Renewable and Sustainable Energy Reviews 15 (2011) 2342–2354

Contents lists available at ScienceDirect

Renewable and Sustainable Energy Reviews

journal homepage: www.elsevier.com/locate/rser

A review of boiling and convective heat transfer with nanofluids

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ARTICLE INFO

Article history:
Received 23 September 2010
Accepted 31 January 2011

Keywords:
Nanofluids
Boiling
Heat transfer coefficient
Critical heat flux
Nanoparticles
Superheat
Reynolds number

ABSTRACT

Nanofluids have evoked immense interest from researchers of multi-disciplines from around the globe due to their fascinating thermophysical properties and numerous potential benefits and applications in important fields such as microelectronics, microfluidics, transportation, and biomedical. However, there are many controversies and inconsistencies in reported arguments and experimental results on various thermal characteristics such as effective thermal conductivity, convective heat transfer coefficient and boiling heat transfer rate of nanofluids. As of today, researchers have mostly focused on anomalous thermal conductivity of nanofluids. Although investigations on boiling, droplet spreading, and convective heat transfer are very important in order to exploit nanofluids as the next generation coolants, considerably less efforts have been made on these major features of nanofluids. In this paper, these important cooling features—boiling, spreading, and convective heat transfers of nanofluids are presented together with exhaustive review of research and development made in these areas of nanofluids.

Contents

1. Introduction

With an ever-increasing thermal load due to smaller features of microelectronic devices and more power output, cooling for maintaining desirable performance and durability of such devices is one of the most important technical issues in many high-tech industries. The conventional method to increase the cooling rate is to use extended heat transfer surfaces. However, this approach requires an undesirable increase in the size of the thermal management system. In addition, the inherently poor thermal properties of traditional heat transfer fluids such as water, ethylene glycol (EG) or engine oil (EO) greatly limit the cooling performance. Thus, these conventional cooling techniques are not suitable to meet the cooling demand of these high-tech industries. Although thermal conductivity of a fluid plays

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1364-0321/$ – see front matter
doi:10.1016/j.rser.2011.02.016

Nomenclature

C_{sf}	an empirical coefficient in Rohsenow correlation
c_p	specific heat (J/kg K)
D	tube diameter (m)
f	friction factor
g	acceleration of gravity (m/s^2)
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
K	a constant in Zuber correlation
L	length (m)
$\dot{m}$	mass flow rate (kg/s)
q	supplied heat (W)
q''	heat flux (W/m^2)
T	temperature (K or °C)

Greek symbols

ϕ	particle volume fraction
ρ	density (kg/m^3)
μ	dynamic viscosity (kg/m s)
σ	surface tension (N/m)

Subscripts

b	bulk
f	fluid
g	vapor phase
i	inner
o	outer
nf	nanofluid
p	particle
s	saturated liquid
v	laminar sublayer
w	wall surface (heater or tube)

Nondimensional numbers

Nu	Nusselt number
Pe	Peclet number
Pr	Prandtl number
Re	Reynolds number

Abbreviations

BHF	burnout heat flux
BHTC	boiling heat transfer coefficient
CHF	critical heat flux
CNT	carbon nanotube
DNB	departure from nucleate boiling
Ni–Cr	nickel–chromium (nichrome)
ONB	onset of nucleate boiling

a vital role in the development of energy-efficient heat transfer equipment it is known that fluids possess order-of-magnitude smaller thermal conductivity than metallic or nonmetallic particulates. For example, thermal conductivities of water (0.607 W/m K) and engine oil (0.145 W/m K) are about 5000 times and 21,000 times, respectively smaller than that of carbon nanotubes (e.g. 3000 W/m K for multi-walled carbon nanotubes, MWCNTs). Fig. 1 compares thermal conductivities of various commonly used liquids and materials at room temperature. It can be seen from Fig. 1 that most of the commonly used metallic and nonmetallic particulates posses orders magnitude higher thermal conductivity than those of liquids. Fig. 1 also shows that metals have higher thermal conductivity compared to ceramics (or metallic oxides). Therefore, the thermal conductivities of fluids that contain sus-

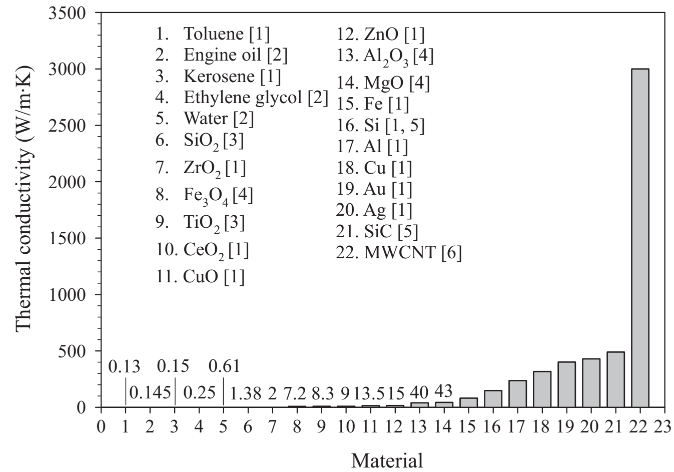

Fig. 1. Thermal conductivities of commonly used liquids and materials at room temperature (some small thermal conductivity values are shown above the numbers of the liquid or material) [1–6].

pended metallic or nonmetallic particles or tubes are expected to be significantly higher than those of traditional heat transfer fluids.

Although nanoparticles-suspensions were used in heat transfer studies as early as 1984 by Yang and Maa [7] and then in 1993 by Masuda et al. [3], it is only in 1995 that Choi [8] at Argonne National Laboratory of USA coined the concept of "nanofluid" to meet the cooling challenges facing many high-tech industries. Apart from Masuda et al. [3], there are other researchers who also used the term "nanofluid" or were involved with research on thermal conductivity of nanoparticle suspensions during 1993. Gass et al. [9] from Switzerland used the same term "nanofluid" to express minute volume of fluid (nanoliter) in microfluidics study in 1993. A German researcher, Grimm [10] also won a German patent on the enhanced thermal conductivity of nano- and micro-sized particles suspensions in the same year. Aluminum particles of 80 nm to 1 μm were suspended into a fluid and about 100% increase in the thermal conductivity of the fluid for loadings of 0.5–10 vol.% was reported in his patent.

The definition of this new class of heat transfer fluids, i.e. nanofluids is the suspensions of nanometer-sized solid particles, rods or tubes in traditional heat transfer fluids. Nanofluids were found to exhibit significantly higher thermophysical properties, particularly thermal conductivity and thermal diffusivity than those of base fluids [11–16]. Nanofluids have attracted great interest from the research community due to their enhanced thermal performance, potential benefits and applications in numerous important fields such as microelectronics, microfluidics, transportation, manufacturing, medical, and so on. Recent record shows (Fig. 2) that there is an exponential growth of annual research publications on nanofluids. According to ISI Web of Knowledge searched results (Fig. 2), total 153 nanofluids-related publications have already appeared in the first half of year 2010 (till July 14). It is also believed that there are more than 300 research groups and companies worldwide who are involved with nanofluids research [17]. Compared to studies on thermal conductivity, few works have been reported on boiling, droplet spreading, and convective heat transfer characteristics of nanofluids although these features are very important in order to exploit nanofluids as the next generation coolants. It is, therefore imperative and timely to provide a state-of-the-art review on the advances in boiling, droplet spreading, and convective heat transfer research of nanofluids.

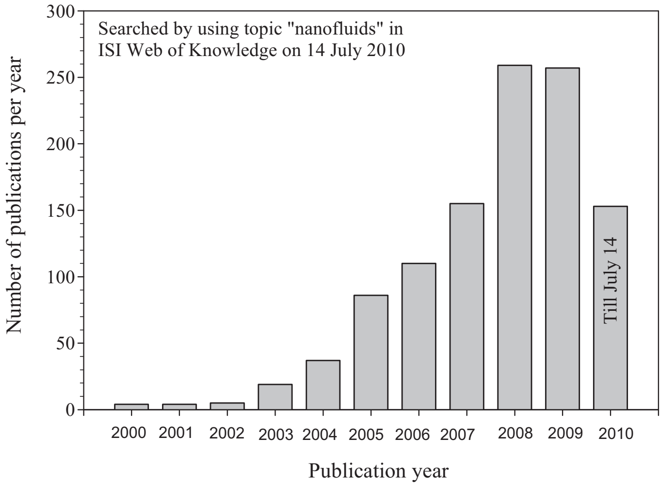

Fig. 2. Annual publications on nanofluids since 2000 (publications including all types of journal and conference articles, patent, news, letter and other).

