}$, $q''=1160 \text{ kW/m}^2$)

The detachment process occurs when a bubble reaches a critical size and buoyancy force overcomes capillary forces. Besides the buoyancy force, lifting forces also initiate the bubble detachment process. Lifting forces arise either from the nucleation

jet, Figure 3.42, or coalescence impact, Figure 3.43. In this process, Marangoni thrust force and contact line force act as the major resistive forces.

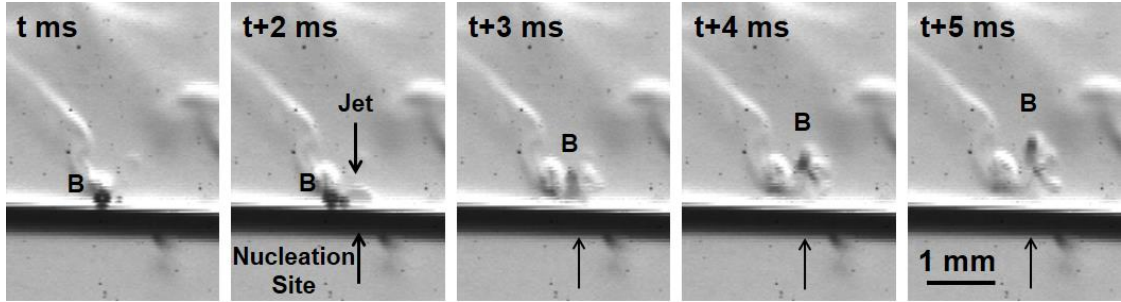


Figure 3.42: Detaching a bubble (B) above the jet nucleation site, and decreasing its diameter after detachment process. ($\phi=0.002$ wt.%, $T_f=35^\circ\text{C}$, $q''=1160$ kW/m²)

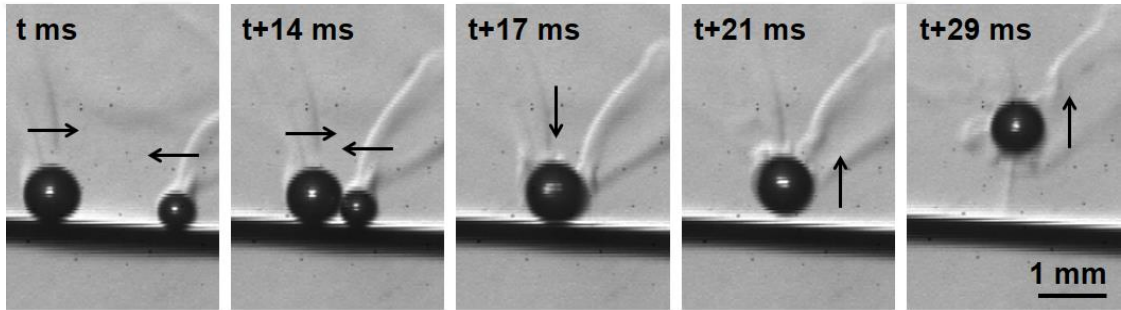


Figure 3.43: Detachment a bubble (t+21 ms) after coalescence (t+17 ms). ($\phi=0.002$ wt.%, $T_f=45^\circ\text{C}$, $q''=745$ kW/m²)

3.2.3. TiO₂ Nanoparticles/Water Nanofluid with Mass Fraction of 0.005 wt.%

Many of the features of bubble dynamics in NF-I are also present in NF-II. Thus, repetition of these observations is avoided. However, it is worth to state that the frequency of the events is not the same. Furthermore, some additional features exist only in NF-II, which will be provided in detail in the following section.

3.2.3.1. Oscillation

In the case of two stationary bubbles close to each other, a bubble in between can oscillate periodically as it goes towards one stationary bubble, collides and reverses its direction to the other bubble. Nanoparticles seem to facilitate this behavior, since deposition of nanoparticles increases the friction force and reduces contact angle, which

raises the contact line force, thereby allowing for the formation of more stationary bubbles. This effect is more obvious in nanofluids with a higher mass fraction. In fact, solid-like ordering of the nanoparticles occurs in the three-phase contact region [76]. An explanatory figure associated with bubble oscillation is given in Figure 3.44 at a NF-II bulk temperature of 35°C and an applied heat flux of $q''=832 \text{ kW/m}^2$. Bubble I oscillates between two stationary bubbles with different sizes. Bubble I approaches the bigger stationary bubble at $t \text{ ms}$ and stops at $t+25 \text{ ms}$. Then it changes its moving direction to the smaller stationary bubble. When it encounters with smaller stationary bubble, again it stops ($t+71 \text{ ms}$) and reverses its direction in the other direction. This cyclic process however terminates suddenly, at $t+230 \text{ ms}$ and bubble I sticks on the wire. Meanwhile, bubble II also oscillates between two stationary bubbles (one stationary bubble is not present in the figure) in the time period between $t+71 \text{ ms}$ and $t+690 \text{ ms}$. By looking at the bubble at $t+117 \text{ ms}$, $t+290 \text{ ms}$ and $t+493 \text{ ms}$, it can be seen that distances are interestingly not the same and the velocity of bubble II changes slightly, probably due to deposition and/or pinning effect of nanoparticles. This event can be observed in different sequences involving different bubble sizes, Figure 3.45. On the other hand, in some cases, bubble oscillations between a stationary bubble and a point without any bubble or nucleation site (probably because of the deposition) is also seen, Figure 3.46.

Additionally, in some other cases, bubble oscillation is observed not only above the wire but also under the wire, Figure 3.47. Furthermore, the inception of oscillations is presented in Figure 3.44. Bubble IV holds still until the time $t+660 \text{ ms}$ under the wire. However, when bubble III returns back after the approach of the stationary bubble, which is above the wire, bubble IV goes upward the wire and starts to migrate and oscillate between the two stationary bubbles.

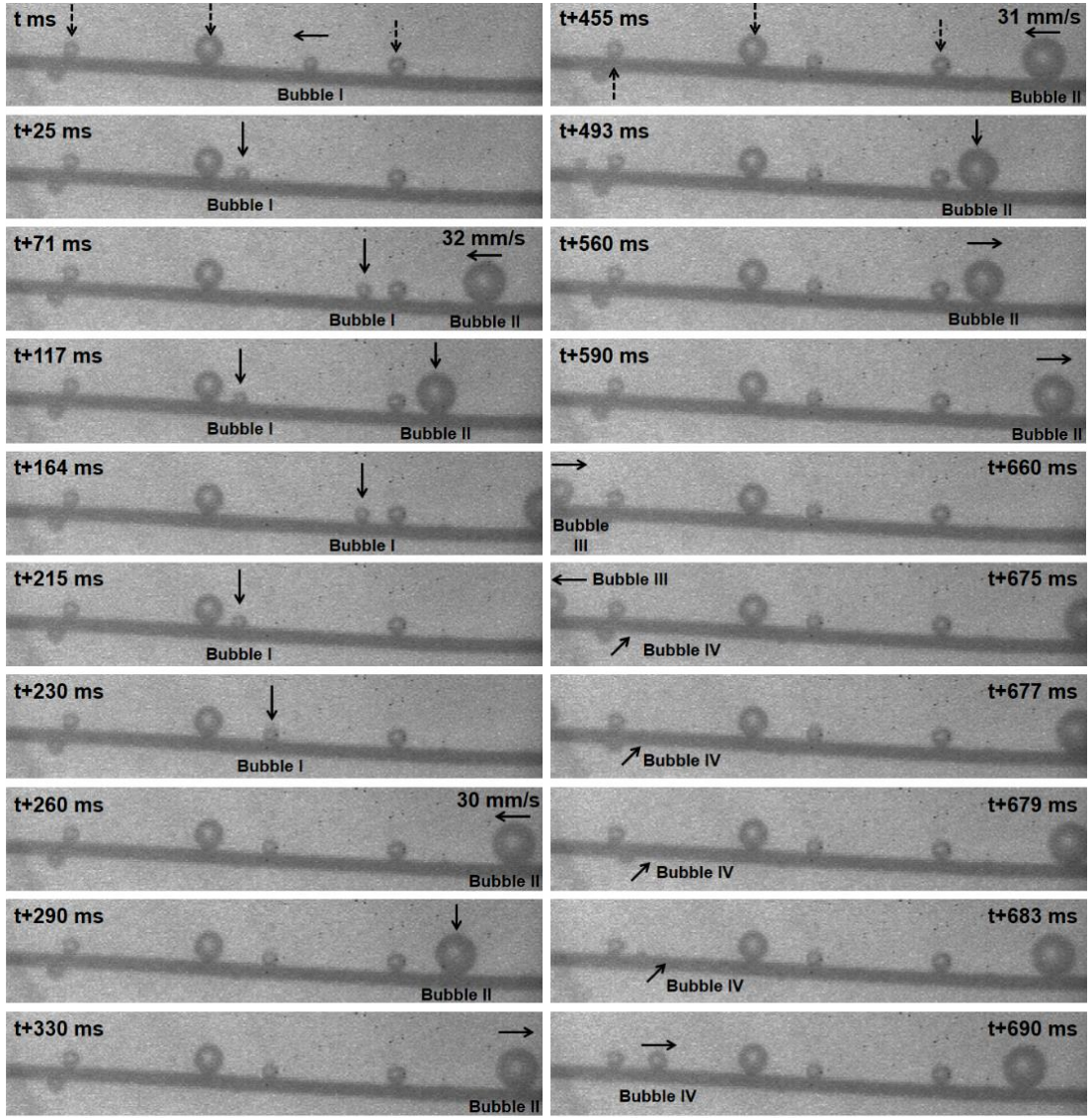


Figure 3.44: Oscillation of bubble I between two stationary bubbles (t ms to $t+215$ ms) and its sticking due to effect of deposition ($t+230$ ms), oscillation of bubble II from different positions due to nanoparticle deposition or pinning effect ($t+290$ ms position versus $t+560$ ms), turning of stopped bubble IV at $t+675$ ms due to Marangoni flow of bubble III ($t+660$ ms). (Dash-lined arrows represent the stationary bubbles.) ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=832$ kW/m²)

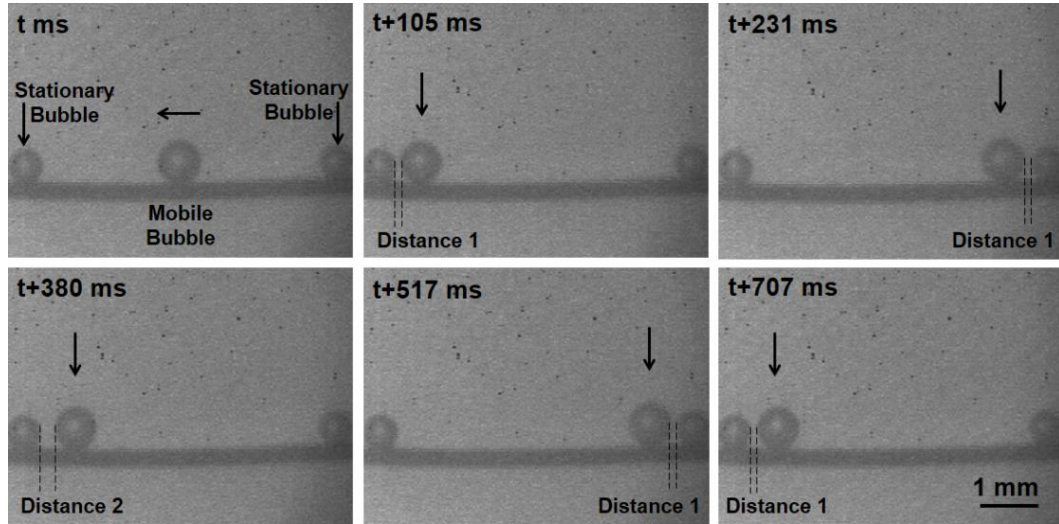


Figure 3.45: Oscillation of a bubble between two stationary bubbles with different distances due to effect of nanoparticles deposition. ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=832\text{ kW/m}^2$)

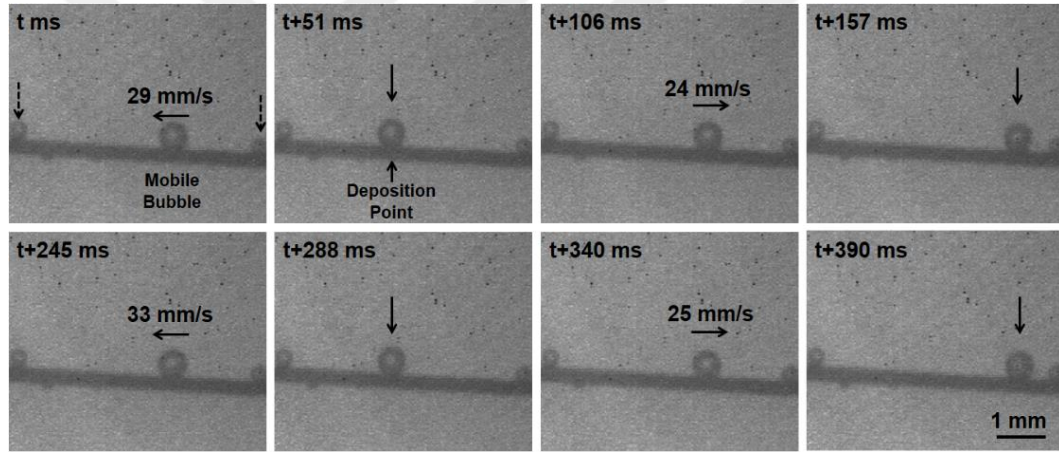


Figure 3.46: Oscillation of a bubble between a stationary bubble and a point of deposition instead of between two stationary bubbles. (Dash-lined arrows represent the stationary bubbles.) ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=832\text{ kW/m}^2$)

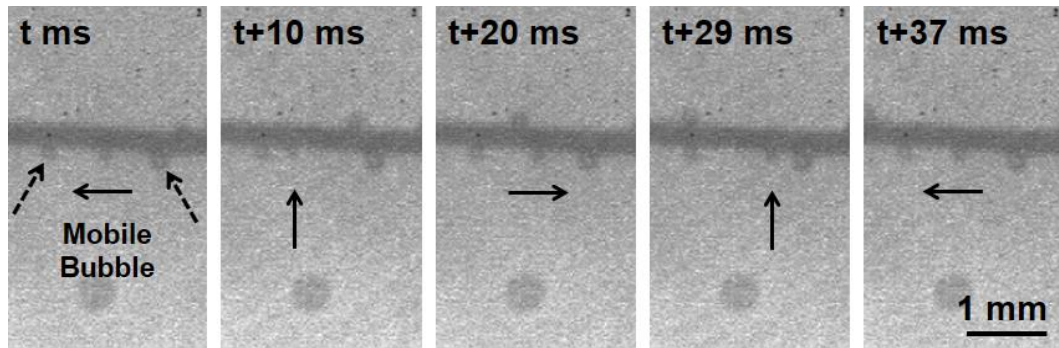


Figure 3.47: Oscillation of bubble between stationary bubbles under the wire in the nanofluid. (Dash-lined arrows represent the stationary bubbles.) ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=1774\text{ kW/m}^2$)

3.2.3.2. Coalescence

Coalescence takes places at relatively higher heat fluxes, $q'' > 700 \text{ kW/m}^2$. This event is more common in NF-I than NF-II since unlike bubbles in NF-I, bubbles in NF-II tend to oscillate and stick more than making translational motion. Regarding to coalescence of bubbles in NF-II, our observations indicate that Case 1 to Case 5, which are explained for NF-I experiments, are also present in NF-II. In addition to the mentioned cases, an interesting coalescence event is recorded for NF-II, as shown in Figure 3.48.

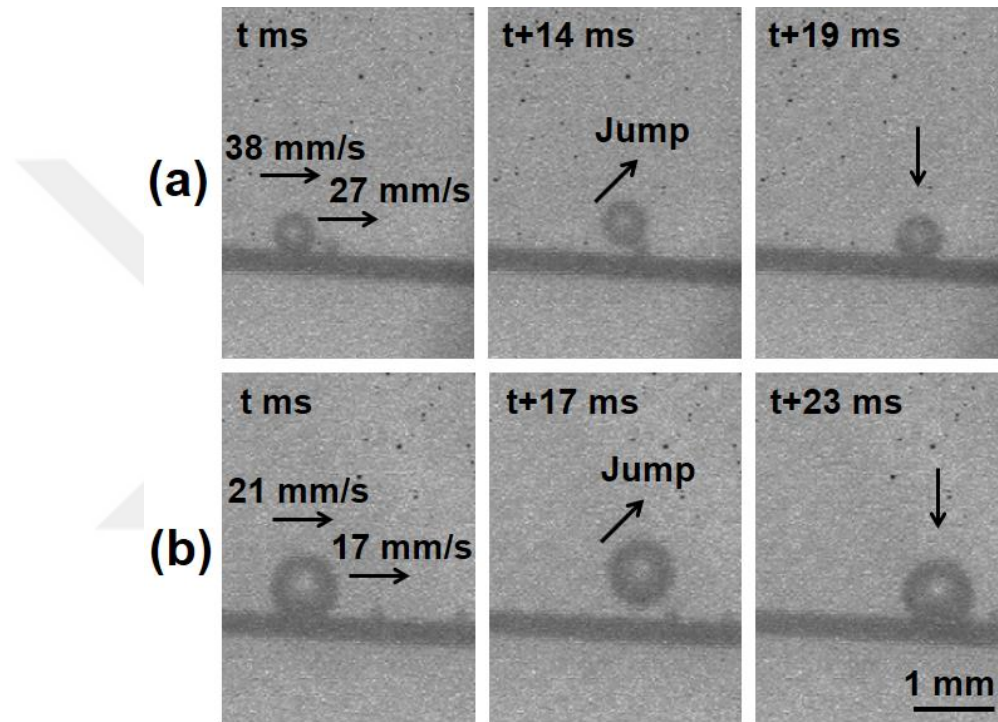


Figure 3.48: Merging by the jump of: (a) big bubble at $q''=976 \text{ kW/m}^2$ and (b) big bubble at $q''=1283 \text{ kW/m}^2$, during lateral migration on the wire. ($\phi=0.005 \text{ wt.}\%$, $T_f=35^\circ\text{C}$)

A bigger bubble with a velocity of 38 mm/s follows a smaller bubble with a speed of 27 mm/s at $t \text{ ms}$ at the NF-II bulk temperature of 35°C and heat flux of $q''=976 \text{ kW/m}^2$ in Figure 22a. When the bigger bubble approaches the smaller one, it suddenly jumps ($t+14 \text{ ms}$) and coalescence occurs at $t+19 \text{ ms}$. This event is also observed at higher heat fluxes ($q''=1283 \text{ kW/m}^2$), as can be seen in Figure 22b. This can be related to the change in Marangoni force and surface tension force due to the effect of nanoparticles deposition in vertical direction, which results in the jump before coalescence.

3.2.3.3. Stick

Another important bubble behavior in NF-II unlike NF-I is bubble sticking behavior on the wire possibly due to augmentation of both friction force associated with nanoparticle deposition and horizontal component of contact line force. Increasing nanoparticle mass fraction gives rise to a decrease in of contact line and subsequently elevates the contact line force. Therefore, Marangoni thrust force is not sufficient to overcome resistive forces of friction and contact line forces. Furthermore, pinning effect generated by disjoining pressure also generates a resistive effect. In Figure 3.49, while bubble I moves with a velocity of 29 mm/s, bubble II sticks on the wire. At $t+18$ ms, bubble I sticks and bubble II migrates with a velocity of 22 mm/s. Finally, after 52 ms, bubble II sticks, too.

