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Nucleate pool boiling heat transfer characteristics of R600a with CuO nanoparticles†

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Abstract

The present research work investigates the effect of CuO nanoparticles on the nucleate boiling heat transfer characteristics of Isobutane (R600a) refrigerant. All the pool boiling experiments are carried with both pure and nano-refrigerants of 0.01, 0.025, 0.05 and 0.1 percentage by volume. The heat flux is varied from 2 $kW.m^{-2}$ to 20 $kW.m^{-2}$ at a regular interval of 2 $kW.m^{-2}$. The heat transfer coefficient values for the pool boiling condition of R600a refrigerant are calculated experimentally, which are less deviating from the established theoretical correlations. The added CuO nanoparticles significantly influenced the nucleate boiling heat transfer coefficient of R600a refrigerant at higher heat flux values. The experiment results reveal that the thermophoretic mobility of nanoparticles play a major role in nanofluids heat transport. In the present work, CuO nanoparticles addition in R600a is optimized and is justified based on gravity and agglomeration effect.

Keywords: Nanoparticles; R600a; Heat transfer; Pool boiling; Thermophoresis

1. Introduction

In global scenario, the heating ventilation and air-conditioning (HVAC) systems consume significant proportion of energy. The energy consumption by buildings is around 40 % of total energy consumptions [1]. Commercial buildings consume nearly 30 % of the total energy for refrigeration and conditioning of air [2]. Hence, energy efficient HVAC systems will more likely be a solution for saving the energy. Nanofluids are the colloidal suspensions having nano scale particles in the common base fluids such as water, oil and ethylene glycol. Nanofluids can be used to enhance the heat transfer performance of electronic devices, nuclear reactors, laser diodes, etc. [3]. Earlier, the nano scale particles were also investigated for its reliability in refrigeration systems and it was found enhancing the performance of refrigerators [4]. The nanoparticles added either in the refrigerant directly [5, 6] or in the lubricant oil [7-9] applicable in the refrigeration systems has been found to have the potential to improve the performance of the refrigeration systems.

Tang et al. [5] investigated the thermal performance of R141b refrigerant with δ-Al_2O_3 nanoparticles. The authors reported that 0.001 vol % of δ- Al_2O_3 nanoparticles in R141b enhanced the pool boiling characteristics of the refrigerant with or without surfactants, and 0.01, 0.1 vol % of δ- Al_2O_3 nanoparticles showed the similar enhancements but along with SDBS surfactants added in R141b. Mahbubul et al. [6] analyzed the effect of thermophysical properties of Al_2O_3 nanoparticles on the coefficient of performance (COP) of R134a refrigerant system. The authors claimed that the addition of 5 vol % of Al_2O_3 nanoparticles with the refrigerant was capable to augment the thermophysical properties of the refrigerant and resulted with a 3.2 % increase in the COP of the refrigerant system.

Kedzierski et al. [7] investigated the influence of CuO nanoparticles mixed lubricant on the pool boiling heat transfer characteristics of R134a refrigerant. At first, the authors mixed CuO nanoparticles with the lubricant (RL68H) in three different mass fractions of 0.5, 1 and 2 %. The three mixtures were tested with R134a refrigerant for its boiling heat transfer coefficient value. Among the three mixtures, the mixture contains 1 % mass of CuO in RL68H enhanced the heat transfer coefficient of R134a to a maximum under pool boiling condition. The nanoparticles mixed in refrigerants should be capable to travel along with the vapor phase to confirm its existence completely. Henceforth, the migration study of nanoparticles is also significant. Ding et al. [10] studied the migration characteristics of CuO nanoparticles from R113/CuO nano-refrigerant during pool boiling process, experimentally and proposed mechanisms for the nanoparticles migration. The authors concluded that the migration tendency of nanoparticles declined with increasing in the volume fraction of nanoparticles in the nano-refrigerant while it increases with the initial mass of refrigerant. This study showed that the mi-

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gration of nanoparticles from the liquid refrigerant to its vapor phase during the boiling process is significantly related to the heat transfer characteristics of nano-refrigerant.

Similar observations were found from the migration study of the refrigerants R141b and n-pentane added with the nanoparticles (Cu, Al, CuO and Al_2O_3) [11]. The studies on nucleate pool boiling heat transfer of refrigerant (R113) with nanoparticles (Cu, Diamond) or CNT's showed that the coefficient of nucleate pool boiling heat transfer of nano-refrigerant could be enhanced to a maximum of 61 % [12, 13]. Presence of surfactants also enhances the nucleate pool boiling heat transfer coefficient of refrigerants, due to reduced surface tension, increased active nucleation sites and by avoiding agglomeration [8]. The enhancement depends on the types of surfactants and the nanoparticles volume fraction.

Mahbubul et al. [14] analyzed the thermal performance of R-134a refrigerant with Al_2O_3 nanoparticles in the range of 1 to 5 % by volume. Thermal conductivity of R134a refrigerant was found increased linearly with Al_2O_3 nanoparticles' volume fraction. Their numerical investigation for 5 to 25 nm size Al_2O_3 nanoparticles revealed that the convective heat transfer coefficient of the R134a was found to be increased with the addition of nanoparticles.

