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An overview of the effect of lubricant on the heat transfer performance on conventional refrigerants and natural refrigerant R-744

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ABSTRACT

This review provides an overview of the lubricant on the heat transfer performance, including nucleate boiling, convective boiling, shell side condensation, forced convective condensation, and gas cooling, for conventional refrigerants and natural refrigerant R-744. Various parameters affecting the heat transfer coefficient subject to lubricant, such as oil concentration, heat flux, mass flux, vapor quality, geometric configuration, saturation temperature, thermodynamic and transport properties are discussed in this overview. It appears that the effect of individual parameter on the Prod. Type: FTP heat transfer coefficient may be different from studies to studies. This is associated with the complex nature of lubricant and some compound effect accompanying with the heat transport process. 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. The lubricant can, increase or impair the heat transfer performance depending on the oil concentration, surface tension, surface geometry, and the like. For the condensation, it is more well accepted that the presence of lubricant normally will impair the heat transfer performance due to deposited oil film. However, the deterioration is comparatively smaller than that in nucleate/convective boiling. For the effect of lubricant on R-744 with convective evaporation, the general behavior is in line with the convectional refrigerant. For gas cooling, the lubricant cast significant effect on heat transfer coefficient especially for a higher mass flux or at a smaller diameter tube.

Contents

1. Introduction

A critical component in all refrigeration and air-conditioning systems is the compressor for compressing the circulating refrigerant vapor. In many applications the lubricant, which is essential for lubrication of the moving parts, is charged during the

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Nomenclature

A_G	flow area occupied by vapor (m^2)
A_L	flow area occupied by liquid (m^2)
A, B, C	constant in Eq. (1)
Cp_l	specific heat capacity for liquid (J/kg K)
D	pipe diameter (m)
g	acceleration of gravitation
G	mass flux, (kg/m^2 s)
h	heat transfer coefficient (W/m^2 K)
H_L	the depth of liquid in the pipe (m)
HTC	heat transfer coefficient (W/m^2 K)
Nu	Nusselt number, dimensionless
i_{fg}	latent heat (J/kg K)
P	pressure (bar)
P_{red}	reduced pressure
Pr	Prandtl number, dimensionless
q	heat flux (kW/m^2)
Re_v	Reynolds number based on vapor phase, dimensionless
T_{bulk}	bulk temperature (K)
T_s	saturation temperature (K)
T_w	wall surface temperature (K)
ΔT_{sub}	wall subcooling (T_s–T_w) (K)
U_{GS}	superficial velocity of vapor phase (m/s)
W	work (J)
x	vapor quality; mass fraction of more volatile component of binary mixture in liquid phase
x_{in}	inlet vapor quality
y	mass fraction of more volatile component of binary mixture in vapor phase
ρ_G	vapor density (kg/m^3)
ρ_L	liquid density (kg/m^3)
σ	surface tension (N/m)
ω	oil concentration ratio (%)

manufacture of the compressor and is expected to work reliably for the entire life of the unit. The main function of lubricant oil is to lubricate the internal moving parts of the compressor. On the other hand, the lubricant provides a seal between the moving parts enabling efficient vapor compression. Gibb et al. [1] had shown that the benefits of the introducing more energy efficient refrigeration lubricants can lead to a reduction in energy consumption as high as 15% and indirect reductions in emissions of the greenhouse gas CO_2. With properly designed lubricants, Gibb et al. [1] estimated that up to 80% of the industrial refrigeration and air-conditioning systems replaced in the next 20 years in the USA could result in annual energy saving up to 200,000 GW h corresponding to 11 million metric tons of carbon in reduced CO_2 emissions. Despite its crucial role in increasing the energy efficiency of compressor, in typical operation of an air-conditioning or refrigeration system, a small amount of lubricant oil may migrate from the compressor and 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. As a consequence, it is imperative to realize its role on the heat transfer performance as far as system efficiency is concerned.

The presence of lubricant alters the physical properties of refrigerant mixtures. Among the difference in thermophysical properties, the influence of viscosity is especially imperative since the viscosity of lubricant oil is about two to three orders higher than that of refrigerant. On the other hand, the corresponding surface tension of lubricant is approximately one order higher than that refrigerant. 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. Fig. 1 is a schematic of the associated properties of R-410A/POE VG68 mixture based on the calculated results of Wei et al. [2] 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, equal, or higher than that of refrigerant. In the meantime, a detectable rise of surface tension and a sharp rise of viscosity is encountered.

There had been many reviews concerning the influences of lubricant oils on the heat transfer characteristics of refrigerant, for instance [3–5], some general behaviors of the lubricants were reported and some controversies still exists. Hence the major objective of this review is to summarize the important findings and clarify some possible causes of the inconsistency. In addition, the second objective is to review the effect of lubricant on the heat transfer performance of the environmentally friendly natural refrigerant—R-744.

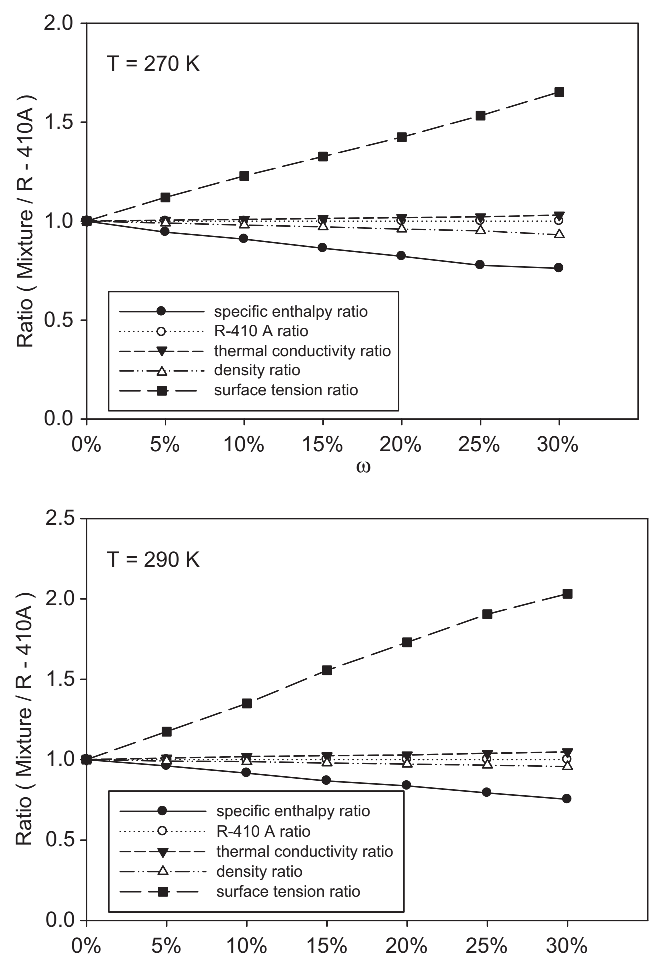

Fig. 1. Schematic of the associated properties of R-410A/POE VG68 mixture based on the calculated results of Wei et al. [2] (a) T=270 K; and (b) T=290 K.

2. Effect of lubricant on the heat transfer of conventional refrigerants

2.1. Effect of lubricant on pool boiling

For pool boiling, the effect of lubricant on heat transfer is rather small at a low weight concentration, i.e., $\omega < 3\%$, and for $2\% < \omega < 4\%$, some types of lubricant can enhance the pool boiling, and with a higher concentration ($\omega > 5\%$), almost all types of lubricant oil reduce pool boiling heat transfer. In practical application, the refrigerants used in 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 decrease the heat transfer accordingly. It is not uncommon for 5–10% oil by weight to be present in the system. There were much data reporting pool boiling heat transfer for smooth and enhanced surfaces for a wide variety of refrigerant–oil mixtures. However, the published data in the open literature about the effect of lubricant oil on the heat transfer performance is quite confusing. Some investigators had reported a consistent decrease of heat transfer coefficient with oil concentration (e.g., [6], [7], [8]). Conversely, some investigators had observed an increase of heat transfer coefficients at low oil concentration and decreases it at higher oil concentration (e.g., [9], [10], [11,12]). Even more interestingly, Zheng et al. [13] conducted boiling of ammonia/ lubricant mixture on a flooded evaporator with inlet quality. Some of their data, 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 is still inferior to pure refrigerant. However, this phenomenon is not seen at an elevated saturation temperature of 23.3 °C. Similar trend is shown for an inlet quality (x_{in}) of 0.2 and for x_{in}=0.4. at $T_s = -9.4$ °C. The authors did not provide explanations of this phenomenon. One possible explanation about this unusual characteristic is associated with foaming. We will defer this possible reasoning in later discussion. Wang et al. [14] investigated the effect of lubricant oil (3GS) on the pool boiling heat transfer performance for plain tube at saturation temperatures of 20, 4.4, and −5 °C for oil concentrations of 0.75%, 1.5%, 3.6%, and 7%. For T_s=20 °C, the heat transfer coefficients is decreased with increase of oil concentration. For q=21.1 kW/m^2, the heat transfer coefficients for ω=3.6% is about 55% that of pure refrigerant while for ω=7%, the heat transfer coefficient is only 45% of pure refrigerants. 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 coefficients is reversed. The heat transfer coefficients with oils are higher than those of pure refrigerants over the range of ω=0–3%. A maximum increase of 20–30% of heat transfer coefficient is observed near ω=1.5%.

In summary of the foregoing results, the heat transfer performance is very complicated when lubricant is added. The experimental results show that the influence of oil on the heat transfer coefficient is not only depending on the oil concentration itself but also on the heat flux and on other physical properties. Yet some of the properties may offset with each other and result in a complex behaviors. In the following, the authors try to summarize the associated influences of lubricant on the pool boiling HTC.