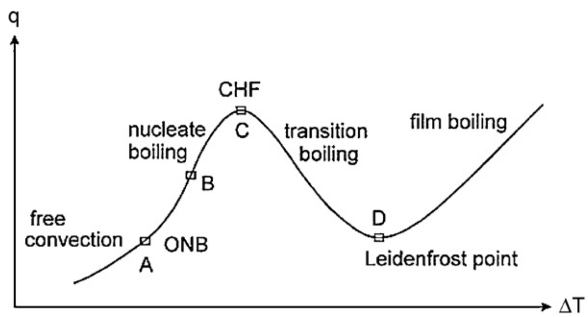

Fig. 3. Typical pool boiling curve.

2. Boiling and droplet spreading of nanofluids

2.1. Boiling heat transfer—basic

Boiling is a phase change (from liquid to vapor) process in which vapor bubbles are formed either on a heated surface or in a superheated liquid layer adjacent to the heated surface. Although boiling is a complex and elusive process, it is very efficient mode of heat transfer in various energy conversion and heat exchange systems as well as cooling of high energy density electronic components. There are two common types of boiling: pool boiling and flow or forced convective boiling. Pool boiling refers to boiling on a heated surface (heater) submerged in a pool of initially quiescent liquid while flow boiling is boiling in a flowing stream of fluid, where the heating surface may be the channel wall confining the flow. Heat flux in any boiling is one of the most important parameters in designing and operating the heat transfer equipments with high heat flux such as boiler, evaporator, electronic equipment and rocket engines. Since there are very few efforts are made on flow boiling of nanofluids, this mode of boiling heat transfer will not be elaborated further instead pool boiling heat transfer is briefly discussed here.

In pool boiling problems, the fluid is initially quiescent near the heating surface, and subsequent fluid motion arises from free convection and the circulation induced by bubble growth and detachment. Two main parameters in boiling problems are the degree of wall superheating or excess temperature (the difference between the wall and the liquid saturation temperature at the local pressure, $\Delta T = T_w - T_s$) and heat flux (q''). The classical pool boiling curve, which is as a plot of heat flux versus excess temperature (ΔT), is shown in Fig. 3. As the value of ΔT increases, the curve traverses four different regimes or modes of pool boiling (Fig. 3): (I) free convection, (II) nucleate boiling, (III) transition boiling, and (IV) film boiling.

As illustrated in Fig. 3, up to point A is the free convection regime which occurs since there is insufficient vapor to cause active boiling. Small temperature differences exist in the liquid, and heat is removed by free convection to the free surface. At point A, isolated bubbles initially appear along the heating surface and this is termed as onset of nucleate boiling (ONB). Nucleate boiling occurs between points A and C. At this regime, isolated bubbles appear, and heat is transferred mainly from the surface to the liquid. As ΔT increases (B–C), more nucleation sites become active and bubbles coalesce, mix, and ascend as merged jets or columns of vapor. At point C, the maximum heat flux or critical heat flux (CHF) occurs. Between points C and D, transition boiling occurs. An unstable (partial) vapor film forms on the heating surface, and conditions oscillate between nucleate and film boiling. Intermittent vapor formation blocks the liquid (higher conductivity) from contacting the surface (lowering the surface heat flux). Film boiling occurs beyond point D. In addition to conduction and convection, heat transfer by radiation is important at these high wall superheat levels. A stable vapor film covers the surface in this regime.

There are numerous factors such as heater or channel surface conditions (smooth or rough), heater size, shape, material, diameter and orientation, degree of surface wetting, subcooling, inclusion of surfactants, and properties of liquid affecting heat transfer performance and bubble dynamics of pool and flow boiling. These factors are widely studied both theoretically and experimentally [18–23]. For instance, effects of several parameters such as tube diameter, length, surface roughness, and inclination on pool boiling heat transfer have been investigated systematically by Kang [21–23]. His results showed that there are significant individual or combined effects of these factors on boiling heat transfer characteristics.

2.2. Boiling heat transfer of nanofluids—literature survey

The boiling critical or burnout heat flux enhancement of particle-suspensions depends on the particle concentration, pH of the solution as well as on the deposition of the particles on the suspended heater surface. It is long back proven that addition of solid particle in base fluid can alter its boiling heat transfer performance. For example, Yang and Maa [7] first used nano-sized Al_2O_3 particles as small as 50 nm in water to study the pool boiling heat transfer characteristics. They reported significant increase in pool boiling performance for very small volumetric concentrations (0.1–0.5%) of nanoparticles. After nanofluids emerged a growing number of research groups have got involved with boiling heat transfer study of nanofluids and it is timely to review their research findings.

For Al_2O_3–water nanofluid with a flat plate heater, You et al. [24] observed a three-fold increase in critical heat flux (CHF). For silica nanofluids, similar three-fold enhancement in CHF was later reported by Milanova and Kumar [25]. Whereas, Das et al. [26] reported deterioration of boiling heat transfer of water in the presence of Al_2O_3 nanoparticles in it. Their outcome was partially attributed to the properties of the nanofluid, boiling surface and the interaction between the two. In contrast to Das et al. [26], Wen and Ding [27] showed that the enhancement of pool boiling heat transfer of the same γ-Al_2O_3/water-based nanofluid was about 40% at 1.25 wt.% of particle loading. In another study, Witharana [28] investigated the boiling heat transfer performance of two types of Au and SiO_2-laden aqueous nanofluids in a cylindrical vessel under atmospheric pressure. They found that the boiling heat transfer increases and decreases for Au-nanofluid and SiO_2-nanofluid, respectively. These conflicting results are not well-explained.

Prakash et al. [29,30] performed two sets of experiments to quantify the effect of heater orientation and heater surface roughness on pool boiling heat transfer of Al_2O_3/water-based nanofluids. Their first set of experiment [29] with vertical tubular heaters of various surface roughness showed that while the rough heater surface increases heat transfer, smooth surface significantly deteriorates the heat transfer. For example, for rough heater surface the heat transfer enhancement was about 70% at 0.5 wt.% concentration of alumina nanoparticles. Whereas, for smooth heater the heat flux reduction reaches up to 45% at a particle concentration of 2 wt.%. Their other set of experiment [30] was mainly to identify the influence of heater orientation on the boiling heat transfer of the same nanofluid (Al_2O_3/water). Their results demonstrated that horizontal and inclined heater orientations showed enhancement and deterioration of boiling heat transfer of this nanofluid, respectively.

Soltani et al. [31] investigated the pool boiling heat transfer performance of Newtonian nanofluids under various heat flux densities. In their study, γ-Al_2O_3 (20–30 nm)/water and SnO_2 (55 nm)/water-based nanofluids were used in a vertical cylindrical glass vessel. Their results showed that except for low concentrations (>0.5 wt.%) of SnO_2 nanoparticles, the boiling heat transfer coefficients of these nanofluids increase with increasing concentration of nanoparticles. These paradoxical results were attributed to the differences in thermal conductivity and size of these two nanoparticles. In a similar pool boiling experiment, this same research group [32] recently used non-Newtonian nanofluids of γ-Al_2O_3 nanoparticles dispersed in carboxy methyl cellulose (cmc)–water solution and the boiling heat transfer coefficient was found to increase by about 25% at 1.4 wt.% loading of nanoparticles. Recently, Truong et al. [33] conducted pool boiling experiments of diamond, ZnO, and Al_2O_3 nanoparticles-laden aqueous nanofluids with modification of sandblasted as well as bare plate heaters. They found up to 35% increase in CHF for pre-coated heaters compared to those of bare and sandblasted heaters.

Among very few studies on flow boiling of nanofluids, Kim et al. [34] reported about 50% enhancement in flow boiling CHF for Al_2O_3/water nanofluids flowing through a vertical stainless steel tube. Very recently, Henderson et al. [35] studied refrigerant-based SiO_2 and CuO-nanofluids in flow boiling experiments in horizontal copper tube. They found that while the boiling heat transfer coefficient (BHTC) of SiO_2/R-134a nanofluid decreases up to 55% in comparison to pure R-134a, the BHTC increases more than 100% for CuO-laden nanofluid over base fluid, i.e. mixture of R-134a and polyolester oil (PO).

