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Influence of Lubricant on the Nucleate Boiling Heat Transfer Performance of Refrigerant—A Review

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This review provides an overview of the lubricant on the heat transfer performance pertaining to nucleate boiling. It appears that the effect of individual parameter on the heat transfer coefficient may be different from study to study. This is associated with the complex nature of lubricant and some compound effects accompanying the heat transport process. Some important effects such as oil concentration, heat flux, geometric configuration, saturation temperature, thermodynamic and transport properties, miscibility, foaming, and additional surface active agent are reported and discussed. In general, the heat transfer coefficient is impaired provided the oil concentration is sufficient high (e.g., >7%), and this is applicable to smooth and structured tubes. But normally structured tubes tend to suffer more from lubricant, yet this is especially conspicuous when the size of the reentrant channel is small. On the other hand, foaming and partial miscibility seem to benefit the heat transfer coefficient.

INTRODUCTION

For vapor compression processes, the role of lubricant oil in a refrigerating or heat pump system is very much essential for lubricating and sealing the sliding parts, and the lubricant also takes part in cushioning cylinder valves. In addition, oil also acts as a coolant in a compressor, transferring heat from bearings and mechanical elements to the crankcase sump, and relieves the noise generated from the moving parts. However, in typical operation of an air-conditioning or refrigeration system, a small amount of lubricant oil may migrate from the compressor into another part of the system, such as the evaporator,

condenser, expansion device, and connecting piping, thereby inevitably altering the heat transfer and frictional characteristics of the refrigerant.

The presence of lubricant alters the physical properties of refrigerant mixtures. This is especially pronounced for viscosity, since the viscosity of lubricant oil is about two to three orders higher than that of refrigerant, whereas the corresponding surface tension of lubricant is approximately one order higher. Hence the presence of lubricant oil would considerably affect the thermodynamic and transport properties of refrigerant, casting a significant impact on the heat transfer characteristics. Figure 1 is a schematic of the associated properties of R-410A/POE VG68 mixture based on the calculated results of Wei et al. [1] with oil concentration ranging from 0 to 30%. Normally, only a very slightly drop of mixture density subject to the rise of lubricant concentration is seen, followed by a moderate decrease of specific enthalpy. It should be mentioned that the density of the lubricant oil can be lower than, equal to, or higher than that of refrigerant. In the meantime, a detectable rise of surface tension and a sharp rise of viscosity are encountered.

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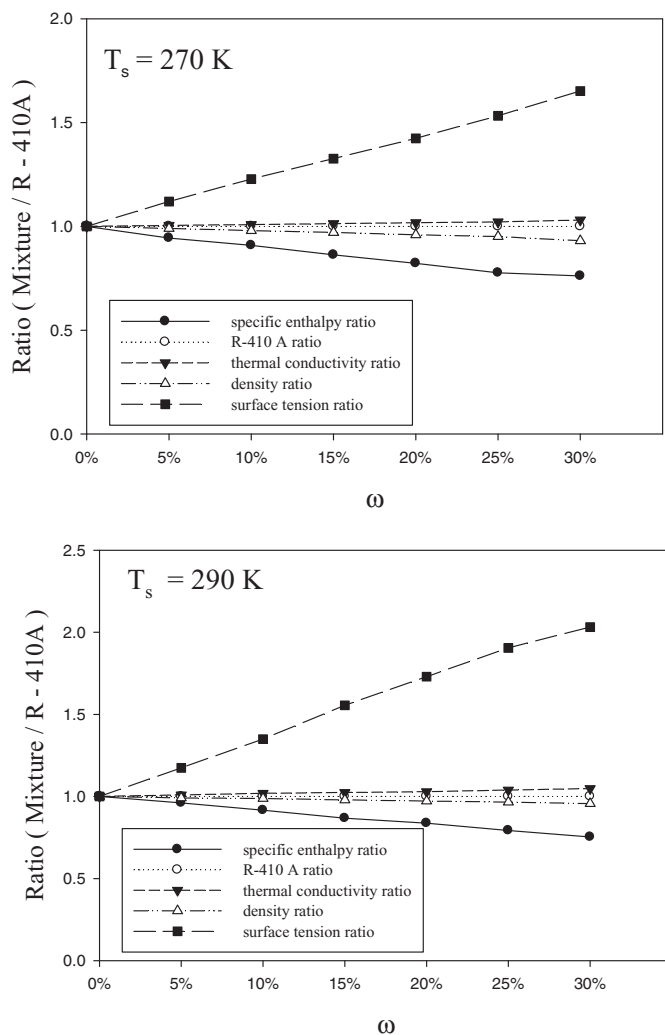


Figure 1 Schematic of the associated properties of R-410A/POE VG68 mixture based on the calculated results of Wei et al. [1] (a) $T_s = 270$ K; and (b) $T_s = 290$ K.

Background of Lubricant Oil

According to ASHRAE [2] classification, the mineral oils can be subdivided into paraffins, naphthenics, aromatics, and nonhydrocarbons. Paraffins consist of all straight-chain and branched-carbon-chain saturated hydrocarbons. Isopentane and *n*-pentane are typical examples of paraffinic hydrocarbons. Naphthenes are also completely saturated but consist of cyclic or ring structure; cyclopentane is a typical example. These two are widely used in the refrigeration, heat pump, and air-conditioning industry, and typical refrigerants like chlorofluorocarbon (CFC)-12, hydrochlorofluorocarbon (HCFC)-22, CFC-502, ammonia, propane, and isobutene had used these types of mineral oils. Aromatics are unsaturated cyclic hydrocarbons containing one or more rings characterized by alternate double bonds (benzene is a typical example). Nonhydrocarbon molecules contain atoms such as sulfur, nitrogen, or oxygen in addition to carbon and hydrogen.

The synthetic oils that had been introduced for the widely used mineral oils are not miscible, and the hydrofluorocarbons (HFCs) were proposed as substitutes for the CFCs. The synthetic lubricant oils are classified as polyol ester (POE), polyalkylene glycol (PAG), alkyl benzene (AB), and poly alpha olefin (PAO). Another distinct and very important characteristic of this kind of synthetic oil (ester oil), when compared to mineral ones, is its capacity to absorb moisture (hygroscopicity), which is normally several orders higher than mineral oil and must be carefully manipulated during operation [3]. This peculiar characteristic is important since a high moisture concentration can cause hydrolysis, transforming the ester oil into an acid. It is quite common to include additives in a lubricant to improve its characteristics. Additive types include (1) pour-point depressants for mineral oil, (2) floc-point depressants for mineral oil, (3) viscosity index improvers for mineral oils, (4) thermal stability improvers, (5) extreme pressure and antiwear additives, (6) rust inhibitors, (7) antifoam agents, (8) metal deactivators, (9) dispersants, and (10) oxidation inhibitors [2]. Some additives provide performance advantages in one area but could raise other problems in another. Apparently, the presence of these additives further complicates the heat transfer performance of the lubricant oils.

