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Review of Nucleate Pool Boiling Heat Transfer using Refrigerant

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Abstract

Nucleate pool boiling has an important place in refrigeration industries. The pool boiling process occurs in the shell side of flooded evaporators and low pressure refrigerants are proposed for industry applications. Enhancement of heat transfer rate depends on different design of heating surface, type of refrigerant and operating parameters like heating surface roughness, surface orientation, operating pressure & temperature. Refrigerant plays a crucial role in the study of nucleate pool boiling heat transfer. Various inferences have been drawn based on the existing parameters by different researchers for enhancement of heat transfer rate. In this paper a detailed study has been carried out using different refrigerants on different surfaces to investigate the optimum value of heat transfer coefficients (HTC). The process could be further investigated using heat transfer enhancement particles such as nanofluids and modification of the heating surface. The surfaces are characterized with contact angle, roughness, film thickness and scanning electron microscopy (SEM) & atomic force microscopy (AFM) etc. The surface modification can be done by nanocoating or polishing the surface area. The addition of lubricant to individual or binary or ternary refrigerants can significantly alter the boiling phenomena for accumulating lubricant at the nanocoating heat transfer surface. Many researchers explained the heat transfer enhancement techniques by the passive methods. Many authors gave empirical correlations (dimensional and nondimensional) for heat transfer coefficient in terms of surface roughness factor, vapour density, liquid density, etc from their experimental results. But there was no theoretical method to get the value of HTC which is a function of bubble density, surface roughness factor, bubble growth, etc.

Keywords: Pool Boiling; Surface; Refrigerant; Heat Transfer.

NOMENCLATURE :

A	Heat transfer area (m^2)
Bo	Bond number
D_b	Bubble diameter
dp	Maximum horizontal length of the cavity
h	Heat transfer coefficient [Wm^2K^{-1}]
M	Molecular mass, kg/kmol
m	Exponent of $\emptyset$ and constant
Pr	Prandtl number, (□α^{-1})
R_a	Arithmetical mean deviation of the profile, µm
$\mathcal{P}$	Pressure
Greek letters	
α	Thermal diffusivity (m^2 s^{-1})
λ	Cavity height (mm)
ϵ	Base open length of cavity (mm)
ϑ	Kinematic viscosity(m^2 s^{-1})
Subscripts	
sat	Saturation

INTRODUCTION

The characterization of nucleate pool boiling heat transfer on surfaces using refrigerant recently becomes more and more important because of the optimal design of the flooded evaporators to save energy and to conserve the natural resources. The mechanisms of pool boiling heat transfer on surfaces using refrigerant have been studied for a long time, since they are related with the design of the more efficient flooded type evaporators (Figure 1) in refrigeration & air-conditioning industries. Most of industrial water chillers are having flooded type evaporator. In the flooded evaporator the refrigerant surrounds the tube in the shell and water to be cooled flows through the tubes. Enhancing heat transfer in evaporators is one of the major approaches to increasing chiller efficiency and heat transfer rate depends on different design of heating surface, type of refrigerant and operating parameters like heating surface roughness, surface orientation, operating pressure & temperature. Refrigerant plays a crucial role in the study of nucleate pool boiling heat transfer. This paper highlights the nucleate pool boiling heat transfer characteristics on surfaces using refrigerants by different authors to predict the nucleate pool boiling heat transfer coefficient of refrigerants and also presents a review of literature for boiling on different surfaces using individual refrigerants, mixture of pure refrigerants, refrigerant with oil mixtures, refrigerant with nanoparticles and recent research on nanocoated surfaces with the aim to identify future research work requirements & objectives.