A summary of studies on boiling heat transfer of nanofluids is presented in Table 1. It can be noticed that besides inconsistent and contradictory results, most of the researchers used alumina-nanofluids. Whereas very few efforts have been made with carbon nanotubes (CNTs)-nanofluids which exhibit much higher thermal performance compared to those of ceramic-nanofluids [14,15]. It is, therefore worthwhile to study boiling heat transfer characteristics of CNTs-nanofluids.

A comparison of heat flux versus superheat results from various groups is shown in Fig. 4. From these representative results (Fig. 4), it can clearly be seen that heat flux (also critical heat flux) data relative to superheat reported by various research groups vary widely. This is probably due to the differences in characterization of nanofluids, different size and concentration of nanoparticles used and different types of heaters used in various research groups. Despite of the fact that some researchers reported deterioration of boiling heat transfer of nanofluids, the significant increase in the critical heat flux of nanofluids is still undisputed.

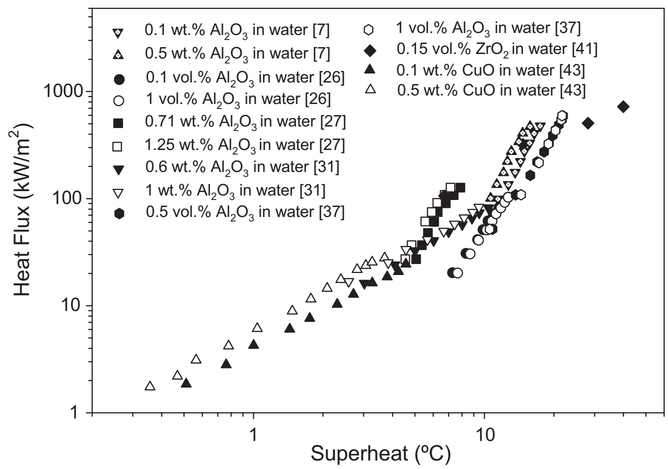

Fig. 4. Comparison of heat flux versus superheat results from various research groups.

2.3. Boiling heat transfer—correlations

Although there are many correlations [18,20] to predict the boiling heat transfer coefficient and critical heat flux, researchers working on boiling heat transfer with nanofluids [27,30,36,37] mainly used predictions from two most popular correlations—Rohsenow [45] and Zuber [46] correlations to compare their experimental data. Thus, without elaborating details of other models or correlations, these two classical correlations are presented here.

Based on dimensional analysis of all relevant factors arising in nucleate boiling and experimental data over a wide variety of conditions Rohsenow [45] proposed one of the first and most widely used nucleate boiling correlations which is commonly expressed in the following form:

$$\frac{c_p(T_w - T_s)}{h_{fg}} = C_{sf}\left[\frac{q''}{\mu h_{fg}}\sqrt{\frac{\sigma}{g(\rho-\rho_g)}}\right]^{0.33}\left(\frac{c_p\mu}{k}\right)^n \tag{1}$$

where c_p, h_{fg}, T_w, T_s, q'', μ, ρ, ρ_g, σ, g and k are specific heat of liquid, the latent heat of the fluid, temperature of heater wall, saturation temperature of liquid, heat flux, viscosity of saturated liquid, density of saturated liquid, density of saturated vapor, surface tension, acceleration of gravity and thermal conductivity of liquid, respectively. The values for empirical coefficient C_{sf} and exponent n for various surface–fluid combinations can be found elsewhere [47]. For example, C_{sf} and n for water–platinum combination are 0.013 and 1, respectively.

The maximum heat flux attainable in the nucleate pool boiling regimes is the critical heat flux (CHF) or point of departure from nucleate boiling (DNB). A widely used correlation which predicts the pool boiling CHF for a flat infinite heater surface (facing upwards) is Zuber's correlation [46]. The Zuber CHF is expressed as:

$$q''_{CHF} = K\rho_g^{1/2}h_{fg}\left[g\sigma(\rho-\rho_g)\right]^{1/4} \tag{2}$$

The value of constant K ranged from 0.138 to 0.157. However, simplifying the formulation analysis, Zuber proposed a value of K = 0.131. Although widely used, most of the cases predictions of pool boiling heat transfer and critical heat flux of nanofluids using these two abovementioned correlations do not agree well with the experimental results [27,36]. There is therefore a strong need to develop new correlations or modify these classical models to be used for nanofluids.

2.4. Present boiling experiments with CNT-nanofluids

For pool boiling experiments, high purity single-walled carbon nanotubes (SWCNTs) are suspended in deionized water to prepare sample nanofluids. As a part of surface treatment, CNT bundles were

Table 1
Summary of boiling experiments with nanofluids.

Researchers	Heater (type of boiling)	Nanofluids	Remarks
Yang and Maa [7]	Horizontal tube heater (pool)	Al_2O_3/water	Boiling heat flux increases considerably.
Witharana [28]	Cylindrical vessels (pool)	Au and SiO_2/water and EG	Boiling heat transfer increases for Au-nanofluids but decreases for SiO_2-nanofluids.
Das et al. [26]	Cylindrical cartridge (pool)	Al_2O_3/water	Deterioration of heat transfer performance.
You et al. [24]	Cartridge (pool)	Al_2O_3/water	Critical heat flux (CHF) increases up to 200%.
Vassallo et al. [36]	NiCr wire (pool)	SiO_2/water	CHF increases significantly.
Bang and Chang [37]	Square flat heater (pool)	Al_2O_3/water	Pool boiling heat transfer deteriorates but CHF increases.
Wen and Ding [27]	Flat disk heater (pool)	Al_2O_3/water	The boiling heat transfer increases up to 40% at 1.2 wt.% of nanoparticles.
Kim et al. [38]	Smooth NiCr wire heater (pool)	TiO_2/water	The maximum enhancement of CHF is 200%.
Kim et al. [39]	Stainless steel wire and flat plate heaters (pool)	Al_2O_3, ZrO_2 and SiO_2/water	The maximum increase in CHF is 80% but the heat transfer rate is deteriorated.
Jackson [40]	Flat copper coupon heater (pool)	Au/water	While the heat transfer decreased about 20%, the maximum CHF increase is 5 times over water.
Prakash et al. [29]	Vertical tubular heaters (pool)	Al_2O_3/water	While rough heater surface increases heat transfer, smooth surface significantly deteriorates.
Prakash et al. [30]	Tubular heaters at various orientations (pool)	Al_2O_3/water	Horizontal and inclined heater orientations showed enhancement and deterioration of heat transfer, respectively.
Chopkar et al. [41]	Flat surface (pool)	ZrO_2/water	Enhanced boiling heat transfer is found at low particle concentration. Tetramethyl ammonium hydroxide surfactant increases the heat transfer rate.
Liu and Liao [42]	Flat horizontal copper surface (pool)	CuO/water and Alcohol (C_2H_5OH)	The boiling heat transfer characteristics of the both nanofluids are somewhat poor and the CHF values are higher than those of the base liquids.
Lv and Liu [43]	Vertical small heated tubes (pool)	CuO/water	Enhancement in CHF and boiling heat transfer rate are observed for surfactant-free nanofluids but addition NaDBS surfactant deteriorates the heat transfer.
Kathiravan et al. [44]	Horizontal tube (pool)	CNTs/water	Boiling heat transfer coefficient (BHTC) increases up to 1.75-fold for 0.25 vol.% of CNT. Sodium lauryl sulfate (SLS) surfactant of 9 wt.% was also added.
Kim et al. [34]	Vertical stainless steel tube (flow)	Al_2O_3/water	Flow boiling CHF increases about 50%.
Soltani et al. [31]	Vertical cylindrical glass vessel (pool)	Al_2O_3 and SnO_2/water	Except for low concentrations (>0.5 wt.%) of SnO_2, the BHTC increases with loading of nanoparticles.
Soltani et al. [32]	Vertical cylindrical glass vessel (pool)	Al_2O_3/cmc–water solution	The BHTC increases by 25% for 1.4 wt.% of Al_2O_3 nanoparticles.
Henderson et al. [35]	Horizontal copper tube (flow)	SiO_2/R-134a and CuO/R134a + polyolester oil	While BHTC of SiO_2/R-134a nanofluid decreases up to 55% in comparison to pure R-134a, it increases more than 100% for CuO-laden nanofluid over base fluid.
Truong et al. [33]	Sandblasted and bare horizontal plate heaters (pool)	Diamond, ZnO and Al_2O_3/water	The CHF of the pre-coated heaters increased by up to 35% with respect to that of the bare, sandblasted heaters.