Effect of physical property: Thome [15] concluded the 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 one. In an investigation of the surface tension of the refrigerant–oil mixtures, Waller and Dick [16] argued that the heat transfer coefficients for refrigerant–oil mixtures are strongly related to the interfacial effects. Based on the theoretical developments of Stephan and Körner [17], Waller and Dick [16] proposed a simplified formula for the work required for the generation of vapor in the mixtures, i.e.,

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

For a weak interfacial interaction of refrigerant–oil mixtures, the bubble generation work is increasing with increasing (y–x). Since refrigerant–oil is considered as a zeotropic mixtures which has a considerable mass transfer resistance. As a result, 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 Fig. 2, the relation of surface tension of refrigerant–oil mixtures, as was classified by Wallner and Dick [16], has three types of characteristics. For type a, the surface tension vs. oil concentrations for refrigerant–oil mixtures did not show a minimum. For type b, the oils contain surface active agent which leads to minimum in surface tension. For type c, the oils contain surface active agent and additional dirt particles in small concentration, this leads to two points of inflexion and a sharp minimum of surface tension. Wallner and Dick [16] argued that type c is responsible to an increase of heat transfer coefficients at lower oil concentrations. However, the explanation is unable to explain the aforementioned Zheng et al.'s data [13] showing a sharp rise of heat transfer coefficient.

Moreover, when oil added to the refrigerant an intensive foaming is produced as heat flux is increased. Several investigators had postulated the increase of heat transfer coefficients at low oil concentration is related to the foaming process (e.g., Stephan [18], Udomboresuwan and Mesler [19]). They had put important efforts to interpret the effect of foaming. According to a review by Mohrlok et al. [20], several other possible mechanisms that cast significant impacts on the pool boiling heat

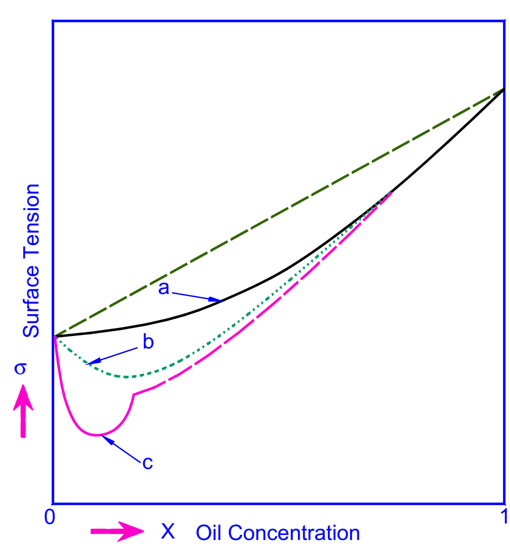

Fig. 2. Possible relationship between surface tension and oil concentration.

transfer performance, and some of their comments are summarized as follows:

Effect of blocking: During 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. This will cause a smaller bubble departure diameter and a higher temperature difference amid the tube wall and the saturation temperature of the liquid. 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.

Effect of nucleation site: 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 constant heat flux condition. Via addition of oil, the surface tension of the refrigerant/oil mixtures rises and the wall temperature, resulting in a lower surface tension nearby the heating surface. Therefore a greater gradient of the surface tension around the bubble which enhances the Marangoni convection. In the meantime, the area influenced by one nucleation site will be reduced. The reduced area influenced by one nucleation site can lead to a higher nucleation site density and an increase of the pool boiling heat transfer coefficient.

Effect of foaming: 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 an 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 ([11,12], [19]) also claimed that the effect of foaming for refrigerant–oil mixtures may significantly increase the heat transfer characteristics. Udomboresuwan and Mesler [19] reported significant enhancement in pool boiling heat transfer in the presence of foam. They assumed two possible enhancement effects caused by the foaming. And they are (1) a thin liquid film was created between the foam and the heated surface which results in a very large heat transfer coefficient; and (2) secondary nucleation caused by the bubble leaving the surface which bursting into the neighboring liquid–vapor region. Wang et al. [14] performed a visualization of the boiling phenomenon to understand further details about the effect of foaming. Fig. 3 shows the foaming characteristics for 5GS oils at T_s=20 °C, ω=0.75% and for q=84, 58.8, and 16.7 kW/m^2 whereas Fig. 4 illustrates the foaming behaviors at similar test conditions except T_s=−5 °C. 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 to the liquid/foam interface may be smaller as the saturation temperature is decreased. The observation substantiates the first conclusion by Udomboresuwan and Mesler [19]. It is also interesting to know that a further increase of heat flux may result in significantly increase of the depth of the forming. In addition, the size of the foaming is getting 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 one. The coarse one is on the top of the fine one. In summary of the foregoing observations, it is concluded that the effect of foaming are more evident in higher oil concentration and at a lower saturation temperature. The results may explain the sharp bounce of heat transfer coefficient of Zheng et al.'s data [13] 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.

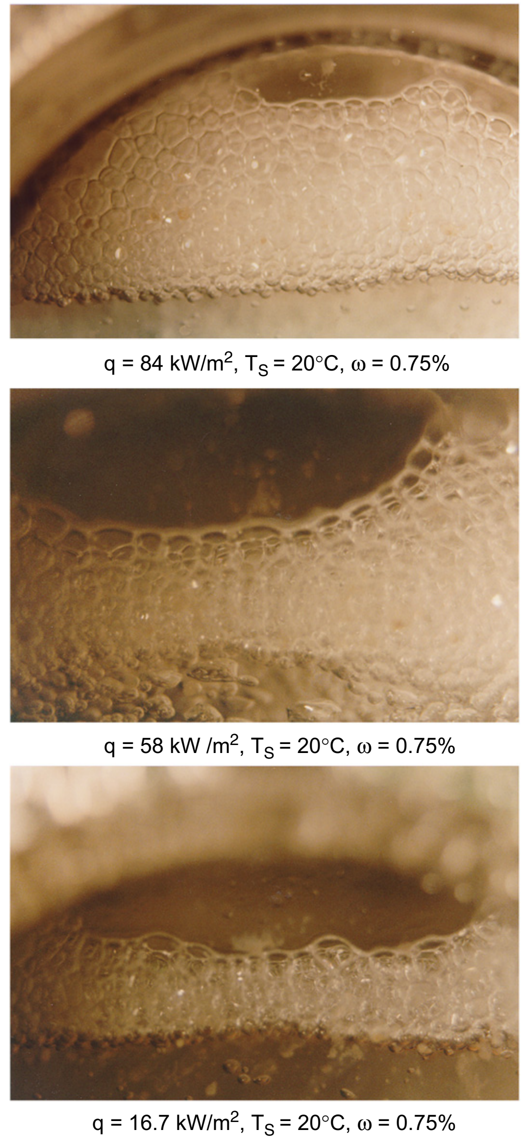

Fig. 3. Boiling pattern for T_s=20 °C, and ω=0.75% [14].

Effect of geometry: For standard tube geometries (smooth and low finned tubes) the heat transfer increases by adding oil to the refrigerant. For enhanced tubes, it is reported that the oil plugs the micro channels 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 micro-channels of the heating surface. This leads to a lower heat transfer performance. Most studies depicted that the presence of oil deteriorate more nucleate boiling heat transfer performance for a highly structured surface than a smooth one. This can be made clear from the studies of Ji et al. [21] who performed R-134a/PVE lubricant for plain, integral fin and four enhanced tubes (the geometry of four enhanced tubes is shown in Fig. 5(a)). Their test results, in terms of ratio relative to plain tube, is shown in Fig. 5(b). Apparently, more structured surface reveals a more pronounced drop with lubricant concentration. Ji et al. [21] also compares some existing results concerning the effect of

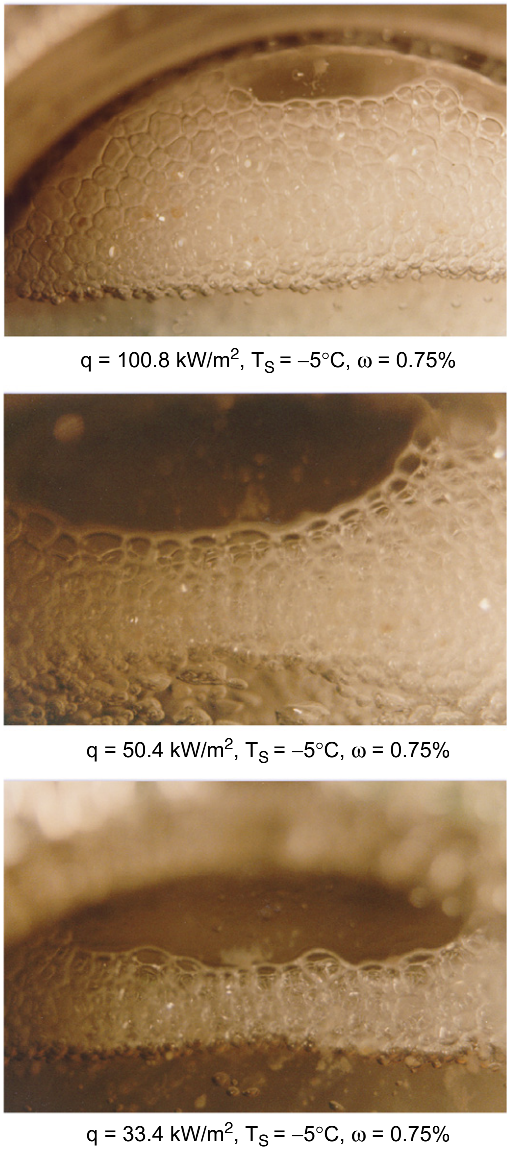

Fig. 4. Boiling pattern for $T_s = -5$ °C, and $\omega = 0.75\%$ [14].