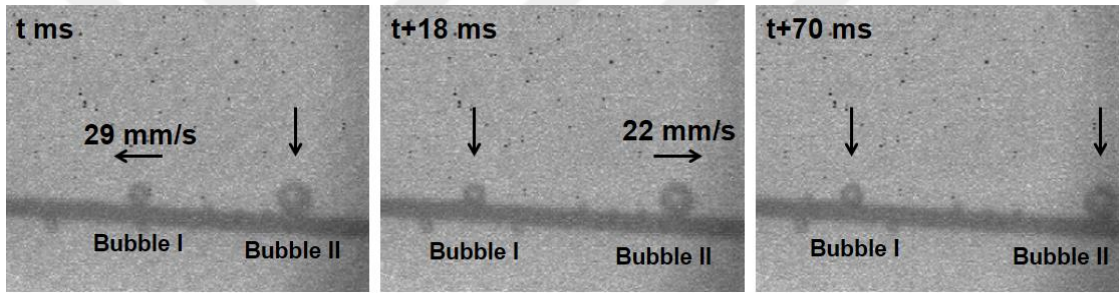


Figure 3.49: Sticking bubbles I ($t+18$ ms) and II ($t+70$ ms) after sliding due to nanoparticles deposition and/or pinning effect. ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=1417\text{kW/m}^2$)

Not only sticking behavior but also stick-slip-like behavior of bubbles is recorded in this study. However, based on the detailed investigation of the motion of bubbles, it can be hypothesized that nucleation jets (Figure 3.50) or tiny bubbles, Figure 3.51, may be responsible factors for this event.

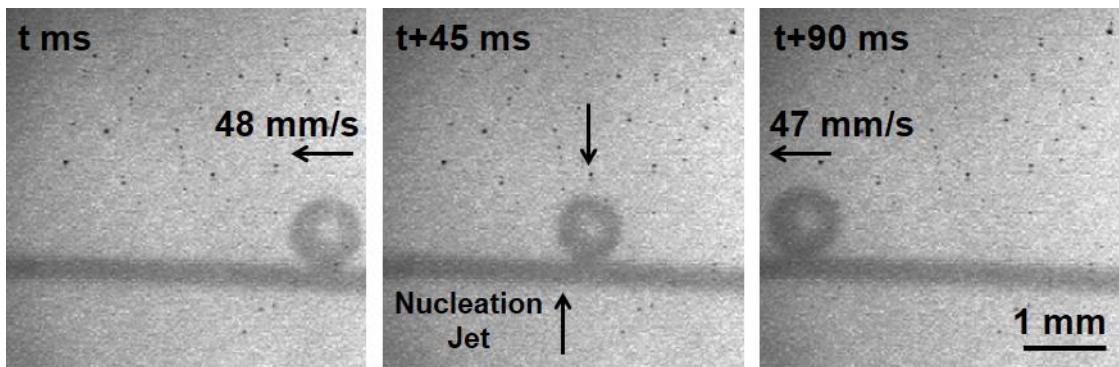


Figure 3.50: Stick-slip-like behavior of a bubble. ($\phi=0.005$ wt.%, $T_f=45^\circ\text{C}$, $q''=770\text{kW/m}^2$)

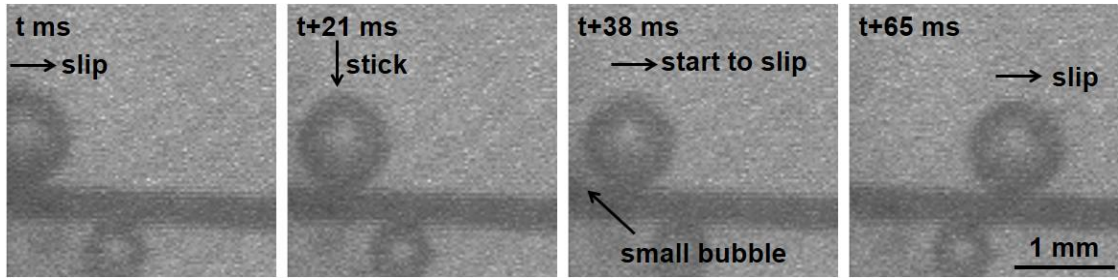


Figure 3.51: Stick-slip-like behavior of a bubble on the wire. ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=976$ kW/m²)

3.2.3.4. Dancing

Bubble-bubble interactions may result in another interesting behavior of bubbles: dancing. Bubbles oscillate along a very small distance even in 3D but not periodically, Figure 3.52(b), or shake, Figure 3.52(a). The main reason can be attributed to another bubble, which oscillates between the dancing bubble and another one or coalesce with a dancing bubble. In case of coalescence, a dancing bubble can be described as oscillating in a very small area, since small bubbles can approach to a dancing bubble from any direction, even from underneath the wire. On the other hand, if a bubble oscillates between two bubbles, one or both of them may exhibit shaking behavior.

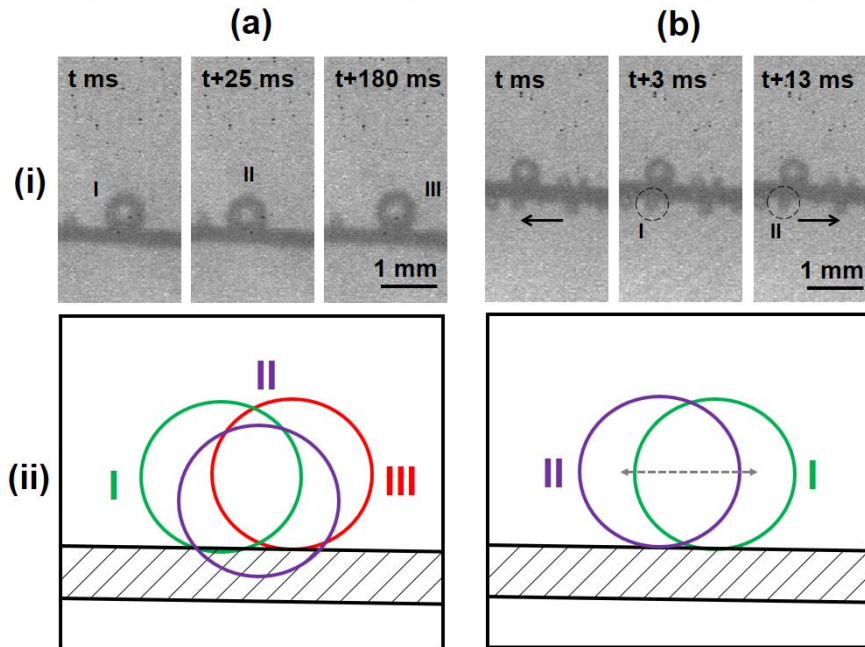


Figure 3.52: (a) Dancing of a bubble on the wire (i) and its motion (ii) positions I, II and III, (b) mini-oscillation of a bubble (i) due to oscillation of another bubble and its motion (ii) positions I and II. ($\phi=0.005$ wt.%, $T_f=35^\circ\text{C}$, $q''=2015$ kW/m²)

The contact angle, θ , is expressed according to Young's equation as:

$$\cos \theta = \frac{\gamma_{SV} - \gamma_{SL}}{\gamma_{LV}} \quad (3.4)$$

where $\gamma_{SV} - \gamma_{SL}$ is the adhesion tension and γ_{LV} is the surface tension. Surface tension of water in 25°C is approximately 72 mN/m, and its value decreases with bulk temperature. It becomes approximately 69.60 mN/m at 40°C [246]. The surface tension of TiO₂ nanoparticles/water nanofluid at mass fractions of 0.002 wt.% and 0.005 wt.% (without introducing any surfactant at the ambient temperature [247]) is approximately the same as that of DI-water. However, a decrease in the contact angle is present for nanofluids (Figure 3.53(b)), which can be attributed to the disjoining pressure and pinning effect. With an increase in mass fraction, this structural disjoining pressure and pinning effect becomes even more obvious, and a further decrease in the contact angle can be observed (Figure 3.53(c)). This trend in the contact angle can be also attained for pure water on the nanoparticle deposited wire (Figure 3.53(a) and Figure 3.53(d)) due to the effect of TiO₂ nanoparticle deposition on the adhesion tension of the surface.

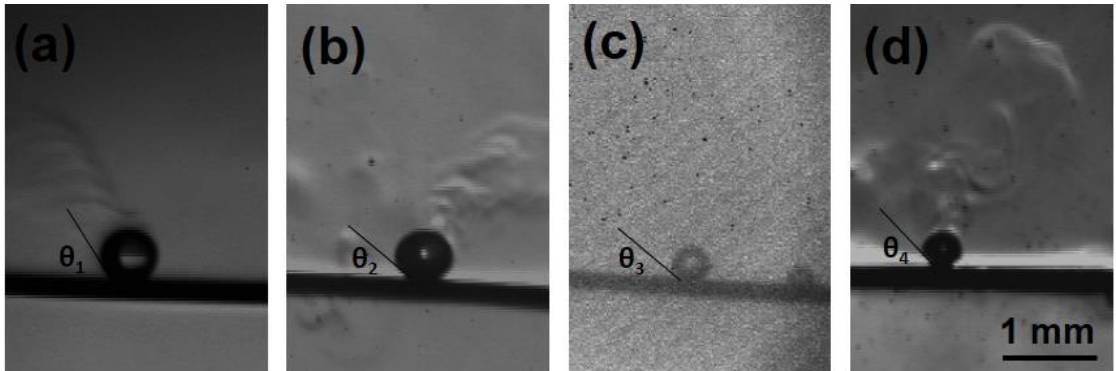


Figure 3.53: Contact angle of vapor bubble during leaping on platinum wire is: a) 60° (DI-water, $q''=1096 \text{ kW/m}^2$), b) 40° ($\phi=0.002 \text{ wt.}\%$, $q''=1291 \text{ kW/m}^2$), c) 35° ($\phi=0.005 \text{ wt.}\%$, $q''=1222 \text{ kW/m}^2$), and d) 48° (DI-water, deposited wire, $q''=1393 \text{ kW/m}^2$). ($T_b=40^\circ\text{C}$)

Major forces acting on a bubble on a wire are Marangoni, buoyancy, surface tension (contact line), and drag forces. The major forces and their direction according to the lateral motion of a bubble on the wire are included in Figure 3.54.

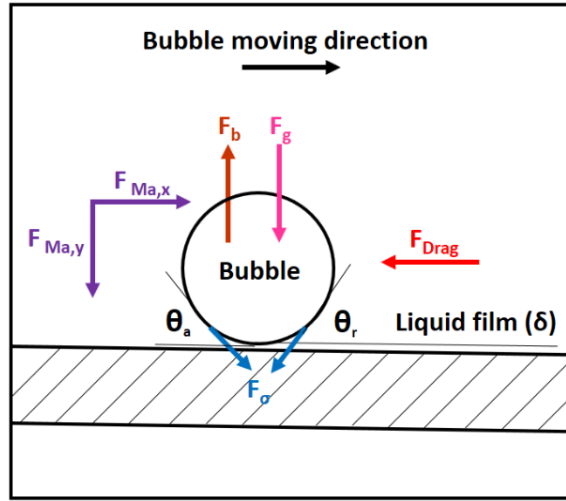


Figure 3.54: Major forces acting on a bubble in pool boiling on a wire.

The Marangoni force, F_{Ma} , is expressed as [248]:

$$F_{Ma} \propto \nabla \sigma \quad (3.5)$$

$F_{Ma,y}$ and $F_{Ma,x}$, as vertical and horizontal Marangoni forces are defined by considering the surface tension gradient between bubble's top and bottom or front and behind, respectively. Surface tension, σ , varies with temperature, T , and concentration, c [249]:

$$d\sigma = \frac{\partial \sigma}{\partial T} dT + \frac{\partial \sigma}{\partial c} dc \quad (3.6)$$

By considering the temperature gradient only, the thermocapillary force, F_{Ma} , is stated as [82]:

$$F_{Ma} = -\frac{2}{3} \pi r_b \gamma \Delta T \quad (3.7)$$

where r_b is the bubble radius, $\gamma = -\partial \sigma / \partial T_i$ is a positive constant number for most of the fluids, T_i is the temperature along the bubble interface and ΔT is temperature gradient. Thus, vertical and horizontal Marangoni forces, $F_{Ma,y}$ and $F_{Ma,x}$, can be calculated by considering the temperature gradient between bubble's top and bottom or front and behind, respectively. It should be noted that although the density of the bubble is much smaller than that of the bulk fluid, bubbles remain on the wire due to the fact that the Marangoni flow generates a thrust force [94] on the bubbles, and bubble rising is thus prevented because of the vertical component of the thermocapillary force.

The surface tension force (contact line force) is given as:

$$F_{\sigma} = \sigma \cos \theta \quad (3.8)$$

where θ is the contact angle of the bubble on the surface. Contact angles of a bubble on the platinum wire for different surfaces, namely plain and TiO₂-nanoparticle deposited surfaces, and working fluids, namely pure water and nanofluids with two mass fractions, were measured (Figure 3.53). As previously mentioned, the surface tension of the working fluids remains almost the same. By implementing the measured contact angles to the contact line force equation and comparing the results with pure water on the plain platinum wire, increases in the contact line force are approximately 30%, 40% and 15% for nanofluids with mass fractions of 0.002 wt.%, 0.005 wt.% and for pure water with a nanoparticle deposited surface, respectively.

The drag force, which acts on a bubble can be categorized as viscous drag force, acting from the fluid on the bubble, and friction force, acting from the surface on the bubble. The viscous drag force, F_D , according to the Hadamard-Rybczynski law is expressed as [250]:

$$F_D = -4\pi\mu_f r_b u_b \quad (3.9)$$

where μ_f is the liquid viscosity and u_b is the bubble velocity. Besides, friction force which acts as a resistive force, also plays a role during bubble movement on the wire. The parameters affecting the friction force are surface roughness, and nanoparticle existence.

The buoyancy force, F_b , is given as:

$$F_b = (\rho_g - \rho_f)g \cos \varphi V_b \quad (3.10)$$

where ρ_g and ρ_f are the gas and fluid densities, respectively, φ is the plane orientation and V_b is the bubble volume.

Figure 3.55 shows different bubble dynamics behavior regarding four cases; Case I) DI-water on a plain platinum wire, Case II) nanofluid with a mass fraction of $\varphi=0.002$ wt.%, Case III) nanofluid with a mass fraction of $\varphi=0.005$ wt.%, and Case VI) DI-water on a wire with TiO₂-nanoparticle deposition. In Case I, slipping of bubbles upon nucleation is dominant due to thermocapillary forces. After bubble nucleation, when thermocapillary force becomes greater than contact line force, the bubble starts slipping. Additionally, collision phenomenon can be observed at $t+234$ ms owing to the fact that

an adjacent bubble acts as a heat sink. Hence, the temperature gradient becomes zero between the front and behind the bubble. Besides, drag force acts as a resistive force. In Case II, with the introduction of nanoparticles in DI-water at the same heat flux, typically coalescence takes place. After the coalescence of two bubbles (Bubble I and Bubble II) ($t+106$ ms), interestingly, a small bubble (Bubble IV) is flung out of the newly formed bubble (Bubble III). Then, Bubble IV continues growing ($t+177$ ms) at the same position, while Bubble III moves away from the growing bubble. According to the observations, it can be claimed that not only momentum of initially moving bubbles but also remaining bubbles could determine the migration direction of the newly formed bubble, because the remaining bubble can alter the wire temperature and change the newly formed bubble direction by changing the balance force act on the bubble. One of the most remarkable phenomenon is stick-like behavior of the bubble in Case III. With a further increase in the nanoparticle mass fraction, the deposition effect of nanoparticles on the surface becomes more than in Case II. Furthermore, sticking bubbles make a bubble periodically oscillate. Deposition of nanoparticles raises the friction force and decreases the contact angle, thereby increasing the contact line force as analyzed earlier. Thus, the oscillation process will be facilitated in presence of nanoparticles due to formation of more stationary sticky bubbles. Solid-like ordering of the nanoparticles occurs in the contact line region, and this effect is more obvious at a higher mass fraction.[76] The bubble behavior of Case IV is similar to Case II. When the measured contact angles are taken into account, it can be recognized that the contact angle of a bubble in Case IV is greater than that in Case II. Thus, contact line force is lower. On the contrary, the contact line force in Case IV is higher than that in Case I, due to a lower contact angle in Case IV.

Figure 3.56 shows the behavior of a bubble under the influence of TiO_2 -nanoparticle deposition on the wire while having oscillation for the nanofluid with a larger mass fraction, namely 0.005 wt.%. As can be seen, a moving bubble approaches to a stationary bubble at t ms. During lateral migration until its stoppage near the stationary bubble, there is a knoll formed as a result of the nanoparticle deposition. Interestingly, sometimes the oscillating bubble climbs up the knoll, $t+58$ ms, and passes the hill and continues with its motion. However, sometimes the bubble cannot climb up to the top and oscillates between the knoll and the other stationary bubble (which is not seen in the figure). The major forces acting on the bubble in this situation are Marangoni, contact line, and drag forces (Figure 3.54). It should be mentioned that besides the nanoparticle

deposition, structural disjoining pressure and pinning should be taken into account. According to the contact line force calculation based on the experimentally measured contact angles, contact line force increases approximately up to 40% for the nanofluid with a mass fraction of 0.005 wt.% in comparison to pure water on the plain wire. Thus, the required driving force, mainly horizontal component of Marangoni force, should be higher which leads to bubble more sticking behavior.

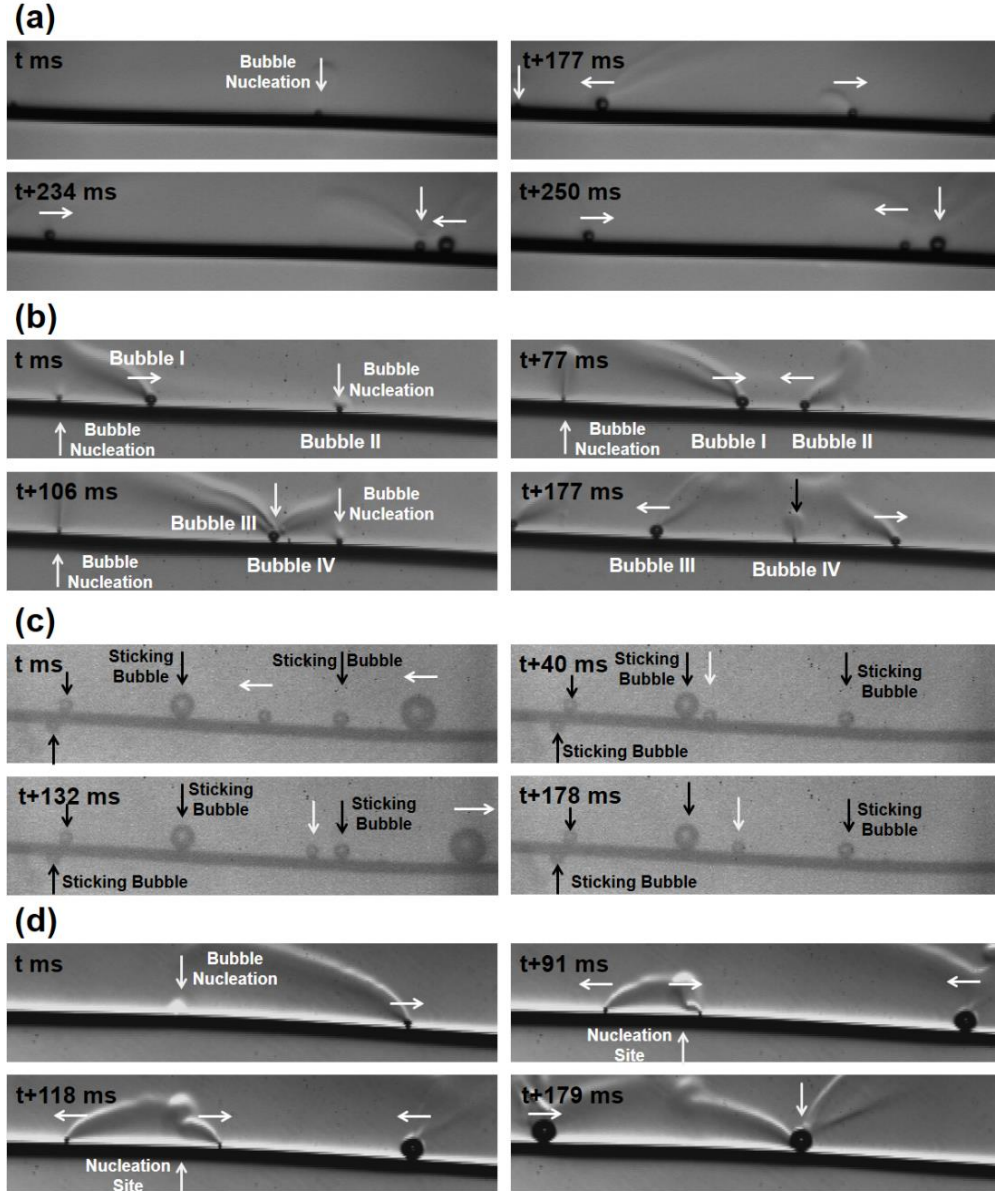


Figure 3.55: Bubble behavior for applied heat flux of approximately $q''=845\pm45$ kW/m^2 , bulk temperature of $T_b=35^\circ\text{C}$ and working fluid of: a) pure deionized water on plain wire, b) nanofluid ($\phi=0.002$ wt.%), c) nanofluid ($\phi=0.005$ wt.%) (Black arrows point the sticking bubbles as marked.), and d) pure deionized water on TiO_2 -nanoparticle deposited wire.