refluxed with hydrochloric acid at 100 °C for several hours. This acid was chosen as a reactive reagent because it removes catalytic particles without reducing the length of the tubes or damaging the side walls. Other acids (HNO_3 and HNO_3:H_2SO_4) tried as refluxing mixtures, were too harsh and damaged the structure. CNTs were separated from the acid by centrifugation for about an hour and left to dry. Different concentrations of sodium dodecyl benzenesulfonate (NaDBS) surfactant are used as dispersing and stabilizing agent for nanotubes in water. The schematic of experimental facilities used is shown in Fig. 5. Details of the experimental facilities and procedures are reported in previous studies [48,49] and will not be discussed here. Representative results of surfactant- and surface tension-dependent pool boiling experiments of CNTs-nanofluids [48] are presented and discussed in this section. Both the critical heat flux (CHF), which is the sudden jump in temperature at one and the same heat flux and the burnout heat flux (BHF), which is the case when the wire is brought to catastrophic failure (i.e. complete burnt out) are determined at constant 0.1% volumetric loading of CNT and for various concentrations of surfactant.

2.4.1. Effect of surfactant concentration on boiling heat flux

The results of pool boiling heat flux with respect to superheat (wire temperature minus saturation temperature of liquid, $T_w - T_s$) are presented and compared with the one for pure de-ionized water in Fig. 6. The NaDBS surfactant to CNTs concentrations are varied from 1:20 to 1:1. Fig. 6 demonstrates that the CNT-nanofluid with any concentration of surfactant exhibits higher CHF value than that of base fluid. The CHF value of pure deionized water is 750 kW/m^2. The effect of increasing the surfactant concentration from 1:20 to 1:5 (NaDBS:CNT) is that the CHF increases as well. However, if the concentration of surfactant is further increased from 1:5 to 1:1, the CHF drops drastically from 4439 kW/m^2 to 1322 kW/m^2, respectively. There is, therefore a critical concentration of surfactant for which the CHF and BHF reach maximum values. The highest CHF and BHF results are obtained for the concentration ratio of 1:5. The deposition of nanoparticles on the heater wire is believed to be one of the key reasons for any enhancement in CHF of this nanofluid up to critical concentration of surfactant. Kim et al. [50] also claimed that the deposition of nanoparticles on the heater wire is the main reason for the enhancement of CHF of their nanofluids. SEM micrographs shown in Fig. 7 demonstrate that with an increase in surfactant the deposition on the wire increases as well. However, the maximum CHF and BHF do not occur at the maximum particle deposition, i.e. at 1:1 NaDBS:CNT ratio.

2.4.2. Effect of surface tension on burnout heat flux

The burnout heat flux is determined at constant 0.1% volumetric loading of CNT and at various surface tensions of nanofluids which

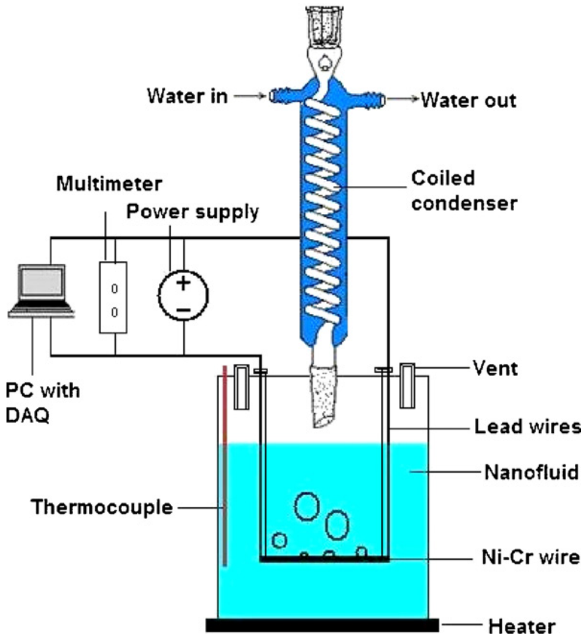

Fig. 5. Schematic of pool boiling experimental setup [48].

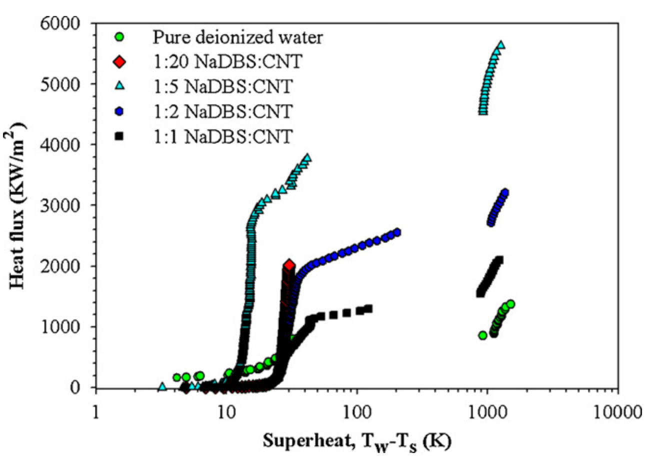

Fig. 6. Effect of surfactant concentration on heat flux of SWCNTs-nanofluid.

was adjusted by varying concentrations of NaDBS surfactant. The surface tensions of nanofluid with surfactant (σ_{NS}) and deionized water with surfactant (σ_{WS}) concentrations are measured. At any concentration of surfactant, the difference between surface tensions of nanofluid and its based fluid (deionized water) is termed as relaxation of surface tension, i.e. $\Delta\sigma_r = \sigma_{NS} - \sigma_{WS}$. This relaxation of surface tension of nanofluids is the driving force for the burnout heat flux of nanofluid, i.e. $(BHF)_N$ in pool boiling as shown in Fig. 8. Based on the best fit of the experimental data, a nonlinear empirical correlation between the BHF and relaxation of surface tension

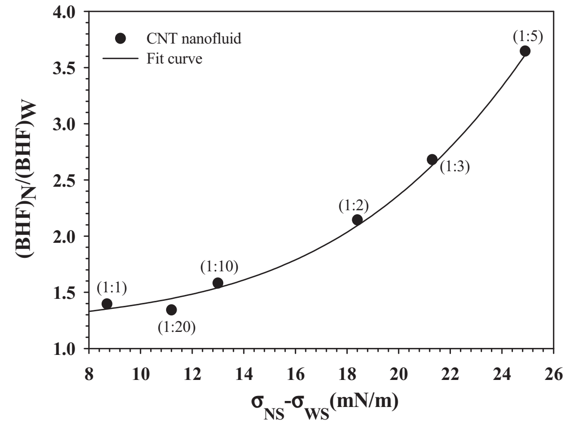

Fig. 8. Effect of surface tension on the burnout heat flux of nanofluid (NaDBS to CNTs concentration ratios are shown in parentheses).

is obtained and has the form

$$\frac{(BHF)_N}{(BHF)_W} = 1 + 0.073\Delta\sigma_r - 0.0065\Delta\sigma_r^2 + 0.0003\Delta\sigma_r^3 \quad (3)$$

where $(BHF)_W$ is the burnout heat flux of water and $\Delta\sigma_r = (\sigma_{NS} - \sigma_{WS})$ in mN/m.

It is demonstrated that regardless of surfactant concentration, the burnout heat flux of nanofluid strongly depends (i.e. nonlinearly related) on its relaxation of surface tension with base fluid. Similar to the CHF, the BHF of nanofluid also reaches its highest value at NaDBS and CNT concentration ratio of 1:5. The highest BHF value of nanofluid is about 265% enhancement over that for the base fluid at $\Delta\sigma_r \approx 25$ mN/m. The burnout heat flux of deionized water $(BHF)_W$ is found to be 1500 kW/m^2. The observed increment of relative BHF of nanofluid (i.e. $(BHF)_N/(BHF)_W$) can be attributed to the Marangoni effect which results from the surface tension differences. Both the CHF and burnout are found to happen at the optimum 1:5 surfactant:CNT ratio. Thus, the pool boiling heat transfer behavior of surfactant-added nanofluid is not only dictated by the deposition of nanotubes on heater surface, but also by the relaxation of surface tension of nanofluid, which is a precursor to the deposition and overall onsets of both the CHF and BHF.