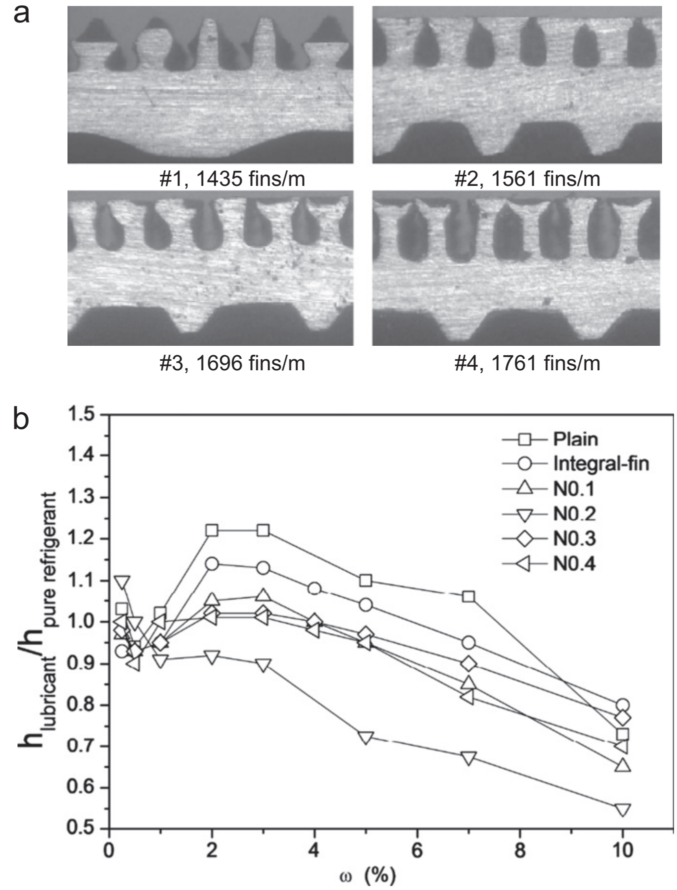

Fig. 5. Test results for Ji et al. [21] (a) Schematic of the enhanced tube structure by Ji et al. [21], enhanced tube geometry; (b) Variation of the relative boiling heat transfer coefficient of R-134a/lubricant. HTC vs. oil concentration.

tube geometry as shown in Fig. 6 where Fig. 6(a) is for smooth and integral fin tube ([22], [11], [20], [23], [24]) whereas Fig. 6(b) is for highly structured surfaces ([22], [11], [20], [24], [25]). It appears that for plain/integral tubes that a detectable enhancement is encountered at low oil concentration ($<5\%$) 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 re-entrant tunnel by lubricant. Test results at a lower oil concentration ($<2\%$) from Kedzierski [26–30] for various working fluids and enhanced surfaces also reveals similar trend. However, a recent report by Zhu et al. [31] who performed nucleate boiling test 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 indicated 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 itself. As shown in Fig. 7 for the metal foam structure, unlike those of highly structured surfaces shown in Fig. 5(a) where adjacent re-entrant channels are normally isolated. By contrast, the open cell structure of the metal foam suggests a continuous re-entrant channel amid the metal foam surface. In this regard, the lubricant oil casts less effective on blocking the pore surfaces of the metal foam. As a result, the heat transfer degradation for metal foam subject to lubricant oil is less pronounced.

Effect of miscibility [20]: 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. In a mixture of refrigerant and lubricating oil the liquid is separated almost in 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. On the other hand the effective oil mass fraction increases around the heating surface since the refrigerant is more volatile. 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 which plugs the nucleation sites and the degradation becomes more pronounced for enhanced tube.

2.2. *Effect of lubricant on convective boiling*

Similar to the influence of lubricant on the pool boiling heat transfer performance, the existing data concerning the effect of lubricant on convective boiling is also inconclusive. The presence

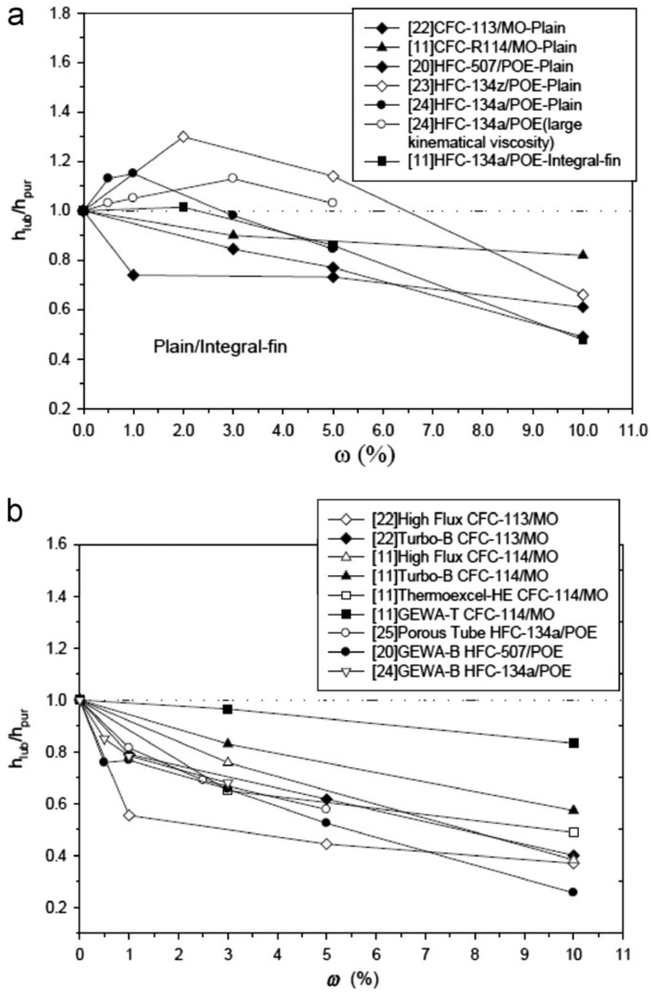

Fig. 6. Variation in the relative boiling heat transfer coefficient of lubricant-mixed refrigerant reported in literatures ([21]) (a) plain/integral tubes; (b) Highly structured surfaces.

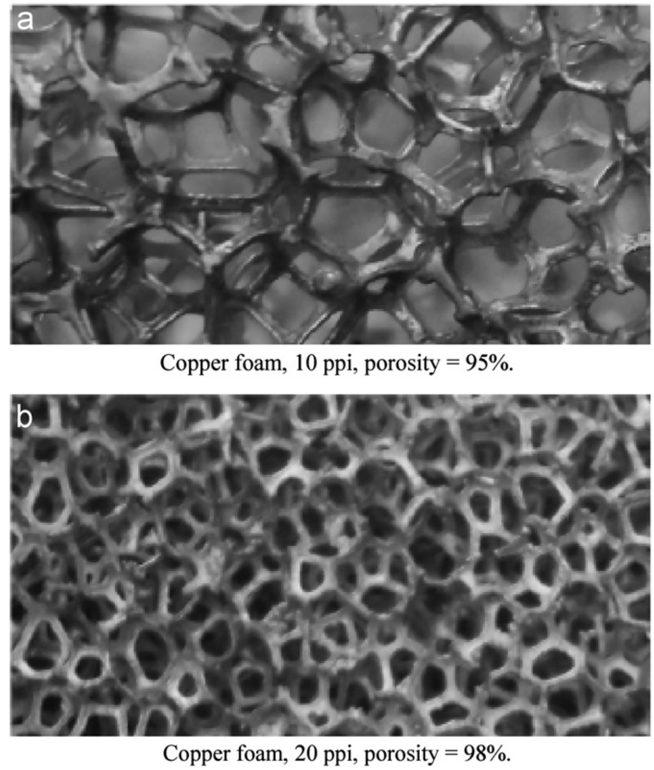

Fig. 7. Photos of the metal foam tested by Zhu et al. [26].

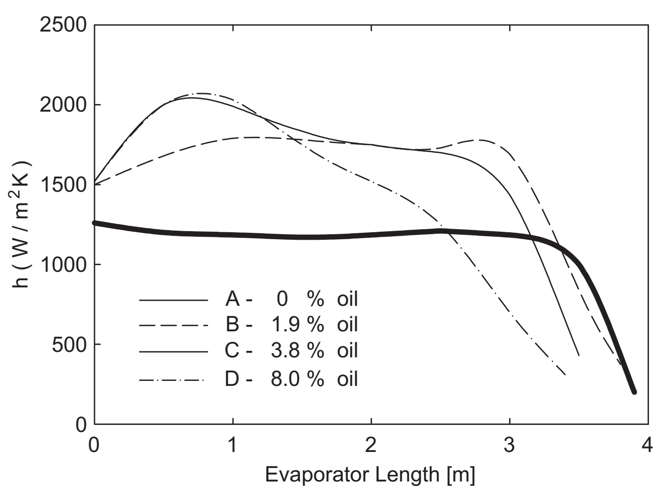

Fig. 8. Local HTC as a function of evaporator length at −15 °C ([35]).

of lubricant has two counter effects which normally offset with each other, resulting in certain inconsistency. First, lubricant promote wettability, causing refrigerant mixture to spread around the periphery and an early transition to annular flow pattern. This is especially beneficial upon stratified/wavy flow pattern. By contrast, the presence of lubricant increases the viscosity which normally lessen the degree of mixing of refrigerant mixtures, leading to a decline of heat transfer performance. There are also other important factors affecting the convective boiling performance and are summarized in the following.

Oil concentration effect: In general, the presence of lubricant oil above 1% in mass reduces the heat transfer coefficient and this degradation increases with the oil concentration. Lubricant oil at high concentrations (above 5%) drastically decreases flow boiling heat transfer. However, as aforementioned previously, some studies (for example, [32,33] and [34]) observed that some lubricant oils increase flow boiling heat transfer when the oil concentrations are less than 3%. The enhancement apparently depends on the type of lubricant oil, heat flux, flow rate, flow patterns, the type of tube and other parameters, yet the exact enhancement mechanism was not clearly identified. Worsoe-Schmidt [35] was the first one to study the effect of lubricant on the flow pattern. His test data depicted in Fig. 8 showed an appreciable rise of heat transfer coefficient with oil concentration, and an heat transfer enhancement of 50% for R-12 with ω=1.9% at a saturation temperature of −15 °C. When ω=8.0%, the heat transfer coefficient presented a strong enhancement in the first meter of the test section, but above this length the HTC dropped drastically while the heat transfer coefficient for pure refrigerant remained unchanged. The significant rise of HTC of Worsoe-Schmidt's data arises from two aspects. First, their test was performed at low mass flux (30–100 kg/m^2 s), a relatively low heat flux, and at a low temperature conditions which resembling pool boiling condition to some extent. From the foregoing discussion of the effect of lubricant on pool boiling, one can see that there exists a positive effect on HTC at a low evaporation temperature provided the concentration is comparatively low.

Second, the mass flux was quite low where stratified/wavy flow prevails as the major flow pattern, and this was also confirmed from his flow pattern observation. As a consequence, the rise of surface tension from lubricant addition normally promotes wetting characteristics and resulting in an early transition to annular flow pattern and a considerable rise of heat transfer coefficient accordingly. Third, with the rise of oil concentration, he also observed detectable foaming which may easily touch and wet the periphery and assists the heat transfer performance. The enhanced level caused by lubricant oil of Worsoe-Schmidt's data is moderately higher than others [32–34]. This is probably due to its hairpin configuration which comprise a series of return bends. The presence of return bend further accentuates the occurrence of annular flow pattern [36], prolonging the influence of lubricant at a higher enhancement level.