There had been many reviews concerning the influences of lubricant oils on the heat transfer characteristics of refrigerant, for instance, references [3]–[5]; some general behaviors of the lubricants were reported and some controversies still exist. The present study is to update and to summarize information about the lubricant in terms of the basic phase change heat transfer characteristics and nucleate boiling. On the other hand, some more rational explanations will be made to clarify some inconsistencies.

OVERVIEW OF THE EFFECT OF LUBRICANT ON THE NUCLEATE BOILING

Effect of Lubricant Concentration

Experimental data reporting the influence of lubricant on the nucleate boiling heat transfer subject since 1980 are tabulated in Table 1, which contains a total of 34 literature reports [6–39]. The associated researchers, surface, operation condition, lubricant concentration, heat flux, and major findings are also tabulated. A first glimpse of this table implies that the presence of oils may increase or impair the heat transfer performance, depending on a variety of conditions. For instance, even for a low oil concentration of 2%, Kim and Min [20] reported a 21% degradation in their smooth tube data. Many investigators reported a consistent decrease of heat transfer coefficient with oil concentration (e.g., Bell et al. [6], Chongrungrong and Sauer [10], Webb and McQuade [35]). Conversely, some investigators observed an increase of heat transfer coefficients at low oil concentration and decreases at higher oil concentration (e.g., Stephan and Mitrovic [29, 30], Wanniarachchi et al. [33, 34], Memory et al. [23,

Table 1 Experimental studies associated with the influence of lubricant on the nucleate boiling performances

Articles	Refrigerant and lubricant	Tested surfaces	Saturation temperature or pressure	Heat flux, kW/m ²	Oil concentration, ω	Primary qualitative observations
Bell et al. [6]; (1987)	R-113/Shell Clavus 68 oil	Smooth tube; low-finned tube	1 atm	5–80	0–10%	HTC decreases with all ω .
Bergles et al. [7]; (1994)	R-113/Suniso 3GS	Smooth tube Four enhanced tubes	103 kPa	0.4–50	0–10%	HTC decreases with all ω . High flux tube shows very huge degradation (40%) even at a concentration of 1%.
Chyu et al. [8]; (2009)	Ammonia/copolymer of polyalkylene glycol (PAG)	Enhanced tubing particularly for shell-side boiling of ammonia	–13.2 to 7.2°C	3.2–32	0–10%	HTC decreases with all ω . Most data in the bottom row are lower than the single-tube heat transfer coefficient data at a low saturation temperature, while at a higher temperature, the single-tube data are lower than or close to that of the bottom row.
Cochin and Boyd [9]; (1992)	R-113/Shell Clavus-68 oil	Smooth single tube; tube bundles	1 atm	5–20	0–20%	HTC decreases with ω for single tube. HTC is slightly increased for tube bundle.
Chongrueang and Sauer [10]; (1980)	R-12/3GS, 4GS, 5GS oil	Copper tube	1–3 atm	11–110	0–10%	HTC decreases with all ω .
Grant et al. [11]; (1989)	R-113/Shell Clavus 68 oil	Smooth and high-flux tube	N/A	3–80	0–10%	HTC decreases with all ω .
Hsieh and Weng [12]; (1997)	Water, R-134a/Emkarate RL Oil	Smooth tube; four rib-roughed tubes	1 atm for water; 343.06 kPa for R-134a	0.8–50	0–10%	HTC decreases with all ω .
Jensen and Jackman [13]; (1984)	R-113/four mineral oils	Smooth tube	47.7°C	10–100	0–10%	HTC decreases with all ω . Diffusion is the key for HTC reduction.
Ji et al. [14]; (2010)	R-134a/PVE oils	Smooth tube; five enhanced tubes	6°C	9–90	0–10	Smooth tube, integral tube: HTC is increased when ω is below certain value. Most highly enhanced tube: HTC continuously decrease with ω .
Kedzierski [15]; (1999)	R-123/N-hexane	Enhanced surface (GEWA-TTM)	4.4°C	10–80	0–2%	<i>n</i> -Hexane reduces the size of bubble and increase the site density and a rise of HTC.
Kedzierski [16]; (2000)	R-123/five hydrocarbons	Enhanced surface (GEWA-T)	4.4°C	10–80	0–1%	A maximum 19% increase was achieved with the addition of 0.5% mass isopentane. Other mixtures of isopentane, pentane, hexane, and cyclohexane exhibited smaller maximums than isopentane.
Kedzierski [17]; (2001)	R-134a/four oils Enhanced surface (Turbo-BIITM-HP)		4.4°C	10–160	0.5–2%	The lubricant that had the largest CST with R134a exhibited the greatest heat transfer.
Kedzierski [18]; (2002)	R-123/naphthenic mineral oil	Enhanced surface (plain roughened)	4.4°C	25–45	0.5–1.8%	HTC increases with an average enhancement of the heat flux of 9 and 5% was achieved for the 0.5 and 1% lubricant mass fractions, respectively, and an average degradation of 5% in the heat flux was obtained for the 1.8% lubricant mass fraction mixture
Kedzierski [19]; (2007)	R-134a/POE; R-123/mineral oil	Roughened plate (plain roughened)	4.4°C	25–45	0.5–1.8%	R-134a/POE: HTC is increased within $5 < q < 22$ kW/m ² R-123/mineral oil: HTC remains unchanged (Continued on next page)

Table 1 Experimental studies associated with the influence of lubricant on the nucleate boiling performances (*Continued*)