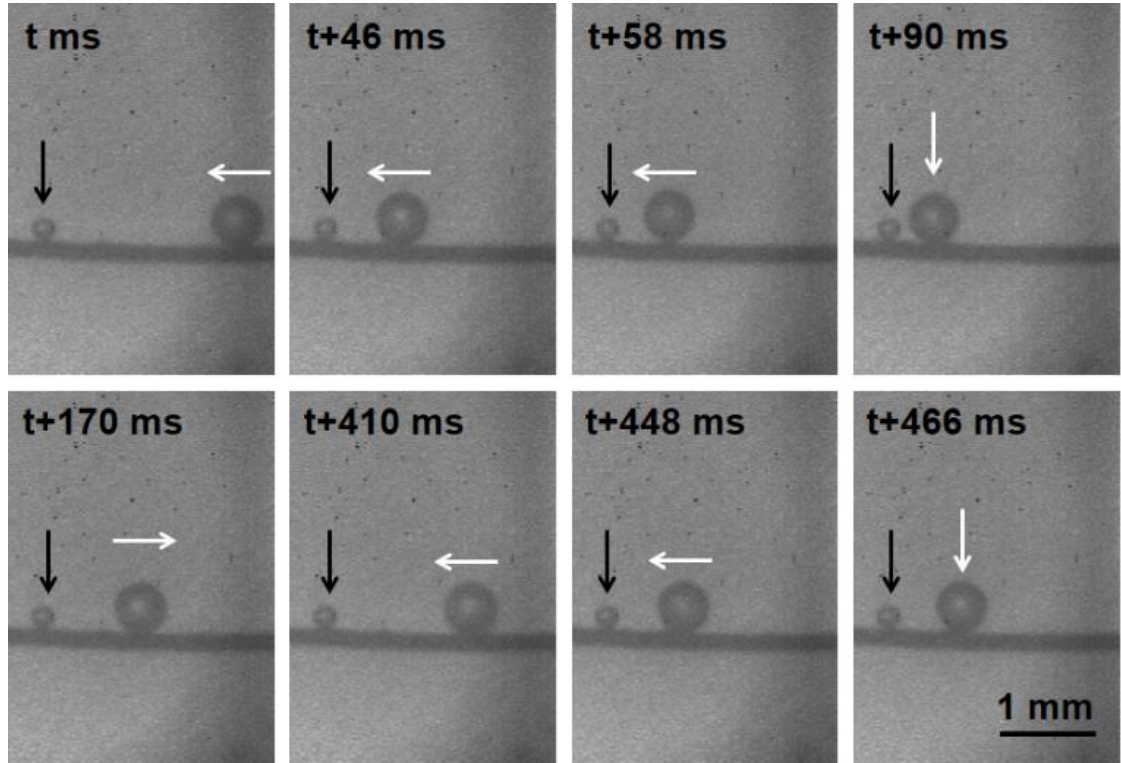


Figure 3.56: Effect of nanoparticle deposition on bubble behavior on wire for nanofluid. ($\phi=0.005$ wt.%, $T_b=35^\circ\text{C}$, $q''=845\text{kW/m}^2$)

Sticking bubbles are observed for nanofluids with a mass fraction of 0.005% at bulk temperatures of 35°C and 40°C at applied heat fluxes of $213\text{--}2015\text{ kW/m}^2$ and $175\text{--}2005\text{ kW/m}^2$, respectively. Besides, oscillatory behavior of the bubbles for the mentioned nanofluids at the mentioned bulk temperatures are seen at heat fluxes of $463\text{--}2015\text{ kW/m}^2$ and $497\text{--}2005\text{ kW/m}^2$, respectively.

3.3. Pool Boiling Heat Transfer

In the experiments related to the pool boiling heat transfer, to validate the experimental procedure, initially the pool boiling heat transfer experiments were conducted using the pure water, and the results were compared against the well-known correlations available in the literature such as Rohsenow et al. [251], Mostinski [252] and Gorenflo [253]. The wall superheats (the difference between the wall and saturation temperatures) corresponding to the experimental data of pure water are shown in Figure 3.57 for different heat fluxes along with the predictions of these correlations. As seen,

overall, a fair agreement with the experimental data and the predictions exists. The experimental results closely match with the predictions of Mostinski [46] and Gorenflo [47] and the predictions of Rohsenow [45] at high and low heat fluxes, respectively. These results serve for a validation of the experimental setup and procedure.

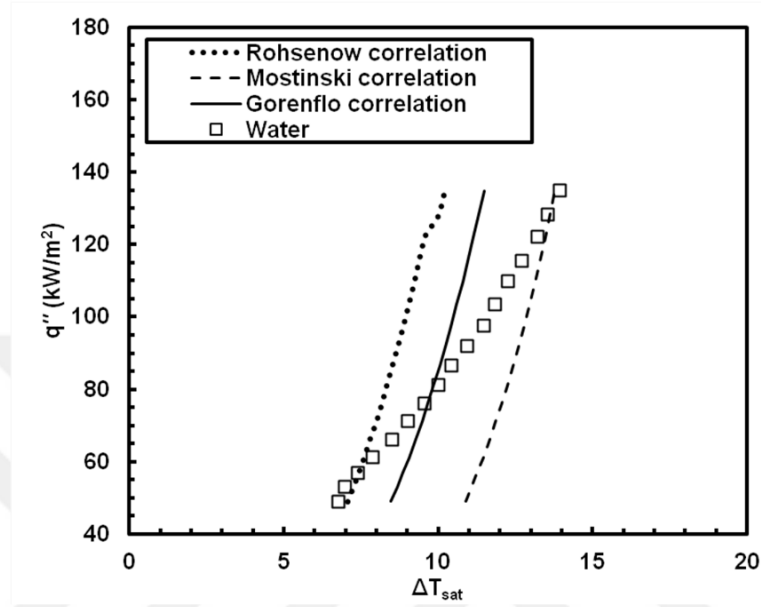


Figure 3.57: Boiling curve of pure water validated by theoretical correlations.

3.3.1. Results of TiO₂ Nanoparticles/Water Nanofluids

Boiling curves and boiling heat transfer coefficients of TiO₂ nanoparticles/water nanofluids are displayed in Figure 3.58 and Figure 3.59 for different mass fractions along with those of the pure base fluid, respectively. It can be seen from Figure 6 that the boiling curves of the nanofluids shift leftwards relative to that of water, which implies a better heat transfer performance. However, an increase in mass fraction beyond 0.01% does not considerably change the wall superheat at a fixed heat flux. Accordingly, the boiling curves become closer to that of the base fluid.

The heat transfer coefficients of dilute TiO₂ nanoparticles/water nanofluid, i.e. 0.001%, are higher than those of pure water irrespective of heat flux even though the amount of the enhancement diminishes at the upper range of the heat flux. The boiling heat transfer coefficients of nanofluids with mass fractions larger than 0.001% are still higher than those of pure water at low heat fluxes (< 90 kW/m²).

The increase in mass fraction from 0.001% to 0.01% leads to a remarkable reduction in heat transfer coefficient, while heat transfer coefficient still remains higher than those of pure water (except high heat fluxes). Nevertheless, a further increase in mass fraction does not significantly alter the heat transfer coefficient.

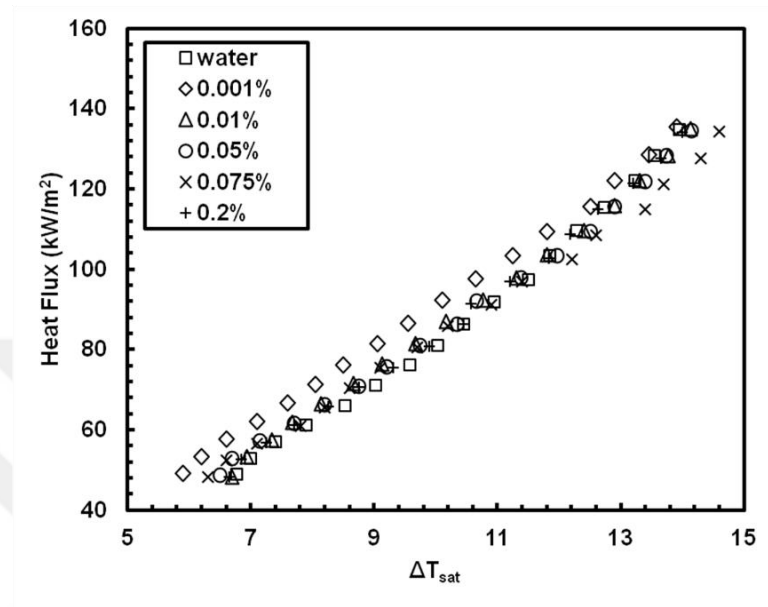


Figure 3.58: Boiling curves of TiO₂ nanoparticles/water nanofluids at different mass fractions.

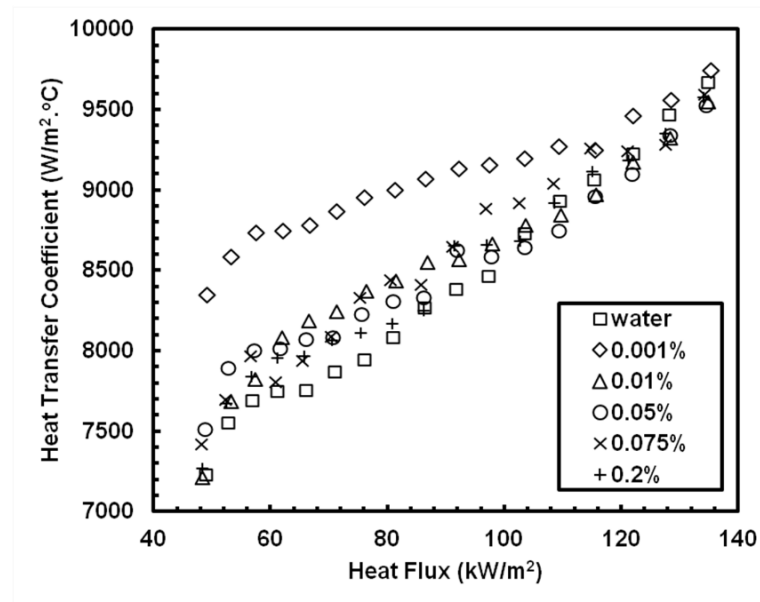


Figure 3.59: Heat transfer coefficient of TiO₂ nanoparticles/water nanofluids at different mass fractions.

Heat transfer coefficient ratios of nanofluids with respect to pure water are displayed for different mass fractions in Figure 3.60. It can be observed that an enhancement up to 15% can be achieved with dilute TiO_2 nanoparticles/water nanofluids at low heat flux. For heat fluxes lower than 100 kW/m^2 , the heat transfer coefficients of nanofluids with nanoparticles mass fractions larger than 0.001% yield a slightly better heat transfer performance compared to pure water ($< 5\%$ enhancement). Beyond the heat flux of 100 kW/m^2 , their heat transfer coefficients become lower compared to pure water. Under this condition, heat transfer enhancement only occurs for the mass fraction of 0.001%. Heat transfer enhancement sharply drops beyond a specific heat flux for all the mass fractions, which is more visible for the mass fraction of 0.001%.

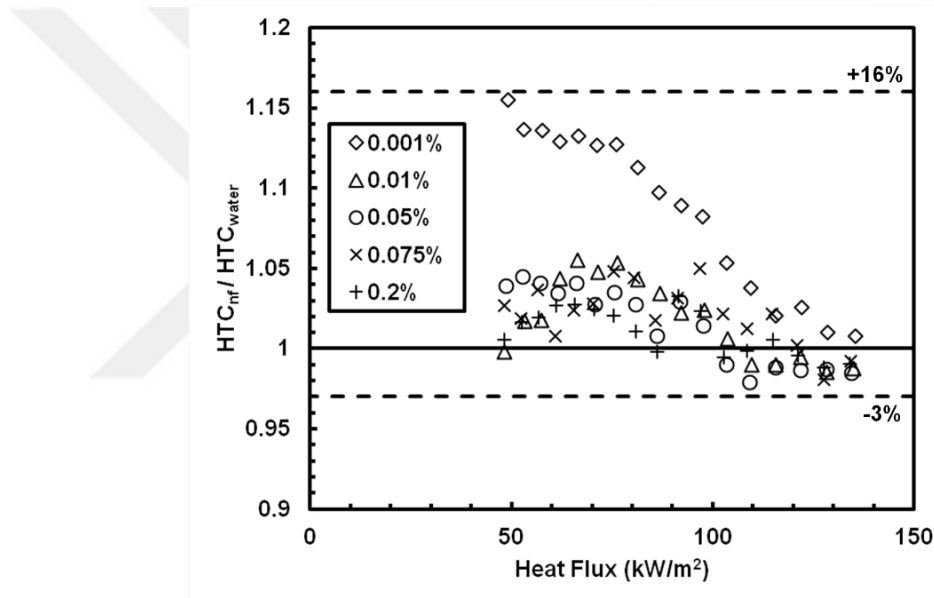


Figure 3.60: Heat transfer coefficient ratios of TiO_2 nanoparticles/water nanofluids to pure water at different mass fractions.

Nucleate pool boiling heat transfer strongly depends on formation, distribution and activity of bubbles on surfaces. Thus, for better understanding pool boiling heat transfer, visualization was performed for different fraction ratios (0.001%, 0.01% and 0.2%) in addition to pure water at heat fluxes of 48.7, 61.5, 75.9 and 91.8 kW/m^2 , as illustrated in Figure 3.61. The images clearly demonstrate that boiling images of pure water are distinctively different from those of nanofluids at different fractions, which also provides explanation for the change in heat transfer as a result of addition of TiO_2 nanoparticles to the base fluid.

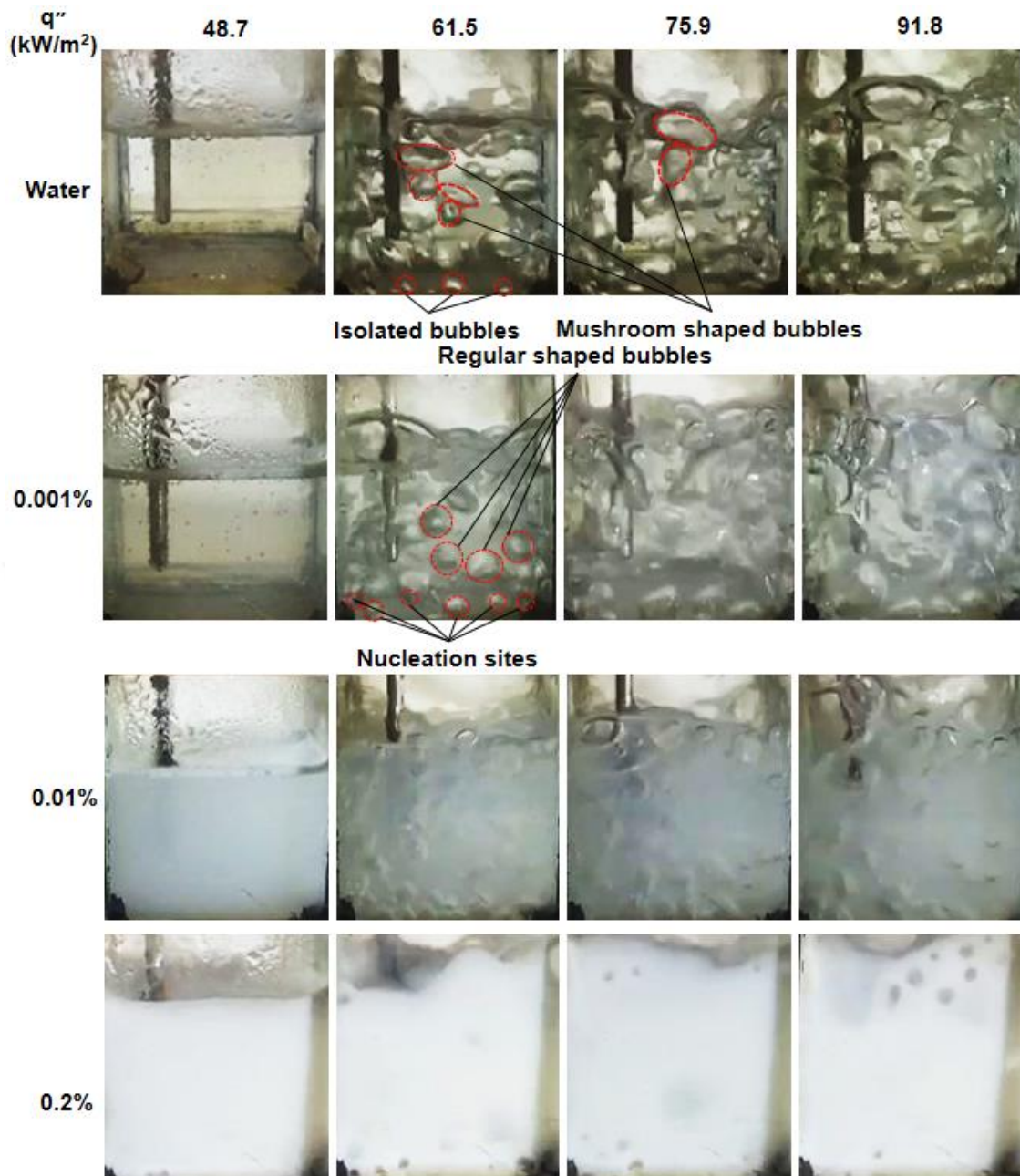


Figure 3.61: Visualization of pool boiling of water and TiO_2 nanoparticles/water nanofluids at different mass fractions and heat fluxes.

As can be seen from the images, some isolated bubbles emerge on the surface for pure water. They then expand and take the mushroom shape, which rises upwards while coalescing at higher heat fluxes. On the contrary, the addition of TiO_2 nanoparticles leads to the increase in bubble nucleation sites, which is associated with heat transfer enhancement. Furthermore, the bubbles take more regular shapes and have a smaller size in nanofluids as compared to pure water. The increase of mass fraction results in a larger bubble density and smaller bubble size, even though it is less observable at the mass

fraction of 0.2% due to its reduced transparency. In this study, visualization was performed for presenting a general view about the nucleation sites and bubble size.

Due to high bubble density and subsequent bubble crowding near the surface beyond a specific mass fraction, liquid replenishment becomes more difficult, which gives rise to deterioration in heat transfer. Since the nanofluid configuration with the best performance corresponds to a low mass fraction, deposition is not expected to have an effect on the heat transfer performance. It should be noted that TiO₂ nanoparticles/water nanofluids are known to be stable at low mass fractions [254].

3.3.2. Results for CuO/Water Nanofluids

Copper oxide nanoparticles were utilized as another type of nanoparticles for the improvement of pool boiling heat transfer. Figure 3.62 depicts the boiling curves of CuO nanoparticles/water nanofluids along with that of pure water. The boiling curves tend to shift leftwards as more CuO nanoparticles are dispersed in pure water, except the lowest mass fraction (0.001%). An increase in mass fraction gives rise to a larger shift in boiling curves. For example, the change in the wall superheat as a result of an increase in mass fraction from 0.01% to 0.2% at the heat flux of 49 kW/m² is 26.7%, while it is 28.6% at the heat flux of 134.9 kW/m².