2.5. Studies on nanofluids droplet spreading

Droplets impingement and spreading studies on heated substrate surface are of great importance for the practical application of nanofluids as an advanced coolant particularly in spray cooling. Unfortunately, very little research efforts have been made to study nanofluids' droplet impingement behavior on solid surfaces under various conditions. Among very few studies, Wasan and Nikolov [51] were the first to investigate the effects of the particle structure formation and the structural disjoining pressure of nanoparticles on the spreading of nanofluids on solid surface. Recently, Duursma et al. [52] studied the effect of aluminum nanoparticles on droplets boil-off by allowing nanofluid drops to fall onto a copper surface at temperature higher than the liquid satura-

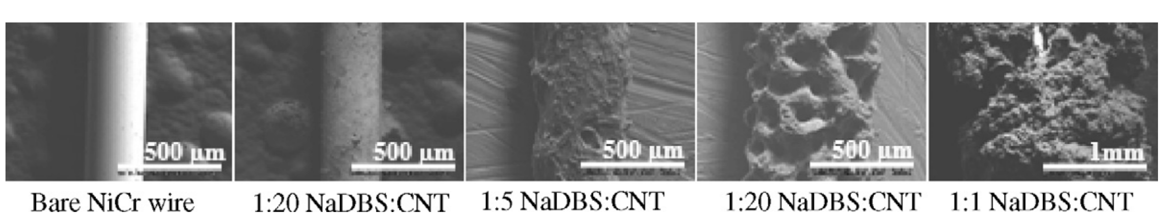

Fig. 7. SEM images of NiCr wires at various surfactant concentrations after pool boiling experiments [48].

tion temperature. They demonstrated that increasing the surface temperature and Weber number promote the receding breakup scenario, while increasing the nanoparticle concentration discourages this breakup. In another recent study, the influence of surface temperature on the hydrodynamic characteristics of water and a nanofluid droplets impinging on a polished and nanostructured surface was investigated by Shen et al. [53]. Their results showed that SWCNTs-nanofluid has larger spreading diameter compared to that of deionized water and using a nanofluid or a nanostructured surface can reduce the total evaporation time up to 37%. Murshed and Nieto de Castro [54] very recently reported the spreading characteristics of a nanofluid droplets impinging on a metallic substrate under the influence of several key factors such as nanoparticles volume fraction, substrate temperature, and the Weber number. Results showed that the transient spreading diameter and height of droplet impacting onto solid surface are greatly influenced by each of these factors. Nevertheless, more studies are needed on dynamics of non-boiling and boiling nanofluid droplets impinging on solid or liquid surfaces as the spreading of liquid droplet plays a key role in many industrial processes like spray cooling, coating, ink-jet printing, and oily soil removal.

3. Flow and heat transfer characteristics of nanofluids

Compared to the reported studies on thermal conductivity, investigations on convective heat transfer of nanofluids are still scarce. However, the practical applications of nanofluids as advanced heat transfer fluids are mainly in flowing systems such as mini- or micro-channels heat sinks and miniaturized heat exchangers. In this section, we critically review both experimental and computational studies on convective heat transfer of nanofluids. In addition, some representative results from our experimental investigation on laminar flow convective heat transfer of TiO_2-nanofluids are also presented and discussed.

3.1. Experimental studies on convective heat transfer of nanofluids

The first experiment on convective heat transfer of nanofluids (e.g. γ-Al_2O_3/water) under turbulent flow conditions was performed by Pak and Cho [55]. In their study, even though the Nusselt number (*Nu*) was found to increase with increasing nanoparticle volume fraction and Reynolds number, the heat transfer coefficient (*h*) actually decreased by 3–12%. The reasons for such paradoxical results might be the observed large enhancement in viscosity. On the other hand, Eastman et al. [56] later showed that with less than 1 vol.% of CuO nanoparticles, the convective heat transfer coefficient (*h*) of water increased more than 15%. The results of Xuan and Li [57] illustrated that the Nusselt number of Cu/water-based nanofluids increased significantly with the volumetric loading of particles as for 2 vol.% of nanoparticles, the Nusselt number increased by about 60%. Wen and Ding [58] investigated the heat transfer behavior of nanofluids at the tube entrance region under laminar flow conditions and showed that the local heat transfer coefficient varied with particle volume fraction (ϕ) and Reynolds number (*Re*). They also observed that the enhancement is particularly significant at the entrance region. Later Heris et al. [59] studied convective heat transfer of CuO and Al_2O_3/water-based nanofluids under laminar flow conditions through an annular tube. Their results showed that heat transfer coefficient increases with particle volume fraction as well as Peclet number. In their study, Al_2O_3/water-based nanofluids found to have higher enhancement of heat transfer coefficient compared to CuO/water-based nanofluids.

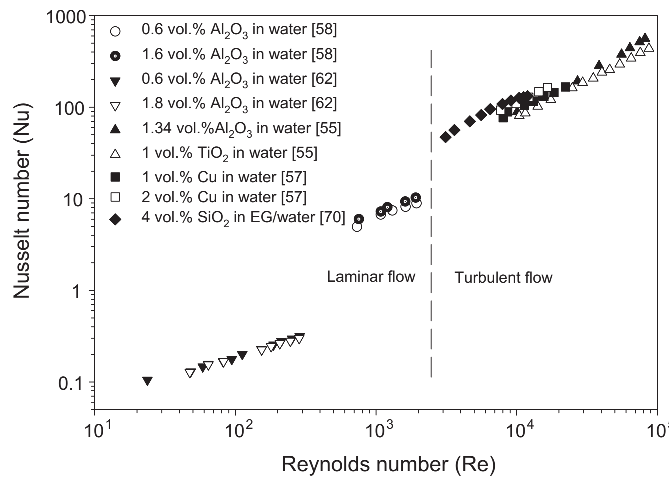

Fig. 9. Comparison convective heat transfer results from various research groups [55,57,58,62,70].

An experimental investigation on the forced convective heat transfer and flow characteristics of TiO_2–water nanofluids under turbulent flow conditions is reported by Duangthongsuk and Wongwises [60]. A horizontal double-tube counter flow heat exchanger is used in their study. They observed a slightly higher (6–11%) heat transfer coefficient for nanofluid compared to pure water. The heat transfer coefficient increases with increasing mass flow rate of the hot water as well as nanofluid. They also claimed that the use of the nanofluid has a little penalty in pressure drop.

In microchannel flow of nanofluids, Faulkner et al. [61] was the first to conduct convective heat transfer experiments with aqueous CNT–nanofluid in a microchannel with hydraulic diameter of 355 μm at Reynolds numbers between 2 and 17. They found considerable enhancement in heat transfer coefficient of this nanofluid at CNT concentration of 4.4%. Later, Jung et al. [62] studied heat transfer performance of Al_2O_3/water-based nanofluid in a rectangular microchannel under laminar flow condition. Results showed that the heat transfer coefficient increased by more than 32% for 1.8 vol.% of nanoparticles and the Nusselt number increases with increasing Reynolds number in the flow regime of $5 > Re < 300$.

An up-to-date overview of the published experimental results on the convective heat transfer characteristics of nanofluids is given in Table 2. A comparison of results of Nusselt number versus Reynolds number for both laminar and turbulent flow conditions from various groups is shown in Fig. 9. From Table 2 and Fig. 9, it can be seen that the results from various groups vary widely and most of the studies lack physics-based explanation of the observed results. Although there are couple of review articles [71,72] that attempted to put together some studies on convective heat transfer of nanofluids, no comprehensive analysis of up-to-date findings are reported.

Among very few research efforts, the work of Putra et al. [73] was the first to investigate the natural convective heat transfer of aqueous CuO and Al_2O_3-nanofluids. They used a horizontal polyoxymethylene cylinder which was heated from one end and cooled from the other. Significant deterioration (decrease) of convection heat transfer for these nanofluids was observed and the deterioration increases with increasing particle concentration, particularly for CuO nanoparticles. Putra et al. ascribed the possible reasons of such deterioration to the effects of particle–fluid slip and sedimentation of nanoparticles. In contrast to Putra et al. [73], numerical simulation of Khanafer et al. [74] showed that in a 2D horizontal enclosure, the natural convection heat transfer coefficient (*Nu*) of nanofluids increases with particle concentration. By taking into account the solid particle dispersion, they [74] developed a model

Table 2
Summary of forced convection heat transfer experimental studies of nanofluids.