Articles	Refrigerant and lubricant	Tested surfaces	Saturation temperature or pressure	Heat flux, kW/m ²	Oil concentration, ω	Primary qualitative observations
Kim and Min [20]; (2010)	R-123/mineral oil	Smooth tube and three specially made enhanced tubes	4.4°C and 26.7°C	1–40	0–5%	HTC decreases with all ω , and significant decrease is seen even at a very small ω . The highest heat transfer coefficient with oil is obtained for pore diameter = 0.23 mm tube
Marvillet and Cavahlo [21]; (1994)	R-113/3GS oil	Planar confined space	1 atm	10–200	1–7%	The oil has a relatively minor effect on low flux nucleate boiling, and it caused a serious degradation of the high flux boiling performance. This deterioration increased with increasing oil concentration and was more severe for smaller gap sizes.
Memory et al. [22]; (1993)	R-124/Alkylbenzene oil	Smooth tube; Four enhanced tubes	2.2°C	1–100	0–10%	HTC increased for smooth/integral tube but decreases with for high flux/turbo B tubes.
Memory et al. [23]; (1995)	R-114/a mineral oil	Smooth tube; five finned tubes; three reentrant cavity tubes	2.2°C	1–100	0–10%	HTC of the finned tubes at first increases (with 3% oil) before dropping off. For structured tubes, a steady drop-off in performance with ω , especially for the porous tube.
Memory et al. [24]; (1995)	R-114/a mineral Oil	Smooth tube bundles; three enhanced tube bundles	2.2°C	1–100	0–10%	HTC for the smooth and finned tube bundles at first increases before dropping off slightly. For the structured and porous bundles, HTC shows a steady decrease, especially for the porous bundle at high heat fluxes.
Moeykens et al. [25]; (1995)	R-134a/two POE Oils	Smooth tube; six enhanced tubes	314.5 kPa	5–40	0–5%	HTC increases for all tubing, mainly because of the effect of foaming.
Moeykens and Pate [26]; (1996)	R-134a/340SUS POE; R-22/300SUS mineral oil	Smooth tube bundles; three enhanced tube bundles	314.5 kPa	20–40	0–2.5%	HTC increases for all tubing, mainly because of the effect of foaming.
Mohrlok et al. [27]; (2001)	R-507/a POE oil	Smooth tube; Enhanced tube (GEWA-B)	–28.6°C –20.1°C	1–80	0–10%	Smooth tube: HTC is increased up to $\omega = 3\%$ at low T_s . At high T_s , HTC decreases with ω for both tubes. HTC decreases with all ω
Monde and Hahne [28]; (1987)	R-11, R-115/a naphthene-basis oil and a synthetic oil	Platinum round wire	1–24 bar	1.6–180	0–35%	
Stephan and Mitrovic [29]; (1981)	R-12/Shell Clavus G68	Smooth and GEWA-T	–20°C–0°C	2.08–21.75	0–10%	At low heat flux, HTC for lubricant is lower than pure refrigerant which is applicable to all ω . But the HTC is increased by raising heat flux and HTC is slightly higher than pure refrigerant at a small range of ω . At low heat flux, HTC for lubricant is lower than pure refrigerant which is applicable to all ω . But the HTC is increased by raising heat flux and HTC is slightly higher than pure refrigerant at a small range of ω .
Stephan and Mitrovic [30]; (1982)	R-12/Shell Clavus G68	Smooth and GEWA-T	–20°C–0°C	2.08–21.75	0–10%	For the sandblasted tube and the SE55 oil the heat transfer coefficients for the refrigerant/oil-mixture can be higher or lower than those for the pure refrigerant, depending on oil mass fraction, boiling temperature, and heat flux. In some cases the highest heat transfer coefficients were obtained at a mass fraction of 3%. For the 170 cSt oil there is a clear decrease in heat transfer for all variations except for a heat flux 4,000 W/m ² and –10.1°C at 0.5% oil content.
Spindler and Hahne [31]; (2009)	R-134a/polyol ester oils (POE) with medium viscosity 55 cSt (SE55) and high viscosity 170 cSt (SE170) were used	Smooth tube (sandblasted) and enhanced tube (GEWA-B tube).	–28.6°C – +20.1°C.	4–20	0–5%	

Wang et al. [32]; (1999)	R-22/3GS and 5GS oil	Smooth tube	-5°C-20°C	16.7-84	0-7%	HTC decrease with increase of oil concentrations when $T_s = 20^\circ\text{C}$. But for $T_s = -5^\circ\text{C}$, HTC may be larger at a lower concentration.
Wanniarachchi et al. [33]; (1986)	R-114/a mineral oil	Smooth and porous coated surface	-2.2°C and 6°C	0.5-95	0-10%	HTC decreases with all ω . Significant decreases especially when ω and heat flux is high.
Wanniarachchi et al. [34]; (1987)	R-114/a mineral oil	Smooth and four enhanced tubes	2.2°C	0.5-91	0-10%	Oil delays onset of nucleate boiling. HTC decreases with all ω .
Webb and McQuade [35]; (1993)	R-11 and R-123/mineral oils	Smooth tube; two enhanced tubes	4.4°C	30	0-5%	Larger reentrant tunnel gives less degradation. HTC decreases with all ω .
Zarnescu et al. [36]; (2000)	R-134a/a POE oil	Four structured surfaces; one porous surface	26.6°C	2-90	0-5%	HTC normally decreases with ω , but pore diameter is not the major cause for degradation. HTC degrades more when the tunnel cross section area increases.
Zheng et al. [37]; (2001)	Ammonia/copolymer of polyalkylene glycol (PAG)	Smooth tube	-23.3°C to 7.2°C	10-60	0-10%	HTC in general is decreased by adding lubricant. The saturation temperature plays a critical role while heat flux only plays minor role.
Zheng et al. [38]; (2008)	Ammonia/copolymer of polyalkylene glycol (PAG)	Enhanced tubing particularly for shell-side boiling of ammonia	-23.3°C to 7.2°C	10-65	0-10%	HTC in general is decreased by adding lubricant. But HTC does not necessarily decrease as the lubricant concentration increases.
Zhu et al. [39]; (2011)	R113/VG68	Open cell metal foam	101 kPa	10-80	0-5%	The nucleate pool boiling heat transfer coefficient on copper foam covers is larger than that on flat heated surface by a maximum of 160% under the present experimental conditions; the presence of oil deteriorates the nucleate pool boiling heat transfer on copper foam covers by a maximum of 15% under the present experimental conditions, and the deterioration of oil on nucleate pool boiling heat transfer on copper foam covers is lower than that on a flat heated surface.