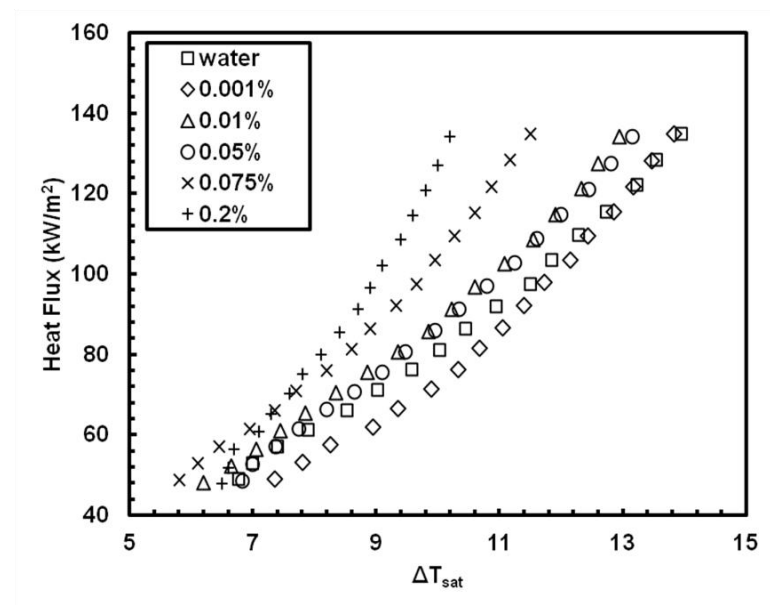


Figure 3.62: Boiling curves of CuO nanoparticles/water nanofluids at different mass fractions.

Pool boiling heat transfer coefficients of CuO nanoparticles/water nanofluids are presented for different mass fractions, namely 0.001%, 0.01%, 0.05%, 0.075 and 0.2%, along with those of the pure base fluid in Figure 3.63. For the lowest fraction ratio (0.001%), the heat transfer coefficients are lower compared to those of the base fluid, mostly at heat fluxes lower than 100 kW/m², where they converge to those of the pure base fluid beyond this heat flux. An increase in mass fraction from 0.001% to 0.01% causes enhancements up to 8% in comparison to pure water at higher heat fluxes, while no considerable change in heat transfer coefficient is seen with the change in mass fraction from 0.01% to 0.05%. Further increases from 0.05% to 0.075% and 0.2% significantly enhance heat transfer.

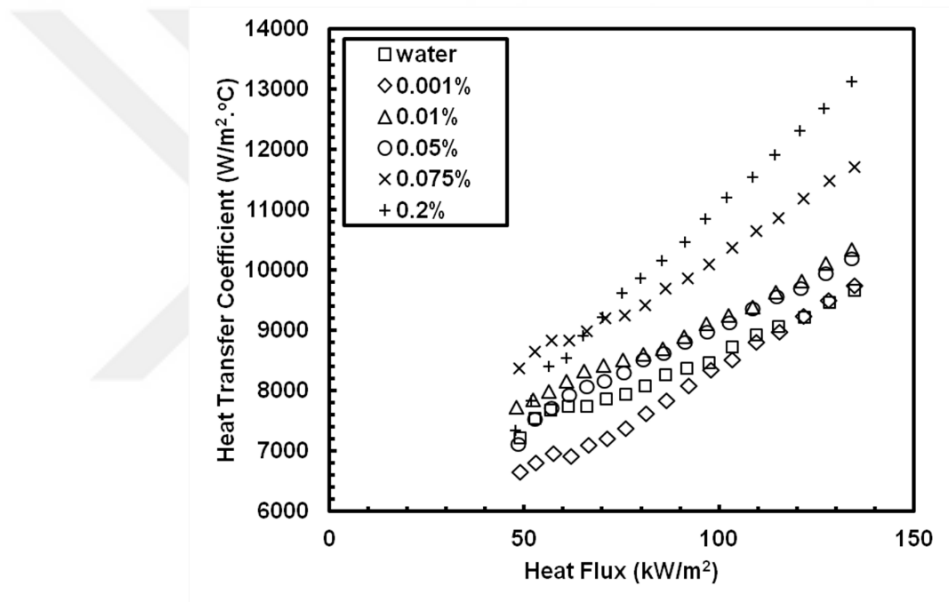


Figure 3.63: Heat transfer coefficient of CuO nanoparticles/water nanofluids at different mass fractions.

The enhancement ratios of the CuO nanoparticles/water nanofluids to pure water are depicted as a function of heat flux in Figure 3.64. It can be realized that the dilute copper oxide nanofluid with the mass fraction of 0.001% leads to degradation in heat transfer, in contrast to the dilute TiO₂ nanoparticle/water nanofluid, which proves the effect of the type of nanoparticles on boiling heat transfer. However, this degradation, which is about 12% at low heat fluxes, decreases with heat flux, until heat transfer coefficients overlap with those of pure water. The increase in mass fraction has a positive effect on heat transfer enhancement. The maximum enhancement (up to nearly 36%) thus

corresponds to the highest mass fraction (i.e. 0.2%) case. Furthermore, it can be observed that heat transfer enhancement is a function of heat flux for the mass fraction of 0.2% (increases with heat flux) and partially for the mass fraction of 0.001%, where the degradation in heat transfer decreases with heat flux.

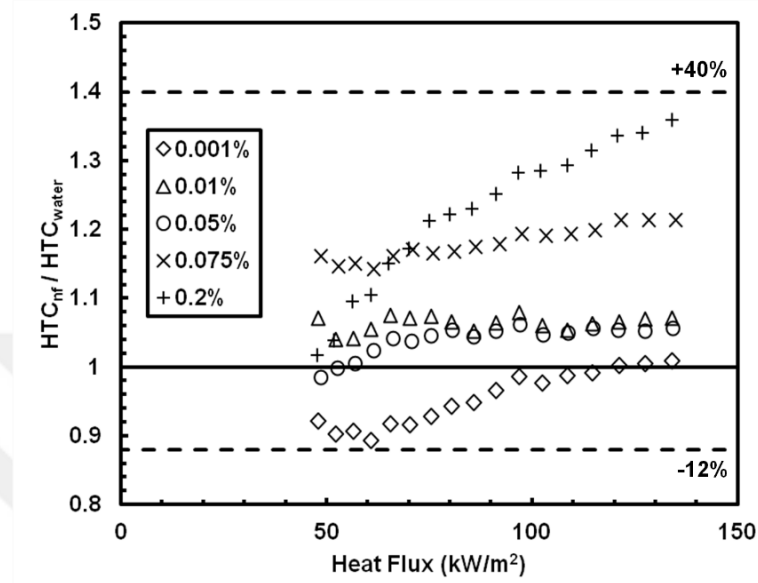


Figure 3.64: Heat transfer coefficients ratio (CuO nanoparticles/water nanofluids to pure water) at different nanoparticle mass fractions.

The deposition of nanoparticles on the surface might cause deterioration or enhancement of heat transfer coefficient, depending on the type of nanoparticles and other factors. Buongiorno and coworkers investigated the effect of different nanoparticles, namely Al_2O_3 , ZrO_2 and SiO_2 nanoparticles, and deposition on pool boiling critical heat flux (CHF) on wire and flat heaters [69,70]. According to their results, dilute nanofluids having the above-mentioned nanoparticles lead to significantly enhanced pool boiling CHF on wire heaters. The main reason of this enhancement was attributed to the deposition of nanoparticles on the heater surface during nucleate boiling and subsequent formation of a layer on the surface and subsequent increase in the wettability of the surface [69,70]. In the present study, the effect of deposition of CuO nanoparticles was investigated by examining pool boiling of pure water just after experimentation with CuO nanoparticles/water nanofluids at mass fractions of 0.01% and 0.2%. The results were compared to those of pure water tested with the plain surface. The results are displayed in Figure 3.65. They reveal that the considerable increase in heat transfer coefficient at

high mass fraction (i.e. 0.2%) is attributed to the deposition of CuO nanoparticles. The increase is less pronounced for lower mass fraction (0.01%) due to less deposition effects.

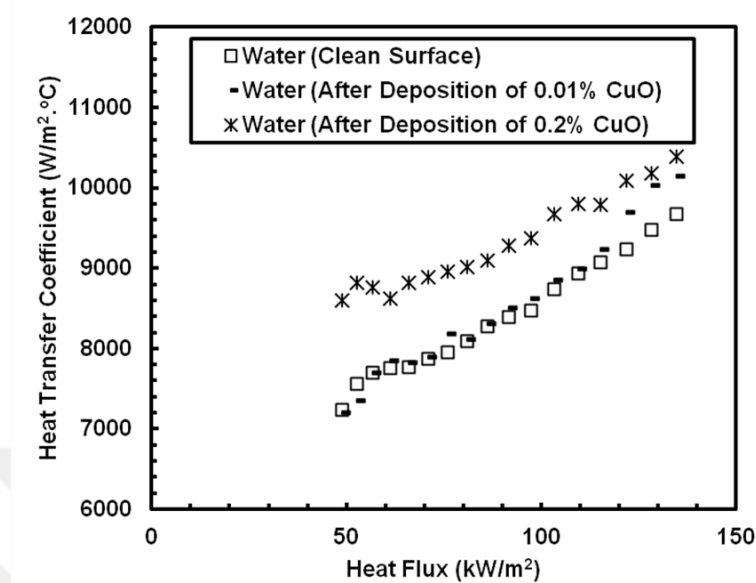


Figure 3.65: Heat transfer results of water on clean and deposited surfaces with the use of CuO nanoparticles/water nanofluids with low and high concentrations.

It should be noted that a visualization study could not be done with CuO nanoparticles/water nanofluids due to the lack of transparency. Due to their lower stability, which was also indicated in the literature [226], examination of deposited surfaces and accompanied pool boiling experiments were considered to be more appropriate. As a result, a new surface was utilized for each set of experiment in this study, except the experiments on deposited surfaces, which reveal the effect of nanoparticles deposition on pool boiling heat transfer.

As seen in the results of both types of nanoparticles, adding the nanoparticles to the base fluid alters the surface tension and contact angle (Figure 3.66). Deposition of TiO₂ nanoparticles makes the nanofluid hydrophilic so that an increase in the concentration leads to a smaller contact angle. As a result of boiling experiments, a porous coating on the surface forms. This porous structure augments bubble nucleation from the surface and facilitates liquid replenishment to the surface after bubble departure, which has an enhancing effect on boiling heat transfer at a low mass fraction. However, wetting of the surface becomes more difficult under the conditions of bubble crowding, which causes deterioration in heat transfer at higher mass fraction for TiO₂ nanoparticles.

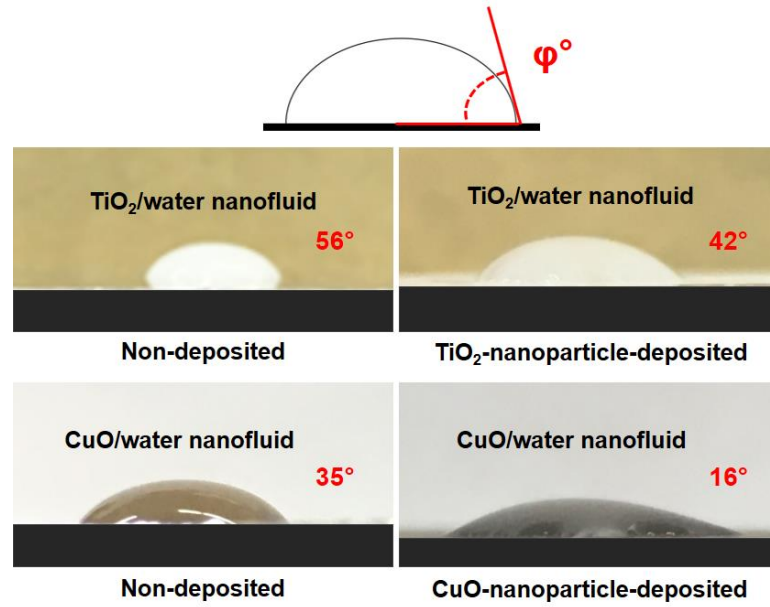


Figure 3.66: Contact angles of TiO₂/water and CuO/water nanofluid droplet on the heater surface with and without nanoparticles deposition.

According to the findings of Buongiorno and coworkers, nanoparticles led to the deposition of a porous layer on the heater surface. Thus, the contact angle and nucleation site density decrease. On the other hand, the deposition of nanoparticles adds an extra thermal resistance. These two reasons were the most important reasons behind the decrease in the pool boiling heat transfer coefficient [69,70]. An increase in CHF was reported with the decrease in the contact angle and improved wettability of the surface with the nanoparticle deposition. However, the nucleation site density could also increase due to the porous layer and increase in the cavities on the surface with the deposition of the nanoparticles, which was mentioned in their study and was also observed in the present study. The increase or deterioration in boiling heat transfer and CHF is highly dependent on nanoparticle size, initial morphology of the substrate, particle deposition rate and duration of the experiment according to the Buongiorno and coworkers [70]. The reasons behind the increase in boiling heat transfer could be attributed to nanoparticle interactions with bubbles [116], which is in agreement with the observations in the present study, and to the increase in thermal conductivity and nucleation site density through the formation of a porous layer caused by the nanoparticle deposition [255].

The coating on the surface with CuO particles as a result of boiling experiments makes the surface hydrophilic, which was also considered as the major reason behind the enhancement in CHF in the literature [256–258]. While the improved hydrophilicity may

at first glance seem to have a negative effect on nucleate boiling, a porous structure with a most likely higher thermal conductivity is formed upon boiling at particularly higher mass fraction. This porous structure assists in raising bubble generation frequency from the surface and the number of active nucleation sites and improvement in liquid replenishment to the surface after bubble departure. As a result, more heat transfer enhancement is present with an increase in mass fraction for CuO nanoparticles, since more porous structure forms at higher mass fraction.



3.4. Bubble Dynamics in Minichannels

3.4.1. Theoretical Background

Understanding the forces acting on a bubble plays an important role in studying the bubble dynamics in channels and assessing the forces on the bubble interface and the surface in the presence of micro/nano-structures or nanoparticle deposition. Generally, the significant forces acting on the gas bubble in the channel are inertia, surface tension, Marangoni, shear, buoyancy and evaporation momentum forces [124,127,145]. The main forces acting on a bubble in the channel are shown in Figure 3.67.

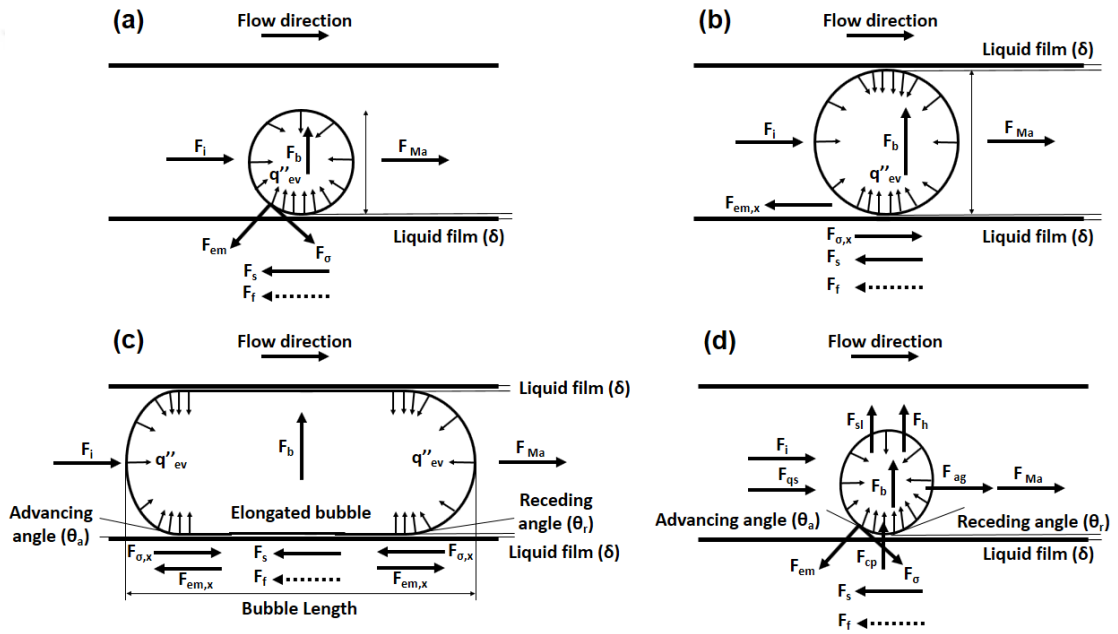


Figure 3.67: Forces acting on: a) a bubble smaller than the channel hydraulic diameter, b) a confined bubble (the diameter is equal to the channel hydraulic diameter), c) an elongated bubble, and d) a stopped bubble smaller than the channel hydraulic diameter. (Dashed arrows are important particularly in the case of TiO_2 nanoparticle deposited channel)

Inertia force, F_i , acts at the channel cross section:

$$F_i = \rho_f U_f^2 A_c \quad (3.11)$$

$$F_i \propto \rho_f U_f^2 D_h^2 \quad (3.12)$$

where ρ_f and U_f are the average density and mean velocity of the fluid, respectively, A_c is the channel cross section area and D_h is the channel hydraulic diameter for the confined bubble, where its diameter is the same as the channel hydraulic diameter, and is replaced with bubble diameter, D_b , for bubbles smaller than the channel hydraulic diameter. Inertia force is in the flow direction. As the main aiding force, it pushes the bubble along the channel. As can be seen from the Eq. (3.11), for a confined bubble, it increases with the flow velocity, and for smaller bubbles, an increase in the flow velocity and the diameter of the bubble causes an increase in this force.

Surface tension force, F_σ , is an interfacial force given as:

$$F_\sigma = \sigma \cos \theta D_b \quad (3.13)$$

$$F_\sigma \propto \sigma \cos \theta \quad (3.14)$$

where σ is the surface tension, θ is the contact angle of the liquid-gas interface, which is assumed as the static contact angle and D_b is the bubble diameter. The modification of the surface, i.e. nanoparticle deposition, alters the contact angle. Using different fluids as the working fluid, such as ethanol versus water or nanofluids with these base fluids, changes the surface tension, thereby changing the total surface tension force. In flow boiling experiments, surface tension force helps in rewetting the surface by moving the liquid from the wetter surface to the dryer surface and enhances critical heat flux. In a non-spherical bubble, the diameter is replaced with Sauter mean diameter (SMD), $D_S = 6V_b / A_b$, where V_b and A_b are the volume and surface area of the bubble, respectively.

By considering the dynamic contact angles, advancing and receding contact angles the proportionalities of the surface tension force at advancing and receding sections are stated as:

$$F_{\sigma,a} \propto L_{c,a} \sigma (\cos \theta_e - \cos \theta_a) \quad (3.15)$$

$$F_{\sigma,r} \propto L_{c,r} \sigma (\cos \theta_r - \cos \theta_e) \quad (3.16)$$

where $L_{c,a}$ and $L_{c,r}$ are the advancing and receding contact line lengths, θ_a , θ_r and θ_e are the advancing, receding and equilibrium contact angles of liquid-gas interface,

respectively. The equilibrium contact angle is calculated by balancing the forces at the triple contact line [259].

While studying a bubble smaller than channel hydraulic diameter, surface tension force acting on the bubble can be categorized according to the acting point: surface tension on the top of the bubble with contact angle of zero and surface tension acting in triple line interface among the solid, liquid and the gas phases with contact angle larger than zero [260]. The surface tensions are then different due to the different temperatures at the triple line interface and fluid temperature at the bubble's top section.

The Marangoni effect appears with surface tension gradient by changing the temperature. Surface tension decreases with a decrease in the temperature for more fluids and decreases up to a specific temperature and increases beyond this temperature for self-wetting fluids. With the application of heat flux to the channel walls, the temperature of the inlet fluid increases along the channel. As a result, the surface tension gradient force, F_{Ma} , becomes important and is expressed as [248]:

$$F_{Ma} \propto \nabla \sigma \quad (3.17)$$

The variation of surface tension can be due to thermal or concentration effects:

$$d\sigma = \frac{\partial \sigma}{\partial T} dT + \frac{\partial \sigma}{\partial c} dc \quad (3.18)$$

where T and c are the temperature and the concentration of the fluid, respectively. Temperature and concentration gradients cause the thermocapillary and solutocapillary Marangoni effects, respectively [249].