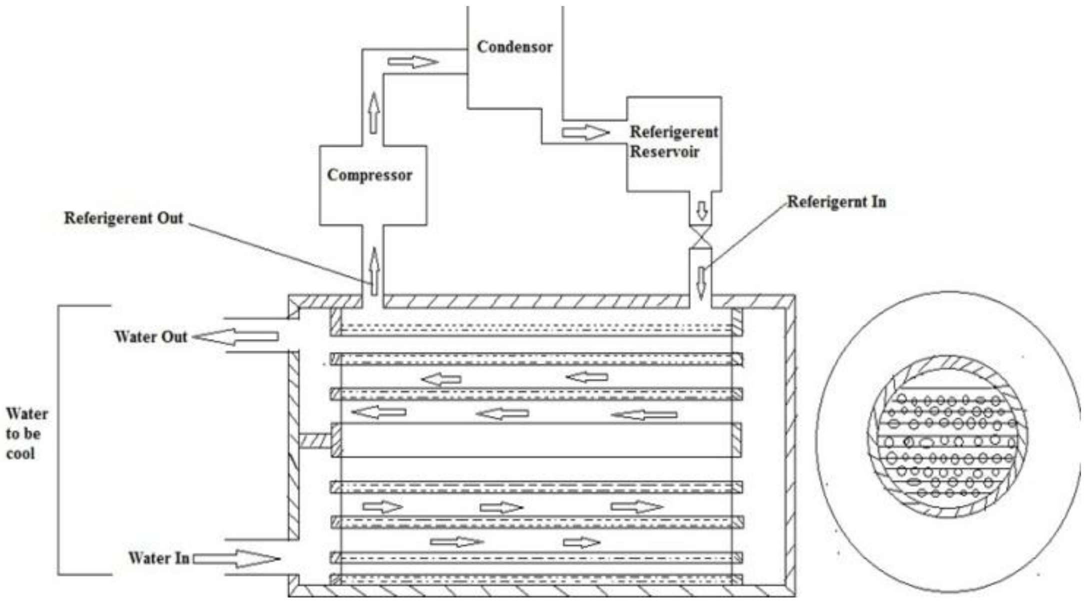

Figure 1. Schematic of a flooded type evaporator

INDIVIDUAL REFRIGERANTS

The experiments on five different horizontal tube geometries using five refrigerants at two saturation temperatures were performed by Webb and Pais [1]. The refrigerants tested are R-11, R-12, R-22, R-123 and R-134a at saturation temperatures of 4.44°C (40°F) and 26.7°C (80°F). The tube geometries tested are a plain tube, a 1024 fins/m integral-fin tube, and three commercially used enhanced tube geometries (GEWA TX19, GEWA SE, and Turbo-B). Except for the Turbo-B with R-11, the authors observed that the boiling coefficients for R-123 and R-134a are within 10% of the values for R-11 and R-12, respectively. For R134a compared to R12 and R22, the boiling curves are quite similar, lying, near to the R12 data and slightly lower the R22 data. Chiou and Lu [2] carried out experimental study on a plain copper tube using refrigerant R-22, R124 and R134a at reduced pressures of 0.1 and 0.2. The saturation temperatures were 4.4^0C and 26.7^0C. For R134a compared to R124 and R22, the boiling curves are similar, lying higher to the R124 data and slightly lower the R22 data except at reduced pressure 0.1 for all heat flux. At reduced pressure 0.1 for R134a, the boiling curve is lying slightly higher to the R22 data at low heat flux and slightly lower to the R22 data low heat flux. Jabardo et al. [3**]** determined the effects of surface roughness of different materials on nucleate boiling heat transfer of refrigerants R-134a and R-123, with cylindrical surfaces of copper, brass and stainless steel. The results were showing significant effects of surface material, with brass being the best performing and stainless steel the worst. Polished surfaces seem to present slightly better performance than the sand paper roughened.Ji et al. [4] studied the pool boiling heat transfer performance of refrigerant R134a on single horizontal tube surfaces sintered with open-celled copper foam at Ts= 6^0 C; Ps=3.62 bar; pore density values: 40, 80 & 130 PPI; two porosity values: 90% & 97% and two thickness values: 1.6 & 2.5 mm. Enhancement of the foam-coated tubes show higher compared with plain tubes at heat fluxes below 30 kW/m^2. A sharp reduction in heat transfer coefficient is encountered for pore density 130 PPI at larger heat fluxes. Tube coated with thin foam and high porosity having pore density 80 PPI, porosity 97% & thickness 1.6 mm, indicates a comparatively higher heat transfer property. Yang et al. [5] studied the pool boiling heat transfer characteristics for enhanced surface tubes using HFC134a. Key parameters were the shape of tube surfaces, the wall superheat and the saturation temperature of the experiment. Outer diameter (d_o = 19.05 mm) of the copper tubes were treated with different helix anglesand the saturation temperatures were varied from 3^0C to 16^0C. The result shows that the pool boiling heat transfer coefficient, h_o decreases with increasing the wall superheat. The boiling heat transfer coefficients for Turbo-II and Turbo-III are 1.5–3.0 times and 1.2–2.0 times higher than that for Turbo-I without the helix angle respectively. The higher heat transfer coefficient from Turbo-II and Turbo-III was found due to ''bubble detention'' phenomenon on the surface without the helix angle for the Turbo-I.

For Type I without the helix angle,

$10,000 \leq q \leq 30,000;\ 2.5 \leq \frac{\lambda}{\epsilon} \leq 7.0$

$$\text{ho} = 4347.524 \left[\frac{K_f}{D_b}\right]\left[\frac{(q/A)D_b}{K_f\,T_{sat}}\right]^{0.283}\left[\frac{\rho_g}{\rho_f}\right]^{1.298} Pr^{0.132}\left[\frac{\lambda}{\epsilon}\right]^{0.241} \quad (1)$$

For Type II and Type III with the helix angle

$20,000 \leq q \leq 25,000;\ 5 \leq \frac{\lambda}{\epsilon} \leq 6.0$

$$\text{ho} = 8{,}134{,}280 \left[\frac{K_f}{D_b}\right]\left[\frac{(q/A)D_b}{K_f T_{sat}}\right]^{0.16}\left[\frac{\rho_g}{\rho_f}\right]^{1.519} Pr^{1.046}\left[\frac{\lambda}{\epsilon}\right]^{-5.3081} \quad (2)$$

where, D_b, and Bo are bubble diameter and Bond number, respectively.