Naphon and Thongjing [15] reported that increasing the volume fraction of TiO_2 nanoparticles reduced the pool boiling heat transfer coefficient of R141b refrigerant. Also they observed that the heat transfer coefficient of R141b was found to increase with the boiling pressure, for their experimental work. Diao et al. [16] conducted experimental investigation on the pool boiling heat transfer characteristics of R141b refrigerant with copper nanoparticles and Sodium dodecylbenzenesulfonate (SDBS) surfactants. Three different concentrations of copper nanoparticles in R141b say 0.008, 0.015 and 0.05 vol % are suspended along with SDBS surfactants. They claimed that the surfactants along with copper nanoparticles in 0.008 vol% decreased the surface tension force in the nano-refrigerant and which could be due to the enhancement in the heat transfer coefficient of R141b under pool boiling, according to the authors. The authors observed that higher volume fractions of nanoparticles in refrigerant have possible tendency to deposit on the heating surface and to increase the thermal resistance, which in further deteriorates the boiling heat transfer coefficient of R141b.

Until now, the majority of explanations from various studies on the heat transfer characteristics of refrigerant based nanofluids are focused on thermophysical properties of the nanoparticles and refrigerants. In the current work the pool boiling heat transfer characteristics of R600a refrigerant along with CuO nanoparticles is investigated. The experimental results are analyzed to corroborate the mechanism behind the physics of heat transfer characteristics of nano based refrigerant. The outcomes of the present study conform that thermophoretic mobility of particles has strong role over the heat transport in nanofluids. Refrigerant R600a belongs to Hydro-Carbon (HC) category which can be used direct or in blend with other suitable refrigerants in many existing applications like commercial, domestic, transport, constructions due to its relatively lower global warming potential than other Chloro-Fluoro-Carbon (CFC) refrigerants (like R11, R12, R134a) and zero ozone depletion potential [17], to benefit a better system performance, with utmost care against its flammability. Copper (II) Oxide (CuO) nanoparticles are found more stable upon dispersed in refrigerant with ultrasonication [18] and commonly used for many technological applications [19]. Refrigerant R600a is currently used in modern refrigeration systems. Yet, limited studies were reported with R600a.

Table 1. Properties of refrigerant, R600a at 40 psi.

Property name	Value	Unit
Saturation temperature	18	°C
Thermal conductivity	0.12	$W.m^{-1}.K^{-1}$
Liquid density	555.56	$kg.m^{-3}$
Vapor density	7.44	$kg.m^{-3}$

Table 2. Properties of Copper oxide nanoparticles.

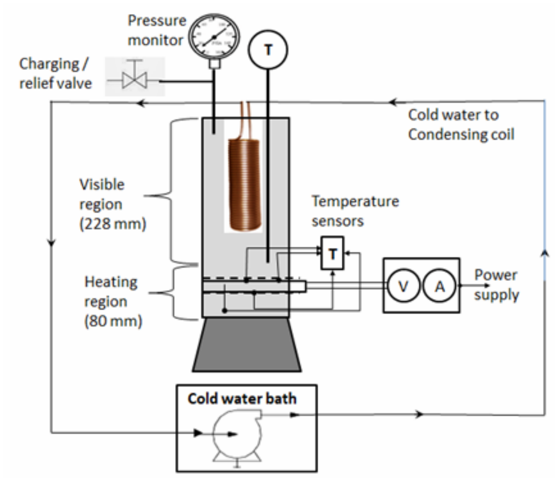
Property name	Value	Unit
Melting point	1326	°C
Density	6.3	$g.cm^{-3}$
Average size	50	nm
Molecular weight	79.55	$g.kmol^{-1}$

Hence, this research work is focused on examining the heat transfer characteristics of the R600a systems with suspension of CuO nanoparticles. Copper (II) oxide nanoparticles of purity > 99 %, 13 $m^2.g^{-1}$ effective surface area were used for the present work. Properties of the refrigerant (R600a) and the nanoparticles are listed in Tables 1 and 2, respectively.

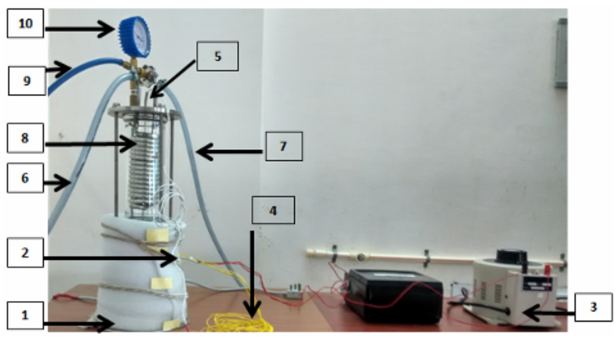
2. Experimental setup

An experimental set-up built to study the nucleate boiling characteristics of R600a+ CuO nanoparticles is schematically shown in Fig. 1(a), and the setup is shown in Fig. 1(b). It consists of a boiling chamber with two proportionate cylinders made from stainless steel (SS304) and Pyrex glass. Pyrex glass cylinder is bonded on top of stainless steel cylinder to ensure visibility of fluid datum. The chamber is sealed with two circular steel discs one at the bottom and the other at the top. A charging valve is fixed at the top of the chamber. A hollow stainless steel cylinder of 22 mm diameter (outer) is welded inside to the boiling chamber in the transverse direction. A brass cartridge heater of maximum rated capacity 200 W is positioned appropriately inside this hollow stainless steel cylinder. Once it is completed, the stainless steel cylinder is properly insulated to avoid heat transfer with environment.