Researchers	Geometry/flow nature	Nanofluids	Findings
Pak and Cho [55]	Tube/turbulent	Al_2O_3 and TiO_2/water	At 3 vol %, h was 12% smaller than pure water for a given average fluid velocity.
Xuan and Li [57]	Tube/turbulent	Cu/water	A larger enhancement of h with volume fraction (ϕ) and Re was observed.
Wen and Ding [58]	Tube/laminar	Al_2O_3/water	Increased h with ϕ and Reynolds number was observed. NaDBS was used as the surfactant
Ding et al. [63]	Tube/laminar	CNT/water	At 0.5 wt.%, h increased by more than 350% at $Re = 800$.
Yang et al. [64]	Tube/laminar	Graphite/automatic transmission fluid	The nanoparticles considerably increase the heat transfer coefficient of the fluid system in laminar flow.
Heris et al. [59]	Tube/laminar	Al_2O_3 and CuO/water	h increase with ϕ and Pe. Al_2O_3 shows higher enhancement than that of CuO.
Lai et al. [65]	Tube/laminar	Al_2O_3/water	Nu increased 8% for $\phi = 0.01$ and $Re = 270$.
Jung et al. [62]	Rectangular microchannel/laminar	Al_2O_3/water	h increased 15% for $\phi = 0.018$.
Williams et al. [66]	Tube/turbulent	Al_2O_3 and ZrO_2/water	h increased significantly.
Hwang et al. [67]	Tube/laminar	Al_2O_3/water	h increased only up to 8% at $Re = 730$ for $\phi = 0.003$.
Xie et al. [68]	Tube/laminar	Al_2O_3, ZnO, TiO_2 and MgO/water	h increased up to 252% at $Re = 1000$ for MgO nanofluid.
Amrollahi et al. [69]	Tube/laminar and turbulent	MWCNT/water	h increased up to 33–40% at concentration of 0.25 wt.%.

to analyze the heat transfer performance of nanofluids inside an enclosure.

Wen and Ding [75] later investigated heat transfer behavior of specially formulated TiO_2/water nanofluid under the natural convection conditions. The results showed that the natural convective heat transfer coefficient decreases with increasing particle concentration. These unexpected results are in contradiction to the numerical findings of Khanafer et al. [74] but are in agreement with the observations by Putra et al. [73]. Along with the reasons ascribed by Putra et al. [73], Wen and Ding [75] added several other reasons for such paradoxical results that include convection induced by concentration difference, modification of dispersion properties and particle–surface and particle–particle interactions.

3.2. Theoretical models for convective heat transfer of nanofluids

Researchers investigating convective heat transfer of nanofluids employed existing conventional single-phase fluid correlation to predict the heat transfer coefficient or proposed new correlations obtained by fitting their own experimental data. However, none of these correlations were validated with wide range of experimental data under various conditions and thus are not widely accepted. In an attempt to establish a strong explanation of the reported anomalously enhanced convective heat transfer coefficient of nanofluid, Buongiorno [76] considered seven-slip mechanisms and concluded that among those seven only Brownian diffusion and thermophoresis are the two most important particle/fluid slip mechanisms in nanofluids. Besides proposing a new correlation, he also claimed that the enhanced laminar flow convective heat transfer can be attributed to a reduction of viscosity within and consequent thinning of the laminar sublayer. Commonly used classical models together with recently proposed correlations for the predictions of convective heat transfer coefficient of nanofluids are summarized in Table 3. Considering all possible key mechanisms such as particle Brownian motion and migration, thermophoresis and inertial a generalized convective heat transfer correlation needs to be developed.

3.3. Numerical studies on convective heat transfer of nanofluids

Several preliminary numerical studies on the convective heat transport of nanofluids were conducted by Khanafer et al. [74] and a research group from Université de Moncton, Canada [79,80]. Results from the Canadian group showed that nanofluids can substantially enhance heat removal in both tube [79] and radial flows [80]. For example, Roy et al. [80] numerically studied the hydrodynamic and thermal fields of Al_2O_3/water-based nanofluids in a radial laminar flow system and reported a two-fold increase in heat transfer for 10 vol.% of nanoparticles in water. They also observed that the wall shear stress increased with an increase in particle volume fraction. For Al_2O_3/water- and Al_2O_3/ethylene glycol-based nanofluids, numerical results of Maïga et al. [79] showed that the heat transfer coefficient increases considerably with increasing nanoparticle concentration. However, the presence of such particles has also induced drastic effects on the wall shear stress that increases appreciably with the particle loadings.

A Lattice Boltzmann model for simulating flow and energy transport processes inside nanofluids was proposed by Xuan et al. [81]. They considered the external and internal forces acting on the suspended nanoparticles and interactions among the nanoparticles and fluid particles. Based on the simulation results, they inferred that the fluctuation of Nusselt number of nanofluids along the main flow direction is due to the unstable distribution of nanoparticles. No further development is made in exploring the use of the Lattice Boltzmann method to explain the anomalous thermal convection of nanofluids.

Palm et al. [82] performed numerical simulations for the laminar forced convection flow of nanofluids with temperature-dependent properties. They found that Al_2O_3/water-based nanofluids with a volume fraction of 4% can produce a 25% increase in the average wall heat transfer coefficient compared to water alone. Significant differences were also found when using constant property nanofluids (temperature-independent) versus nanofluids with temperature-dependent properties. The use of temperature-dependent properties resulted in larger heat transfer predictions with corresponding decrease in wall shear stresses when compared to predictions using constant properties nanofluids. With an increase in wall heat flux, they found that the average heat transfer coefficient increases while the wall shear stress decreases.

Based on Khanafer et al.'s [74] model and taking into account particle dispersion, Jou and Tzeng [83] numerically studied convective heat transfer performance of nanofluids inside two-dimensional rectangular enclosures. Their results showed that increasing volume fraction and buoyancy parameter can cause an increase in average heat transfer coefficient.

The cooling performance of a microchannel (silicon) heat sink under forced convective laminar flow with nanofluids is numerically investigated by Jang and Choi [84]. Results showed that at a

Table 3
Correlations used for predicting the convective heat transfer coefficient of nanofluids.

Origin/reference	Correlations	Flow regimes/conditions/remarks
Shah equation [77]	$Nu = \begin{cases} 1.953\left(Re\,Pr\dfrac{D}{x}\right)^{1/3}; & \text{for } \left(Re\,Pr\dfrac{D}{x}\right) \geq 33.3 \\ 4.364 + 0.0722 Re\,Pr\dfrac{D}{x}; & \text{for } Re\,Pr\dfrac{D}{x} < 33.3 \end{cases}$	Laminar flow and constant heat flux condition. It is popular for a thermal entrance region.
Dittus and Boelter [78]	$Nu = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4}$	Heating of fluid and used for $0.7 < Pr < 100$. It is very popular in pipe flow cases and is not recommended for temperature dependent property variation.
Sieder–Tate equation [77]	$Nu = 0.027Re^{0.8}Pr^{1/3}(\mu_b/\mu_s)^{0.14}$ where μ_b and μ_s are the fluid viscosity at bulk fluid temperature and at heat-transfer boundary surface temperature, respectively.	Valid for $0.7 < Pr < 16{,}700$ and $Re > 10{,}000$. It is used for temperature-dependent property variation cases.
Pak and Cho [55]	$Nu = 0.021\mathrm{Re}^{0.8}\mathrm{Pr}^{0.5}$	Turbulent flow
Xuan and Li [57]	Laminar flow: $Nu = 0.4328(1 + 11.285\phi^{0.754}Pe^{0.218})Re^{0.33}Pr^{0.4}$ Turbulent flow: $Nu = 0.0059(1 + 7.6286\phi^{0.6886}Pe^{0.001})Re^{0.9238}Pr^{0.4}$	*Pe* is the particle Peclet number. It considered particles concentration.
Maïga et al. [79]	Averaged *Nu* for constant heat flux: $\overline{Nu} = 0.086Re^{0.55}Pr^{0.5}$ Averaged *Nu* for constant temperature: $\overline{Nu} = 0.28Re^{0.35}Pr^{0.36}$	Correlations are proposed from numerical study and valid for $Re \leq 1000$ and $6 \leq Pr \leq 753$.
Jung et al. [62]	$Nu = 0.0095\phi^{0.01}Re^{0.42}Pr^{0.55}$	For microchannel laminar flow regime. It considered the volume fraction of nanoparticles.
Buongiorno [76]	$Nu = \frac{(f/8)(Re_b - 1000)Pr_b}{1 + \delta_v^+\sqrt{f/8}(Pr_v^{2/3} - 1)}$ where f is the friction factor and δ_v^+ is an empirical constant.	Turbulent flow

fixed pumping power (2.245 W) the cooling performance of their heat sink with aqueous-diamond (1 vol.%) nanofluid is enhanced by about 10% compared to that of with water alone. They also reported that nanofluids reduced both the thermal resistance and the temperature difference between the heated microchannel wall and coolant.