$$D_b = \left[\frac{Bo + 2404(96/Bo - 3)^{1/2}}{192 - 6Bo}\right] d_p$$

and

$$Bo = \left[\frac{{d_p}^2(\rho_l - \rho_v)g}{\sigma}\right] d_p$$

Following experimental correlations for the pool boiling heat transfer on the studied tubes were developed with the error bands of 30%. In this experiment, dp is defined as the maximum horizontal length of the cavity. The aspect ratio (λ/ϵ) is the ratio of the cavity based on the height (λ) and the base open length on the surface (ϵ) for the cavity. In the helix angle, the bubble remains in the cavity and the heat transfer rate increases as the height of the cavity increases. In the helix angle, the bubble escapes easily from the cavity and the heat transfer rate increases as the height of the cavity.Ribatski and Thome[6]reported the results of an experimental investigation on pool boiling heat transfer of R134a on Gewa-B, Turbo-CSL and Turbo-BII HP tubes at high heat flux. The range of heat fluxes from 20 to 70 kW/m^2 at saturation temperatures of 5, 10 and 20^0C. At most of the experimental result shows that, there are a negligible effect of the saturation temperature on the heat transfer coefficient except for Turbo-CSL. The heat transfer enhancement ratios of Gewa-B, Turbo-CSL and Turbo-BII HP tubes were 4.9–21.3, 2.4–5.2, 2.4–2.9 and 1.8–7.0, respectively, compared to a plain tube at high heat flux. Furberg and palm [7] carried out a visualization study to understand the boiling mechanism on dendritic and micro-porous copper structure surface. Here pool boiling tests were conducted using R134a & dielectric fluid FC- 72 separately and images were visualized with a high speed system. The data collected on bubble size, bubble frequency density, HTC, latent and sensible heat flux contributions and calculated at heat flux varying from 2 to 150Kw/m^2. The enhanced surface produces smaller bubbles and sustains a high bubble frequency density in both fluids, even at low heat flux. The enhanced latent heat transfer mechanism of up to 10 times compared to that of a plain reference surface. The high nucleation bubble frequency density leads to increase bubble pumping action and therefore single phase convection up to 6 times compared to plain reference surface. Ribatski and Jabardo [8] investigated the saturated pool boiling of halocarbon refrigerants on cylindrical surfaces of different materials with different finishing conditions under wide range of reduced pressures and heat fluxes. The results indicate that the nucleate boiling HTC of high pressure refrigerants (R-12, R-22 & R-134a) is higher than that of lower pressure refrigerants (R-11 & R-123) and the following correlation was proposed. The correlation was compared with experimental data and found ±20% range of error.

$$\frac{h}{\phi^m} = f_w p_r^{0.45}[-\log(p_r)]^{-0.8} R_a^{0.2} M^{-0.5} \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

$$Where, \quad m = 0.9 - 0.3p_r^{0.2}$$

The heating surface material parameter, f_w is assumed as copper: 100; brass: 110; and stainless steel: 85.