Four calibrated K-type (Nickel-Chromium) thermocouples of accuracy ±1 °C are brazed on the circumference of the brass cartridge heater. Each thermocouple is located at an



(a)



(b)

Fig. 1. (a) Schematic of pool boiling experimental setup; (b) experimental setup (1. Stand, 2. Heater unit, 3. Power supply, 4. Surface temperature readings (heater), 5. Fluid temperature reading, 6. Water outlet, 7. Water inlet, 8. Condensing coil, 9. Chargeing valve, 10. Pressure gauge).

angle of 90° with respect to each other. Surface temperature of heater recorded at a constant pressure is the average of all four thermocouple readings. The brass cartridge heater is covered by a copper tube to provide uniform heat input to fluid. The input heat supplied by electrical means is controlled by a Watt meter, and the power is calculated using the readings obtained from calibrated Voltmeter and Ammeter connected in series with the heater. A condenser coil made of copper is inserted in to the boiling chamber whose inlet and outlet passages are brought outside through the two holes made on the top circular disc. Cold water is circulated inside the coil which helps to maintain a constant pressure through the removal of excess heat from the liquid refrigerant inside the chamber. The vaporized refrigerant from the boiling chamber is condensed back by the cold water circuit, which is attached to a water bath separately. The boiling chamber pressure can be monitored through a pressure gauge mounted on top of the set-up. The fluid temperature is measured using a PT.1 type RTD of accuracy 0.1 °C located inside the boiling chamber through a separate hole made on the top circular disc. A minimum depth of 100 mm liquid datum is maintained for the RTD measurements. The focusing point of study is the boiling of R600a nano refrigerant at the heater surface.

Initially, a vacuum condition of 0.02 mbar is achieved to remove air or any gas present inside the boiling chamber by using a vacuum pump [FD6 model – HHV pumps]. Pure refrigerant R600a is charged in to the chamber through a charging nozzle. Once charging is completed, the charge valve attached in the setup is closed and the level of liquid refrigerant inside the chamber can be seen through the Pyrex glass. All the experiments in the set up are conducted at 40 psi constant pressure. A slender gauge pressure variation of 2 psi is found to occur at high surface temperatures (45 – 50 °C). This small pressure variation is approximated for constant pressure throughout the study. The cold water circuit helps to maintain the liquid refrigerant in saturated condition at the pressure of 40 psi. The heat is supplied to the cartridge heater by means of electrical heat input. The heat flux supplied to the heating surface inside the boiling chamber can be obtained using Eq. (1). At first 20 $kW.m^{-2}$ heat flux is supplied to the refrigerant through the heating surface. The surface temperature and the refrigerant temperature values are monitored. If the temperature change is less than 1 °C in 10 min, then the system is considered to be reached a steady state. The nucleate boiling heat transfer coefficient of the refrigerant, for the supplied heat flux can be calculated based on Eq. (2), as quoted by Trisaksri et al. [20]. The heat flux value is varied from 20 $kW.m^{-2}$ to 2 $kW.m^{-2}$ and the temperatures were noted down for each heat flux values. The heat flux value was reduced gradually at an interval of 2 $kW.m^{-2}$ to avoid hysteresis.

The same procedure is repeated for studies with nanoparticles suspended at different mass fractions in the refrigerant. The sequence of the experiments is fixed based on the concentration level of nanoparticles. The experiments sequence is then randomized and conducted on different time. The measurements of heat transfer coefficient at all four volume concentrations (0.01, 0.03, 0.05 & 0.1) of nanoparticles in the refrigerant were repeated for 8 times on different days. Before starting the experiment, the setup would be vacuumed each time. A good agreement of data is observed at each level of measurement.

$$q" = \frac{V * I}{\pi * D * L} \tag{1}$$

$$h_{nb} = \frac{q"}{T_w - T_{sat}} . \tag{2}$$

3. Experimental error analysis

The measurements may involve uncertainties due to the instruments used, environment, operating conditions and human observations. The accuracy of the experimental instruments (given by the manufacturer) is detailed in Sec. 2. Additionally, the uncertainty in measurements of data is estimated by choosing 5 % significant level for the data sample. A *t*-based confidence interval for the population mean of data measurements including upper and lower limits (two tails) is given by Eq. (3),

$$\bar{x} \pm 2.365\left[\frac{\sigma}{\sqrt{n}}\right]. \tag{3}$$

Here, $\bar{x}$ and σ are the mean and standard deviation respectively. The measurements are repeated for 8 different times at each concentration. Hence, the sample size 'n' for Eq. (3) is taken as 8 and the degree of freedom will be (n-1) = 7. The measured values are found to be deviated from the mean value by an acceptable margin, except for the surface temperature of 50 °C with a deviation of > 5 %.