The early transition to annular flow pattern for refrigerant/oil mixture had been observed by Wongwises et al. [37] who conducted two-phase flow visualization experiments with R-134a with lubricant oil mixtures inside a 7.8 mm horizontal tube. A schematic of the progress of flow pattern subject to the influence of lubricant oil is shown in Fig. 9. They found that the small amount of the liquid layer is pushed up around the tube wall perimeter by the momentum of vapor when the surface wave rolls along the flow direction, and the presence of foaming, when comparing to that of pure refrigerant, results in a higher effective liquid level that may easily reach the upper wall. This liquid level and the high viscosity of the attached "oil-rich" layer Kim and Katsuta [38] may result in an "annular-like" flow pattern. It should be mentioned that Worsoe-Schmidt [35] and Manwell and Bergles [39] also reported that the oil was found to increase the wetted portion of the tube wall.

The above-mentioned results may imply an early transition from wavy to annular flow pattern at a lower value of U_{GS}. According to Taitel and Dukler [40], the transition from stratified to intermittent or annular flow will occur when

$$U_{GS} > \left(1-\frac{H_L}{D}\right)\left\{\frac{(\rho_L-\rho_G)gA_G}{\rho_G(dA_L/dH_L)}\right\}^{1/2} \tag{2}$$

where D is the pipe diameter and H_L the depth of liquid in the pipe. From Eq. (2), with an increase of effective liquid level, one can see the transition boundary from wavy to annular flow region may be shifted to a lower value of U_{GS}.

Viscosity Effect: The decrease in the convective evaporation due to the presence of oil can be attributed to the increased mixture viscosity and the oil mass transfer resistance effect. On the other hand, the high viscosity accelerates the formation of annular flow ([4]). This factor is likely to benefit the flow boiling at low and intermediate qualities, while it impairs the flow boiling at high quality. Hambraeus et al. [41] studied three ester-based lubricants mixed with R-134a. They observed that the largest flow boiling degradation corresponded to the lubricant with the largest viscosity. McMullan et al. [42] reported the flow boiling heat transfer of R-12 mixed with three lubricants. During the tests, the authors found that at an oil concentration of 1%, the overall evaporator performance degraded with an increase of the oil viscosity. However, at an oil concentration of 3%, the trend was reversed. The authors claimed that the reversed trend is associated with the contribution of surface tension which provides better wetting, albeit increased viscosity impairs the heat transfer performance in both concentrations. The optimum oil concentration for heat transfer was determined by the trade-off between the reduced convective heat transfer and the increased wetted surface. In addition, the miscibility of the refrigerant mixture also plays certain role in this phenomenon. As depicted in Fig. 10 which is taken from Kim and Katsuta [38], despite the presence of lubricant promote an early transition to annular which normally enhances the corresponding heat transfer, the upper liquid layer that is an oil-rich layer containing very few refrigerant. The flow pattern is known as tear flow pattern and were observed from some studies at high oil concentration (e.g., [38] and [35]). In this regard, the heat transfer performance is comparable or even lower than the oil-free case even the flow pattern is similar to annular flow.

Mass velocity effect: Some authors verified that high mass velocities promote a more uniform refrigerant/oil mixture, which can reduce the performance loss caused by the lubricant oil and lessen its non-equilibrium effects. Many researchers had reported that HTC increased with the mass velocity, implicating the same behavior as seen in pure refrigerants. It is also important to note that at low mass velocities (stratified/wavy flow), the foaming formation happens on the liquid–vapor interface. In the meantime, at high mass velocities (annular flow), it is possible to observe froth flow like those observed by Wongwises et al. [37].

Vapor quality effect: Since the lubricant oil is a nearly nonvolatile component, its partial pressure in the vapor phase is usually neglected. The lubricant oil remains in the liquid phase and its concentration increases with vapor quality as the refrigerant evaporates. With a nominal lubricant oil concentration around 5.0%, at the end of the evaporation process the local concentration can reach values of the order of 90% in the remaining liquid. Therefore, the increase of the viscosity of refrigerant/oil mixture and the non-equilibrium effect become rather significant. This becomes especially pronounced when the mass flux is low and high oil concentration regime where stratified/wavy flow pattern prevails. With the presence of tear like pattern, the oil-rich layer in the upper portion of the horizontal tube may impair the heat transfer performance considerably. Thome [43] observed that in many experimental results the effect of lubricant oil concentration was not important at vapor qualities below about 85%. On the other hand, in direct expansion evaporators all the refrigerant is supposed to be evaporated and hence the evaporation process passes through the high vapor quality region where the adverse oil effect is very profound. In fact, it is not feasible to completely drive all of the refrigerant out of the solution into the vapor phase unless elevated temperatures are applied (>300 °C).

Effect of geometry of tube: Since microfin tubes promote the annular flow pattern even at low mass velocities (or pseudo annular, because the grooves tend to convey liquid to the upper regions of the tube, promoting its wetting and the flow pattern thus has a close resemblance with the annular flow pattern in smooth tubes, with a liquid layer of thickness possibly higher than that of the microfin covering the whole surface of the tube), the presence of lubricant oil in the refrigerant can lose its benefit to induce annular flow as observed in smooth tubes. Ha and Bergles [44] performed experiments for R-12/oil mixtures in an electrically heated microfin tube with ω ranging from 0% to 5%. They found a much larger HTC degradation for microfin tube. On the other hand, Nidegger et al. [45] also reported an appreciable falloff of HTC for microfin tube at the mass flux of 100 kg/m^2 s with the presence of lubricant oil, but the trend is reversed where oil increased the HTC relative to pure R-134a when tested at the highest mass flux (G=300 kg/m^2 s). They argued that the degradation of HTC at low mass flux is associated with the holdup of the lubricant oil at the microfins. The comments seem feasible for it also can explain the reversed trend shown in higher mass flux. And this degradation is even worse for highly porous coated surface as reported in the recent study by Dawidowicz and Cieśliński [46].

Correlations to predict the convective boiling heat transfer coefficient subject to influence of lubricant: Hu et al. [47] provided an

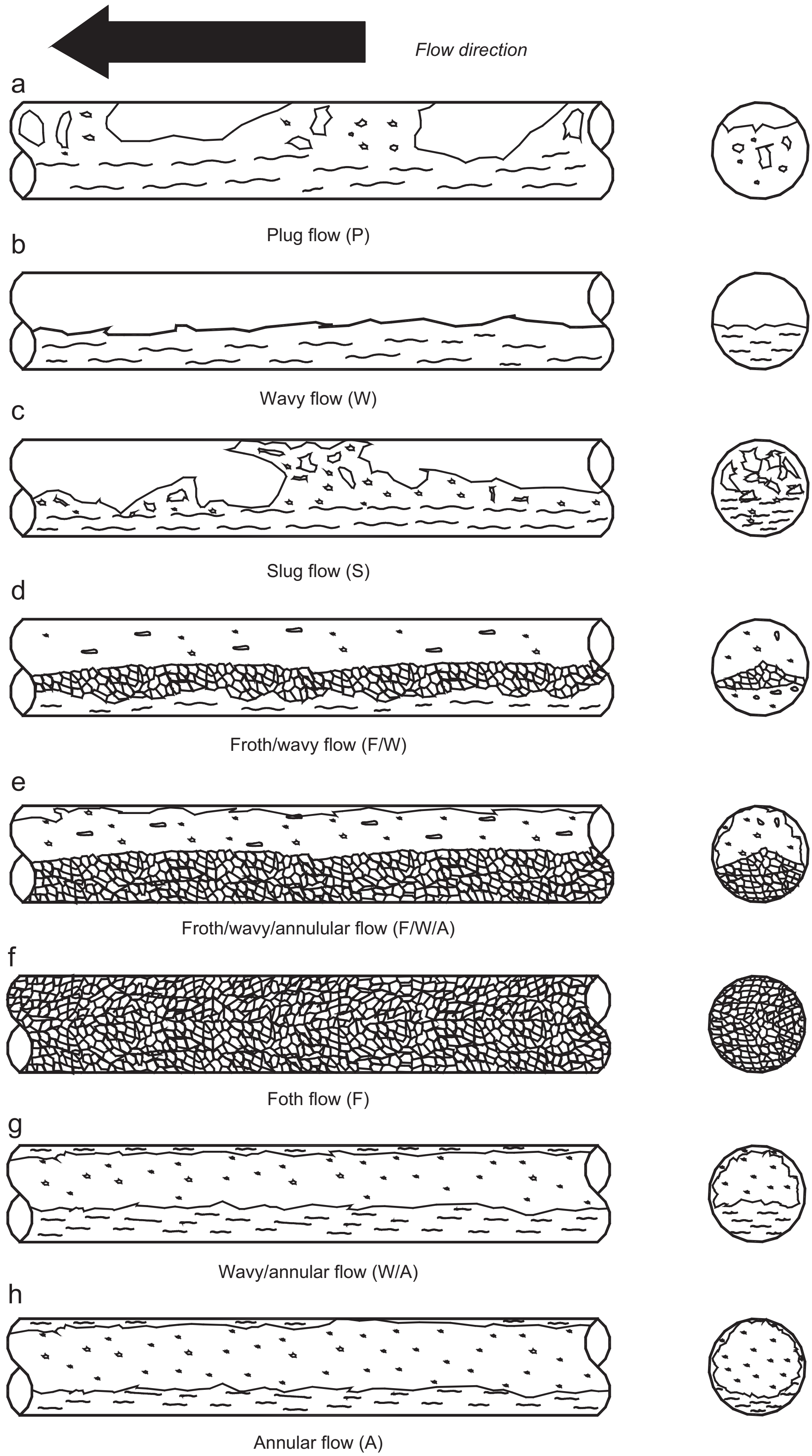

Fig. 9. Sketch of flow patterns subject to influence of lubricant [32].

extensive comparison of their measured R-410A/oil data against existing correlations. These correlations are all empirically based and the correlations were developed based on a specific database, extending the applicability outside their database is generally not recommended. One of the major reasons of the limited applicable range of the existing correlations is due to lack of rational

