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Critical review of flow boiling heat transfer of CO_2–lubricant mixtures

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A R T I C L E I N F O

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A B S T R A C T

This paper presents a comprehensive review on the flow boiling heat transfer of CO_2–lubricant mixtures. Some of the immiscible lubricants in CO_2 include alkyl naphthalene/alkylbenzne (AN/AB) and polyalphaolefin (PAO), while polyalkylene glycol (PAG) is partially miscible, and polyol ester (POE) is completely miscible. The effect of oil concentration, vapour quality, heat and mass fluxes and saturation temperature is addressed. One database has been created by collecting the experimental data from the open literature on the flow boiling heat transfer of CO_2–lubricant mixtures, along with empirical correlations. A simple simulation model has been developed in EES software package to compare the empirical correlations with the CO_2–lubricant mixtures experimental database. Most empirical correlations fail to predict the flow boiling heat transfer coefficient in good agreement with the experimental data. Hence, further research is needed to develop appropriate correlations for the flow boiling heat transfer of CO_2–lubricant mixtures.

1. Introduction

Carbon dioxide (CO_2) is emerging as one of the most promising environmentally friendly and energy efficient refrigerants due to the environmental concerns and corresponding regulations. Recent research on CO_2 heat exchangers has been related to the development of heat transfer empirical correlations for evaporation and gas cooling processes. Considerable research [1–4] has been performed on the flow boiling heat transfer of pure CO_2, particularly at high saturation temperatures, which is suitable for air conditioning systems. However, due to the varying experimental conditions, it is difficult to compare the experimental data from different papers. The review papers [3,4] have revealed that the flow boiling heat transfer coefficient of pure CO_2 at the same saturation temperature, is higher than the other traditional refrigerants due to its low surface tension, while the pressure drop is considerably lower due to its low viscosity. Several correlations have been developed specially for CO_2, e.g. Yoon et al. [5], Choi et al. [6] and Thome and Hajal [7]. These correlations agree well with their own experimental database, but disagree considerably with the other CO_2 databases. Meanwhile, other empirical correlations, such as Kandlikar [8], Liu and Winterton [9], and Gungor and Winterton [10], are still widely used to predict the flow boiling heat transfer of CO_2.

Evaporators of a vapour compression refrigeration system are subject to the flow boiling of refrigerant–lubricant mixtures, which tend to have high viscosity and preferential evaporation of pure refrigerant in comparison to the lubricant. The flow boiling heat transfer of CFC/HCFC refrigerant–lubricant mixtures has been reviewed by Shen and Groll [11]. The flow boiling heat transfer coefficient was found to drop rapidly when oil concentration exceeded 5%. This effect is particularly pronounced at high vapour qualities. However, there is no common agreement on the effect of oil at lower oil concentrations. Literature review [12–20] proposed different empirical correlations to predict the flow boiling heat transfer of various refrigerant–lubricant mixtures, however, none of these studies presented any experimental data of CO_2–lubricant mixtures.

This paper addresses the limitations of the currently available lubricants for CO_2 systems, and reviews the experimental studies on the flow boiling heat transfer of CO_2–lubricant mixtures, including the effect of oil concentration, vapour quality, heat and mass fluxes. Furthermore, the prediction correlations for refrigerant–lubricant mixtures are reviewed, along with the comparisons of the empirical correlations and the experimental database using EES [21] simulation model.

2. Lubricants for CO_2 refrigeration systems

2.1. Lubricants for CO_2 refrigeration systems

Proper lubricant selection depends on the operation of the proposed system. 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. The poor solubility of the synthetic hydrocarbons is compensated by their excellent

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Nomenclature

Bo	Boiling number
Co	convective boiling number
Cp	specific heat (J/kg K)
d_i	inner tube diameter (m)
$E_{l,mix}$	two-phase heat transfer multiplier
EF	enhanced factor
F	multiplying factor
Fr	Froudle number
G	mass flux (kg/m^2 s)
h	boiling heat transfer coefficient (W/m^2 K)
Ja	Jacob number
K	constant number
$\dot{m}$	mass flow rate (kg/s)
M	molecular weight (kg/kmol)
P_c	critical pressure
p_r	reduced pressure
q	heat flux (W/m^2)
Re	Reynolds number
s	specific gravity
T	temperature (°C or K)
T_b	oil NBP temperature
T_c	critical temperature
w	oil mass fraction (kg/kg)
w_o	nominal oil mass fraction (kg/kg)
x	vapour quality
X_{tt}	Lockhart–Martinelli parameter

Greek symbols

ρ	density
θ_{dry}	dry angle
δ	film thickness (m)
μ	dynamic viscosity (N s/m^2)
ν	kinematic viscosity, (mm^2/s)
ε	void fraction
λ	thermal conductivity (W/m K)

Subscripts

bub	bubble
cb	convective boiling
exp	experimental data
l	liquid state
local	local parameter
mix	refrigerant–lubricant mixture
nb	nucleate boiling
o	oil
pre	prediction by empirical correlations
ref	refrigerant
sat	saturation
tp	two phase
v	vapour phase
wet	wet wall

low temperature flow properties, which can be improved further by blending with more miscible lubricants [22]. In low-temperature NH_3/CO_2 cascade system, PAO oils are generally used with very large oil separators on the compressor discharge [23].

Kawaguchi et al. [24] reported that polyalkylene glycol (PAG) was the primary lubricant for CO_2 systems since it is partially miscible with CO_2. Seeton and Fahl [25] found that PAG seems to give the best lubricity for trans-critical applications, and at the same time, PAG is not miscible with CO_2 at high concentrations.