24]). Even more interestingly, Zheng et al. [38] conducted boiling of ammonia/lubricant mixture on a flooded evaporator with inlet quality. For instance at a saturation temperature of 7.2°C, they found the heat transfer coefficient (HTC) is decreased by adding lubricant up to 5% concentration, and there is a significant increase of HTC from 5% to 10%, but the HTC is still inferior to pure refrigerant. However, this phenomenon is not seen at an elevated saturation temperature of 23.3°C. A similar trend is shown for an inlet quality (x_{in}) of 0.2 and for $x_{in} = 0.4$ at $T_s = -9.4^\circ\text{C}$. The authors did not provide explanations of this phenomenon. One possible explanation about this unusual characteristic is associated with foaming. We defer this possible reasoning in later discussion. Wang et al. [32] investigated the effect of lubricant oil (3GS) on the pool boiling heat transfer performance for a plain tube at saturation temperatures of 20°C, 4.4°C, and -5°C for oil concentrations of 0.75, 1.5, 3.6, and 7%. For $T_s = 20^\circ\text{C}$, the heat transfer coefficient is decreased with increase of oil concentration. For $q = 21.1 \text{ kW/m}^2$, the heat transfer coefficient for $\omega = 3.6\%$ is about 55% that of the pure refrigerant, while for $\omega = 7\%$, the heat transfer coefficient is only 45% of that of the pure refrigerant. For a lower saturation temperature of 4.4°C, the effect of lubricant on the heat transfer coefficient becomes less profound. There is no significant decrease of heat transfer coefficients for $\omega < 2\%$. For a saturation temperature of -5°C , the effect of lubricant oil on the heat transfer coefficient is reversed. The heat transfer coefficients with oils are higher than those of pure refrigerants over the range of $\omega = 0\text{--}3\%$. A maximum increase of 20–30% of heat transfer coefficient is observed near $\omega = 1.5\%$.

In summary of the foregoing results, the effect of concentration on HTC is very complicated. This is because additional parameters may be statically or dynamically incorporated with the oil concentration, for example, foaming, temperature, heat flux, viscosity, miscibility, and the like. In the following, the authors try to summarize the associated influences of lubricant on the pool boiling HTC.

Effect of Physical Property

Thome [40] concluded that flow boiling heat transfer coefficients subject to the influence of lubricant oil are affected by (1) an increase in the nucleate boiling contribution via favorable refrigerant–oil properties and/or foaming; (2) a decrease in convective contribution due to local increase of liquid viscosity; and (3) an adverse effect of mass transfer on the evaporation process. For nucleate boiling of the refrigerant–oil mixtures, the overall heat transfer performance is a complicated combination between (1) and (3). Among the most influential physical properties, surface tension and viscosity may be the most important. In an investigation of the surface tension of refrigerant–oil mixtures, Wallner and Dick [41] argued that the heat transfer coefficients for refrigerant–oil mixtures are strongly related to the interfacial effects. Based on the theoretical developments

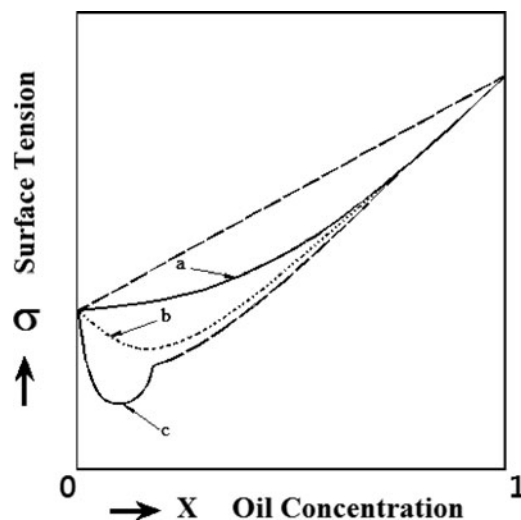


Figure 2 Possible relationship between surface tension and oil concentration. Type a represents when the surface tension vs. oil concentrations did not show a minimum. Type b denotes the oils that contain surface-active agent, which leads to minimum in surface tension. The oils for type c contain surface-active agent and additional dirt particles in small concentration, which leads to two points of inflexion and a sharp minimum of surface tension.

of Stephan and Körner [42], Wallner and Dick [41] proposed a simplified formula for the work required for the generation of vapor in the mixtures:

$$W = \frac{A\sigma^3}{[B - C(y-x)]^2} \quad (1)$$

For a weak interfacial interaction of refrigerant–oil mixtures, the bubble generation work is increasing with increasing $(y-x)$. Since refrigerant–oil is considered a zeotropic mixture that has a considerable mass transfer resistance, an appreciable reduction of heat transfer coefficient is often encountered. However, by adding a surface-tension-reducing substance, a reduction of the bubble generation work can be achieved as calculated from Eq. (1). This corresponds to an improvement of the heat transfer performance. As shown in Figure 2, the relation of surface tension of refrigerant–oil mixtures, as was classified by Wallner and Dick [41], has three types of characteristics. For type a, the surface tension versus oil concentrations for refrigerant–oil mixtures did not show a minimum. For type b, the oils contain a surface-active agent, which leads to minimum in surface tension. For type c, the oils contain surface active agent and additional dirt particles in low concentration, and this leads to two points of inflexion and a sharp minimum of surface tension. Wallner and Dick [41] argued that type c is responsible for an increase of heat transfer coefficients at lower oil concentrations. Based on the ideal mixture behavior and homogeneous oil distribution, Mitrovic [43] derived an expression that connects the temperature, oil concentration, and the size of the bubble in equilibrium with the mixture:

$$\frac{h}{h_o} = \frac{-\sigma_o r}{\sigma r_o} \frac{x}{\ln(1-x)} \quad (2)$$

where h and r denote the heat transfer coefficient and equilibrium bubble radius, respectively. The subscript o refers to the state of lubricant mixtures. By using this expression, it is shown that when oil is added to a refrigerant it can facilitate the bubble formation and improve the heat transfer. For this to occur, the oil must contain some surface-active components.

For refrigerant–oil mixtures, a higher wall temperature is required to accommodate the same nucleation site density as compared to the pure refrigerant. This leads to a decrease of HTC at a constant heat flux condition. Via addition of oil, the surface tension of the refrigerant/oil mixtures rises and so does the wall temperature, resulting in a lower surface tension near the heating surface. Therefore there is a greater gradient of the surface tension around the bubble, which enhances the Marangoni convection.

The lubricant viscosity is considerably higher than that of the refrigerant. A large lubricant viscosity can benefit HTC by promoting a thicker thermal boundary layer [17]. A thicker thermal boundary layer provides a larger active site density [44], which improves the nucleate boiling performance.