Using a dispersion model Heris et al. [85] later performed numerical simulations for the laminar-flow convective heat transfer of aqueous nanofluids in a circular tube with constant wall temperature. Effects of concentration and size of various types of nanoparticles (Al_2O_3, CuO, and Cu in water) as well as Peclet number on heat transfer coefficient are investigated. They reported that *Nu* increases with increasing particle concentration and with decreasing nanoparticles size. Their numerical predictions were found to be in considerable agreement with their experimental data.

A numerical study on laminar mixed convection of Al_2O_3/water nanofluid was reported by Mirmasoumi and Behzadmehr [86]. Two-phase mixture model was used to study the effects of nanoparticles mean diameter on the flow parameters. Their calculated results showed that like Heris et al.'s [85] findings, the heat transfer coefficient significantly increases with decreasing the nanoparticles mean diameter. However particle size does not notably change the skin friction coefficient.

In a separate numerical study, Ho et al. [87] attempted to identify the effects of uncertainties in various effective viscosity and thermal conductivity models adopted on buoyancy-driven convection heat transfer of alumina–water nanofluid in a square enclosure. Based on their simulation results from a comparative study of four different models, they demonstrated that the effect of nanoparticle concentration on the averaged Nusselt number significantly depends on the effective viscosity and thermal conductivity models used.

Oztop and Abu-Nada [88] recently numerically investigated the buoyancy-driven natural convection heat transfer characteristics of different types of water-based nanofluids containing Cu, Al_2O_3, and TiO_2 nanoparticles in a partially heated enclosure. An increase in mean Nusselt number was observed with the volume fraction of nanoparticles for the whole range of Rayleigh number ($10^3 \leq Ra \leq 5 \times 10^5$) and heat transfer also increases with increasing of height of heater.

A very recent numerical work on a 3D laminar flow heat transfer with Al_2O_3, CuO/mixture of ethylene glycol and water-based nanofluids circulating through flat tubes of a radiator was undertaken by Vajjha et al. [89]. Their results demonstrated that the convective heat transfer coefficient in the developing and developed regions along the flat tubes with the nanofluid flow showed remarkable improvement over the base fluid and for the local and the average friction factor and convective heat transfer coefficient show an increase with increasing volumetric concentration of nanoparticles.

Above review on numerical studies demonstrated that convective heat transfer coefficient of nanofluids considerably increases with increasing concentration of nanoparticles and with decreasing nanoparticle size. However, more works on forced and natural convective heat transfer of nanofluids are to be performed by using advanced numerical methods such as Lattice Boltzmann, Molecular and Brownian dynamic simulations.

3.4. Present experiments on laminar flow convective heat transfer of nanofluids

Sample nanofluids for convective heat transfer experiments were prepared by dispersing different volume percentages, i.e. 0.2–0.8% of titanium dioxide (TiO_2) nanoparticles (15 nm diameter) in deionized water (DIW). As a dispersant agent, Cetyl Trimethyl Ammonium Bromide (CTAB) surfactant was added in the sample. To ensure proper dispersion of nanoparticles, sample nanofluid was homogenized by using an ultrasonic dismembrator.

The effects of nanoparticles concentration and Reynolds number on the convective heat transfer coefficient of TiO_2/deionized water based-nanofluids were previously investigated [90] and some representative results are presented. An experimental setup was established to conduct experiments on heat transfer of nanofluids at laminar flow regime in a cylindrical channel. The schematic of experimental facilities used is shown in Fig. 10. Details of the exper-

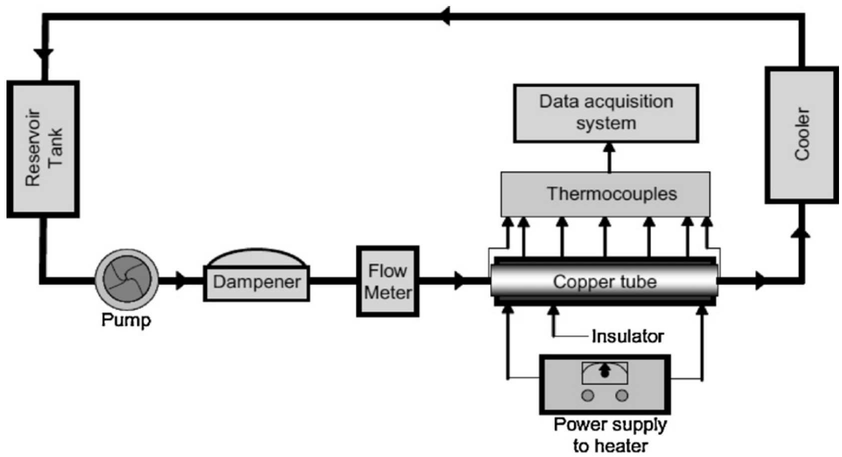

Fig. 10. Schematic of convective heat transfer experimental setup [90].

imental facilities and procedures are reported elsewhere [90] and will not be discussed here. Instead formulations for obtaining the heat transfer coefficient (h) together with a most commonly used existing correlation for predicting the Nusselt numbers of nanofluids in laminar flow conditions are presented.

As detailed in previous paper [90], applying first law (energy balance) in control volume of flow channel, the following formulation for the local heat transfer coefficient is obtained

$$h_{nf\text{-}x} = \frac{q''}{\{T_{o,w}(x) - ((q[2D_o^2 \ln(D_o/D_i) - (D_o^2 - D_i^2)])/4\pi(D_o^2 - D_i^2)k_s x)\} - \{T_i + ((T_o - T_i)/L)x\}} \quad (4)$$

where $T_{o,w}(x)$ is the outer wall temperature of the tube (measurable), q'' is the heat flux of the test section (W/m^2), q is the heat supplied to the test section (W), k_s is the thermal conductivity of the copper tube (W/m K), D_i and D_o are the inner and outer diameters of the tube, respectively and x represents the longitudinal location of the section of interest from the entrance. L is the length of the test section and T_i and T_o are the inlet and outlet fluid temperature, respectively.

Once the local heat transfer coefficient is determined and the thermal conductivity of the medium is known, the local Nusselt number is calculated from

$$Nu_{nf\text{-}x} = \frac{h_{nf\text{-}x} D_i}{k_{nf}} \quad (5)$$

where k_{nf} is the effective thermal conductivity of nanofluids. The classical Hamilton–Crosser model is used for the determination of k_{nf} which is given by [91]

$$k_{nf} = k_f \left[\frac{k_p + (n-1)k_f - (n-1)\phi(k_f - k_p)}{k_p + (n-1)k_f + \phi(k_f - k_p)} \right] \quad (6)$$

where k_f and k_p are the thermal conductivities of the base liquid and the nanoparticles, respectively, ϕ is the volume fraction of nanoparticles and n is the empirical shape factor, which has a value of 3 for spherical particle.

The Nusselt number can also be determined from the existing correlations. The well-known Shah's correlation for laminar flows under the constant heat flux boundary conditions is used and reproduced as [77]

$$Nu = 1.953\left(Re\,Pr\frac{D_i}{x}\right)^{1/3} \quad \text{for} \quad \left(Re\,Pr\frac{D_i}{x}\right) \geq 33.3 \quad (7)$$

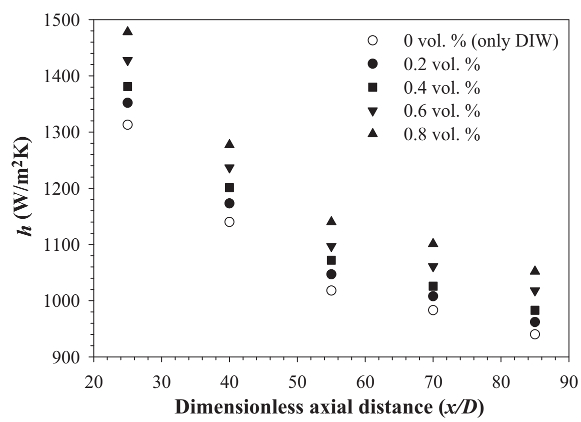

Fig. 11. Axial profiles of local heat transfer coefficient of nanofluid at $Re = 1100$ [90].