The local fluid temperature, $T_{L,f}$, is given as:

$$T_{L,f} = T_{in} + \frac{2q''(W_{in} + h_{in})}{c_p G A_c} x \quad (3.19)$$

where T_{in} is the inlet temperature, q'' is the applied heat flux, W_{in} and h_{in} are the inner cross sectional dimensions of the channel, c_p is the specific heat capacity, G is the mass flux, A_c is the channel cross sectional area and x is the location related to the inlet.

Shear force, F_s , is an interfacial force and acts against the flow is expressed as:

$$F_s = \frac{\mu_f U_f A_{pl}}{\delta} \quad (3.20)$$

$$F_s \propto \frac{\mu_f U_f}{\delta} \quad (3.21)$$

where μ_f is the fluid viscosity, δ is the film (microlayer) thickness under the bubble and A_{pl} is the shear plane area over which the force acts. The main factor affecting the shear force is the change in the flow rate of the fluid and accordingly the fluid velocity.

Buoyancy force, F_b , or gravity force is a body force and results from the difference in the gas and fluid densities:

$$F_b = (\rho_g - \rho_f)g \cos \varphi V_b \quad (3.22)$$

$$F_b \propto (\rho_g - \rho_f)g \cos \varphi D_b^3 \quad (3.23)$$

where ρ_g and ρ_f are the gas and fluid densities, g is the gravitational acceleration, φ is the heating surface orientation, D_b and V_b are the bubble diameter and volume, respectively.

Evaporation momentum force, F_{em} , acting at the evaporation interface is expressed as:

$$F_{em} = \left(\frac{q''_{ev}}{h_{fg}} \right)^2 \frac{1}{\rho_g} A_i \quad (3.24)$$

$$F_{em} \propto \left(\frac{q''_{ev}}{h_{fg}} \right)^2 \frac{1}{\rho_g} \quad (3.25)$$

where q''_{ev} is the evaporative heat flux at the interface, which is assumed as effective applied heat flux and increases with the applied heat flux, h_{fg} is the latent heat of vaporization and A_i is the bubble interface area. In case of an elongated bubble, extended meniscus can be divided into three sections; non-evaporating section (thickness in micro scale), thin-film section due to long-term molecular disjoining pressure forces and intrinsic meniscus section, where the capillary forces are dominant [261]. With increasing heat flux, the elongated bubble's nose grows faster with respect to its tail. Thus, effective evaporation momentum force is much more in the nose related to the tail. On the other hand, the tail section experiences inertia force.

According to the study of Raghupathi and Kandlikar [262] on net evaporation momentum force acting on an asymmetrical heated bubble, the forces acting on a bubble, which is subjected to two temperatures, are considered. Since constant heat flux was applied during the experiments in this study as shown in Figure 3.68, an asymmetric temperature distribution is obtained. As a result, a bubble experiences different evaporation momentum forces by having a smaller value at a lower temperature and a larger quantity at a higher temperature so that the total evaporation momentum force becomes the resultant of these two forces.

For confined bubbles (elongated bubble), the advancing and receding sections experience the same above-mentioned forces. Buoyancy and shear forces act on the bubble and the contact plane with the wall, respectively. However, inertia, surface tension and evaporation momentum forces act differently at advancing and receding sections. At the advancing interface section, inertia force as the aiding force pushes the bubble along the channel in the flow direction, while surface tension force helps inertia force and wetting the downstream section of the channel, and evaporation momentum force act as the resisting forces. On the other hand, at the receding interface section, the pushing force originates from inertia force indirectly by pushing the gas inside the bubble slug, evaporation momentum force helps the inertia force and surface tension force resists the motion. Besides, Marangoni force and evaporation momentum force are in the flow direction due to the fact that constant heat flux is applied, and the fluid temperature has an increasing profile along the channel.

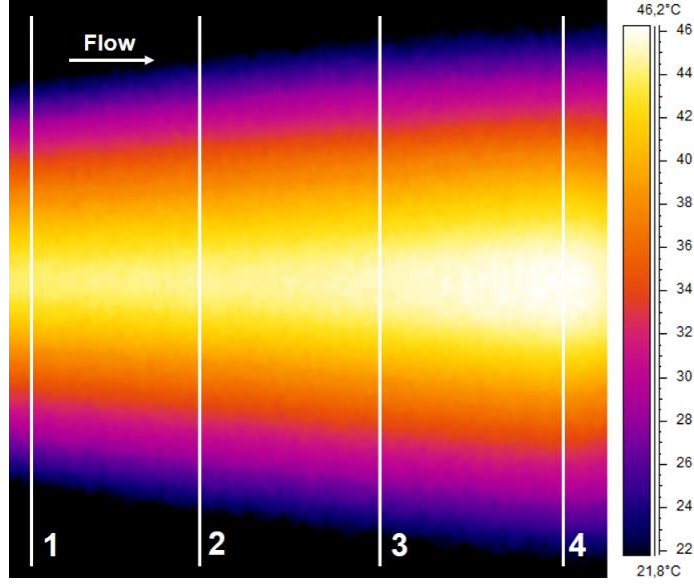


Figure 3.68: A sample temperature distribution on channel outer surface at the heat flux of 4.5 kW/m^2 showing four sections at along a length of 4 cm of the end of channel. Max temperatures at these sections are: $T_{\text{Max},1}=44.1^\circ\text{C}$, $T_{\text{Max},2}=44.7^\circ\text{C}$, $T_{\text{Max},3}=45.3^\circ\text{C}$ and $T_{\text{Max},4}=46.2^\circ\text{C}$

In the case, where a bubble stops on the modified surface in the channel due to the force balance, the diameter of the bubble increases to a specific diameter, at which aiding forces for the movement become large enough to push the bubble along the channel. In this case, some additional forces acting on the bubble, such as unsteady drag force, due to asymmetrical growth of the bubble and dynamic effect of unsteady liquid, hydrodynamic force, contact pressure force, quasi-steady drag force and shear lift force in the flow direction are present [146,248].

Force due to the asymmetrical growth of the bubble, F_{ag} , is given as:

$$F_{ag} = \frac{3}{2} m U_f \frac{dr_b / dt}{r_b} \quad (3.26)$$

$$F_{ag} \propto m U_f \frac{dr_b / dt}{r_b} \quad (3.27)$$

where m is the mass of liquid displaced by the bubble, U_f is the uniform flow velocity and r_b is the bubble radius.

Hydrodynamic force, F_h , is calculated by considering an inviscid flow over a sphere in an unbounded flow field as:

$$F_h \propto \rho_f U_f^2 d_c^2 \quad (3.28)$$

where d_c is the contact diameter of the bubble.

Contact pressure force, F_{cp} , due to the pressure difference between the inside and the outside of the bubble at a reference point over the contact area is expressed as:

$$F_{cp} = \frac{\pi d_c^2}{4} \frac{2\sigma}{r_r} \quad (3.29)$$

$$F_{cp} \propto d_c^2 \frac{\sigma}{r_r} \quad (3.30)$$

where r_r is the curvature of the bubble at the reference point.

Quasi-steady drag force, F_{qs} , is stated as:

$$F_{qs} = 3C_D \mu_f \Delta U \pi d_s \quad (3.31)$$

$$F_{qs} \propto C_D \mu_f \Delta U d_s \quad (3.32)$$

where C_D is the drag coefficient, ΔU is the relative velocity between the bubble center of mass and liquid and d_s is the sliding bubble diameter.

3.4.2. Results of Bubble Dynamics in Minichannels

The characteristics of single air bubble dynamics in flows through a rectangular glass channel are examined with the assistance of visualization and velocity measurements. Moreover, discussion about the forces and underlying mechanisms are included by considering the effects of heat flux, mass flux, fluid type, nanoparticle deposition and bubble size. The dynamics of the bubbles is highly dependent on the combined effect of acting forces, Figure 3.67.

The effect of heat flux on a single injected air bubble in pure water is displayed in Figure 3.69. While the bubble diameter (3.9 mm) in Figure 3.69(a) (bubble I) is approximately the same as that in Figure 3.69(b) (bubble II), the speed of bubble II, corresponding to the heated case, is slightly lower than the bubble I corresponding to the adiabatic case. In the light of force analysis, it can be asserted that, in both cases, inertia and surface tension forces act on the bubble as aiding forces for moving while shear force acts on the bubble as a resistive force. By the application of heat flux, evaporation

momentum and Marangoni forces are introduced to the bubble. Evaporation momentum force acts on the bubble as a resistive force against the flow direction and Marangoni force acts in the reverse direction. On the other hand, an increase in the temperature causes a decrease in the surface tension force due to the decrease in the surface tension. As an outcome of acting forces, the bubble velocity slightly decreases with heat flux.

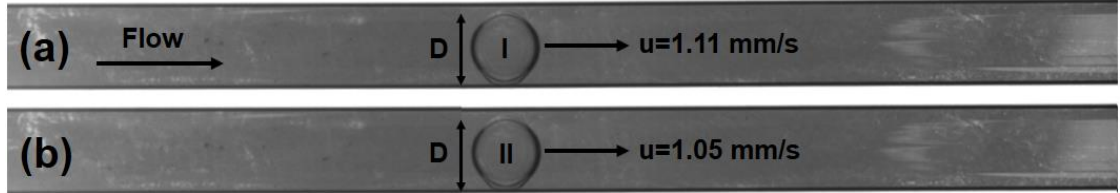


Figure 3.69: Velocity measurement of a bubble in DI water at the flow rate of 1 ml/min for: a) no heat flux and b) heat flux of 1.4 kW/m^2 . ($D=3.9 \text{ mm}$)

Figure 3.70 exhibits the comparison of the speeds of single bubbles in ethanol as the working fluid with almost the same diameter at different flow rates, namely 1, 3, 5, and 10 ml/min. As expected, bubble speed increases with the flow rate due to increasing inertia force.

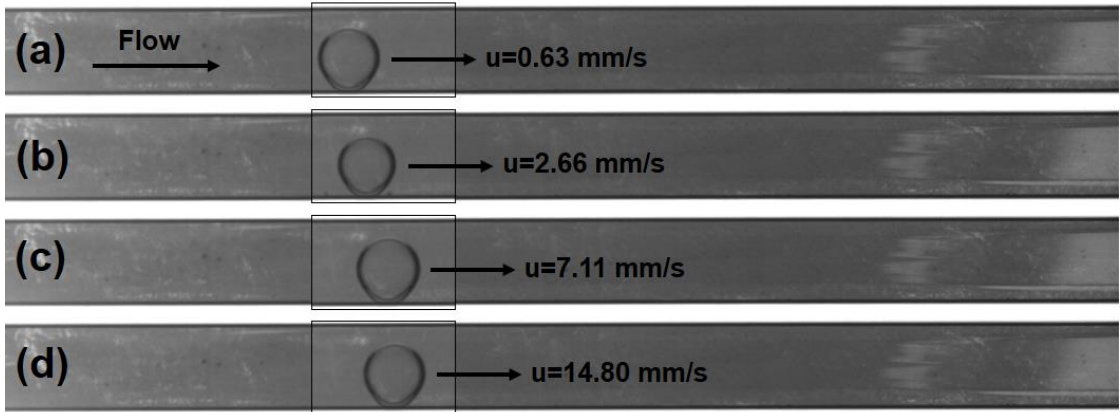


Figure 3.70: Bubble speed in ethanol as working fluid under adiabatic conditions for flow rate of: a) 1 ml/min, b) 3 ml/min, c) 5 ml/min and d) 10 ml/min.

From heat transfer point of view, the effect of nanoparticles has not fully understood. The possible underlying mechanisms of enhancement or deterioration of heat transfer along with the usage of nanofluids can be found elsewhere [32,263], while the effect of nanoparticles on bubble dynamics has not been assessed in detail. The speed of bubbles in water and water based nanofluid with a fraction ratio of 0.005 wt.% at different

heat fluxes and a constant flow rate (10 ml/min) is demonstrated in Figure 3.71. As seen in Figure 3.69, increasing heat flux leads to a slight reduction in bubble speed for water at low flow rate (1 ml/min) (Case 1). However, bubble speed is larger with increasing in heat flux at high flow rate (10 ml/min) (Case 2). Comparing Case 1 to Case 2, the magnitude of major forces is the greatest at the maximum applied heat flux in Case 2, except the surface tension force. Inertia force as aiding force is greater in Case 2 than Case 1 due to higher mass flux. Evaporation momentum force, which is a resistive force, increases with heat flux. Hence, this force is greater for Case 2 than Case 1. Similarly, high applied heat flux results in an increase in Marangoni force, which is aiding force. Thus, Marangoni force is greater in Case 2 with respect to Case 1. However, surface tension force diminishes with temperature. Besides, nanoparticle effect becomes more apparent at high applied heat fluxes and high flow rates and retards the bubble motion. Due to increasing pinning effect at higher heat fluxes, a drop in bubble speed occurs. In another study [264], accumulation of the TiO_2 nanoparticles at the contact line augments the pinning effect, which causes slip-stick behavior during evaporation of a sessile nanofluid droplet due to deformation of the contact line [31], based on theory of wetting hysteresis, pinning and depinning effects during droplet evaporation [265].

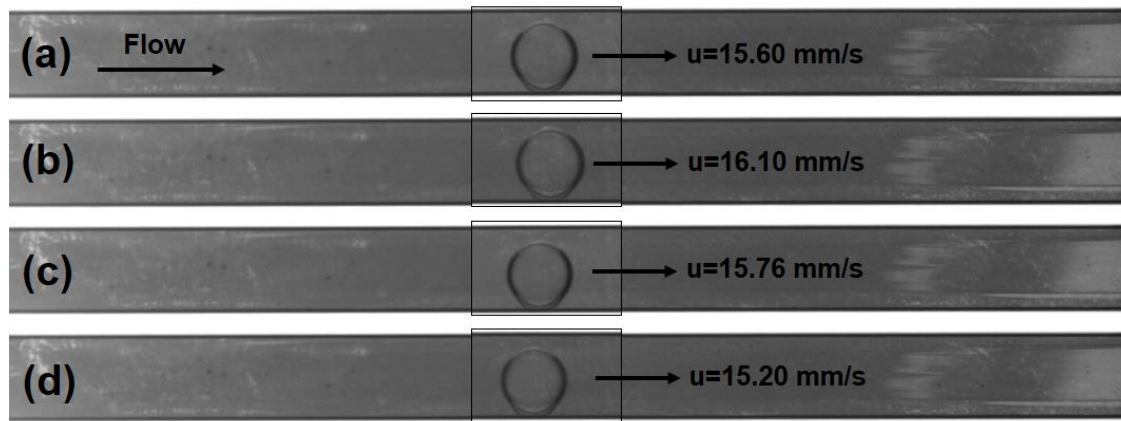


Figure 3.71: Bubble speed at flow rate of 10 ml/min, heat flux of 0.4 kW/m^2 and working fluids of: a) pure water, b) water/ TiO_2 nanofluid and heat flux of 9 kW/m^2 and working fluids of: c) pure water, d) water/ TiO_2 nanofluid.

Figure 3.72 displays the impact of nanoparticles on elongation process of the bubbles in ethanol. Under adiabatic conditions, a striking velocity difference of bubbles for pure ethanol and ethanol-based nanofluid exists (Figure 3.72(a-b)). With increasing heat flux, elongation process in ethanol-based nanofluid is weakened with respect to that

in pure ethanol, so that bubble nose speed in the pure ethanol becomes greater than that in ethanol-based nanofluid (Figure 3.72(c-d)). Clearly, nanoparticles induce a resistive effect on elongation process. As shown in Figure 3.71, at high heat flux, the pinning effect influences the bubble having a diameter smaller than the channel hydraulic diameter by decreasing its velocity. Similarly, in Figure 3.72, for elongation process at a higher heat flux, this effect causes bubble nose velocity to decrease and shortens the bubble elongation length.

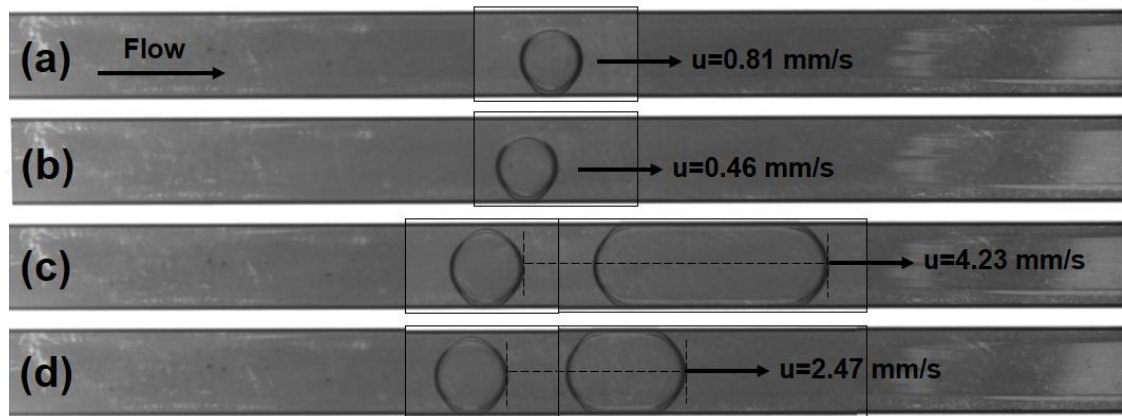


Figure 3.72: Bubble speed for the case of without applied heat flux in working fluid of: a) pure ethanol, b) TiO_2 /ethanol nanofluid and for the case of with 2.3 kW/m^2 applied heat flux in working fluid of: c) pure ethanol and d) TiO_2 /ethanol nanofluid in non-deposited channel.

Another important parameter for assessing the nanofluid effect on boiling heat transfer is nanoparticle deposition [55,266–268]. In the partially deposited channel, bubble dynamics depicts interesting features for ethanol. Initially, single small air bubble starts sliding until it reaches a critical position. If heat flux is applied to the system, the bubble starts to grow by absorbing heat from the channel walls. After reaching a critical diameter, it moves along the channel again. However, this critical diameter strongly depends on flow rate in such a way that an increase in flow rate results in a decrease in the critical diameter (Figure 3.73). On the other hand, a change in heat flux does not influence bubble critical diameter. The velocity of the bubbles after departure from the critical position is measured near the outlet section (Figure 3.73). At the flow rates of 1 and 3 ml/min, bubbles grow along the channel while moving, whereas at the flow rate of 5 ml/min, the diameter of the moving bubble from the critical position increases relatively less because of less time for growth. At the largest flow rate, namely 10 ml/min, this

stopping behavior is not observed because the inertia force alone provides a high enough aiding force on the bubble.

The deposition of TiO_2 nanoparticles enhances the wettability of the surface owing to the reason that with an increase in the ratio of actual surface area to the projected surface area, the surface becomes more hydrophilic and the contact angle decreases. The increase in actual surface area occurs most likely filling the cavities on the surface [215]. Considering a sliding bubble in deposited channel, inertia, Marangoni and surface tension forces act on the bubble as aiding forces, while evaporation momentum, shear and friction forces act as resistive forces on the bubble. As a result of interplay among these forces, there are cases, where the bubble stops moving due to force equilibrium. When the stopping bubble at the critical position, which continues growing comes into play, additional repulsive forces are introduced as quasi-steady and asymmetrical growth forces. Deposition of TiO_2 nanoparticles changes the contact angle and surface characteristics. As a result, surface tension and friction forces vary, leading to an equilibrium position of bubble at a critical position, for the deposited channel in the experiments with ethanol. At a constant flow rate and applied heat flux, the bubble continues growing at this position, hence, surface tension force increases while shear force decreases until the resultant of the major forces pushes the bubble along the channel again.

The reason why the above-mentioned stopping bubble behavior is not observed for pure water or nanofluids may be that the diameter of initially produced air bubbles is not small enough to stick before it reaches the deposited area, where the bubble size is bigger than the critical size for stopping.