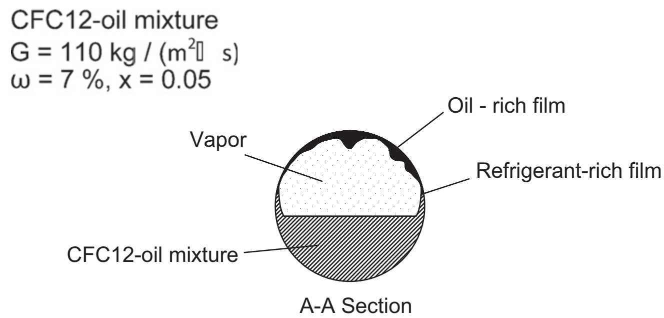

Fig. 10. Schematic of the two-phase annular flow pattern of R-12/oil mixtures with oil-rich tear flow pattern at the upper part of the tube [33].

parameters. For instance, as aforementioned in previous section that the presence of lubricant may affect the flow pattern, thereby affecting the heat transfer performance. In this regard, Hu et al. [47] developed a flow pattern map for R-410A/oil mixture which was originally developed by Wojtan et al. [48,49] using the mixture properties to replace the refrigerant properties. They found that the presence of oil promotes the transition from "Slug" to "Intermittent", while it delays the transition from "Slug+SW (stratified wavy)" to "Slug", "Intermittent" to "Annular", from Annular" to "Dryout" and from "Dryout" to "Mist". Using the information, they developed a correlation that can correlated their measurements for the 4.18 mm horizontal smooth tube with a deviation of $\pm 30\%$, and it agrees with 96% of their experimental data in the 6.34 mm horizontal smooth tube, within a deviation of $\pm 20\%$. However, it should be noted that the correlation is, again, applicable to their own database. Some important effects, like surface tension, locally oil-rich tear flow pattern, and the like are still missing in the correlation. In summary, there is no well accepted correlations available that can describe the influence of lubricant on the convective boiling heat transfer performance up to now.

2.3. Effect of lubricant on shell side condensation

The effect of lubricant on shell side condensation is generally consistent. Most of the studies indicate a negligible or only very small deterioration of heat transfer coefficient subject to lubricant. Though the increased viscosity may deteriorate heat transfer performance, the increased surface tension may assist the wetting characteristics and help to drain the liquid condensate which is very helpful for shell side condensation. In addition, the lubricant is normally entrained with the refrigerant and does not necessary rest on the tubing, lessening the influence of lubricant oil. Williams and Sauer [50] performed condensation tests for R-11 with 150 and 550 SSU at 37.8 °C, and they found no change in heat transfer coefficient when the concentration is less than 3% but an appreciable decrease of HTC is encountered when the oil concentration is above 7%. Sauer and Williams [51] also investigated the condensation of R-11/oil mixtures on low finned tubes, using the same apparatus of [50]. Their results indicated a negligible influence of lubricant oil on the condensation heat transfer coefficient. Wang et al. [52,53] conducted experimental study for R-12 and R-22 on the external surface of single and multiple horizontal plain tubes, and they found the average decrease in the condensing heat transfer coefficient is 2% for every 2% increase of oil concentration (R-12) and 3% for R-22. In the meantime, the miscibility may also play certain role in the condensation heat transfer performance. Adbul'Manov and Mirmov [54] performed experiments for ammonia–oil mixtures, and they found that the presence of oil may result in partial dropwise condensation, and an enhancement of HTC. On the other hand, Wang et al. [55] conducts R-11/oil mixtures for plain and low fin tubing with ω ranging from 0% to 15.2%. Their test results for plain tube is in line with that of Wang et al. [52,53] but the data of low fin tubing subject to the influence of lubricant oil shows a reversed trend at low wall subcooling ($\Delta T_{sub} < 5$ °C). Part of the explanation is due to higher surface tension of the from the lubricant contributes the condensate drainage. This explanation is feasible for the plain tube is more on the gravity drainage while low fin tube is more related to surface tension drainage [56].

2.4. Effect of lubricant on convective condensation

Unlike those in convective boiling, the effect of lubricant on the convective condensation heat transfer performance are quite consistent ([5]). Almost all the literatures reported a decrease of heat transfer coefficient with addition of lubricant except Shao and Granryd [57] who measured the heat transfer and pressure drop performance of R-134a/oil mixtures. They found that the refrigerant/oil mixture had a higher condensation coefficient at the beginning of the condenser while a lower heat transfer coefficient toward the latter part of the condenser is observed. A possible reason for this phenomenon is that the refrigerant/oil mixture has a higher condensation temperature compared to the pure refrigerant. And consequently it begins to condense earlier in the condenser compared to the pure refrigerant. In the meantime, there is another possible explanation about the inconsistency of Shao and Granryd's data against general observation. Notice that the lubricant they used is SW32 whose viscosity at 40 °C is about one-fifth of other studies (normally use SUS 150 grade). The low viscosity lubricant implied that a more turbulent condensate film occurring at the refrigerant/SW32 mixtures, and result in a higher condensation heat transfer performance. Moreover, Shao and Granryd [57] and Tichy et al. [58] indicated that the definition of the condensation reference temperature is important and that the heat transfer coefficient calculated using the pure refrigerant saturation temperature is larger than the one calculated using the mixture dew temperature. Other important factors on the condensing HTC summarized from Shen et al. [5] and the review article of Gidwani et al. [59] are as follows.

Effect of Oil Concentration: In general, increase oil concentration degrades the condensation heat transfer performance. However, this relationship is nonlinear. At small oil concentrations (e.g., less than 3%), the lubricant influence is negligible. Data from Schlager et al. [60] and Eckels et al. [61] showed that the HTC of refrigerant/oil mixtures is very close to each other and reveals slight decrease with the rise of oil concentration. Similar trend is observed from Bassi and Bansal [62], they also developed a correlation to describe the influence of oil concentration:

$$\frac{h_{oil}}{h_{no-oil}} = e^{-2.2\omega} \tag{3}$$

Shao and Granryd [57] modified correlation that is originally applicable for pure refrigerant R-134a from Tandon et al. [65] correlations. The developed correlation for refrigerant–lubricant mixture is:

$$Nu = 15.9\mathrm{Pr}_l^{1/3}\left(\frac{i_{fg}}{Cp_l(T_s - T_w)}\right)^{1/6} Re_v^{0.15} e^{-5\omega} \quad \text{When } Re_v < 24000 \tag{4}$$

$$Nu = 15.9\mathrm{Pr}_l^{1/3}\left(\frac{i_{fg}}{Cp_l(T_s - T_w)}\right)^{1/6} Re_v^{0.67} e^{-5\omega} \quad \text{When } Re_v > 24000 \tag{5}$$

Effect of Tube Geometries: Generally, the effect of tube geometry (microfin vs. smooth tube) subject to the influence of lubricant is about the same. This can be made clear from the reports of Eckels et al. [61,64,63], Sur and Azer [66], and Cavallini et al. [67]. Their explanation for this observation may be associated with the fin tips were not flooded by the condensate/lubricant mixtures. Schlager et al. [60] reported that the presence of oil reduces the condensation heat transfer in smooth tubes and micro-fin tubes at the same level, while the presence of oil is less important low-fin tubes for the low-fin tubes have a larger fin height than the micro-fin tubes, thereby the fin tip of the low-fin tube may be free from oil contamination. In addition, the higher surface tension of lubricant often assist condensate drainage provided the fin tip is not flooded. In this regard, the effect of lubricant becomes less profound for low fin tube as compared to microfin tubes. The presence of lubricant often lead to a higher pressure drop when compared to oil free conditions. However, Schlager et al. [60] found that the addition of lubricant decreased the condensation pressure drop but the effect is not found in microfin tube. A recent study by Huang et al. [68,69] who conducted R-410A/oil mixture inside small diameter microfin tubes (4 mm) had shown that a transition quality exists about the influence of lubricant on pressure drop. The frictional pressure drop of R-410A/oil mixtures may be reduced by a maximum of 18% when the vapor quality is lower than 0.6, and is increased by a maximum of 13% when the vapor quality exceed 0.6. Huang et al. [68] argued that the unique phenomenon is caused by the significant drop of turbulent flow intensity pertaining to high viscosity of lubricant which is likely to change the turbulent condensate liquid film into a laminar one. The arguments seem plausible for the data of Eckels et al. [61] had conducted condensation of R-134a with SUS169 & SUS369 lubricant, and their data clearly showed that the condensation pressure drop for SUS 369 is nearly identical to the oil-free tests (with ω up to 5%) whereas the condensation pressure drop for SUS 169 is much higher than the oil-free tests (55% for $\omega=5\%$ and $G=200\,\mathrm{kg/m^2\,s}$). For Eckels's data, it is interesting to know that the pressure drop penalty ratio (in terms of $\Delta p_{oil}/\Delta p_{oil\text{-}free}$) for SUS 169 lubricant is appreciably reduced when G is increased to $300\,\mathrm{kg/m^2\,s}$. In this regard, there must be other explanation for this situation. From the present authors aspect, it is quite likely due to another influence—entrainment. With the rise of mass flux, the significant vapor shear may give rise to considerable lubricant entrainment. In the meantime, the tube size for Huang et al. [68] is only 4 mm whereas it is 9.53 mm for most prior studies. The small diameter and a comparatively high mass flux may entrain more lubricant droplet alongside the tube side rather than deposit on the wall surface, this is especially noteworthy for condensation due to its relatively high vapor velocity from the entrance. Hence, it will reduce the direct contact of lubricant and wall surface, and as a result a smaller frictional pressure drop.

Effect of Mixture Viscosity: Shao and Granryd [57] implied that the degradation of the condensation heat transfer coefficient is due to the higher viscosity of the refrigerant/oil liquid film as compared to the pure refrigerant. The higher viscosity reduces the molecular and turbulent transport in the condensate film. From the foregoing comment, it may reach a conclusion that the degradation in the condensation heat transfer coefficient increases as the viscosity of the oil increases. However, the conclusion seems not to be in line with some experimental results such as Chitti and Anand [70] and Eckels et al. [61]. It indicates that the increased mixture viscosity may not be the only factor affecting the condensation heat transfer coefficient.