Jung et al. [9] determined the nucleate boiling heat transfer coefficients of HCFC22, HFC134a, HFC125 and HFC32 on a low fin, Turbo-B and thermoexcel-E tubes, at pool temperature of 7^OC. The heater surface is a horizontal tube of length 152 mm and 18.6-18.8 outside diameter. The heat flux ranges from 10-80 kw/m^2 with an interval of 10 Kw/m^2. This paper concluding that, high vapour pressure refrigerants showing higher boiling heat transfer coefficients(HTCs) for plain and low fin tubes, due to the wall superheat required to activate given size cavities became smaller as pressure increased. Barthau and Hahne [10] studied experimentally the nucleate pool boiling of R-134a in the reduced pressure range $0.03 \leq p/pc \leq 0.5$ (1.2bar $\leq p \leq$ 20.3bar) for heat fluxes from q=100000 w/m^2 down to single phase natural convection. In addition to the heat transfer measurements, nucleation site density, up to $(N/A)_{max} \approx 6000$ sites/cm^2 is measured by an optical method. The observations were: At high heat fluxes, the local heat flux behaves radial, symmetric and at lower heat fluxes, the local heat flux distribution seems to be governed by the macroscopic two-phase flow around the tube. Gorenflo et al. [11] investigated the pool boiling heat transfer from a single horizontal copper tube (8 mm OD) to HFC-refrigerants (R32, R125, R134a, R143a, R152a and R227) and hydrocarbons (propane, i-butane). The experimental results were compared to the experimental data from the literature and discussed, how to incorporate the data in semi-empirical correlations to describe the influence of thermophysical properties of the fluids on the heat transfer performance. The dependence of HTC on the thermophysical properties is also influenced by the microstructure of the heating surface. Gorgy and Eckels[12] studied experimental results of nucleate pool boiling of R-134a & R-123 on smooth, enhanced tube TBIIHP and enhanced tube TBIIHP at a saturation temperature of 4.44^{0}C. The outer diameter was 19.05 mm of 1 m length for each tube. Experimental results showed the enhanced tubes significantly enhanced the refrigerant side heat transfer coefficients. The present study also continued to the widest heat flux ranges with this type of tubes and showed significant structure to the pool boiling curve that was not traditional.Tatara and Payvar [13] performed experiment on Turbo-BII-HP surface using R-134a refrigerant and found superior performance. The heat coefficient range from 19,294 to 28,117 W/m^{20}C that were increased with heat flux. The corresponding saturation temperatures were 4.6, 5.4, 5.2, 4.2, 4.1 and 5.1^{0}C at heat fluxes of 8,224, 12,335, 16,445, 24,669, 32,893 and 41,114 W/m^2, respectively. Hsieh and Weng [14] carried out experiment for saturated boiling of R-134a and R-407c on copper surface, coated with porous aluminium, copper and molybdenum. The coating techniques were plasma spraying, flame spraying and pitted spraying. Copper and molybdenum were coated by plasma spraying with coating thickness of 35μm and 100 μm, respectively. Aluminium and zinc were coated by flame spraying with a thickness 50-300 μm and 150 μm, respectively. Four pitted coating thickness, viz 18, 30, 31 and 32 μm were done by sand blasting technique. They observed that the R-134a was performed better heat transfer than R-407c at $q > 10$ kw/m^2 and also find out that pitted coating surfaces performed best with R-134a while plasma spraying coated surface well with R-407c.Zhou and Bier [15] carried out experiment to study nucleate pool boiling heat transfer on a horizontal copper tubes which are coated with 0.2mm of aluminum oxide-titanium oxide ceramics using refrigerant R-12, R-113, R-114 and R-134a and three hydrocarbons i.e. propane, n-butane, and n-pentane. The heat transfer coefficient shows a similar dependence on heat flux and normalized saturation pressure as with a metallic heating tube. For hydrocarbons, the absolute values of heat transfer coefficient are just as high as for a sand bla tube of similar surface roughness at normalized saturation pressure $p/p_c \geq 0.1$ and lower saturation pressure even higher. The negative influence of the low thermal conductivity of the ceramics are completely compensated or ever over compensated by positive influence of the microstructure, which results in a higher nucleation site density which is especially effective in pool boiling heat transfer. Hsieh et al [16] conducted experiments for nucleate pool boiling heat transfer from plasma coated copper tube bundles with porous copper (Cu) using saturated R-134a. The Number of tubes were 15 (of which the number of heated/ instrumented tubes was varied) arranged in four types like, (i) one, two, and three vertical-in-line tubes; (ii) one, two and three horizontal-in-line tubes; (iii) two rectangular type 2×2 and 2×3 array; and (iv) triangular type (three tube bundle and six tube bundle) with a pitch-to-diameter ratio of 1.5. The bundle arrangements and heat flux were varied. Tests were conducted with both increasing and decreasing the heat flux. The result indicated that at low heat fluxes, the vertical-inline tube bundles were the highest in heat transfer performance.Md Chowdhury and Kaminaga[17]experimentally measuredboiling heat transfer characteristics of Freon R-113 in a vertical small diameter tube, D =1.45 mm and L = 100 mm at a wide pressure range of 19–269 kPa under natural circulation condition. The flow regime was annular except the entrance region of the test section. Enhancement of heat transfer coefficient was not found although D/B is less than 1.5.Seo et al. [18] studied, the evaporation heat transfer coefficients and pressure drops for R- 22 and R-410A were measured and analyzed as a function of heat flux, mass flux, evaporating temperature, and tube diameter. The experiments were conducted for the smooth and micro-fin tubes with outside diameters of 9.52 mm and 7.0 mm. For both refrigerants, the evaporation heat transfer coefficient was enhanced as the mass flux increased for the smooth and micro-fin tubes. The micro-fin inside a tube was more effective in the larger tube diameter in the range of present test conditions. For R-22, the heat transfer enhancement factor (EF) for 9.52 and 7.0 mm tubes varied from 2.3 to 3.3 and from 1.3 to 1.6, respectively. For R-410A, the EF for 9.52 and 7.0 mm tubes ranged from 1.8 to 2.9, and from 1.1 to 1.5, respectively. The heat transfer coefficients of R-410A were higher than those of R-22 in the range of lower mass fluxes, higher heat fluxes and higher evaporating temperatures. The pressure drop increased with a decrease of evaporating temperature and increase of mass flux. The pressure drop for R-410A was smaller than that for R- 22 due to lower viscosity and velocity. As the evaporating temperature decreased and the mass flux increased, the difference of pressure drop between R-22 and R-4I0A was reduced. Pool nucleate boiling heat transfer experiments from coated surfaces with porous copper (Cu) and molybdenum (Mo) and spirally wrapped with helical wire on copper surfaces with micro-roughness immersed in saturated R-134a and R-600a were conducted byHsieh and Yang [19]. The key parameters were influence of coating thickness; porosity, wrapped helical angle, and wire pitch on heat transfer and boiling characteristics including bubble diameters, contact angle. The heat transfer enhancement was ranged from 1.16 to 2.37 for R-600a and R-134a refrigerant. The enhanced surface heat transfer coefficients with R-600a as refrigerant were performed 2.4 times higher than those of the smooth surfaces.The key parameters were influence of coating thickness; porosity, wrapped helical angle, and wire pitch on heat transfer and boiling characteristics including bubble diameters, contact angle. The heat transfer enhancement was ranged from 1.16 to 2.37 for R-600a and R-134a refrigerant. The enhanced surface heat transfer coefficients with R-600a as refrigerant were performed 2.4 times higher than those of the smooth surfaces. Tehver et al. [20] conducted experiments for pool boiling of R-113 at atmospheric pressure on a wide range of porous surfaces to study relationship between the effectiveness of heat transfer and structural parameters of a