Effect of Blocking

During the evaporation process, refrigerant is regarded as the more volatile phase, resulting in an oil enrichment at the phase interface and next to the heating surface. Figure 3a depicts a schematic showing the excess oil rich layer accumulated around the growing bubble; this will cause a smaller bubble departure diameter and a higher temperature difference between the tube wall and the saturation temperature of the liquid [45]. In fact, the bubble detachment radius decreases by more than a factor of 3 as the oil fraction ω increases from 0 to 10% [43, 45], and the effect is much stronger at a higher boiling temperature. For a smaller bubble departure diameter and a higher viscosity of the oil, the contribution of convective heat transfer due to bubble motion is lessened. As a result, the oil jeopardizes the pool boiling heat transfer pertaining to the effect of blocking. Irrespective of the influence of blocking, Mitrovic [43] noted that the oil molecules are often much larger than that of refrigerant molecule, as illustrated in Figure 3b. Hence the complex shaped oil molecules along its larger size could manage themselves in the interfacial region such that a permeable oil layer builds up at the bubble surface, thus creating a new phase within the mixture as an interphase, depicted in Figure 3b. The properties of this interphase are neither those of the bulk mixture nor those of the pure oil.

Effect of Nucleation Site

The presence of lubricant oil may facilitate smaller bubble departure diameter, which can lead to poorer heat transfer [17]. On the other hand, the area influenced by one nucleation site will be reduced. The reduced area influenced by one nucleation

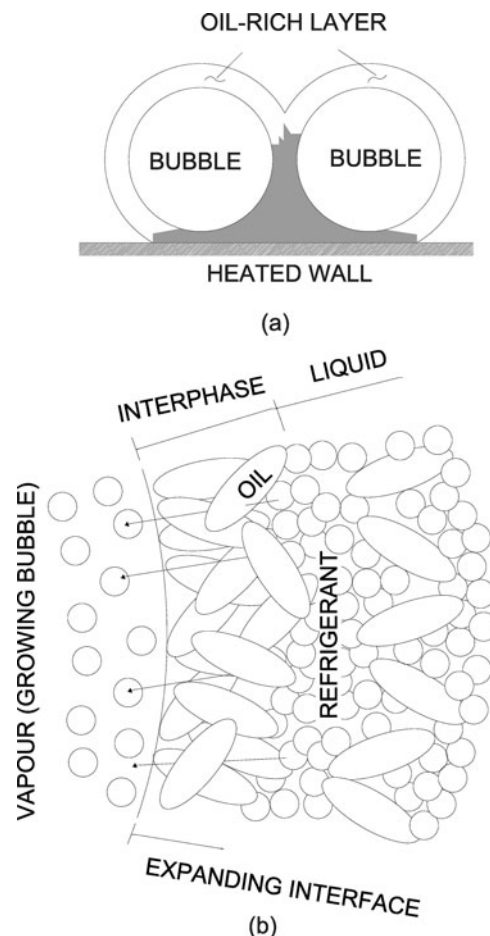


Figure 3 Schematic of the oil rich layer at the interface of bubbles expected to form under real boiling conditions, and the shaded area in (a) indicates regions with a largely hindered mass transfer. (b) A snapshot of the state at bubble surface channeling of refrigerant molecules through the interphase.

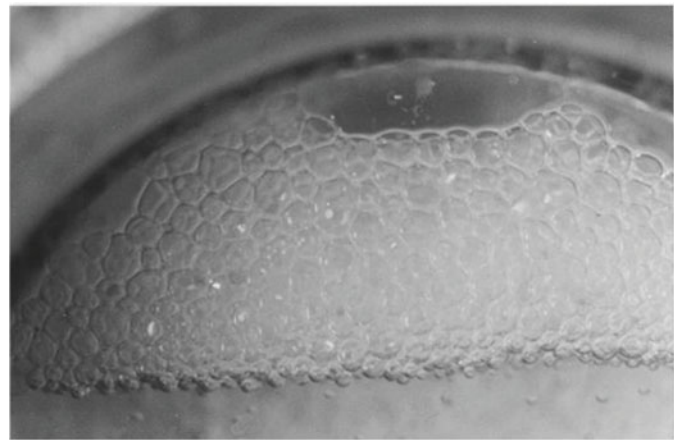
site can lead to a higher nucleation site density and an increase of the pool boiling heat transfer coefficient [17]. Hence an enhanced heat transfer is encountered if the rise of nucleation site surpasses the opposite effect smaller bubble departure diameter. However, based on Mikic and Rohsenow [46], Kedzierski [17] pointed out that the influence of the bubble size on the heat transfer is greater than that of the site density. Hence the HTC eventually degrades with increased lubricant concentration.

Effect of Foaming

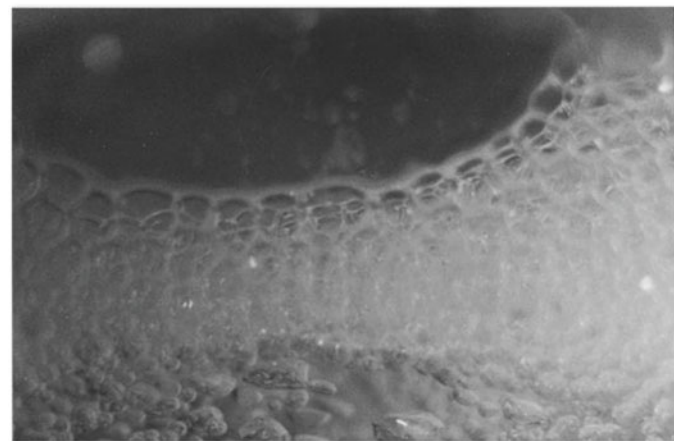
When oil is added to the refrigerant, an intensive foaming is produced as heat flux is increased. Several investigators postulated that the increase of heat transfer coefficients at low oil concentration is related to the foaming process (e.g., Stephan [47], Udombobesuwana and Mesler [48]). They put forth important efforts to interpret the effect of foaming. According to a review by Mohrlok et al. [27], several other possible mechanisms have significant impacts on the pool boiling heat transfer

performance, and some of their comments are summarized as follows.