Effect of Vapor Quality: Cho and Tae [71] found that the heat transfer coefficient of R-22/oil and R-407C/oil decreased with an increase in inlet quality. The author attributed this to an increase in average liquid phase viscosity. In the meantime, they also found a substantial rise of the second section after the return bend, this is particularly noteworthy at a low mass flux like $100\,\mathrm{kg/m^2\,s}$ and normally more lubricant will give rise to higher condensation HTC. Cho and Tae [71] ascribed this results to turbulence engendered by the return bend. However, this phenomenon is more related to flow pattern transition as aforementioned discussions. Cawte [72] indicated that the presence of oil at low concentrations has an insignificant effect on the heat transfer coefficient in low quality region whereas a more pronounced deterioration is encountered at higher qualities. However, at high oil concentrations of 10% or more, the oil presence affects the heat transfer coefficient uniformly over the entire quality range due to the noteworthy increase of liquid viscosity. Fukushima and Kudou [73] presented the condensation heat transfer of refrigerant/oil mixtures vs. quality. In their general observation, the pure refrigerant condensation coefficient decreases at lower vapor qualities region because the liquid film is getting thicker. Upon refrigerant/oil mixtures, the authors observed a maximum in heat transfer coefficient occurred at certain vapor quality. This is caused by two opposite effects. At high vapor qualities, the liquid film is thin but oil-rich, which results in a higher mass transfer resistance effect and higher viscosity in the film. As the vapor quality decreases, the liquid film becomes thicker, but more dissolved refrigerant decreases the effects of the lubricant in the film. Hence, there was a plateau of the condensing heat transfer coefficient when the vapor quality is decreased, and this phenomenon becomes more apparent at a higher oil concentration. The results agree with the visual observation by Kim and Katsuta [38] in Fig. 10. In general, there is little agreement about the lubricant influences at high vapor quality regions. At high qualities, the viscosity and mass transfer resistance effect in the liquid film are more pronounced, which may reduce the heat transfer. However, high oil concentrations at high vapor qualities result in a significant increase in the dew temperature, which increases the heat transfer driving potential.

Effect of U Bends for refrigerant/lubricantmixtures. For typical evaporator and condenser in refrigerators and air-conditioners for HVAC and R application, consecutive U-type wavy tube (hairpin) is very common. Notice that the hairpin is a combination of straight tubes and return bends. The presence of U bends prompts early transition to annular flow patterns [74]. On the other hand, the effect of lubricant on heat transfer and pressure drop with the presence of U-bends are all negative [70,75–77], impairing heat transfer performance and increasing the pressure drop with the oil concentration.

3. Effect of lubricant on the thermofluids characteristics of R-744

When compared with the convectional refrigerants, literatures concerning the effect of lubricant on the heat transfer performance of R-744 (CO_2) is comparatively few. A variety of lubricants can be used in CO_2 refrigeration systems. In certain systems, synthetic hydrocarbons such as alkylbenzenes (ABs) and polyalphaolefins (PAOs) can still be used even though they have poor solubility with CO_2 [78]. The poor solubility of the synthetic hydrocarbons is compensated by their excellent low temperature flow properties, which can be improved further by blending with more miscible lubricants ([79]). Kawaguchi et al. [80] reported that polyalkylene glycol (PAG) was the primary lubricant for CO_2 systems since it is partially miscible with CO_2. Seeton and Fahl [81] found that PAG reveals the best lubricity for trans-critical applications, and PAG is not miscible with CO_2 at high concentrations. Li and Rajewski [82] found that polyol ester (POE) lubricant was completely miscible with CO_2. Ma et al. [83] suggested that POE was better than other lubricants for trans-critical CO_2 system, while Renz [84] reported that POE was particularly suitable for semi-hermetic reciprocating and screw compressors for CO_2

cascade systems. They have a high viscosity index, good lubrication behavior, acceptable solubility properties and favorable miscibility. For the effect of lubricant on the boiling heat transfer performance, Zhao and Bansal [78] had presented a comprehensive review and some of relevant details are summarized as follows:

3.1. Effect of lubricant on convective boiling of R744

Effect of oil concentration: Dang et al. [85] investigated the flow boiling heat transfer of CO_2/PAG mixture in horizontal smooth tubes with oil concentrations varying from 0.5% to 5.0%. For a test tube size of 6 mm, they found that the addition of a small amount of lubricant resulted in a sharp decrease in the heat transfer coefficient. For instance, the HTC is reduced from 8–9 to 3–5 kW/m^2 K when oil concentration is increased from 0% to 0.5%. However, a further increase of oil concentration from 0.5% to 5% casts almost negligible effect on heat transfer coefficient.

Similar influences are observed for smaller diameter tubes like 2 or 4 mm but the corresponding critical oil concentration is 0.5% for 2 mm inner diameter tube and 1% for 4 mm inner diameter tube. Furthermore, the addition of lubricant seems to have no influence on the dryout quality and the post-dryout heat transfer coefficient. A possible explanation of this result may be attributed to most oil lubricant entrained at the post dry out region subject to high vapor shear after the post dry our region. and only a fixed amount of lubricant is deposited on the surface irrespective the lubricant concentration. As a result, the HTC remains unchanged for various oil concentration.

Analogous degradation of HTC were also reported by other investigators. For example, Gao et al. [86,87] found that HTC is decreased by about 50% (compared with pure CO_2), when PAG oil concentration was more than 0.11% in a horizontal tube with 3 mm inner diameter at a saturation temperature of 10 °C. However, an interesting feature is observed by Gao et al. [87] who found that the heat transfer coefficient reveals a strong dependence of heat flux for the near pure CO_2, indicating a dominate nucleate boiling. The results are applicable either for a smooth tube (Fig. 11(a)) or a microfin tube (Fig. 11(b)). But the strong dependence of heat flux is gone with the addition of PAG lubricant, suggesting a change of boiling mechanism from nucleate boing to convective evaporation. Gao et al. [86] noticed that in their test the PAG oil is separated from CO_2 liquid and becomes oil droplets and an oil film forms on the tube wall. This is due to PAG oil being immiscible or partially miscible with CO_2. The cavities applicable for nucleate boiling are thus filled by oil film and a suppress of nucleate boiling is encountered accordingly. Therefore, the flow boiling heat transfer changes from nucleate boiling dominated regime to convective evaporation dominated regime, and the HTC decreases significantly due to the existence of PAG oil. Tanaka et al. [88] also observed that oil concentration more than 0.7% caused a drastic deterioration of HTC about 50%. Katsuta et al. [89] investigated the flow boiling heat transfer of CO_2–PAG mixture, and found that the HTC at 5% oil concentration is about 30% lower than that at 1% oil concentration.

Effect of vapor quality: The local oil concentration increases with vapor quality due to the nonvolatile nature of the lubricant despite its partial pressure in the vapor phase is usually negligible. In the low vapor quality region, the lubricant may increase the wetted surface due to its high surface tension and viscosity or due to the foaming effect. This is quite similar to the foregoing discussion about the effect of lubricant on the conventional refrigerant. But in high vapor quality region, the mixture viscosity and local oil concentration effect are quite significant. Once an oil-rich sublayer is formed near the heating surface, it may not only suppress boiling but also may introduce additional thermal

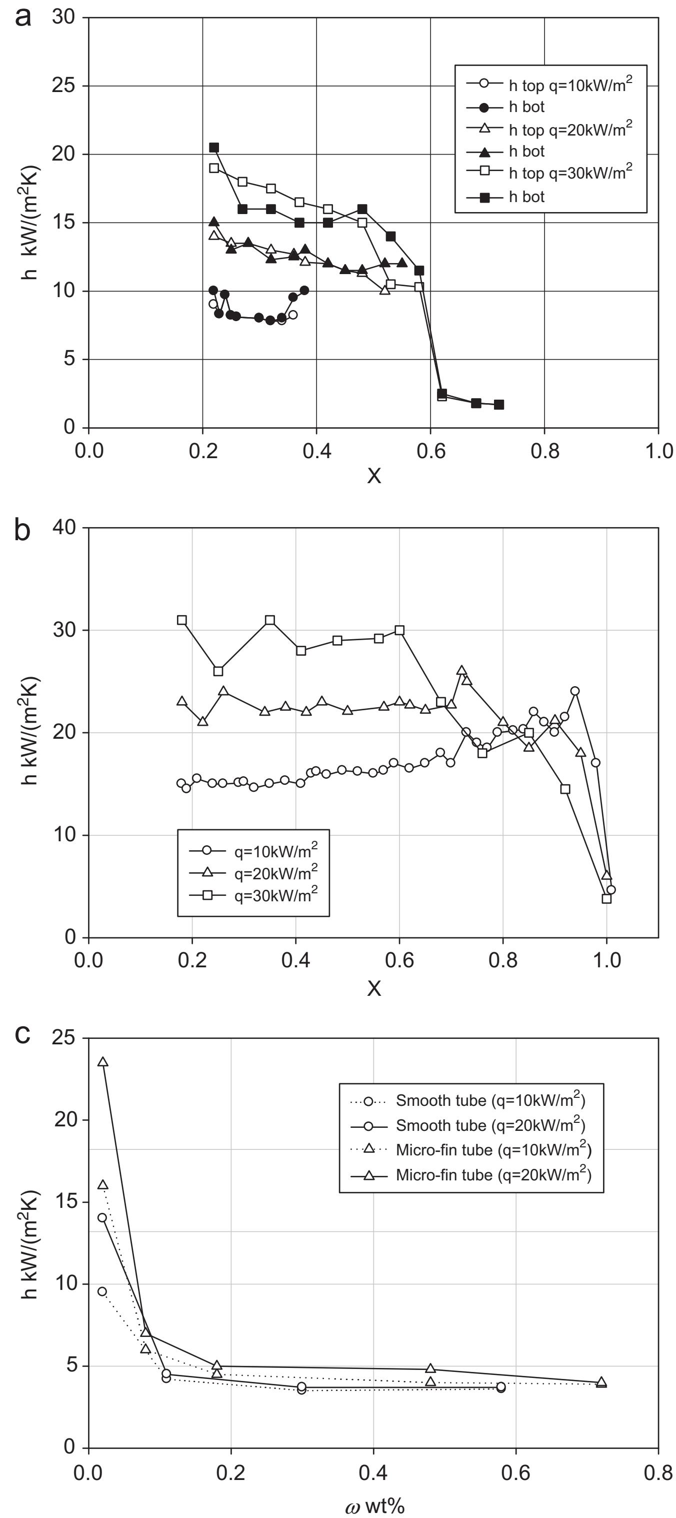

Fig. 11. Effect of heat flux on the smooth/microfin tube with and without lubricant oil [87]. (a) Effect of heat flux for smooth tube, G=780 kg/m^2s, (b) Effect of heat flux for microfin tube, G=380 kg/m^2s and (c) Effect of heat flux for smooth tube, subject to oil concentration.

resistance to the heat transfer process ([4]). Zhao et al. [90] reported considerable decrease of the boiling heat transfer coefficient of CO_2–lubricant mixture in high vapor quality region, especially for low oil concentrations less than 3%. Test results from Gao and Honda [87] and Dang et al. [85] also confirmed that the influence of oil concentration on HTC was higher at high vapor quality region. But the dryout quality and post-dryout heat transfer were not influenced by the addition of oil. At high oil concentration around 7%, Zhao et al. [90] found that HTC was nearly independent of vapor quality. They explained that as the oil concentration increases, a rich oil layer forms on the wall along the whole test tube, which prevents the contact of liquid-phase refrigerant with the wall. Similar results are also seen for Gao et al. [86].