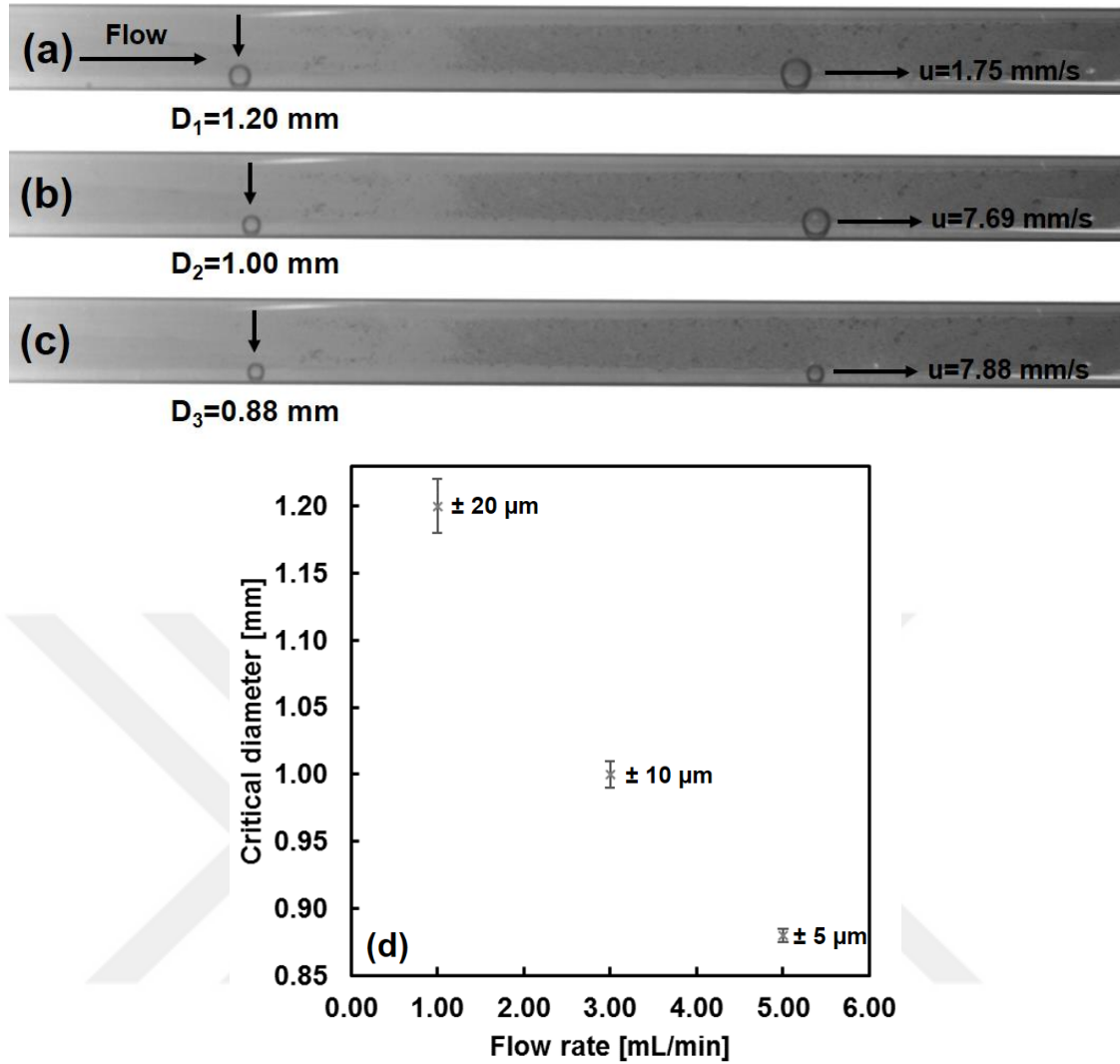


Figure 3.73: Critical diameter of bubbles on surface with TiO_2 nanoparticles deposition for pure ethanol as working fluid at heat flux of 0.8 kW/m^2 with flow rates of: a) 1 ml/min, b) 3 ml/min, c) 5 ml/min and d) critical diameter for above flow rates.

The effect of contact angle is a significant parameter in heat transfer. As decrease in contact angle leads to an increase in wettability, which is the main reason for critical heat flux enhancement in boiling studies [69]. Bubble motion for pure water and water-based nanofluid flowing on plain and TiO_2 nanoparticle deposited channels are shown in Figure 3.74.

In Figure 3.74(a), three different bubbles can be seen in the inlet, middle and outlet sections of the channel under adiabatic conditions. The change in the advancing angles of the bubbles in pure water is the evidence of the deposition effect on contact angle (bubbles I and III) while this is not observed for the plain channel (Figure 3.74(b)). On the other hand, advancing and receding contact angles of the bubbles in water based nanofluids are

almost the same, unlike the pure water case (Figure 3.74(c)). There is no considerable difference between the advancing and receding contact angles of the bubbles at different applied heat fluxes and different flow rates for the same working fluid, which can be seen when comparing Figure 3.74(a) to Figure 3.74(d) (bubbles III and IV). Furthermore, adding nanoparticles to water also alters the advancing angle with respect to pure water, Figure 3.74(d) and Figure 3.74(e) (bubbles IV and V).

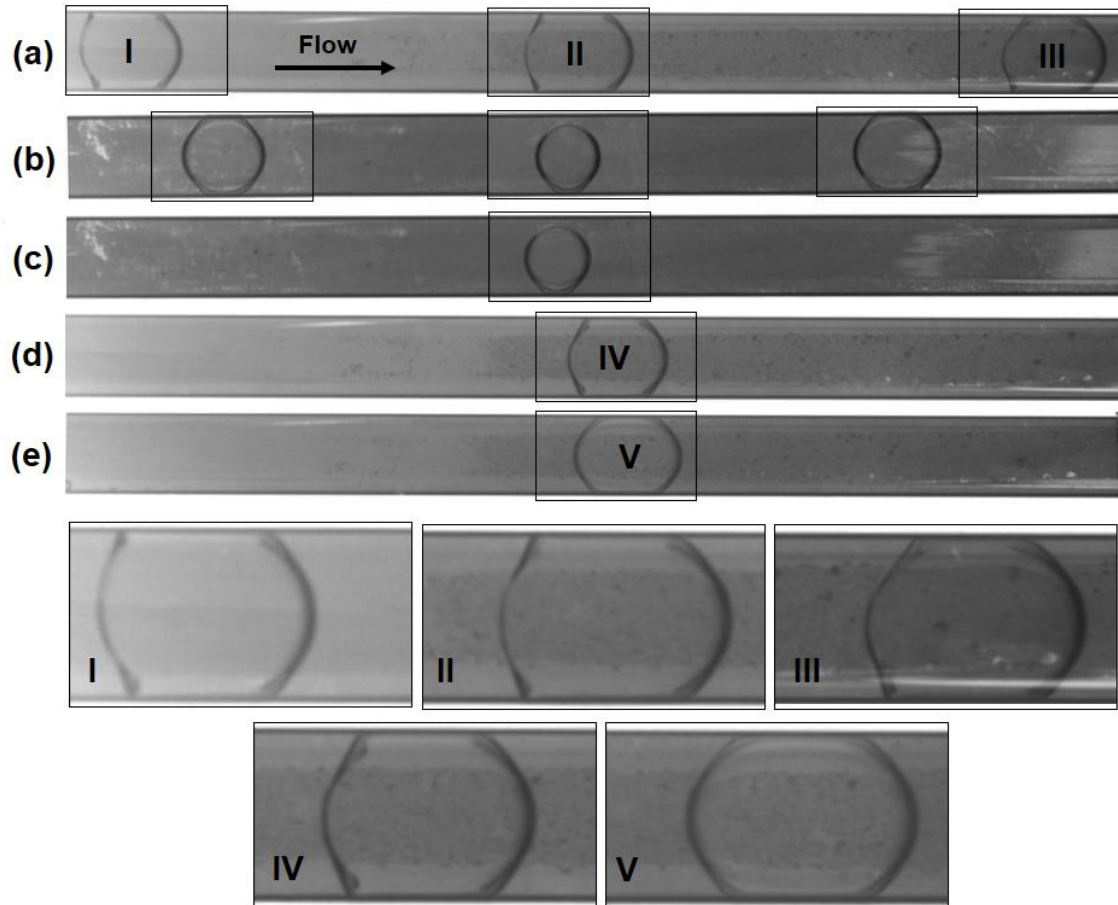


Figure 3.74: Bubble shapes for pure water and water-based nanofluid at flow rate of 1 ml/min. (a), (d) and (e) titania deposited surface. (b) and (c) bare surface. (a), (b) and (d) pure water. (c) and (e) water based nanofluid. (a), (b) and (c) adiabatic, (d) and (e) at heat flux of 1.4 kW/m^2 .

Similarly, bubble shapes for ethanol and ethanol-based nanofluids in plain and TiO_2 nanoparticle deposited channels are shown in Figure 3.75. Three different bubbles at the inlet, middle and outlet sections of the channel are demonstrated in Figure 3.75(a). Unlike the contact angles of the bubbles in pure water, Figure 3.74(a) (bubbles I and III), the deposition along the channel does not seem to too much affect the contact angle of a

bubble in pure ethanol (Figure 3.75(a)). However, significant differences can be observed when comparing the contact angle of the bubbles in the deposited channel with the plain one. The bubble shape in the plain channel is more like spherical for ethanol (Figure 3.75(c)), while this shape becomes slightly elliptical in the deposited channel (Figure 3.75(a)). With the addition of nanoparticles, this shape becomes even more elliptical in deposited channel (Figure 3.75(b)). It should be noted that, according to our observations, nanoparticles have no effect on bubble shape in the plain channel (Figure 3.75(d)).

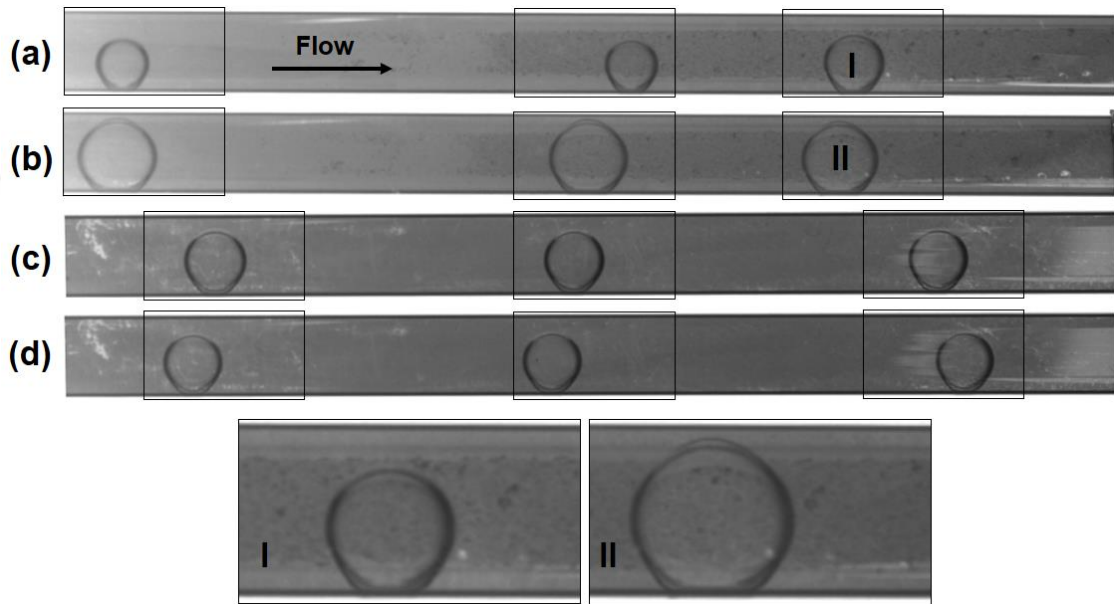


Figure 3.75: Advancing and receding contact angles for pure ethanol and ethanol-based nanofluid with 1 ml/min flow rate and adiabatic condition. (a) and (b) titania deposited surface. (c) and (d) bare surface. (a) and (c) pure ethanol. (b) and (d) ethanol-based nanofluid.

3.5. Subcooled Flow Boiling Heat Transfer

The experiments were performed in different test sections with heated lengths of 7 cm and 12 cm. First, validations of the experimental setup were done using the pure base fluid. For this, heat transfer coefficients corresponding to the single-phase conditions were compared with the predictions of Shah and London [47], which were obtained using the Eqs. (3.1) to (3.3) as previously mentioned:

$$h_{theo} = \frac{Nu_{dev} \cdot k_{bf}}{D_i} \quad (3.1)$$

$$Nu_{dev} = 0.517(f \cdot Re)^{1/3} (x^*)^{-1/3} \quad (3.2)$$

$$x^* = \frac{x_{dev}}{D_i \cdot Re \cdot Pr} \quad (3.3)$$

where k is thermal conductivity and f_{theo} is theoretical friction factor which can be calculated by applying the classical laminar flow formula of $f_{theo} = 64 / Re$ for hydrodynamically fully developed laminar flows in the circular channels. The aforementioned correlation is recommended for thermally developing and hydrodynamically developed flow conditions, which match with the working conditions of this study. As can be seen from Figure 3.76(a) and Figure 3.76(b), the single-phase heat transfer data of the pure base fluid can be predicted within $\pm 15\%$ for mass fluxes of 1200 kg/m²s and 3400 kg/m²s.

The experimental setup was further validated by performing pressure drop experiments with pure water to acquire friction factors. As can be observed from Figure 3.76(c), experimental friction factors can be predicted within $\pm 20\%$ by the classical theoretical friction factor expression recommended for fully developed laminar flows in circular channels ($f_{theo} = 64 / Re$).

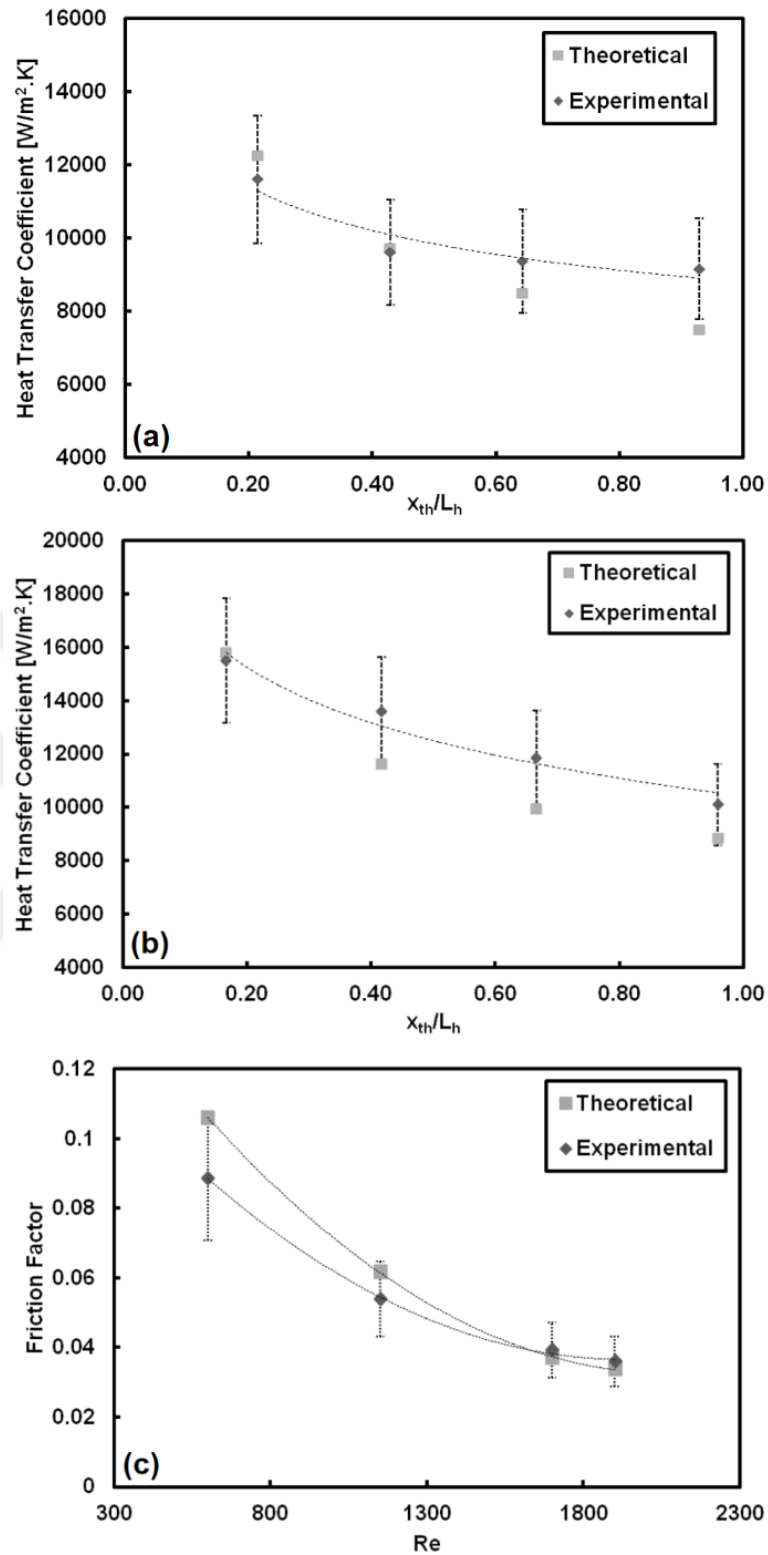


Figure 3.76: Comparison between experimental and theoretical heat transfer correlation results for pure water single phase heat transfer coefficients at: a) $G=1200 \text{ kg/m}^2\text{s}$, and b) $G=3400 \text{ kg/m}^2\text{s}$. c) Comparison of theoretical and experimental friction factors for pure water.

3.5.1. Results for Low Mass Fractions

The results of subcooled flow boiling heat transfer of dilute $\text{Al}_2\text{O}_3/\text{water}$ nanofluids with mass fractions of 0.05 wt.% and 0.2 wt.% in a horizontal microtube of an inner diameter of 502 μm are presented in this section for different mass fluxes. Figure 3.77 displays local heat transfer coefficients along the heated length at different heat and mass fluxes. The region, where the surface temperatures are smaller than the saturation temperature, is considered as the single-phase region, while surface temperatures are higher than the saturation temperature in the boiling region. In the single phase region, heat transfer coefficient has a decreasing trend, which is expected in thermally developing flows, where heat transfer coefficient is the highest at the inlet. With the increase in thermal boundary layer thickness along the channel starting from the inlet, heat transfer coefficient decreases, until subcooled boiling conditions are attained. A rise in applied heat flux at a fixed mass flux extends the boiling region and shifts the minimum of the profile toward the inlet. The same trends exist for the other mass flux. The effect of mass flux on local heat transfer coefficient is also presented in the same figure. Higher mass fluxes lead to enhancement in convective heat transfer. This trend is another indication of the role of single-phase component in subcooled boiling heat transfer and presence of single-phase flow effects [48,49].

In the subcooled boiling region, heat transfer coefficient has two components, namely single-phase and nucleate boiling components [50]. In the present study, local vapor qualities vary between -0.07 to -0.048. The rise in heat transfer coefficient along the boiling region is due to the increasing dominance of nucleate boiling component with increasing superheat along the channels, which leads to more bubbles and enhanced nucleate boiling.

The ratio of heat transfer coefficients of nanofluids to those of pure base fluid is demonstrated in Figure 3.78. As can be seen, there is not any considerable change in heat transfer at the low mass flux. However, the change in heat transfer becomes visible with increasing mass flux. Moreover, the ratio becomes bigger (up to 7%) at larger mass fraction, namely 0.2 wt.%. The enhancement in heat transfer at higher mass fluxes is associated with growing convective effects. Since the inlet Reynolds number at high mass flux is 1700, the flow conditions are close for the transition to turbulent flow. Therefore, mixing and relative particle motion more effectively occurs at higher mass flux, 3400 $\text{kg/m}^2\text{s}$. Such observations were also reported for single-phase nanofluid flows [181].