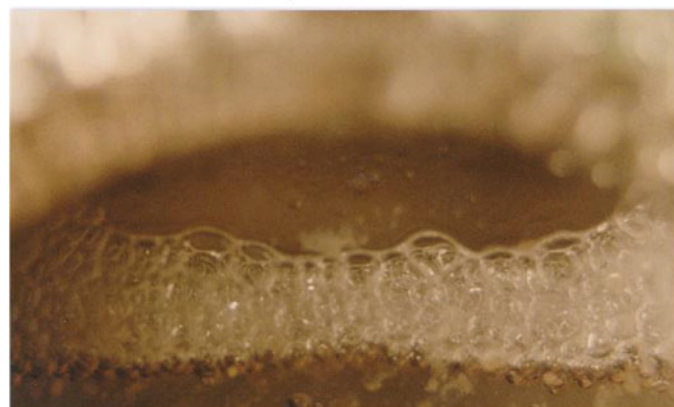
The presence of foaming gives rise to more phase interface area through which the latent heat of evaporation is transferred. For a growing bubble, the temperature of the phase interface is decreasing. Heat is transferred at lower temperature differences and with higher heat transfer coefficients. On the other hand, there is oil enrichment next to the heating surface and in the foam. The foam inhibits the flow of liquid refrigerant to the heating surface. This increases the local oil concentration. The foam can also increase the heat transfer depending on the geometry of the heater surface. The foaming becomes more pronounced with higher heat fluxes and higher oil concentrations. Several investigators (Memory et al. [23, 24], Udombobesuwana and Mesler [48]) also claimed that the effect of foaming for refrigerant–oil mixtures may significantly increase the heat transfer characteristics. Udombobesuwana and Mesler [48] reported significant enhancement in pool boiling heat transfer in the presence of foam. They assumed two possible enhancement effects caused by the foaming. These are (1) that a thin liquid film was created between the foam and the heated surface, which results in a very large heat transfer coefficient, and (2) that secondary nucleation is caused by the bubble leaving the surface, bursting into the neighboring liquid–vapor region. Wang et al. [32] performed a visualization of the boiling phenomenon to understand further details about the effect of foaming. Figure 4 shows a typical diagram for the foaming characteristics for 5GS oils at $T_s = -5^\circ\text{C}$, $\omega = 0.75\%$, and for $q = 84, 58.8, \text{ and } 16.7 \text{ kW/m}^2$. For the same oil concentration, the depth of the foaming increases when the saturation temperature is decreased. The size of the foaming is increased as the saturation temperature is decreased. This is probably related to its lower pressure, which produces a larger bubble size. The results imply that the distance between heater surface and the liquid/foam interface may be smaller as the saturation temperature is decreased. The observation substantiates the first conclusion by Udombobesuwana and Mesler [48]. It is also interesting to know that a further increase of heat flux may result in significant increase of the depth of the foaming. In addition, the size of the foaming gets finer as heat flux is increased. A close examination of the foaming shows that the size of the foaming can be roughly classified into coarse and fine. The coarse one is on top of the fine one. In summary of the foregoing observations, it is concluded that the effect of foaming are more evident at higher oil concentration and at a lower saturation temperature. The results may explain the sharp bounce of heat transfer coefficient of the Zheng et al. data [38] when oil concentration is raised from 5% to 10% at a saturation temperature of 7.2°C , whereas this phenomenon is not seen at a higher saturation temperature of 23.3°C . In essence, the foaming effect casts a positive role in augmentation of heat transfer. Hsieh and Weng [12] also postulated that foaming promotes secondary motion in pool boiling and is helpful in moving the oil out of the heated surface. The foregoing explanation about the effect of foaming may also be confirmed with the data of Stephan and Mitrovic [29, 30], who showed that the HTC is increased by raising heat flux



$$q = 100.8 \text{ kW/m}^2$$



$$q = 50.4 \text{ kW/m}^2$$



$$q = 33.4 \text{ kW/m}^2$$

Figure 4 Boiling pattern for $T_s = -5^\circ\text{C}$, and $\omega = 0.75\%$ for different heat fluxes [32]. (Color figure available online.)

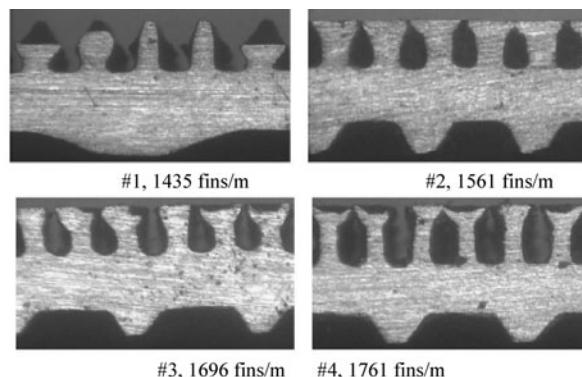
and HTC could be slightly higher than pure refrigerant at a small range of ω . Moeykens et al. [25] and Moeykens and Pate [26] examined the spray evaporation performance of R-134a/340-SUS (POE) and R-22/300-SUS (alkyl-benzene) on the single tube as well as on tube bundles. Foaming was observed due to the oil

dissolved in the refrigerant, and appreciable enhancements were observed.

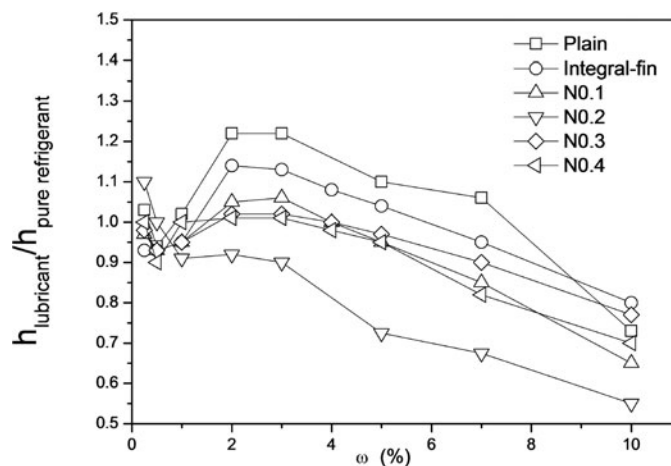
Effect of Geometry

Some test data for the standard tube geometries (smooth and low finned tubes) show that the heat transfer increases by adding oil to the refrigerant, and this is especially pronounced at low oil concentration. For enhanced tubes, it is reported that the oil plugs the microchannels of the surface structure for oil mass fractions greater than 5% and high heat fluxes. The pores of the heating surface inhibit the diffusion of the refrigerant oil mixture into the microchannels of the heating surface. This leads to a lower heat transfer performance. Though the presence of oil may block the pore and thus give rise to lower heat transfer performance, Zarnescu et al. [36] found that the pore diameter plays only a minor role in the deteriorating heat transfer performance. Instead, the tunnel cross-section area (reentrant channel) can significantly affect the heat transfer process when oil is added, and the effect of degradation increases as the tunnel cross-section area decreases.