Effect of heat and mass fluxes: Most studies on the flow boiling heat transfer of pure CO_2 refrigerant showed that nucleate boiling dominates at low/moderate vapor quality prior to dryout (e.g., Gao et al. [86] and Yun et al. [91]). However, addition of lubricant in CO_2 results in higher convective boiling contribution in the low vapor quality region. The comparison of the four synthetic lubricants for CO_2 refrigeration. Zhao et al. [90] found that the HTC of CO_2–oil mixture increased with mass flux, and this trend was more apparent at higher oil concentrations. Small oil concentrations enhanced the HTC more significantly at large mass fluxes. This phenomenon is similar to the conventional refrigerant. Test results from Gao et al. [86] also supported the foregoing statement but their test results for CO_2/lubricant mixtures using microfin tube shows even more effect of mass flux. Dang et al. [85] also reported the effect of heat flux with 1% oil concentration at two heat flux conditions. At a low heat flux of 18 kW/m^2, HTC increased significantly with mass flux in the pre-dryout region. However, no obvious difference was observed at high heat flux of 36 kW/m^2.

Effect of saturation temperature: Zhao et al. [90] experimentally investigated the effect of saturation temperature on the flow boiling heat transfer coefficient of CO_2/lubricant mixture from 0 to 15 °C. They found that a high concentration of oil (> 3%) caused a larger heat transfer degradation at high saturation temperatures. For example, at 0 °C and a vapor quality of 0.05, the difference of HTC between 1% mixture and 7% mixture was only 0.5 kW/m^2 K, but it is increased to about 4 kW/m^{2}K at 10 °C and 7 kW/m^{2}K at 15 °C. The experimental data of Hassan [92] also found that the effect of lubricant on HTC decreased with decreasing saturation temperature. It is well known from many monographs (e.g., Collier and Thome [93]) that the nucleate boiling HTC increased with the increasing saturation temperature, indicating higher contribution of nucleate boiling. On the other hand, the ratio of surface tension amid refrigerant/lubricant mixtures and pure refrigerant is also increased with the saturation temperature as shown in Fig. 1. In this sense, the increased surface tension jeopardized the nucleate boiling as explained in Eq. (1), thereby leading to the deterioration of HTC increases with increasing saturation temperature.

Effect of tube configuration: Koyama et al. [94] compared the oil effect on the flow boiling heat transfer of CO_2 in smooth and micro-fin tubes. They found that the deterioration rate of the heat transfer coefficient in micro-fin tube is smaller than that in smooth tube. They argued that micro-fin tubes can activate annular flow or semi-annular flow and suppress the foaming effect, leading to a gradual decrease of HTC with oil concentration.

Effect of microchannel: Siegismund [95] found a detectable reduction of the heat transfer coefficient of CO_2 with respect to POE lubricant during evaporation at 5°C in microchannels (13x ID 0.8 mm). Fig. 12 shows that the reduction is dependent on the amount of oil, and at ω=3% the heat transfer is reduced by around 20%, while the reduction is around 50% when the oil concentration is increased to 9%. Though most studies for CO_2 reported a decline of HTC with addition of lubricant. However, an unusual phenomenon has been observed by Zhao et al. [90] for CO_2–lubricant mixture flow boiling in a micro-channel at T_s=10 °C. They found that large oil concentrations degrade the heat transfer coefficient significantly, for example, HTC with 7% oil concentration was 60% lower than that of pure CO_2, and by contrast, smaller oil concentrations (< 3%) at low vapor qualities (x < 0.45) a marginally rise of the heat transfer coefficient (about 5% to 10%) is seen (shown in Fig. 13). The moderate augmentation of the heat transfer coefficient may be attributed by the presence of oil for (i) promoting an earlier onset of annular flow and (ii) enhancing the nucleate boiling.

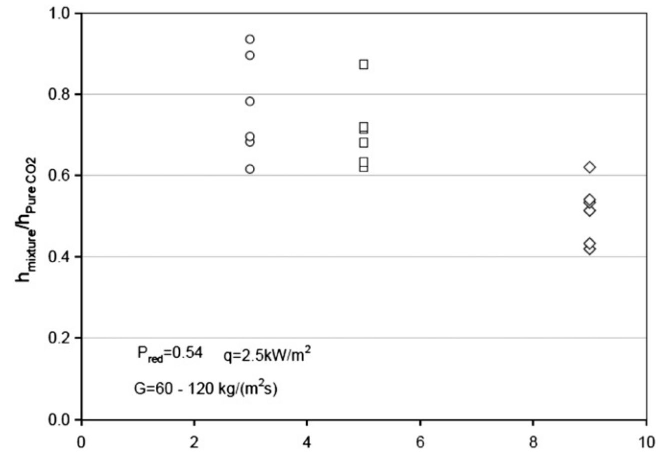

Fig. 12. Reduction of heat transfer as a function of oil content (by mass) at reduced pressure of 0.54 (=5 °C evaporation temperature), mass flux 60–120 kg/m^2 s and heat flux of 2.5 kW/m^2. Comparison (ratio) at equal vapor fractions [96].

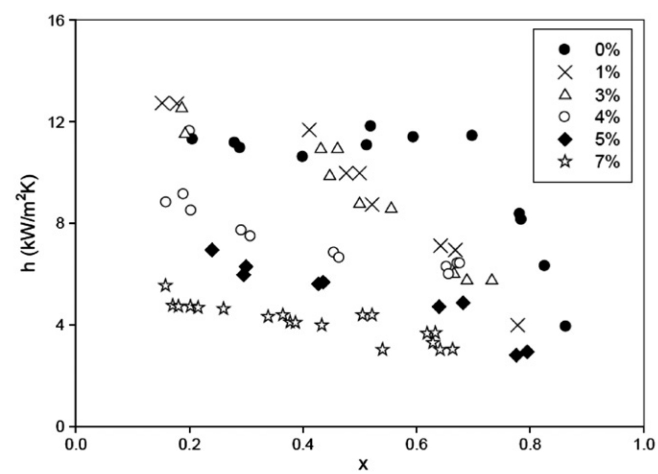

Fig. 13. Effect of oil concentration on HTC at G=300 kg/m^2 s, T_s=10 °C and q=11 kW/m^2 [90].

3.2. Effect of lubricant on R744 at supercritical state

Dang et al. [96–100] had conducted a systematic study about the effect of lubricant (PAG) on the heat transfer performance of CO_2 at supercritical state. Their test tube size were, 2, 4, and 6 mm, respectively with mass flux ranging from 200 to 1200 kg/m^2 s and oil concentration from 1% to 5%. They also performed flow visualization pertaining to lubricant influence. A short summary of their findings are given as follow:

Effect of lubricant on flow pattern: Fig. 14 illustrates the observed flow pattern of supercritical CO_2 flowing with PAG oil under the above-mentioned experimental conditions. Here, "V" denotes supercritical CO_2, "D" denotes oil droplet, and "F" denotes oil film. The observed flow pattern are: (a) mist flow (M), where a small amount of oil droplets flow with CO_2 and no oil film is observed; (b) annular-dispersed flow (AD), where both oil droplets and an oil film are observed; (c) annular flow (A), where no or few oil droplets are observed; (d) wavy flow (W), where the oil film only exists at

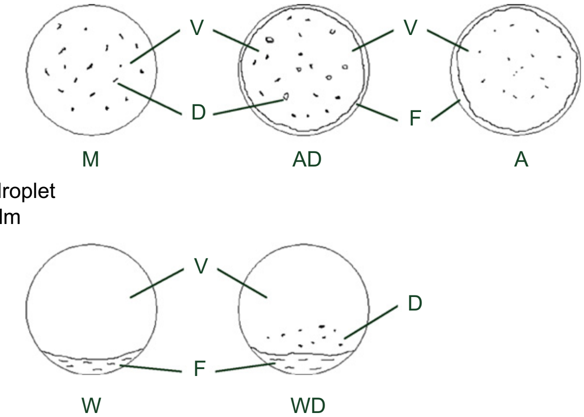

Fig. 14. Classification of flow pattern for supercritical CO_2 with lubricant. M: mist flow; AD: annular-dispersed flow; A: annular flow; W: wavy flow; WD: wavy-dispersed flow. [99].

the bottom of the cross-section; (e) wavy-dispersed flow (WD), where oil droplets are observed flowing along with an oil film at the bottom of the cross-section. The flow pattern was found changing with the temperature, pressure, and oil concentration, as well as with the tube diameter.