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plasma sprayed coating. They used 113 different porous surfaces with base material of copper and aluminium plate. The heating surfaces were coated by various combinations of materials such as aluminium bronze, copper-bronze, copper-copper, aluminium-copper, aluminium-corundum and aluminium-aluminium using plasma spraying technique. The parameters of surface include porous layer thickness from 0.01 to 0.60 mm, porosity from 5% to 61 %, and mean pore diameter from 2 to 31.4 µm. They generated data up to burn out heat flux point. They studied that porous surface parameters such as porosity, mean pore radius and porous layer thickness of porous coating have profound effect on heat transfer performance and determined optimal values of these three parameters analytically. They also remarked that porous coating material of higher thermal conductivity provides higher rate of heat transfer. Scurlock [21] carried out experiments for saturated boiling of liquid nitrogen, argon and R-12 on enhanced porous flat surfaces at atmospheric pressure. Heating surfaces were coated with pure aluminium or a mixture of aluminium/silicon powder (90/10) and polyester on to a 5 mm thick aluminium back plates using plasma spraying technique. Coating thickness was ranged from 0.13 to 1.32 mm. They observed that heat transfer coefficient enhances up to 10 times than those for smooth surfaces. They found that there is an optimum thickness of plasma sprayed coating for each liquid and selected heat flux in order to achieve maximum heat transfer coefficient. Further, they also found that the effect of fouling by impurities and found that smooth surfaces may show greater degradation in heat transfer performance than porous surfaces.Chien and Webb [22] investigated boiling characteristics of R-123 on five different structured enhanced surfaces at atmospheric pressure. They found that a) bubble growth mechanism on enhanced surfaces is different from that on plain surface, b) a significant fraction of vaporization occurs at menisci in the corner of the tunnels which control bubble frequency and nucleation site density, c)Evaporation and bubble growth occurs after the bubble emerges from the surface pores, d) Smaller bubbles are generated on the enhanced surfaces at a greater frequency as compared to those on plain surface for the same heat flux condition and the enhanced surface has greater nucleation site density than that on plain surface.

MIXTURE OF PURE REFRIGERANTS

Zhao et al. [23] measured the heat transfer coefficients in nucleate pool boiling for refrigerants under saturated conditions at 0.9 MPa on a horizontal copper surface. The refrigerants were the pure components of HFC-134a, HFC-32 and HFC-125 and two kinds of binary mixtures: nonazeotrope mixture HFC-32/134a and azeotrope mixture HFC-32/125. Both binary mixtures showed lower heat transfer coefficients as compared with pure refrigerants. Also with the increased heat flux this deterioration was observed to be more.Sun et al. [24]executed series of experiments to measure heat transfer coefficients in nucleate pool boiling on a smooth flat surface using refrigerant R-134a, propane, isobutene and their binary mixtures at pressure 0.1 to 0.6 Mpa. They applied different heat flux and different mixture concentrations and studied the influences of pressure and heat flux on the heat nucleate pool boiling for pure refrigerants. Isobutene and propane were applied for making binary mixtures. As compare to the pure component, they found that binary mixture showed lower heat transfer coefficient and the reductions were more effect at high heat fluxes.Sun et al. [25]carried out experiments to measure the heat transfer coefficients in nucleate pool boiling using fluids HFC234a, HC290, HC600a and their binary and ternary mixtures at pressure 0.1 to 0.6 Mpa and heat flux ranged from 10 Kw/m^2 to 300 Kw/m^2. They applied different heat flux on different mixture concentrations. They found different heat transfer coefficients of binary mixtures and ternary mixtures according to their vapor-liquid phase equilibrium behaviors as compared to smooth surface.Fujita and Tsutsui [26]experimentally determined the heat transfer coefficients in nucleate pool boiling using fluids R-134a, R-142b, R-123 and their binary and ternary mixtures on the upward facing surface of 40 mm diameter. They observed that heat transfer coefficients of mixture were reduced in a comparison with ideal coefficients interpolated between individual refrigerants and also more reduction at higher heat flux.Fujita and Tsutsui [27]investigated the heat transfer coefficients in nucleate pool boiling using fluids R-134a, R-142b, R-123 and their binary and ternary mixtures on a smooth flat surface under the saturation condition at pressure 0.6 Mpa applying wide range of heat flux and mixture concentration. They observed that heat transfer coefficients of binary and ternary mixtures showed lower as compared to the ideal coefficients calculated from a mole fraction average of the wall superheat of pure component and also more reduction at higher heat flux.Jung et al. [28] carried experiments to measure the nucleate boiling heat transfer coefficients of binary and ternary mixture for HFC134a, HFC125 and HFC32 on horizontal tube of 19.0 mm outside diameter with decreasing heat flux from 80-10 kw/m^2 with an interval of 10 Kw/m^2 at pool temperature of $7^{O}C$.The heat transfer coefficients of non-isotropic mixtures of HFC32-HFC134a, HFC125-HFC134a and HFC32-HFC125-HFC134a showed a reduction of heat transfer coefficients as much as 40% from ideal value but the azeotropic mixture of HFC32- HFC125 did not performed reduction. Nahra and Naess [29], carried out an experimental work to determine heat transfer coefficients in nucleate pool boiling of binary and ternary non-azeotropic hydrocarbon mixtures using a vertical electrically heated cylindrical carbon steel surface at atmospheric pressure with several surface roughness. The fluids used were Methanol/1-Pentanol and Methanol/1-Pentanol/1,2-Propandiol at constant 1,2-Propandiol mole percent of 30%. Heat fluxes were varied ranged from 25 to 235 kW/m^2 . Comparison of his experimental data with that predicted from others correlations showed that the correlations available in literature based on the boiling range are in better qualitative agreement than correlations based on the phase envelope. He observed that increasing surface roughness resulted in an increase in the heat transfer coefficient, and the effect was observed to be dependent on the heat flux and fluid composition. Jungnickel et al. [30] studied heat transfer of nucleate pool boiling for the mixtures R-12/R-113, R-22/R-12, R-13/R-12, R13/R-22 and R-23/R-13 and also that of for the respective five pure refrigerants. Dependent upon the mixture, the measurements