The heat loss associated to cartridge heating is estimated separately before determining the heat flux value from Eq. (1). The heat can dissipate from the heater along its end faces and its circumference; of which, the heat transfer through its faces contribute the heat loss. One end face of the heater is butt with the cylinder wall which then is covered with a foam sheet (k = 0.25 $W.m^{-1}.K^{-1}$) while the other end is directly insulated with the foam sheet. At the heat flux of 20 $kW.m^{-2}$, the average heater surface temperature is found to be 50 °C. Hence, the temperature of the end faces is assumed to be 50 °C, at 20 $kW.m^{-2}$. The heat loss through the end face of the heater is estimated using Eq. (4). For an average ambient temperature of 35 °C which suits for the present experimental condition, the total heat loss is found to be approximately 1 % of the input heat at the higher heat flux value.

$$q_l = \frac{(T_\infty - T_{face})}{R_{total}}. \tag{4}$$

Finally, the uncertainty associated with the heat flux and heat transfer coefficient (excluding the constants L, D and T_{sat}) are calculated using Eq. (5) given by Holman [21].

$$\omega_F = \sqrt{\left(\frac{\partial F}{\partial x_1}\Delta x_1\right)^2 + \left(\frac{\partial F}{\partial x_2}\Delta x_2\right)^2 + \left(\frac{\partial F}{\partial x_3}\Delta x_3\right)^2 + \cdots +} \tag{5}$$

where F is the function of interest; x_1, x_2, x_3 are the independent variables in the experiment. Based on Eq. (5), the error in heat transfer coefficient is found to be a maximum of 4.7 % at 8 $kW.m^{-2}$ and a minimum of 2 % at 20 $kW.m^{-2}$ and it is shown in Fig. 2 using error bars.

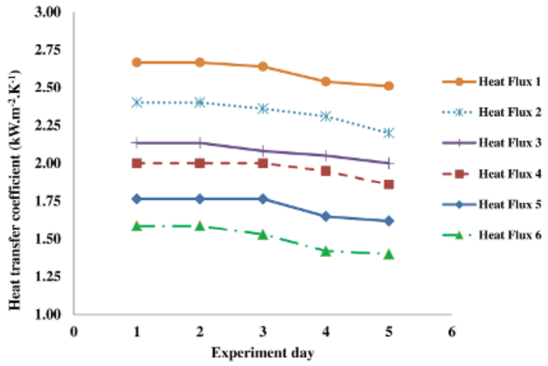


Fig. 2. Heat transfer coefficient-time history.

4. Methodology and validation

The hydrocarbon, iso-butane, is found to be environment friendly and possess good potential for refrigeration applications. However, conducting boiling studies on liquid iso-butane and dispersion of nanoparticles in it are really challenging due to its very low boiling point. The present work is so novel, since both type of experiments are possibly conducted and the results are reported. Generally, the nanofluids are made by dispersion followed by or simultaneous ultrasonication of nanoparticles in the base fluid which are not practically feasible with refrigerants. Henceforth, in a different method, the CuO nanoparticles are dispersed directly in to the chamber along with the refrigerant R600a in the present work. Initially, the quantity of CuO nanoparticles to constitute a specific volume fraction in R600a is estimated and filled in the charging valve (attached to boiling chamber) inlet. Once the charging valve connected to the iso-butane container is opened, the liquid refrigerant is ensured to mix with the nanoparticles in the charging tube and falls inside the boiling chamber in continuous droplets, under gravity. The falling droplets impact the liquid pool and splashes inside the chamber. This effect is considered to promote the continuous particles movement in the liquid refrigerant and visually the particles suspension is conformed during experiments. The visual inspection conformed the suspension of CuO nanoparticles in liquid iso-butane even after 12 h. The tests are conducted after 12 h of charging the refrigerants. Since no separate process is carried out for stable dispersion of CuO nanoparticles, the highest volume fraction for the present work is limited to 0.1 % in a total volume of 600 ml of R600a to ensure stability.