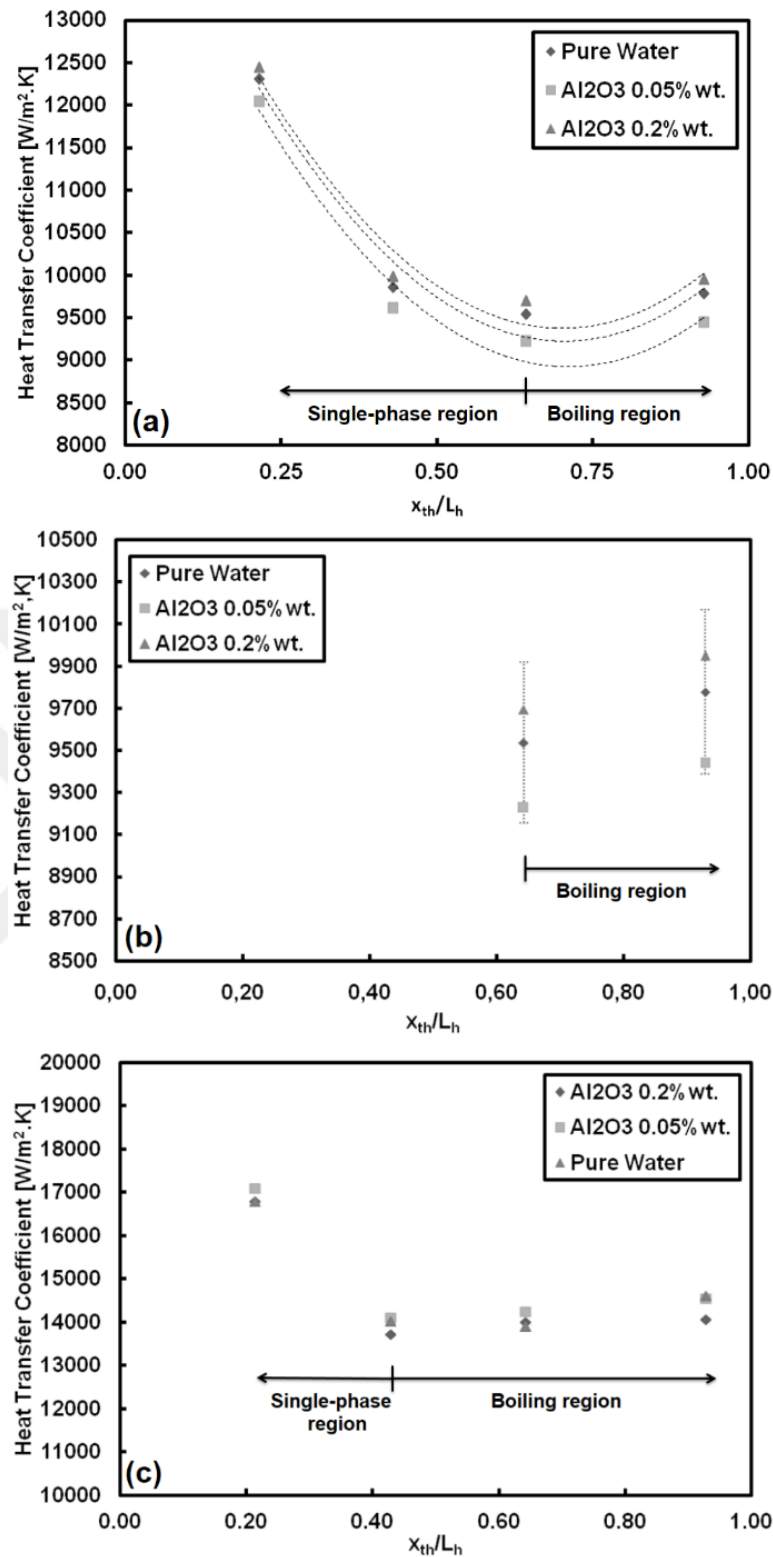


Figure 3.77: Subcooled heat transfer coefficients: a) $G=1200$ kg/m²s, $q''=344$ kW/m², b) subcooled region at $G=1200$ kg/m²s and $q''=344$ kW/m², and c) $G=3400$ kg/m²s and $q''=628$ kW/m², which show no considerable alteration for low mass fractions (0.05 wt.% and 0.2 wt.%).

As can be seen in Figure 3.77 and Figure 3.78, for nanofluids with low mass fractions, although heat transfer is enhanced with the increase in the concentration from 0.05 wt.% to 0.2 wt.%, it is still not considered as a large enhancement.

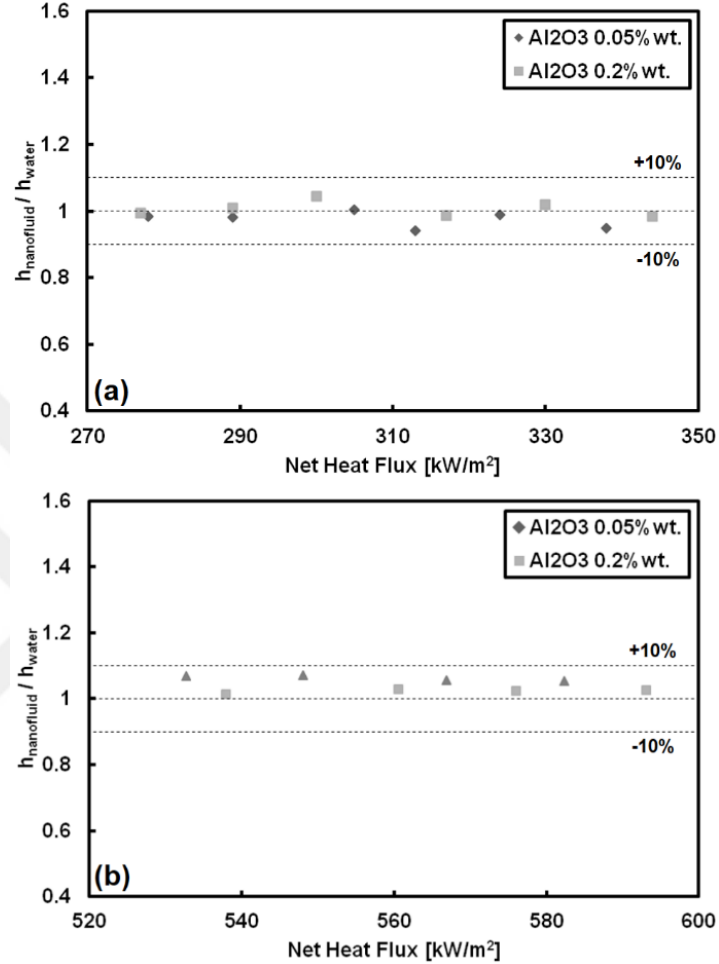


Figure 3.78: Ratio of heat transfer coefficients of nanofluids to those of pure water at: a) $G=1200 \text{ kg/m}^2\text{s}$, and b) $G=3400 \text{ kg/m}^2\text{s}$. No considerable change in heat transfer can be seen at low mass flux, while a slight increase of 7% (still not considerable) is obtained at high mass flux.

Pressure drop profiles are included in Figure 3.79 as a function of heat flux. In the single-phase region, pressure drop has a decreasing trend due to decreasing viscosity of the working fluid. The pressure drops are comparable to the adiabatic pressure drop, which is typical for subcooled or low quality flow boiling and was also documented in the literature [51].

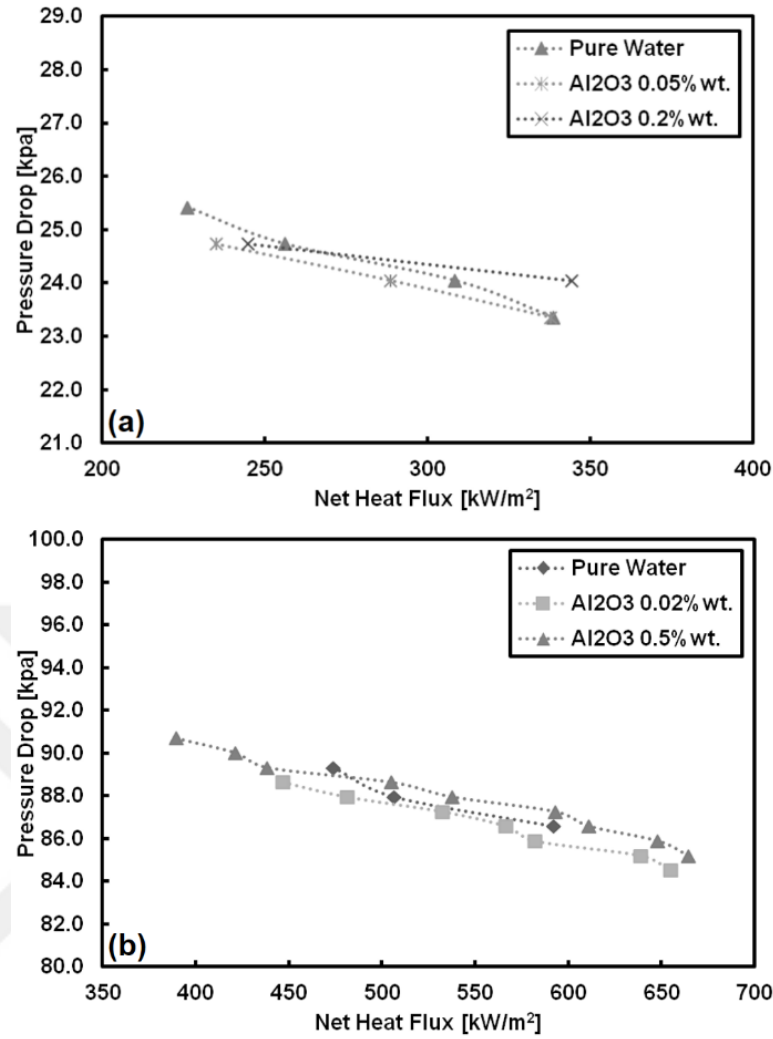


Figure 3.79: Pressure drop profiles as a function of heat flux at: a) $G=1200 \text{ kg/m}^2\text{s}$, and b) $G=3400 \text{ kg/m}^2\text{s}$.

3.5.2. Results for High Mass Fractions

The comparison between local heat transfer coefficients of nanofluids with high mass fractions ($\geq 0.5\%$) and those of pure base fluid is shown in Figure 3.80. As can be seen, heat transfer deteriorates at high mass fraction. This is due to agglomeration, clogging and deposition of nanoparticles upon boiling and subsequent deterioration in heat transfer [20,34]. Such effects are less dominant in subcooled flow boiling because of less evaporation of base fluid and more pronounced relative motion of nanoparticles within the fluid at rather high mass fluxes, where subcooled boiling is more likely to exist.

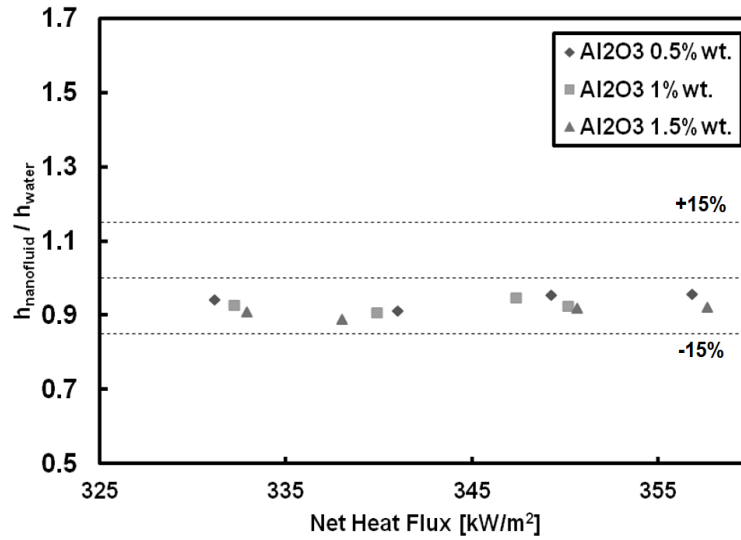


Figure 3.80: Ratio of heat transfer coefficients of nanofluids to those of water at $G=3400 \text{ kg/m}^2\text{s}$, which show heat transfer deterioration at high mass fractions (0.5wt.%, 1 wt.%, 1.5 wt.%)

As a result, stability is restored to a certain extent as can be observed in almost unchanged DLS results after boiling experiments for nanofluids with the mass fraction of 0.2 wt.% (Figure 2.8). The nanofluid samples of fractions of 0.05 wt.% and 0.5 wt.% were examined before and after the heat transfer experiments. For the low fraction ratio (0.05 wt.%), almost the same major peak is observed for the samples corresponding to before and after heat transfer experiments. This implies that agglomeration did not occur at low mass fraction (Figure 3.81(a)). At a high mass fraction (0.5 wt.%), however, both minor and major peaks can be seen (Figure 3.81(b)). Minor peak in this profile indicates agglomeration and decreased stability. As the nanoparticles agglomerates at high mass fractions, as evident in the minor peak in Figure 3.81(b), heat transfer is affected resulting in reduced heat transfer coefficients. In other words, DLS measurements reveal that the nanoparticle stability is not affected at low mass fraction, while both nanoparticle agglomeration and decrease in the stability start to occur at the high mass fraction, which is also linked to the deterioration in heat transfer.

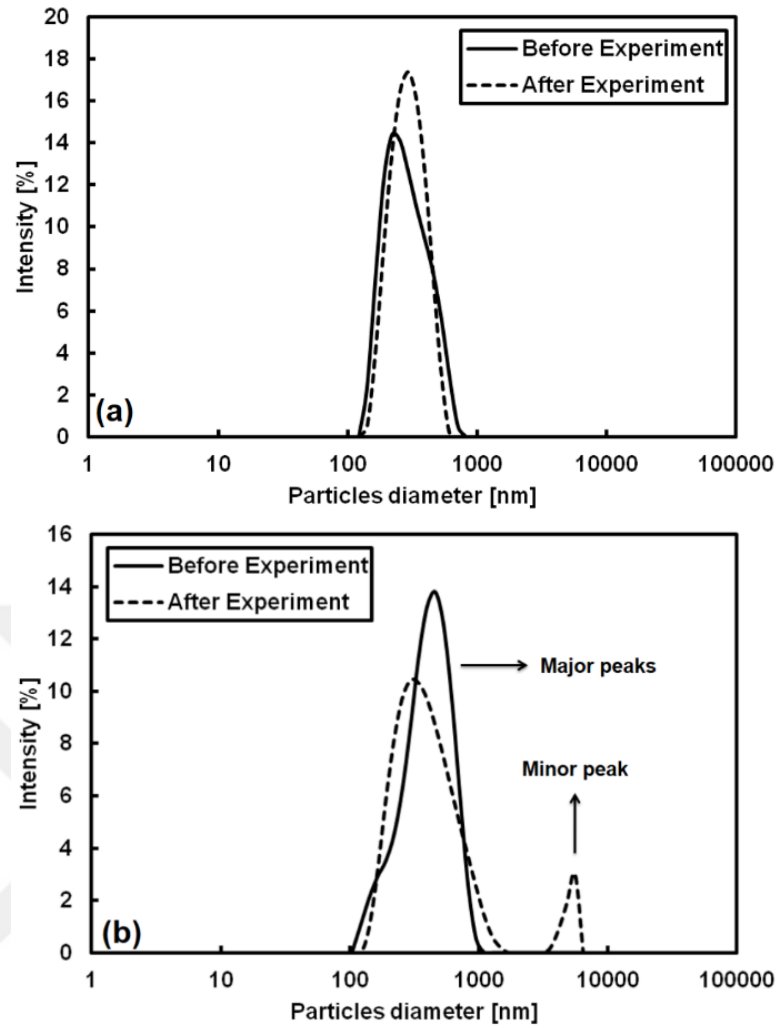


Figure 3.81: DLS measurements of the nanofluids for: a) dilute (0.05 wt.%), and b) high fraction ratio (0.5 wt.%); before and after subcooled flow boiling experiments. Minor peak for the nanofluid with a higher mass fraction revealed the agglomeration of the nanofluid after the experiments, which is a significant cause for heat transfer deterioration.

3.5.3. Surface Characterizations

In addition, the surfaces were characterized after the experiments to further provide explanation about heat transfer enhancement at low mass fractions. SEM images of the microtube surface after boiling experiments are displayed in Figure 3.82. In Figure 3.82(a), the cross-section of the tube can be seen, while the inner surface of a longitudinally cut microtube is depicted in Figure 3.82(b). The SEM images of the inner surface reveal that there is no detectable deposition of nanoparticles after the experiments at low mass fractions (0.02 wt.%, at the heat flux and mass flux of 590 kW/m^2 and $G=3400$

kg/m²s, respectively). This finding clearly shows the connection between heat transfer and nanofluid stability.

Besides the agglomeration of nanoparticles upon boiling, deposition of nanoparticles on the surface as a result of the lack of stability is another important parameter, which negatively affects heat transfer at high mass fraction ratios. To demonstrate nanoparticle deposition, microscopy images of the cross section of the microtube and the inner surface of the longitudinally cut microtube after boiling experiments are shown in Figure 3.82(c) and Figure 3.82(d) for the nanofluid sample with the mass fraction ratio of 1.5 wt.% at the heat flux of 590 kW/m² and the mass flux of 3400 kg/m²s.

In addition, the SEM (JEOL JIB-4601F MultiBeam Platform) SE-images are taken from the test tube after the experiments at the high mass fraction ratio, namely 1.5 wt.% and are displayed in Figure 3.82(e-h). It can be clearly observed that the nanoparticles deposit on the surface upon boiling. The SEM-SE images are obtained in different magnifications in order to show the distribution and the morphology of the agglomerated nanoparticles, which are adhered, adsorbed and deposited on the inner surface of the microtube.

Energy Dispersive Spectrometry (EDS) analyses were performed (Oxford Xmax-N EDS System on JEOL JIB-4601F MultiBeam Platform) by determining the elemental composition of the particles in order to prove that nanoparticles, which deposited on the inner surface of the tube, are Alumina nanoparticles. EDS spectra were taken from the regions on the deposition of nanoparticles labeled as Spot 1. The SEM images and the corresponding EDS spectra-quantifications charts are included in Figure 3.82(j). The EDS spectra and related quantitative analysis reveal that the nanoparticles deposited on the inner surface are indeed agglomerated Alumina nanoparticles. The elements other than Al and O in the chart are collected signals from the substrate below the nanoparticles and correspond to the stainless steel tube inner surfaces.

The change in wettability of the surface with deposition of nanoparticles, especially at high mass fractions, has a big impact on boiling heat transfer. Therefore, the contact angles are measured on both the plain surface and the surface, where deposition of nanoparticles (at the highest fraction ratio, namely 1.5 wt.%) occurred upon boiling. Figure 3.82(k) and Figure 3.82(l) show the contact angles of pure water on the sample surfaces including the plain surface and the surface with nanoparticle deposition.