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were made at boiling pressures of 0.1 to 2 MPa within the temperature region of 198 to 333 K and at heat fluxes of 4 to 100 kW/m^2. A horizontal, electronically heated copper plate of 3 cm^2 was used. The following quantities were measured: pressure; temperature difference between the heating surface and the boiling liquid; composition and temperature in the liquid and vapor phases; and heat flow rate. They observed deterioration in the heat transfer coefficient for an evaporating mixture as compared to the pure components.Shen et al. [31] conducted experiment to determine the nucleate pool boiling heat transfer for refrigerant mixture R32-R125 in a wide range of pressure and heat flux at saturation conditions using a horizontal platinum wire (d = 0.1 mm). The platinum wire used as both heating element and resistance thermometer. The results indicated that the pressure and the heat flux dependence of the heat transfer coefficient for the R32-R125 mixture does not differ from those of pure components.Inoue et al. [32] conducted experiments on a horizontal platinum wire during nucleate pool boiling in non-azeotropic binary mixtures of R12-R113, R134a-R113, R22-R113 and R22-R11 at pressure range of 0.25 to 0.7 MPa with heat fluxes up to critical heat flux. They photographically studied the features of boiling phenomenon occurring during boiling of the mixtures and the pure substances. They observed deterioration in the heat transfer coefficient of mixtures as compared to the pure components.Koster et al [33] conducted experiments to obtain the heat transfer cofficients for nucleate pool boiling of the binary and ternary refrigerant mixtures R404A, R407C and R507 on a horizontal tube with emery ground surface for wide range of pressures and heat fluxes. The values from result were used to comparative study of the influence of heat flux on the heat transfer coefficient as predicted by various correlations of nucleate boiling of mixtures. At comparatively high saturation pressures with experimental heat transfer values are smaller than the molar average of the pure components. They also observed that the predicted values of heat flux and heat transfer coefficient by various relationships significantly differ from the experimental values.