Most studies depict that the presence of oil deteriorates nucleate boiling heat transfer performance more for a highly structured surface than for a smooth one. This can be made clear from the studies of Ji et al. [14], who examined R-134a/PVE lubricant for plain, integral fin, and four enhanced tubes (the geometry of the four enhanced tubes is shown in Figure 5a). Their test results, in terms of ratio relative to plain tube, are shown in Figure 5b. Apparently, a more structured surface reveals a more pronounced drop with lubricant concentration. Ji et al. [14] also compare some existing results concerning the effect of tube geometry as shown in Figure 6, where Figure 6a is for smooth and integral fin tube [7, 23, 27, 12, 31] whereas Figure 6b is for highly structured surfaces [7, 23, 27, 31, 36]. It appears that for plain/integral tubes a detectable enhancement is encountered at low oil concentrations (<5%), provided the viscosity of the lubricant is not so small. On the other hand, the highly structured surface all reveals a significant performance drop with oil concentration. Part of the explanation is associated with blockage of the reentrant tunnel by lubricant. However, a recent report by Zhu et al. [39], who performed nucleate boiling testing on metal foam with porosity ranging from 90% to 98% as well as flat plate using R-113/VG68, shows an opposite trend. Their results clearly indicate that the deterioration of metal foam pertaining to lubricant oil is less profound than that of the smooth plate. A possible explanation of this opposite trend may be attributed to the surface structure itself. The metal foam structure shown in Figure 7 is quite different from those of highly structured surfaces where adjacent reentrant channels are normally separated, as shown in Figure 5a. By contrast, the open cell structure of the metal foam suggests a continuous reentrant channel amid the metal foam surface. In this regard, the lubricant oil casts less influence on blocking the pore surfaces of the metal foam. Therefore, the blockage of the reentrant channel is less pro-



(a) Schematic of the enhanced tube structure by Ji et al. [14]



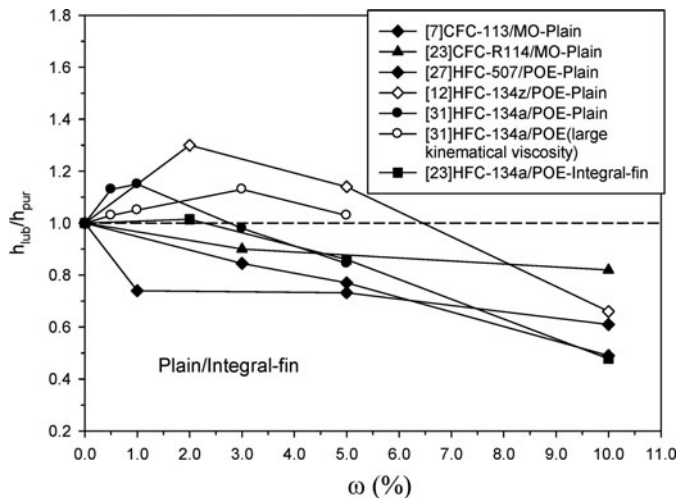
(b) Variation of the relative boiling heat transfer coefficient of R-134a/lubricant.

Figure 5 The test tubes and test results of Ji et al. [14]: (a) enhanced tube geometry; (b) HTC vs. oil concentration.

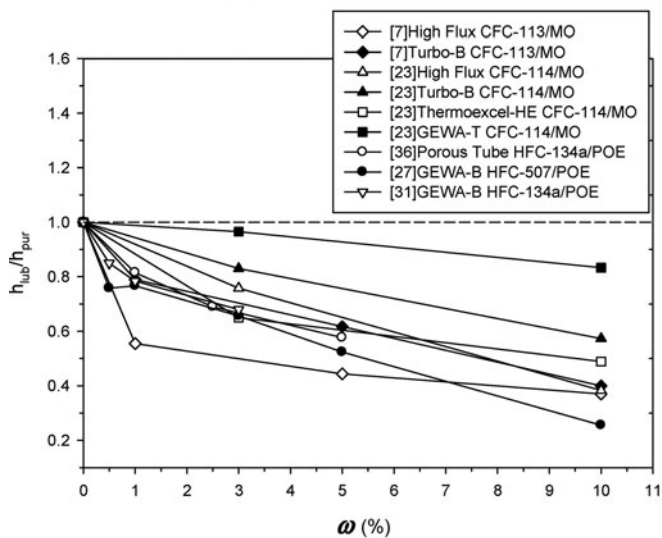
found and accordingly the heat transfer degradation for metal foam subject to lubricant oil is less pronounced.

Effect of Miscibility

In practical application, the refrigerants used in a flooded evaporator are usually miscible in lubricant oil. Miscibility is very important since an immiscible oil may form a film on the evaporator surface and may jeopardize the heat transfer. The refrigerant/lubricant mixtures may not be fully miscible in the whole range of temperature, pressure, and concentration. If immiscibility occurs then two liquid phases exist separately. However, with partial miscibility, the oil-rich layer may promote the heat transfer to some extent. In a mixture of refrigerant and lubricating oil the liquid is almost separated into an oil-enriched and a refrigerant-enriched liquid phase. The phases are stratified due to a lower density of the lubricating oil compared to the density of the liquid refrigerant. The oil-enriched phase is above the refrigerant-enriched phase. Normally the temperature of the fluid next to the heating surface is higher than in the bulk region; the state of the refrigerant-oil mixture may change from miscibility to immiscibility [27].