For steady and incompressible flow of nanofluids in a tube of uniform cross-sectional area, the Reynolds number and Prandtl number are defined as follows

$$Re = \frac{4\dot{m}}{\pi D_i \mu_{nf}} \quad \text{and} \quad Pr = \frac{c_{p\text{-}nf}\mu_{nf}}{k_{nf}} \quad (8)$$

where $\dot{m}$ is the mass flow rate and μ_{nf} and $c_{p\text{-}nf}$ are the viscosity and specific heat of nanofluids, respectively.

While the specific heat of nanofluids is calculated using the following volume fractioned-based mixture rule [55,79,84]

$$c_{p\text{-}nf} = \phi c_{p\text{-}p} + (1-\phi)c_{p\text{-}f}, \quad (9)$$

the viscosity of nanofluids is determined from Batchelor's model given by [92]

$$\mu_{nf} = \mu_f(1 + 2.5\phi + 6.2\phi^2) \quad (10)$$

where ϕ is particle volume fraction and μ_f is the base fluid viscosity. It is noted that other classical models for calculating the viscosity of mixture also yield similar results [13].

3.4.1. Axial profiles of local heat transfer coefficient

Fig. 11 illustrates the local heat transfer coefficient against the axial distance from the entrance of the test section at Reynolds number (Re) of 1100. The results show that nanofluids exhibits considerably enhanced convective heat transfer coefficient which also increases with volumetric loadings of TiO_2 nanoparticles. For example, at 0.8 vol.% of nanoparticles and at position $x/D_i = 25$ (where tube diameter $D_i = 4$ mm), the local heat transfer coefficient of this nanofluid was found to be about 12% higher compared to deionized

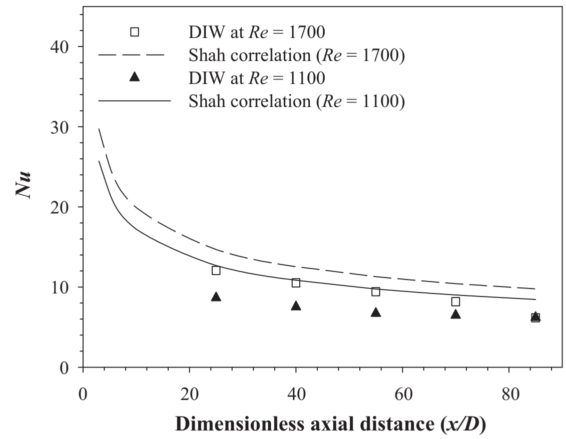

Fig. 12. Comparison with Shah's correlation along axial distance at $Re = 1100$ and 1700 [90].

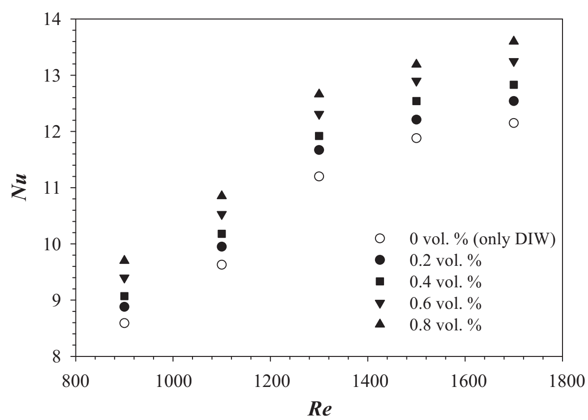

Fig. 13. Effect of Reynolds number and nanoparticle volume fraction on Nusselt number at axial location of $x/D_i = 25$ [90].

water at the same *Re*. The enhancement in heat transfer coefficients of nanofluids is believed due to the enhanced effective thermal conductivity and the acceleration of the energy exchange process in the fluid due to the random movements of the nanoparticles. Another reason for such enhancement can be the migration of nanoparticles in base fluids due to shear action, viscosity gradient and Brownian motion in the cross section of the tube.

Fig. 12 compares experimentally determined Nusselt numbers with the predictions by Shah's correlation, i.e. Eq. (7) along the axial distance. Shah's correlation slightly over-predicts the present results. The difference in tube size may be one of the reasons for such over prediction. A relatively small tube (4 mm diameter) was used in this experiment, whereas the Shah's equation was developed on the basis of laminar flow in large channels. Nevertheless, similar over prediction by Shah's equation was also reported by Wen and Ding [58].

3.4.2. *Effect of Reynolds number and particle volume fraction on Nusselt number*

The effect of Reynolds number on Nusselt number is shown in Fig. 13. It can be seen that the measured Nusselt numbers for nanofluids are higher than those of water and they increase remarkably with Reynolds number. The observed enhancement of the Nusselt number could be due to the suppression of the boundary layer, viscosity of nanofluids as well as dispersion of the nanoparticles. Fig. 13 also demonstrates particle volume fraction dependence of Nusselt number. The Nusselt number of this nanofluid is found to increase almost linearly with the particle volume fraction. The nanofluid behaves more like a fluid than a conventional solid (micrometer or millimeter)–fluid mixture. The effects of several factors such as gravity, Brownian force, and friction force between the fluid and the ultra-fine particles may coexist in the main flow of nanofluids.

4. Concluding remarks

A comprehensive review on three major cooling features—boiling, droplet spreading and convective heat transfer of nanofluids together with representative results from own experimental investigations on these areas are presented and analyzed in this paper.

From the review of available experimental results for boiling heat transfer, it can be conferred that despite of contradictory and inconsistent data on boiling heat transfer, there is undisputed substantial increase in the boiling critical heat flux of nanofluids. However, reported data are still limited and scattered to clearly understand the underlying mechanisms and trend of boiling heat transfer performance of nanofluids. In addition, only a couple of efforts are made on flow boiling of nanofluids. Thus, it is imperative to conduct more research on flow boiling of nanofluids under the influence of various factors such as pressure, mass flux and subcooling besides performing more systematic experimental and theoretical investigations on pool boiling features of nanofluids. Without providing physical-chemistry based details explanation, there are common presumptions such as deposition of nanoparticles or tubes on heat transfer surface and surface wettability used as reasons for the observed results on boiling heat transfer of nanofluids.

This paper also presents investigations of the effects of surfactant concentration and surface tension on the pool boiling heat transfer of carbon nanotubes-nanofluid and results showed that large enhancement of boiling heat flux is possible and would depend on the concentration of the surfactants. The effect of relaxation of surface tension of nanofluid on its burnout heat flux is found to be significant.

Although spreading of liquid droplet plays a key role in many industrial processes like spray cooling, coating and ink-jet printing, only a couple of research efforts have been made to study droplet impingement characteristics of nanofluids on solid surfaces under various conditions. Thus more studies are needed on dynamics of non-boiling and boiling nanofluids droplets impinging on solid or liquid surfaces.

The representative results and the review of the findings from the literature on convective heat transfer of nanofluids demonstrated that nanofluids exhibit an enhanced heat transfer coefficient compared to its base fluid and it increases significantly with increasing concentration of nanoparticles as well as Reynolds number. The review also shows a considerable chaos and randomness in the reported data on convective heat transfer coefficient from various research groups. Therefore, more careful and systematic investigations on nanofluids preparation and the convective heat transfer measurements are needed. A clear understanding of the convective heat transfer mechanisms in nanofluids is also not yet established.

Researchers working on nanofluids have shown tremendous attention to the static thermal properties, particularly thermal conductivity of nanofluids rather than the boiling and convective heat transfer features. Although the applications of nanofluids as advanced cooling medium appears to be promising, the advancement toward concrete understanding on observed properties and features of nanofluids as well as their development for commercial applications remain challenging mainly due to lack of agreement in the data from different research groups, lack of understanding of the mechanisms, and unsystematic measurements and sample preparation. Hence, proper sample preparation and repeatable and more systematic experimental studies on measuring any properties of nanofluids are worthwhile.

In renewable energy sector, nanofluids could potentially be used to enhance heat energy storage from solar collectors and to increase the energy density. Thus, future research should also extend to such renewable energy-based applications of nanofluids.

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