REFRIGERANT WITH OIL MIXTURES

Kedzierski [34] Investigated the effect of bulk lubricant concentration on the non-adiabatic lubricant excess surface density on a roughened, horizontal flat pool-boiling surface. The lubricant excess surface density measurements were used to modify an existing dimensionless excess surface density parameter so that it is valid for different reduced pressures. Here both the excess measurements and heat transfer data are provided for pure R-134a and three R-134a/lubricant mixtures at 277.6K. The lubricant excess layer causes an average enhancement of the heat flux of approximately 24% for 0.5% lubricant mass fraction mixture relative to pure R-134a heat fluxes between 5 and 20 kw/m^2. Both 1 and 2% lubricant mass fraction mixtures experienced an average degradation of approximately 60% in the heat flux relative to pure R-134a heat fluxes between approximately 4 and 20 kw/m^2. Kedzierski [35] investigated the effect of an additive on boiling performance of R134a/polyolester lubricant (POE) mixture and R-123/naphthenic mineral oil mixture on a roughened, horizontal flat surface. For R134a/POE mixture, the pool boiling heat transfer data and lubricant excess surface density data are given before and after use of the additive. The additive causes an average and maximum enhancement of the R134a/POE heat flux between 5 Kw/m^2 and 22 Kw/m^2 of 73% and 95%, respectively. For nearly the same range, the additive caused no change in the pool boiling heat flux of R123/mineral oil mixture. The key parameters of lubricant excess surface density and interfacial surface tensions were used to form the basis of a hypothesis for predicting, whether the additives will enhance or degrade refrigerant/lubricant pool boiling.Ji et al. [36]studiedpool boiling heat transfer coefficients of R134a with polyvinyl ether lubricant of mass fractions 0.25%, 0.5%, 1.0%, 2.0%, 3.0%, 5.0%, 7.0%, and 10.0%, respectively on one smooth tube, one integral and four enhanced tubes at a saturation temperature of 6°C. The heat flux range was from 9000 W/m^2 to 90,000 W/m^2. They observed first, the boiling heat transfer coefficient of the smooth tube is significantly less than the integral and four enhanced tubes and the differences decrease with an increase in the heat flux. Second, the heat transfer coefficients vary with the heat flux linearly in log-log coordinates for both smooth and integral-fin tubes but for the four enhanced tubes the *h-q* curves are more or less bent in the high heat flux region. Third, the four enhanced tubes behaved in different nature and one of the enhanced tubes was the worst performance. The pool boiling heat transfer coefficients for plain tube were increased at 3% of lubricant beyond that drastically reduced. The pool boiling heat transfer coefficients for the plain tube, integral-fin tube, and one enhanced tube within oil mass fraction range of 0 to 1% were reduced a bit and these were appreciably increased with further increase in the lubricant mass fraction. The oil mass fraction of 5% and further always reduce the pool boiling heat transfer coefficient significantly.Kim et al[37]investigated the effect of enhanced geometry (pore diameter, gap width) on convective boiling of R-123/oil mixture on three enhanced tube bundles as well as a smooth tube bundle with oil concentration and observed significant heat transfer degradation for the present enhanced tubes. Compared with the pool boiling counterpart, however, the heat transfer degradation is much smaller.Kedzierski [38] predicted the pool boiling heat transfer of refrigerant/lubricant mixtures on a roughened, horizontal, flat boiling surface. The excess layer forming on the heating surface influences the boiling performance, i.e., the excess layer formation will enhances or degrades the heat transfer. Here the predictive model was purely based on the mechanisms involved in the formation of the lubricant excess layer that exists on the heat transfer surface. In order to generalize the model to other refrigerant/lubricant mixtures, a dimensionless excess layer parameter and a thermal boundary layer constant were fitted to excess surface density and heat transfer data. For predicting this model the transport properties, thermodynamic properties, lubricant composition, viscosity and critical solution temperature with the refrigerant are required as input. This model predicts the boiling heat transfer coefficient of three different mixtures of R-123 and lubricant to within ±10%. Memory et al. [39] investigated the measurements of pool boiling heat transfer coefficients in pure R114 and R114-oil mixtures for a smooth tube and eight enhanced tubes (five finned and three re-entrant cavity). Experiments were conducted at atmosphere pressure while decreasing the heat flux. For pure R114, the finned tubes provide heat transfer enhancement (due to improved bubble dynamics within the channels), while the re-entrant cavity tubes provide even better heat transfer enhancements. For R114-oil mixtures, performance of the finned tubes at first increases (up to 3% oil), further increase in oil leads to a steady decrease in performance. For the re-entrant cavity tubes, any addition of oil leads to drop-off in performance. This is especially significant for the porous coated tube at high heat fluxes high oil concentrations (up to 10%).