4.1 Stability analysis

In pool boiling studies, stable dispersion of nanoparticles in the base fluid is essential to claim the merits or de-merits of nanoparticles addition. The properties of the refrigerant (R600a), considered for the present work, limit the use of ultrasonic agitation process for dispersion of CuO nanoparticles in it at normal room conditions. The process of ultrasonication of the liquid refrigerant with CuO nanoparticles at high pressure (inside the boiling chamber) is not feasible with the present setup, as well. Trisaksri et al. [20] reported that the nanoparticles present in the base fluid in low volume fractions can be stable for a period of 3 to 4 weeks. Furthermore, the stability information of nanofluids can also be obtained by recording the variation in thermal conductivity of the fluid, which might occur due to particles sedimentation [22]. In a similar way to 3ω-method [22], the stability analysis is carried out through the heat transfer coefficient measurement, in the

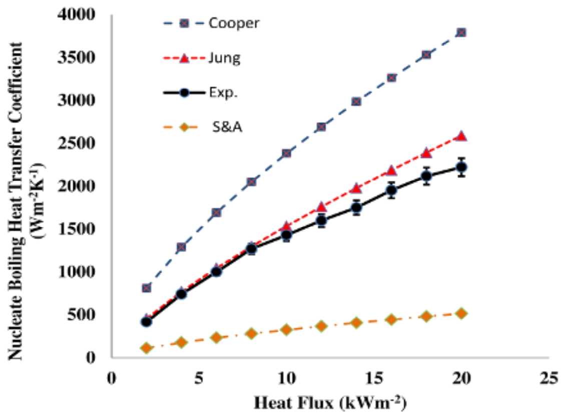


Fig. 3. Comparison of experimental heat transfer coefficient values with theoretical correlations.

present work. For analyzing the stability in R600a, 0.1 % CuO nanoparticles which is larger out of all concentrations in the present work, is taken. The heat transfer coefficients of the nano-refrigerant measured on five consecutive days at the same set of six different heat flux values are plotted in Fig. 2. It is evident from the graph that heat transfer characteristics of the nano-refrigerant starts deteriorating significantly after 72 h in most of the cases. The values measured on day 3 after 72 h of charging, reduced to a maximum of 3.6 %, while the values measured on day 4 after 96 h of charging, reduced to a maximum of 10.7 % from the initial readings. The effect of reduction in heat transfer coefficient values of the nano-refrigerant with time could be attributed to particles agglomeration and progressive sedimentation. Henceforth, the nanoparticles mixed in the refrigerant at all concentrations (0.01, 0.03, 0.05 & 0.10 %) are considered to be stable up to 72 h from the time of charging.

4.2 Experimental setup validation

Preliminary boiling tests on pure R600a are conducted with the experimental apparatus and the test results are plotted as shown in Fig. 3. The nature of variation of heat transfer coefficients of R600a with heat fluxes obtained from the experiments is compared with the predictions given by Cooper [23] as shown in Eq. (6), Stephan and Abdelsalam [24] shown in Eq. (7). The physical meaning of the parameters involved in Eqs. (6)-(8) are provided in Table 4, in the last page of this article.

$$h_{cooper} = 5(P_r)^{0.12-0.4343*\ln(R_p)}\left[-0.4343*\ln\left(R_p\right)\right]^{-0.55} * M^{-0.5}\left(\frac{q}{A}\right)^{0.67} \tag{6}$$

$$h_{S\&A} = 0.0546\frac{K_f}{D_b}\left[\left(\frac{\rho_g}{\rho_f}\right)^{0.5}\frac{\frac{q}{A}D_b}{K_f * T_{sat}}\right]\left(\frac{\rho_f-\rho_g}{\rho_f}\right)^{-4.33}\left(\frac{h_{fg}D_b^2}{a_f^2}\right)^{0.248} \tag{7}$$

where D_b in Eq. (7) is given as follows,

$$D_b = 0.0146\beta\left[\frac{2\sigma}{g\left(\rho_g-\rho_f\right)}\right]^{0.5}.$$

Eq. (6) is valid for the reduced pressure in the range of 0.001 – 0.9 with the molecular weight between 2 and 200 kg/k.mol. Surface roughness can be taken as 1 micron for unknown values for the surfaces.

Eq. (7) is valid for the hydrocarbon refrigerants with reduced pressure in the range of 0.0057 – 0.9. The value for contact angle used in the equation is 35° [24].

$$h_{Jung} = 41.4\frac{K_f}{D_b}\left[\frac{\frac{q}{A}D_b}{K_f T_{sat}}\right]^C\left(-\log_{10}P_r\right)^{-1.52}\left(1-\frac{\rho_g}{\rho_f}\right)^{0.53} \tag{8}$$

where $C = 0.835\left(1-P_r\right)^{1.33}$.

The data measured in our experiments are deviating to a maximum of 41 % from Cooper correlation and 76 % with Stephan & Abdelsalam correlation as experienced by Jung et al. [25]. The possible reason for the variations could be due to the different types of fluids used other than flammable hydrocarbon refrigerants to formulate their correlations as expressed by Jung. The present values for experimental nucleate boiling heat transfer coefficient are compared with the correlation as given by Eq. (8). The deviation of average value of experimental heat transfer coefficients with the predicted values is in the range of 4 % at 2 kWm^{-2} and 14 % at the heat flux 20 $kW.m^{-2}$. This deviation could be attributed to the fact that the type of heating surface is not accounted in Eq. (8). Furthermore the trend closely approaches to the results reported by Jung and confirms the validity of the present results.

5. Results and discussion

In this section, the experimental results from the nucleate boiling heat transfer studies of pure R600a and R600a with CuO nanoparticles are presented. Nano-refrigerant studies are carried out at four different volume fractions viz. 0.01, 0.025, 0.05 and 0.1 % CuO nanoparticles in R600a.