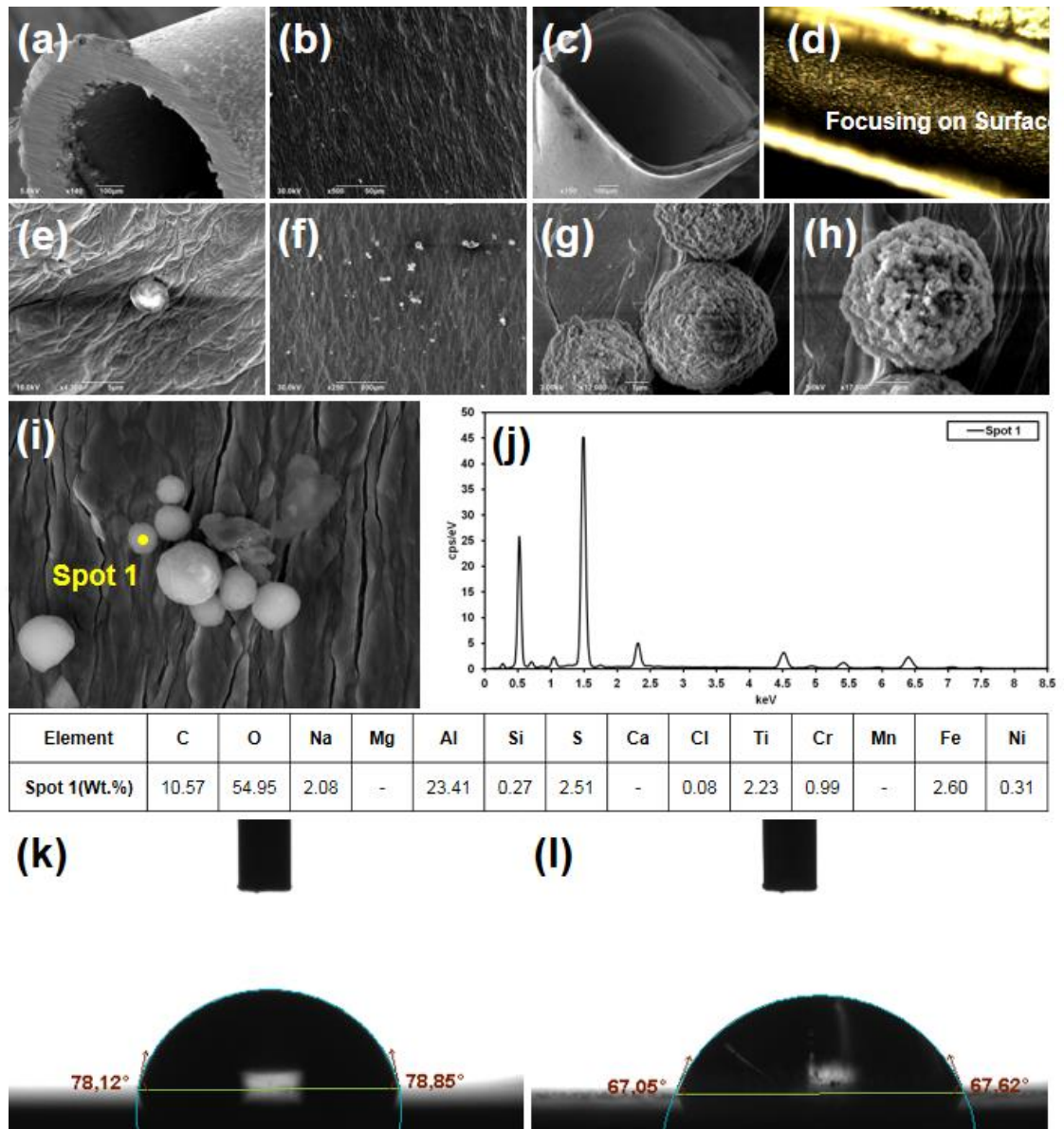


Figure 3.82: Characterization of the microtube surfaces after the experiments. SEM images of the microtube: (a) cross-section, and (b) inner surface after the experiment ($\phi = 0.02$ wt.%, $q'' = 590$ kW/m², $G = 3400$ kg/m²s) which show no nanoparticle deposition, thus no deterioration for nanofluids at low mass fractions, (c) SEM image of the microtube cross section, and (d) light microscopy image of the microtube inner surface ($\phi = 1.5$ wt.%, $q'' = 590$ kW/m², $G = 3400$ kg/m²s), SEM images of the inner surface along with nanoparticles inside the microtube: (e) cross-section of the tube and (f-h) inner surface of the microtube showing the deposition of nanoparticles ($\phi = 1.5$ wt.%, $q'' = 590$ kW/m², $G = 3400$ kg/m²s), (i-j) SEM image and the corresponding EDS spectra-quantifications plot and chart of the region of deposition on the microtube inner surface ($\phi = 1.5$ wt.%, $q'' = 590$ kW/m², $G = 3400$ kg/m²s), which is a cause for heat transfer deterioration for nanofluids with high mass fractions (due to the nanoparticle deposition on the surface), and contact angles of surface: (k) before, and (l) after the nanoparticle deposition ($\phi = 1.5$ wt.%), which demonstrates the surface wettability change due to nanoparticle deposition, a possible reason for heat transfer deterioration for nanofluids with high mass fractions.

For the plain surface, the contact angle is approximately 78°, while it decreases to approximately 67°, when the surface with nanoparticle deposition is utilized. As a result, it can be concluded that the surface becomes more hydrophilic with Alumina nanoparticle deposition, which has a negative effect on boiling heat transfer due to reduced active nucleation sites and provides additional explanation for deterioration in boiling heat transfer at high mass fractions.

3.6. Stability of Nanofluids

The first set of results are demonstrated in Figure 3.83. According to the results, the average hydrodynamic diameter of nanoparticles mixed in water increased up to more than three times after heating the sample up to 99.8°C as a consequence of the agglomeration of nanoparticles during heating. However, some of the nanoparticles are deagglomerated at the inlet pressure of 10 bars, which corresponds to a cavitation number of 1.37. The intensity of cavitation along the channel is characterized by cavitation number, $\sigma = (P_{ref} - P_v) / 0.5\rho V_{ref}^2$, where P_{ref} is reference pressure, which is upstream pressure in this work, P_v is vapor saturation pressure (vapor saturation pressure for the working fluid is approximately 3500 Pa), and V_{ref} is the reference velocity at the outlet of the microchannel, which varies according to the flow conditions. The average hydrodynamic diameter decreases to 75% (from 1718 nm to 1281 nm) with respect to that of the heated sample. With a further increase in the pressure to 60 bars (corresponding to a cavitation number of 0.6), the deagglomeration is even more enhanced and a size reduction of 52% (from 1718 nm to 825 nm) can be achieved.

The average hydrodynamic diameters of nanoparticles in the second set of experiments involving higher pressure tests are illustrated in Figure 3.84. According to the mean diameter measurements of the prepared and heated nanofluids, the same trend of the size increase after the heating process can be observed. The size becomes approximately three times bigger than the size of nanoparticles of recently prepared nanofluids. After the cavitation experiments at a pressure of 60 bars (at a cavitation number of 0.6), the mean hydrodynamic diameter of nanoparticles significantly decreases with respect to that of the heated nanofluids and even that of the recently prepared

nanofluid. The nanoparticle size drops to 515 nm and 255 nm after the experiments at pressures of 60 bars and 90 bars (corresponding to cavitation numbers of 0.6 and 0.2), respectively, which correspond to reductions of 73% and 87% with respect to the size of particles in the heated samples. The results show that the proposed method could prevent agglomeration of nanoparticles and could improve stability and reusability of nanofluids. This effect is more pronounced at increased intensity in cavitating flows, which is characterized by the cavitation number.

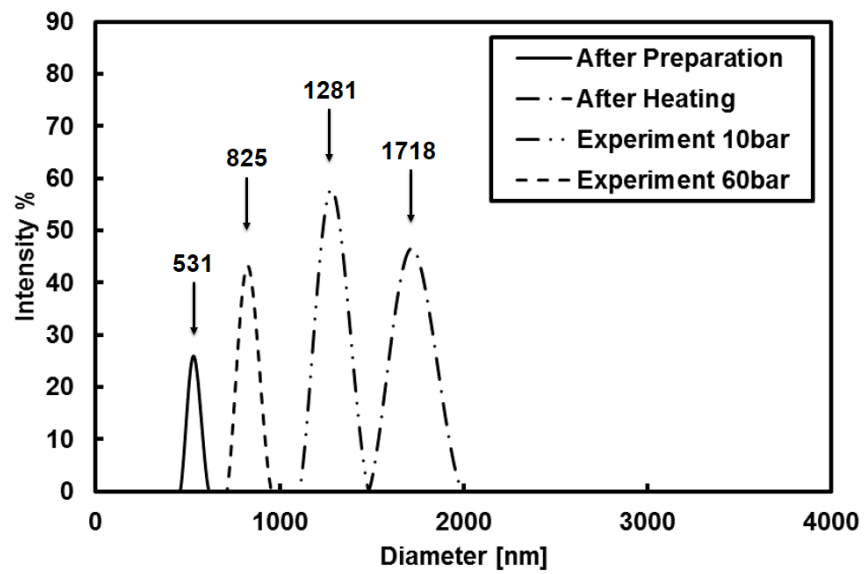


Figure 3.83: Dynamic light scattering results of nanofluid samples for the first set of experiments.

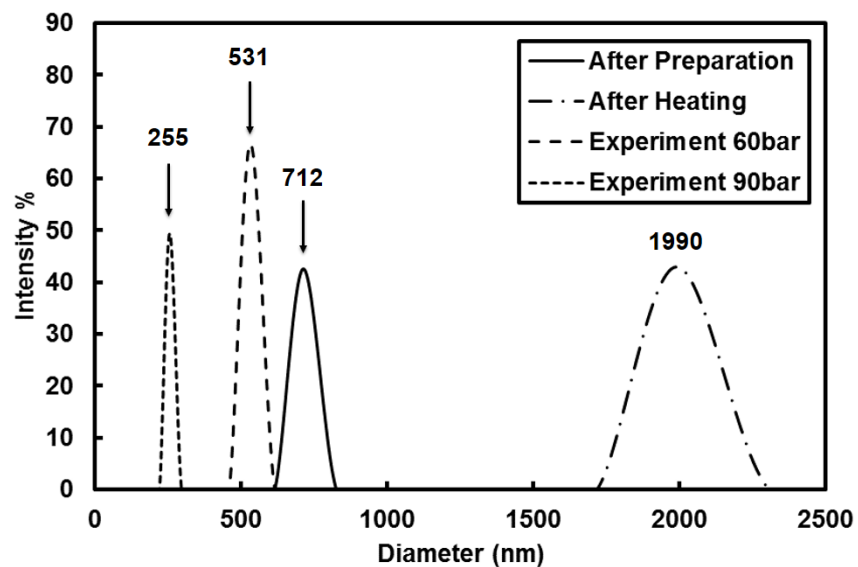


Figure 3.84: Dynamic light scattering results of nanofluid samples for the second set of experiments.

In the micro scale, cavitation typically incepts toward the inlet of the flow restrictive element due to potential low pressure zones and more pronounced surface effects in the small scale [272]. As a result, surface nuclei are the key parameter for the inception of hydrodynamic cavitation. In this study, the length of the micro orifice was kept as short as possible so that the emerging bubbles would collapse downstream of the orifice. Their collapse generates shock waves as well as jet flows, which exert very high stress on the exposed surfaces/structures [273]. This results in a destructive effect on the agglomerated structures of nanoparticles and recovers the stability of nanofluids at the same time.



CHAPTER 4

4. CONCLUSIONS

In this thesis, major conclusions related to heat transfer and bubble dynamics by employing the nanofluids with various nanoparticles and fraction ratios were concluded. The most important ones are as follow:

1) In the study related to the single-phase heat transfer, pressure drop and heat transfer characteristics of water-based nanofluids with TiO_2 and Al_2O_3 nanoparticles are investigated in horizontal smooth hypodermic microtubes over a wide variety of Reynolds numbers and nanoparticle mass fractions under thermally developing flow conditions.

- Experimental friction factors and heat transfer coefficients could be predicted within $\pm 10\%$ and $\pm 15\%$, respectively, by considering appropriate thermophysical properties of nanofluids and using existing correlations in the literature.
- Experimental results show that mass fraction is not effective over heat transfer coefficient for Reynolds number less than 1200, beyond which heat transfer coefficients increase with mass fraction.
- As Reynolds number increases, flow starts to transition to turbulent flow, and as a result, heat transfer enhancement becomes detectable, which is due to enhanced mixing of nanoparticles at higher Reynolds numbers.
- While the stability of nanofluids is preserved after the experiments, a gradual agglomeration of nanoparticles in nanofluids might occur with time and might be accelerated with their repetitive use in the experimental setup and at elevated temperatures. Therefore, methods for improving their stability should be developed so that implementation of nanofluids to commercial products such as refrigerators and heat exchangers would be possible.

2) In the study of bubble dynamics in pool boiling in the presence of nanoparticles, subcooled boiling experiments were performed on a fine platinum wire with a diameter of 250 μm at fluid bulk temperatures between 30-50°C and atmospheric pressure. According to our experimental observations, it can be concluded that mechanisms such as bubble-bubble and nucleation site-bubble interactions, surface tension, Marangoni convection, accumulation of non-condensable gases inside the bubbles, structural disjoining pressure, pinning and nanoparticles deposition and underlying forces play an important role in bubble dynamics and pool boiling. Bubble dynamics events including migration, coalescence, leaping and detachment are common in the nanofluids with mass fractions of 0.002 wt.% and 0.005 wt.%. On the other hand, oscillation, dancing and sticking processes are observed only in the nanofluid with a higher mass fraction of 0.005 wt.%, which presents the nanoparticle effect on bubble dynamics. Major conclusions drawn from this study can be summarized as follows:

- In the coalescence process, the size of the eventually formed bubble after the coalescence is smaller than the sum of the sizes of original bubbles, sometimes even smaller than the size of the originally bigger bubble due to the mass transfer with the form of vapor pack or small bubble flung out of the merged bubble.
- Vertical leaping of the bubble is initiated by nucleation jet flow.
- In the leaping process, the combination of lift force applied by the jet flow and the buoyancy force causes bubble rise above the wire. After the bubble rise, its diameter starts to decrease due to condensation. Thereafter, the Marangoni thrust force pushes the bubble back to the wire.
- In some cases, when a bubble reaches a nucleation jet, the bubble stops and starts to decrease in size, then reverses its direction and continues with its normal growth process.
- When a bubble reaches a critical diameter, the buoyancy force overcomes the capillary force. Thus, the detachment process occurs. Besides, lifting force arises either from the nucleation jet or coalescence impact and initiates the bubble detachment.
- Nanoparticles seem to facilitate the oscillation behavior of a moving bubble between two stationary bubbles, due to an increase in the friction force,

thereby decreasing the contact angle and increasing the contact line force. Thus, more stationary bubbles form.

- In few cases, deposition of nanoparticles on the surface, changes the Marangoni force and surface tension force and causes bubble jump while approaching a smaller one just before coalescence.
- Increasing the nanoparticles concentration causes the bubble sticking behavior on the wire possibly due to the increase in both friction force associated with nanoparticle deposition and horizontal component of contact line force.
- At a higher mass fraction, stick-slip-like behavior is observed. Nucleation jets or tiny bubbles are responsible for this event.

3) In the study for enhancing the pool boiling heat transfer, pool boiling heat transfer of water-based nanofluids consisting of TiO_2 and CuO nanoparticles with average diameters of 20-50 nm and 10-20 nm, respectively, and of different mass fractions, were experimentally investigated on an aluminum plane heater. The aim was to assess the effect of adding nanoparticles of different types to the base fluid (water) on the pool boiling heat transfer performance. Visualization was performed to support the heat transfer results.

- According to the results, we observed that mass fraction plays an important role on pool boiling heat transfer.
- For TiO_2 nanoparticles/water nanofluids, the most dilute nanofluid (mass fraction of 0.001%) enhanced heat transfer (up to 15%), which was attributed to more emerging bubbles with a smaller size and more spherical shape, while the amount of enhancement decreased down to below 5% with the increase in mass fraction.
- For CuO nanoparticles/water nanofluids, the increase in mass fraction led to the augmentation in heat transfer. The highest enhancement relative to pure water (up to 36%) was attained at the highest mass fraction (0.2%).
- Deposition of CuO nanoparticles after the experiments due to their typical limited stability had a positive effect on heat transfer, which was proven with the repeated pure base fluid tests on deposited surfaces. Therefore, the deposition occurring after each experiment might alter the results and should be carefully taken into consideration.

4) In the study of bubble dynamics in minichannels, experimental results on injected single air bubble dynamics in rectangular channels were obtained under adiabatic and diabatic conditions with the aid of visualization. The characteristics of bubble movement were analyzed by taking the effect of fluid type, heat flux, flow rate and deposition into consideration.

- When heat flux was applied to the system, bubble speed become slightly lower than the adiabatic case for pure water at low flow rate. However, an increase in the heat flux raised the bubble speed at high flow rate. In the deposited channel, bubbles stopped at a critical position for pure ethanol as a result of force equilibrium.
- At fixed applied heat flux, the stopping bubble continued growing until it reached a critical diameter and then it restarted with the sliding motion. This critical diameter was dependent on flow rate but not heat flux. Bubble nose speed was larger for pure ethanol than that for ethanol-based nanofluid, which indicated that nanoparticles induced a resistive effect on elongation process.
- Nanoparticles deposition altered the advancing contact angle for pure water and it altered the shape of the bubble for pure ethanol and ethanol-based nanofluid and made the bubble more elliptical.

5) In the study for introducing a novel study for increasing the stability of the nanofluids, the feasibility of decreasing the nanoparticle cluster size and improving nanofluid stability with cavitating flows in micro orifices was experimentally investigated using the agglomerated Al_2O_3 nanoparticles dispersed in water.

- The proposed method was proven to prevent agglomeration and deposition of nanoparticles in nanofluids and could contribute to the extensive use and enhanced durability and reusability of nanofluids in many applications such as electronics cooling, microreactors, biotechnology, air conditioning, refrigerators, and thermophotovoltaic cells as next generation working fluids. This approach is also more economical since no expensive chemicals are required.
- The integration of flow restrictive elements could be accomplished by standard fabrication tools and techniques and standards fittings. To amplify the effectiveness, the nanofluids could be passed through a number of such flow restrictive elements configured in a series.

- The experimental setup consists of metallic tubes/fittings, which can withstand high pressures. Since surface nuclei are significant in small scale, surface modifications (near low pressure zones) can be made on the channel surface via coatings at desired locations (such as hydrophobic coatings), which can lead to cavitation incipience at lower pressures. There is therefore room for improving the conditions with possible surface modifications.

6) Due to scarcity of studies in the literature on micro scale subcooled flow boiling in horizontal microtubes for Al_2O_3 nanoparticles/water nanofluids, the experiments related to this topic were performed for nanofluids flows in stainless steel microtubes with inner and outer diameters of $\sim 502\text{ }\mu\text{m}$ and $\sim 717\text{ }\mu\text{m}$, respectively.

- Subcooled boiling heat transfer coefficients for dilute nanofluids having mass fractions of 0.05 wt.% and 0.2 wt.% had no considerable change (with a slight increase up to 7%) at low and high mass fluxes and applied heat fluxes.
- For nanofluids with a higher concentration, such as mass fractions of 0.5 wt.%, 1 wt.% and 1.5 wt.%, subcooled boiling heat transfer deteriorated down to 12% at the highest mass fraction.
- The results of dynamic light scattering (DLS) measurements for a dilute sample with mass fraction of 0.05 wt.% revealed that the hydrodynamic diameter of dispersed nanoparticles in the nanofluid remained approximately the same for the samples before and after the experiments.
- For a sample with a higher mass fraction, namely 0.5 wt.%, DLS measurements results showed a minor peak besides the major peak for the sample after the experiments representing the agglomeration of the mixed nanoparticles in the nanofluid, which is one of the possible reasons of heat transfer degradation for nanofluids with higher mass fraction ratios.
- Scanning Electron Microcopy (SEM) images of stainless steel microtube's inner walls showed that while there was no nanoparticle deposition for dilute nanofluids after the experiments, agglomerated nanoparticles were deposited on the inner walls for nanofluids with higher mass fractions (with mass fractions larger than 0.2 wt.%), pointing out the other possible reason for heat transfer deterioration at higher mass fractions.

The research related to this thesis has the following contributions to the scientific community: a) presentation of conditions for heat transfer enhancements with Al_2O_3 and TiO_2 nanoparticles/water nanofluids, b) revealing potential of pool boiling heat transfer enhancement by employing different concentrations of TiO_2 and CuO nanoparticles /water nanofluids and corresponding conditions, c) presentation and explanation of the effect of Al_2O_3 nanoparticles with low and high fraction ratios on subcooled flow boiling, d) comprehensive investigation of bubble dynamics for TiO_2 nanoparticles/water nanofluids on fine wires for pool boiling and nanofluids flows along the rectangular minichannels and presentation and explanation of new bubble behavior, especially for pool boiling on fine wires for nanofluids with different concentrations, and e) presentation of a novel method for increasing the stability of nanofluids by using cavitating flows was suggested and was proved for Al_2O_3 nanoparticles/water nanofluids for different pressure conditions.

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