At a low temperature of 25 °C and an oil concentration of 1 wt%, it is clear that the flow pattern is mist flow, with oil droplets flowing along with the bulk CO_2 at a slip ratio of about 0.7. The average diameter of the oil droplets ranges from 50 to 100 μm. Under this condition, no distinct oil film flowing along the inner wall is observed. Since the solubility of CO_2 in the oil decreases with an increase in temperature, the viscosity and surface tension of the oil droplets increase with the temperature. As a result, the separated oil appears to adhere to the inner wall and forms an oil-rich layer at a high temperature with 20–60 wt% CO_2 dissolved inside it; this layer is visible as stripes at a temperature of 30 °C. With a further increase in temperature, the oil-rich layer becomes much thicker, and the oil droplets in the bulk region is not shown. The layer moves at a very low speed, resulting in a decrease in the heat transfer coefficient. A comparison of the flow pattern at tube diameters of 2 and 6 mm tubes is made and it is obvious from the comparison that the oil film for 2 mm tube is comparatively thicker than that of 6 mm tube, and the oil droplets are much larger with smaller number density. The high viscosity of the deposited lubricant film give rise to the decreased turbulent disturbance, and becomes more severe with the decrease in tube diameter when compared under the same mass flux, temperature and pressure condition. As a consequence, the thicker oil film for small sized tube corresponds to a much more significant effect of increased thermal resistance from lubricant. On the other hand, the oil droplets flowing in the bulk region do not contribute significantly to the heat transfer deterioration [99].

Effect of lubricant on HTC subject to tube diameter: The effect of tube diameter on the HTC can be seen from Fig. 15. By introducing lubricant into CO_2, it is seen that smaller diameter tube (2 mm) suffers more deterioration than the larger diameter tube (6 mm). And the decline is especially vivid near the pseudocritical temperature. For an oil concentration of 5% and a smaller diameter of 2 mm (Fig. 15(a)), it appears that the drop in the heat transfer coefficient at 50 °C is much larger than that at 30 °C. From the visual observation, it is found that the flow pattern at low temperature is mist flow with a considerable amount of oil flowing alongside the bulk area and a very thin oil film. At high temperature, a thick oil film is observed, which corresponds to a sharp decrease in the heat transfer coefficient. The distinct difference of the flow pattern is also related to comparatively rise of surface tension of lubricant which inevitably improve the wettability of the refrigerant/lubricant mixtures and lead to an oil film. This visual observation implies that the heat resistance due to the formation of the oil film is the main reason for the heat transfer deterioration. Fig. 15(b) and (c) also shows a comparison of the heat transfer coefficient and flow pattern for the 6 mm ID tube at two mass fluxes of 200 and 800 kg/m^2 s. The heat transfer coefficient decreases significantly at 800 kg/m^2 s due to the heat

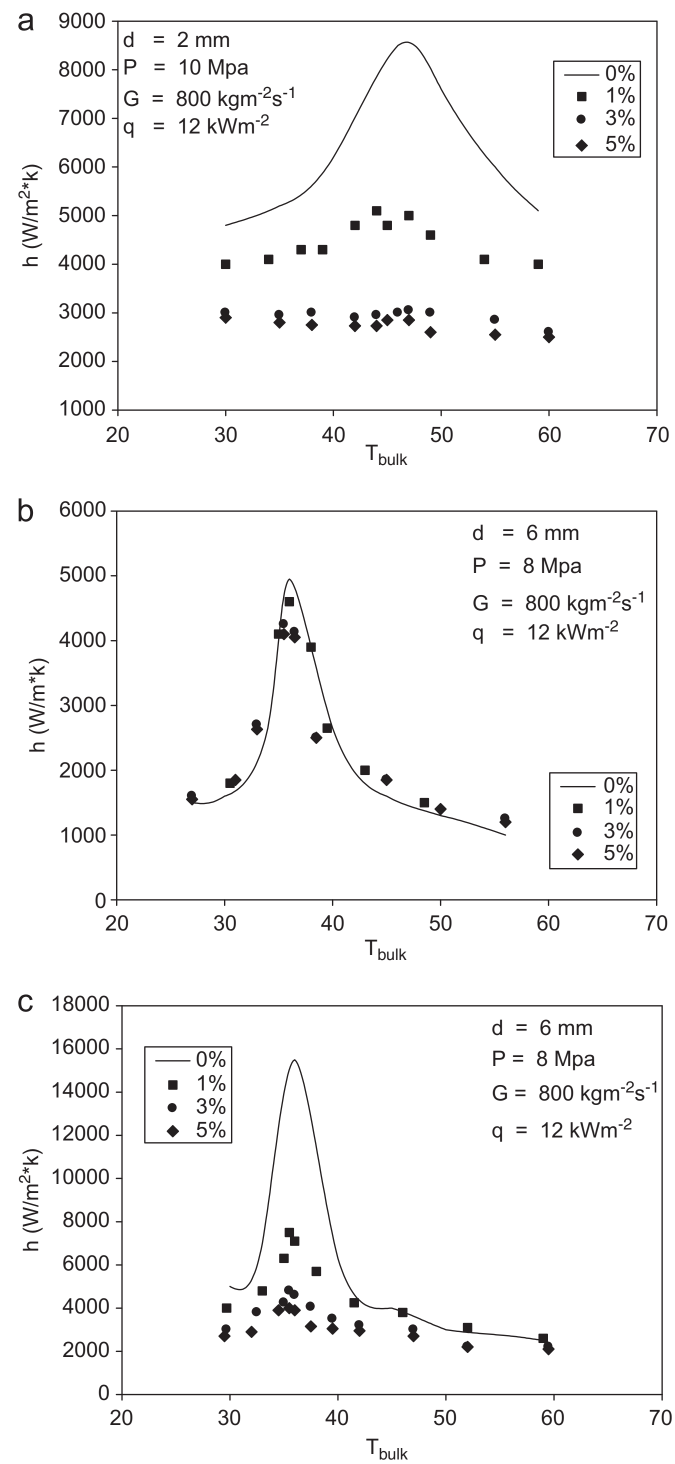

Fig. 15. – Effect of tube diameter on the heat transfer performance of CO_2–oil [99]. (a) 2 mm ID tube, (b) 6 mm ID tube, G=200 kg/m^2s and (c) 6 mm ID tube, G=200 kg/m^2s.

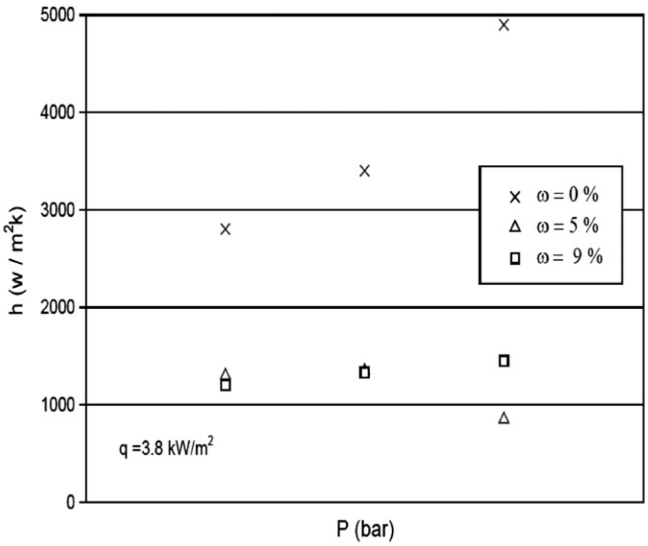

Fig. 16. – Heat transfer coefficient at gas cooling at 3.8 kW/m^2 and 60 kg/m^2 s at 0%, 5% and 9% PAG oil by mass [95].

transfer resistance of the oil film flowing along inner wall. By contrast, the drop of HTC is less pronounced when G=200 kg/m^2 s even for an oil concentration of 5%. From their flow observation, it can be attributed to the presence of wavy flow at a low mass flux and the oil film is present only at the bottom of the tube as opposed to entire film flow around the perimeter at a higher mass flux. In this regard, a significant drop in heat transfer performance is seen at a higher mass flux.

Effect of lubricant on HTC for grooved tube: Dang et al. [100] performed an experimental study to examine the PAG oil on a 2 mm grooved tube having a helix angle of 6.3°. with G ranging from 400 to 1200 kg/m^2 s. The drop in the heat transfer coefficient was 30–50% and 50–70% at oil concentrations of 1% and 3%, respectively. The heat transfer coefficient of CO_2 in the grooved tube was higher than that in the smooth tube at all tested oil concentrations. There are several reasons for the comparatively small deterioration of HTC in microfin tube, including a large heat transfer area, breaking up of the oil film by the grooved configuration, and the oil film is normally so thin to entirely flood the grooved fin.

Effect of lubricant on HTC for microchannel: Siegismund [95] investigated the heat transfer in microchannels at various POE oil concentrations. As shown in Fig. 16, the heat transfer is drastically reduced when oil is present during heat rejection. In Fig. 16, one can see the heat transfer coefficient is only 20% of the pure CO_2 at a pressure of 100 bar. The results are in line with the findings of Dang et al. [99] where smaller tube shows more reduction on HTC due to comparatively thicker of lubricant film. And test results for mini-channels from Yun et al. [101] also revealed similar characteristics.

4. Conclusions

The present review provides an overview of the lubricant on the heat transfer performance, including nucleate boiling, convective boiling, shell side condensation, forced convective condensation, and gas cooling, for conventional refrigerants and for natural refrigerant R-744. There are various parameters affecting the heat transfer coefficient subject to the presence of lubricant, such as oil concentration, heat flux, mass flux, vapor quality, geometric configuration, saturation temperature, thermodynamic and transport properties. 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 effect accompanying with the heat transport process. In addition, albeit only tiny amount of lubricant, introduction of lubricant may lead to change of flow pattern. This is because significant difference of properties such as viscosity and surface tension amid refrigerant and lubricant. 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. For the effect of lubricant on conventional refrigerant (CFC, HCFC, and HFC), the lubricant effect on HTC is quite complex, especially for pool and convective boiling. The lubricant can, increase or impair the heat transfer performance depending on the oil concentration, surface tension, surface geometry, and the like. Some possible explanations are summarized in the study but qualitative influence upon the heat transfer coefficient is still limited. For the condensation, it is more well accepted that the presence of lubricant normally will impair the heat transfer performance due to deposited oil film. However, the deterioration is comparatively smaller than that in nucleate/convective boiling. For the effect of lubricant on the heat transfer performance of R-744, reported literatures include the lubricant effect on convective evaporation and on supercritical gas cooling. For convective evaporation, the general behavior is in line with that of the convectional refrigerant. For gas cooling, the lubricant cast significant effect on heat transfer coefficient especially for a higher mass flux or at a smaller diameter tube.

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