It is evidenced from the initial trials of the present experimental work that the nanoparticles addition could not show an appreciable change in the heat transfer value at very low heat flux values (< 2 $kW.m^{-2}$). Also, the present experimental setup conditions limit the close study of static subcooled boiling phenomenon. Hence, the entire experimental data are recorded in the nucleate boiling regime and focused to discuss in that regime only.

The experimental heat flux values at various concentrations of nanoparticles in the refrigerant against wall superheat are plotted using polynomial trend line (Fig. 4). It clearly shows a

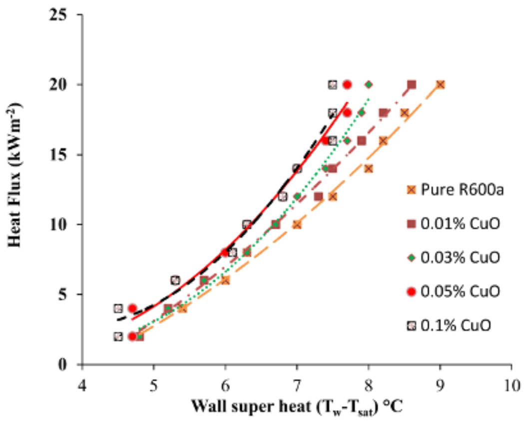


Fig. 4. Influence of nanoparticles concentration on wall superheat.

positive trend that addition of nanoparticles in refrigerant minimized the wall superheat (temperature difference). The value of wall superheat is less at higher volume fractions of nanoparticles, whereas it is high at lower volume fractions. It indicates that heat transfer rate is faster when adding nanoparticles with refrigerant. The degree of superheat closely remains same at the low heat flux value of 2 kW.m^{-2} for all concentrations of nanoparticles. Furthermore, the high volume fraction of nanoparticles which is 0.1 % in the present work, contributes a less significant decrease in wall superheat among the other volume fractions.

During the initial phase of the nucleate boiling regime where the bubbles collapse within the liquid itself, it is expected that the added nanoparticles interact and collapse the bubbles on the heater surface. By doing so, it tends to reduce the wall superheat by quickly making the liquid in contact with the heater surface (bubbles act like thermal barrier). The same effect is observed from the graph shown in Fig. 4.

Dispersion of copper oxide nanoparticles in R600a refrigerant has shown significant enhancement in the heat transfer characteristics of R600a as observed from Fig. 4. At low heat flux value of 2 kW.m^{-2}, the addition of nanoparticles produces no appreciable effects on the heat transfer coefficient of R600a. As seen from the graph, the nanoparticles at any concentrations from 0.01 % to 0.05 % by volume in R600a are not apparently enhancing the heat transfer coefficient of R600a at 2 kW.m^{-2}.

Nanoparticles seeded in the refrigerant in various concentrations, showed a positive trend of enhancing the heat transfer coefficient of R600a refrigerant at increased values of heat flux. Subsequent investigation with the nano-refrigerants revealed that the heat transfer coefficient increases with increase in nanoparticles concentration at higher heat flux values, which resembles the trend observed by research studies on refrigerants [8, 9]. Significant increase in heat transfer coefficient of about 20 % was observed with 0.1 % volume fraction of nanoparticles at high heat flux values but with a little enhancement was observed at heat flux value, 2 kW.m^{-2}.

The confidence interval is set to 95 % for the heat transfer

Table 3. Diffusion mechanisms and governing parameters.

S. No.	Diffusion mechanism	Parameters
1	Mass (particle collision)	Particle size & fluid viscosity
2	Momentum (heat convection)	Fluid viscosity and density
3	Energy (heat conduction)	Thermal conductivity, density and specific heat of fluid

coefficient values obtained from the measurements. Significant enhancement in the heat transfer coefficient of the refrigerant is observed with 0.10 % concentration of nanoparticles at all heat flux values, while, it is observed with 0.03 % and 0.05 % concentration only at higher heat flux values (> 10 kW.m^{-2}). At lower heat flux values (< 10 kW.m^{-2}) the confidence intervals set for each concentration are found to overlap.

The percentage enhancement of heat transfer coefficient of R600a contributed by the addition of 0.1 % volume fraction of CuO nanoparticles in the refrigerant is reported to be higher than the contributions made by other low volume fractions of CuO nanoparticles at various heat flux values as shown in Figs. 5 and 6. Nevertheless, the addition of 0.05 % volume fraction of CuO nanoparticles in R600a resulted closely 17 % and 2.3 % enhancement in heat transfer coefficient at high heat flux and low heat flux respectively. The present experimental result shows significant improvement in the heat transfer coefficient of R600a with 0.05 % dosage of CuO nanoparticles. It is more significant when compared to the enhancement obtained for 0.1 % volume fraction of CuO nanoparticles.