REFRIGERANT WITH NANOPARTICLES

Visineeand Somchai [40] investigated nucleate pool boiling heat transfer of a refrigerant-based-nanofluid at different TiO_2 nanoparticle concentrations and pressures. Nanoparticles were mixed with the refrigerant HCFC 141b at 0.01, 0.03 and 0.05 vol%. They conducted the experiment within the pressure range of 200-500Kpa. The experiment was performed using a cylindrical copper tube as a boiling surface. They observed that the heat transfer coefficient increases with increasing heat flux for both pure refrigerant and nanofluid and also observed, the effect of pressure on the heat transfer coefficients at higher heat flux, i.e., the heat transfer coefficients were much higher for a higher heat flux than for a lower heat flux. The results indicated that the nucleate pool boiling heat transfer deteriorated with increasing particle concentrations. The heat transfer coefficients were deteriorated at 0.05 vol% TiO_2–R141b nanofluid compared to that of pure refrigerant. Kedzierski and Gong [41] quantified the influence of Cuo nanoparticles on the boiling performance of R134a/polyolester mixtures on a roughened, horizontal, flat surface. Here a lubricant based nanofluid was made with a synthetic ester and Cuo particles. At 0.5% nano lubricant mass fraction, the nano particles caused a heat transfer enhancement relative to the heat transfer of pure R134a/polyester (99.5/0.5) of between 50% and 275%. Also a smaller enhancement takes place for the R134a/nanolubricant (99/1) mixture, which had a heat flux that was on average 19% larger than that of the R134a/polyolester (99/1) mixture. For the 2% increment in nanolubricant mass fraction, a smaller boiling heat transfer improvement of approximately 12% has been observed. From the results, although the nanoparticles increased the thermal conductivity of the lubricant, the increase in thermal conductivity responsible for only a small portion (potentially 20%) of the boiling heat transfer enhancement. Kedzierski[42] investigated the effect of Al_2O_3 nanoparticles on pool boiling performance of R134a/polyolester mixtures on a roughened, horizontal, flat surface. Here a nano-lubricant containing nanoparticles (diameter 10 nm) at 1.6% volume fraction with a polyolester lubricant was mixed with R134a at three different mass fractions. The heat flux enhancement for all of the mixtures increased with respect to decreasing heat flux. The average heat flux improvement for heat fluxes less than 40 kw/m^2 was approximately 105%, 49%, and 155% for the 0.5%, 1%, and 2% mass fractions respectively. Also a semi-empirical model was developed to predict the enhancement of refrigerant/lubricant pool boiling caused by nanoparticles. Here it was assumed that the transfer of momentum from the nanoparticles to the bubbles is responsible for the boiling heat transfer enhancement. The model predicts that the maximum performance was approached for volume fraction and mass fractions nearing unity, and suggests small particle size and large nanoparticle volume fraction improve boiling enhancement.

RECENT RESEARCH ON NANOCOATING SURFACE

Stutz et al. [43] investigated the effect of nanostructured surface coatings on boiling heat transfer and critical heat flux (CHF). The coated surface of nanostructured was prepared by deposition of charged γ-Fe_2O_3nanoparticles (average diameter 10nm) on the platinum wire. The nanostructured surface was characterized by using Scanning Electron Microscopy(SEM) and Atomic Force Microscopy(AFM). The deposition of nanoparticles onto the heated surface induced a significant increase of the boiling critical heat flux (CHF) and wettability. It also indusedto reduce the heat transfer coefficient, when the wire was fully covered with nanoparticles. Kwark et al. [44] carried out an experimental work to determone the pool boiling behavior of nanoparticle coated (Al_2O_3) surfaces in pure water. The nanocoating was developed during the boiling of nanofluid (Al_2O_3-Water/Ethanol). Comparing to the coating created in water nanofluid the coatings created by ethanol nanofluid were uniform. The SEM pictures revealed that nanoparticle deposition is a function of heating duration i.e. longer duration implies more nanoparticle deposition. The effect of different nanocoating on surface wettability was found by measuring the contact angles between the droplets of water and different nanocoated surfaces. They observed that for the water-based nanofluids, the contact angles decrease with increase of the nanocoating thickness and stabilizes at 15-20°. Hedge et al. [45] studied the heat transfer characteristics using low concentrations (0.1-0.5 g/l) of Alumina-nanofluid at atmospheric pressure in distilled water, the effect of nanoparticle coating on vertical test surface exposed to multiple heating cycles, heat transfer characteristics of nanoparticle coated surface in distilled water and pool boiling behavior of Alumina nanofluid subjected to transient characteristics. There is deterioration in boiling HTC with increased nano-particle concentration. A porous layer of nanoparticles was revealed by the SEM images. The surface roughness values can were used in the analysis were given by the SEM. The nanoparticle coated heater, when tested in pure water showed significant enhancement in CHF as compared to CHF of bare heater. Tang et al. [46] studied the nucleate pool boiling heat transfer performance of a nanoporous copper surface which was fabricated by the facile hot-dip galvanizing/dealloying (HDGD) process with saturated deionized water. They observed that there was a reduction of wall superheat and improvement of heat transfer coefficient (HTC) for nanostructured surface compared to unstructured surface particularly at low heat fluxes. With the increasing heat flux, the heat transfer on both structures (plain and naoporous surfaces) were in stable stage of nucleate boiling and the number of bubbles generated on the nanostructured surface. They also observed that the bubbles were converted into larger size and then collapse into the vapor film. Ray et al. [47] experimentally investigated a Physical Vapor

Deposition method to fabricate the Titanium Dioxide (TiO_2) nanowire arrays on copper surfaces. The investigation was carried out with R-134a on nanowire arrays surface at 6^0C saturation temperature and found heat transfer augmentation.

CONCLUSION

Many researchers carried out experiments to explain the heat transfer enhancement techniques by the surface roughness with different finish condition, enhanced surface, surface coating (like plasma spraying, flame spraying, pitted coating etc.) and nanoparticles & nanolubricant with the refrigerant. Many authors proposed the empirical correlations to find out the heat transfer coefficient in terms of surface roughness factor, vapour density, liquid density,etc. from their experimental results. However, modifying the surfaces by nanocoating process to augment boiling heat transfer and critical heat flux is a new concept. Recently, researchers are searching to obtain better pool boiling heat transfer performances on nanocoated surfaces using refrigerant, as the pool boiling process occurs in flooded evaporators in refrigeration industries.

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