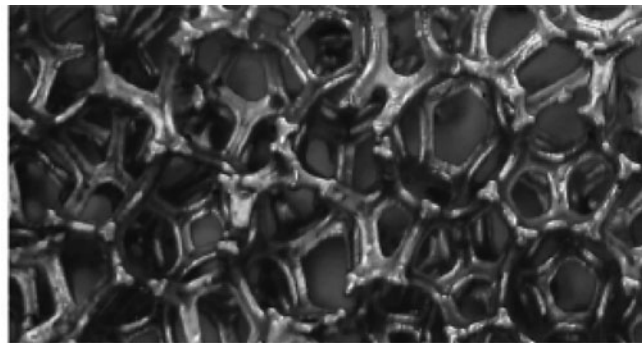


(a) Plain/integral tube

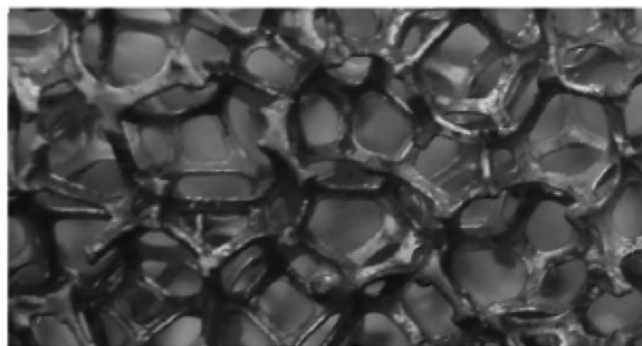


(b) Highly structured tube

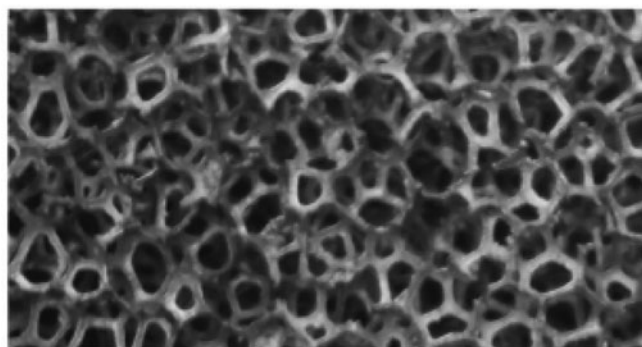
Figure 6 Variation in the relative boiling heat transfer coefficient of lubricant-mixed refrigerant reported in literatures (Ji et al. [14]): (a) plain/integral tubes; (b) highly structured surfaces.



Copper foam #1: ppi=10, Porosity=90%



Copper foam #2: ppi=10, Porosity=95%



Copper foam #3: ppi=20, Porosity=98%

Figure 7 Photos of the metal foam tested by Zhu et al. [39].

Kedzierski [17] explained the heat transfer augmentation reason with lubricants that are partially miscible, due to the liquid/liquid phase separation engendering additional nucleation sites. Figure 8 depicts a schematic from reference [17] concerning the critical solution diagram for a refrigerant/lubricant mixture and a bubble on a heated wall. The critical solution diagram gives the temperature and the compositions for which the lubricant and refrigerant are miscible. Below the dome, the lubricant/refrigerant are regarded as only partial miscible, whereas they are completely miscible above the dome. The arrow shows how the state of the liquid in the immediate vicinity of the bubble transitions to a two-phase state by a combination of composition shifting and evaporation at the liquid–vapor interface. Once this happens, two thin layers of different lubricant/refrigerant solutions will sit on top of the

bubble, leading to a large curvature gradient, and therefore a larger film pressure gradient that transports superheated liquid to the periphery of the bubble and a heat transfer augmentation accordingly. It is unclear whether the fluid next to the heated surface is in the region of total miscibility or in the region of immiscibility, or changing states between these two regions. Generally, in the range of immiscibility, the ratio of enhancement is less than 1.0, caused by the oil that plugs the nucleation sites, and the degradation becomes more pronounced for an enhanced tube.

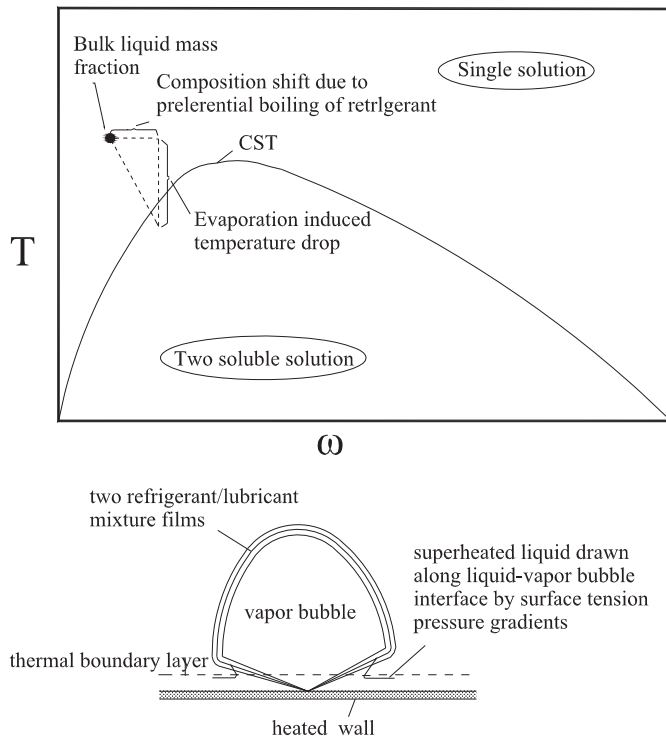


Figure 8 Schematic of the influence miscibility on nucleate boiling performance [17].

CONCLUSIONS

This review provides an overview of the lubricant on the heat transfer performance subject to nucleate boiling. There are various parameters affecting the heat transfer coefficient when oil is added. The effects of some very important parameters, such as oil concentration, heat flux, geometric configuration, saturation temperature, thermodynamic and transport properties, miscibility, foaming, and additional agent, are reported in this review. It appears that the effect of individual parameter on the heat transfer coefficient may be different from studies to studies. This is associated with the complex nature of lubricant and some compound effects accompanying the heat transport process. In addition, although there is only a tiny amount of lubricant, the associated heat transfer performance can be augmented or impaired. In this review, the authors try to summarize the general trend of the lubricant on the heat transfer coefficient, and to elaborate discrepancies of some inconsistent studies. Despite many efforts made during the past several decades, the lubricant effect on nucleate boiling HTC is still quite complex. This is especially pronounced when some lubricants may contain certain surface-active agents. The tiny amounts of active agents may give rise to appreciable change of surface tension, contact angle, and viscosity, and this effect is especially evident adjacent to the heat transfer surfaces. Therefore, it is recommended that future works can be more focused on the fundamental aspect of these parameters. In general, the HTC is impaired provided the oil concentration is sufficient high (e.g., >7%), and this is

applicable to smooth and structured tubes. Although normally structured tubes tend to suffer from lubricant, this is especially conspicuous when the size of the reentrant channel is small. On the other hand, foaming and partial miscibility seem to benefit the HTC.

NOMENCLATURE

A, B, C	constant in Eq. (1)
h	heat transfer coefficient ($\text{W}/\text{m}^2\text{-K}$)
h_o	heat transfer coefficient for refrigerant/lubricant mixture ($\text{W}/\text{m}^2\text{-K}$)
HTC	heat transfer coefficient ($\text{W}/\text{m}^2\text{-K}$)
q	heat flux (kW/m^2)
r	actual bubble radius, equilibrium radius (m)
r_o	actual bubble radius for refrigerant/lubricant mixture, equilibrium radius (m)
T_s	saturation temperature (K)
W	work (J)
x	vapor quality; mass fraction of more volatile component of binary mixture in liquid phase
y	mass fraction of more volatile component of binary mixture in vapor phase

Greek Symbols

σ	surface tension (N/m)
σ_o	surface tension of the refrigerant/lubricant mixture (N/m)
ω	oil concentration ratio (%)

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