For all concentrations of CuO nanoparticles in the refrigerant, it was observed that the heat transfer coefficient enhancement is linear and very progressive with 0.05 % of CuO nanoparticles. At 10 kW.m^{-2}, a marginal increase of 4 % in the heat transfer coefficient is resulted with both 0.01 % and 0.03 % volume fraction of nanoparticles in R600a, whilst an effective increase of 11 % is resulted with both 0.05 % and 0.1 % volume fraction of nanoparticles. Furthermore, at 20 kW.m^{-2}, the heat transfer coefficient enhanced by 5 %, 12 % and 17 % with the addition of 0.01 %, 0.03 % and 0.05 % volume fraction of CuO nanoparticles in the refrigerant. Whereas at the same heat flux value, the addition of 0.1 % volume fraction of nanoparticles showed only 20 % enhancement in the heat transfer coefficient. Interpreting the present results at high heat flux values in particular, as evidence from Fig. 6, it can be concluded that the percentage improvement of heat transfer coefficient with 0.1 % of CuO nanoparticles is not significant for its volume fraction when compared with the percentage improvement resulted with 0.05 % volume fraction of nanoparticles in R600a.

The effect of nanoparticles on the heat transport behavior of refrigerant can be explained by comparing the nature of three different diffusion mechanisms; energy, mass and momentum. Heat diffusion, collision of particles and convection nature of

Table 4. Parameters involved in Eqs. (6)-(8).

S. No.	Parameter (Unit)	Description	Meaning
1	P_r (No unit)	Reduced pressure	Ratio of experiment pressure to liquid saturated pressure of refrigerant
2	R_p (μm)	Surface roughness	Roughness of heating surface
3	K_f ($W.m^{-1}.K^{-1}$)	Thermal conductivity	Thermal conductivity of liquid refrigerant
4	D_b (m)	Bubble diameter	Diameter of bubbles depart from heater surface
5	a_f ($m^2.s^{-1}$)	Thermal diffusivity	Ratio of heat conductivity to volumetric heat capacity
6	h_{fg} ($kJ.kg^{-1}$)	Heat of vaporization	Latent heat of vaporization at a temperature
7	ρ_g & ρ_f ($kg.m^{-3}$)	Densities of vapor and liquid	Density values of saturated vapor & liquid refrigerant

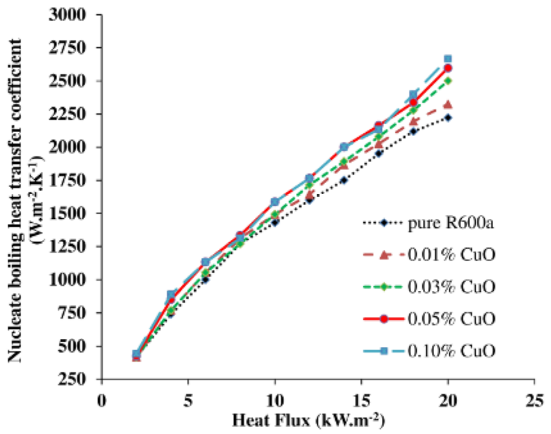


Fig. 5. Heat transfer coefficient variation with heat flux at different concentrations of nanoparticles.

heat transport are the basic physics to constitute the respective diffusion mechanisms, aforementioned. The governing parameters for each diffusion mechanism are presented in Table 3. Most of the previous studies on nano based refrigerants quoted improvement in thermal conductivity as the reason behind the heat transfer enhancement.

To compare the rate of particles motion in a suspension and the rate of heat diffusion, their time scales are analyzed [26] in literature. Here, the distance equal to the nanoparticle size is utilized to compare the time taken by nanoparticles and the heat to diffuse in the base fluid. Heat (energy) diffusion is reported to be faster than particles (mass) diffusion based on simulation studies [27]. This finding may support one to claim that addition of nanoparticles enhanced the heat transfer characteristics of refrigerant due to its improved thermal conductivity only and the particles movement and collision have no significant role in it. Nevertheless, the particles motion is equally significant to its thermal conductivity in nanofluids heat transport and it is supported through the findings that the

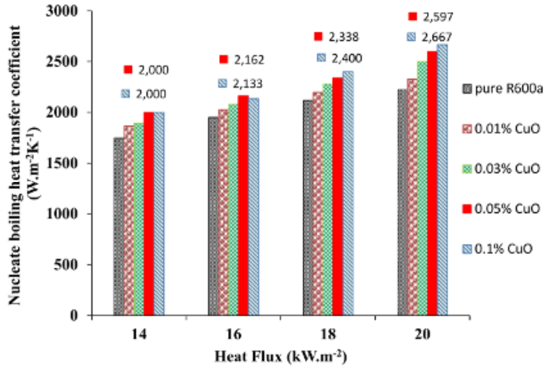


Fig. 6. Enhancement of heat transfer coefficient at higher heat flux values.

time scales for conduction and convection diffusions are comparable [26]. Henceforth, the particles motion is considered for interpreting the results of the present work, apart from its thermophysical properties. Furthermore, the thermophoresis effect is likely considered to be the reason for promoting the particles motion (Brownian effect) under pool boiling conditions and hence, the heat transfer in nanorefrigerant is explained with the help of thermophoresis effect.

Thermophoretic force arises in a fluid due to the existence of temperature gradient between the hot and cold regimes within the fluid. This force favors the motion of highly conductive particles from hot to cold region with a velocity [28] and thereby enhances the heat transfer in nanofluids. The effect of thermophoresis exists until the fluid reaches thermal equilibrium with the source and it is reasonable to conclude that the conductivity of particles has no significant role on thermophoretic effect. The fluid that surrounds the heater surface imparts more energy to the particles present in that region and the high energy particles travel towards the colder region to lose its heat energy to the fluid and gains kinetic energy along the negative temperature gradient. The mobility of particles increases with increase in the heater surface temperature, due to thermophoresis. This can be attributed as a reason to increase in heat transfer coefficient of CuO added refrigerant at higher heat flux values when compared at low heat flux, as given in Fig. 5. Thermophoresis can then be considered an augmenting phenomenon of both heat and convection diffusion in nanofluids.

At higher heat flux values of more than 10 $kW.m^{-2}$ in the present study, 0.05 % CuO nanoparticles contributed to a minimum enhancement of 10.3 % and maximum enhancement of 16.9 % in the heat transfer rate, as shown in Fig. 6. Also, the calculations performed in a similar way revealed that a minimum of 10 % and a maximum of 20 % enhancement in heat transfer rate is achieved with 0.1 % CuO nanoparticles. As it can be witnessed from Fig. 6, dispersion of 0.1 % CuO nanoparticles in the refrigerant performed well over 0.05 % CuO nanoparticles merely by 4 %. This observation with two different concentrations of nanoparticles in refrigerant could be explained with the help of gravity and particles' interactions.

In general, nanoparticles, when suspended in any base fluids are subjected to the influence of gravity, particle-particle interactions and particle-fluid interactions. The nanoparticles suspended in a base fluid involve thermophoretic action at any heater surface temperature, and the chances of interacting among them increases with the heat flux values. Also, the mean-free path of travel reduces at large volume fraction of particles. It is therefore, the addition of nanoparticles further to an optimum level may bound to face the agglomeration issues and leads to form particles of micron sizes. Gravity force, by its nature, plays a major role in settling down the particles in the base fluid and its tendency proportionate to the mass of the particles. It is understood that larger the volume fraction of nanoparticles suspended in a refrigerant more the chances of nanoparticles sedimentation in it. Furthermore, optimum amount of nanoparticles for a prolonged stable suspension should be identified by doing a trade-off between the input quantity of nanoparticles and the output performance achieved. Based on the experimental outcomes and considering the stability issues (especially for refrigerants), the amicable concentration of CuO nanoparticle, to achieve a good level of heat transfer improvement is suggested as 0.05 %.

6. Conclusions

In the present work, the effects of CuO nanoparticles on the nucleate pool boiling heat transfer characteristics of refrigerant R600a has been experimentally investigated. The following are the main outcomes from the study.

(1) The experimental set up has been developed and the heat transfer coefficient values estimated from the parameters taken from the set up are validated with standard correlations.

(2) Dispersion of nanoparticles at 0.01, 0.03, 0.05 and 0.1 % by volume in refrigerant R600a is found to be increasing the heat transfer coefficient of the refrigerant at heat flux values ranging from 2 kW.m^{-2} to 20 kW.m^{-2}.

(3) CuO nanoparticles of 0.1 % volume fraction in R600a established nearly 20 % enhancement in heat transfer coefficient of refrigerant at higher heat flux.

(4) The optimum concentration of CuO nanoparticles in stable dispersion for a prolonged time and contribute for significant enhancement in the heat transfer coefficient of R600a refrigerant is experimentally found as 0.05 % by volume.

On the whole, the present work proposes the optimum volume fraction of CuO nanoparticles which could give effective enhancement in the heat transfer characteristics of R600a refrigerant. This work can be extended to form a numerical model of simulating thermophoresis of nanoparticles and its behavior during pool boiling conditions of the refrigerants.

Nomenclature

V : Voltage [V]
I : Electric current [A]
D : Diameter [m]
L : Length of heater [m]
q : Heat supplied [W]
q" : Heat flux [W.m^{-2}]
A : Surface area [m^2]
h : Heat transfer coefficient [W.m^{-2}.K^{-1}]
T : Temperature [K]
R : Resistance [K.W^{-1}]
Pr : Reduced pressure
Rp : Surface roughness [μm]
M : Molecular weight [kg.k.mol^{-1}]
K : Thermal conductivity [W.m^{-1}.K^{-1}]
g : Gravitational acceleration [m.s^{-2}]

Greek symbols

α : Thermal diffusivity [m^2.s^{-1}]
β : Contact angle [°]
ω : Uncertainty [%]
ρ : Density [kg.m^{-3}]
σ : Surface tension force [N.m^{-1}]
∞ : Ambient condition [-]

Subscripts

w : Heater surface
sat : Saturated liquid
l : Heat loss
f : Liquid refrigerant
g : Vapor refrigerant
b : Bubble
nb : Nucleate boiling

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