Li and Rajewski [26] performed a screening study and found that polyol ester (POE) lubricant was completely miscible with CO_2. Ma et al. [27] suggested that POE was better than other lubricants for trans-critical CO_2 system, while Renz [28] 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 behaviour, acceptable solubility properties and favorable miscibility.

The comparison of the four synthetic lubricants for CO_2 refrigeration systems is summarized in Table 1, along with the advantages and disadvantages of each lubricant.

2.2. Thermal properties of refrigerant–lubricant mixtures

The addition of oil in refrigerants changes the thermal properties of refrigerants. The important properties which are related with the flow boiling heat transfer are the surface tension, viscosity, heat conduction and heat of solution. Generally, if the oil is miscible with the refrigerants and the mixture of oil and refrigerant behaves as an ideal mixture, the addition of oil into the refrigerant will increase its density, surface tension, viscosity and heat conductivity. But Wallner and Dick [29], Henrici and Hesse [30] and Hambræus [31] presented a decrease in the surface tension at low oil concentrations. This decrease is used to explain the increase of the heat transfer coefficient at low oil concentrations. Unfortunately, there was no paper in the literature that deals with the thermal properties of CO_2–lubricant mixtures. Therefore, this paper presents the empirical equations to calculate lubricants and refrigerant–lubricant mixture properties in Appendices A and B. These correlations reveal that the CO_2–lubricant mixture surface tension and viscosity increase greatly with oil concentration. At oil concentration 5%, the surface tension is about 3.5 times than that

Table 1
Comparison of four synthetic lubricants for CO_2 systems

Oil type	Miscibility/solubility	Lubricity	Suitable systems	Challenges of CO_2 systems
Alkyl naphthalene/ alkylbenzne (AN/AB)	Completely immiscible	Good	–	–
Polyalphaolefin (PAO)	Completely immiscible	Normal	Cascade systems	• Large oil separator on the compressor discharge is needed
Polyalkylene glycol (PAG)	Partially miscible (miscible with CO_2 at high concentration)	Good	Trans-critical systems	• Density of CO_2 is higher than oil, oil separation and return systems are needed to avoid long term oil accumulation
Polyol ester (POE)	Miscible	Good	1. Trans-critical systems 2. Cascade systems	• Dry carbon dioxide is needed • Mixture viscosity reduces significantly with temperature increase • Long-term stability is low

of pure CO_2, and the viscosity is about 1.5 times. In addition, the mixture density increases by 1% and 10% when oil concentration increases from 0% to 10% and 70%, respectively. The mixture specific heat decreases by about 5% when oil concentration is increased to 10%.

3. Current experimental studies on CO_2–lubricant mixtures

Table 2 summarizes the current experimental studies on the flow boiling heat transfer of CO_2–lubricant mixtures in horizontal tubes. Experiments were performed at saturation temperatures ranging from −30 to 15 °C with tube diameters ranging from 1.0 to 10.06 mm, including micro-fin tubes and micro-channels. In most of the experimental studies, the lubricants were miscible or partially miscible with CO_2. Unfortunately, except viscosity, other oil properties were not given in the open literature.

3.1. Effect of oil concentration

Dang et al. [32] investigated the flow boiling heat transfer of CO_2–PAG mixture in horizontal smooth tubes with oil concentrations from 0.5% to 5.0%. They found that the addition of a small amount of lubricant resulted in a sharp decrease in the heat transfer coefficient. h_{tp} reduced from 8–9 to 3–5 kW/m^2 K when oil concentration increased from 0% to 0.5%. However, further increase in oil concentration from 0.5% to 5% had almost negligible effect on h_{tp}, as can be seen in Fig. 1. They suggested the critical oil concentration to be 0.5% for 2 mm inner diameter tube and 1% for 4 mm inner diameter tube. Furthermore, the addition of lubricant did not seem to influence the dryout quality and the post-dryout heat transfer coefficient.

Gao et al. [33,34] experimentally investigated the flow boiling heat transfer of CO_2–PAG mixture in a horizontal tube with 3 mm inner diameter and 2.185 m in length at saturation temperature of 10 °C. The test section was heated using a direct heat method. They found that h_{tp} decreased by about 50% (compared with h_{tp} of pure CO_2), when oil concentration was more than 0.11% (see Fig. 2).

Tanaka et al. [35] also observed that oil concentration more than 0.7% caused a drastic deterioration in the heat transfer and h_{tp} dropped by about 50% (compared with h_{tp} of pure CO_2). Katsuta et al. [36] investigated the flow boiling heat transfer of CO_2–PAG mixture, and found that h_{tp} at 5% oil concentration is about 30% lower than that at 1% oil concentration.

Hassan [37] investigated the flow boiling heat transfer of CO_2–PAG mixture in a horizontal tube with inner diameter 10.06 mm and 1.12 m in length, and saturation temperature ranging from −10 to −30 °C. In their experiments, when the oil concentration was increased from 0% to 7%, h_{tp} decreased continually at low saturation temperatures of −25 and −30 °C. Somehow, the sharp drop with oil concentration did not occur, as was observed by other researchers. At these temperatures, the surface tension and viscosity of CO_2 are higher than that at high saturation temperatures, so the effect of oil decreases.

Unusual phenomenon has been observed by Zhao et al. [38] for CO_2–lubricant mixture flow boiling in a micro-channel at $T_{sat} = 10$ °C. They found that large oil concentrations degrade the heat transfer coefficient significantly, for example, h_{tp} with 7% oil concentration was 60% lower than that of pure CO_2, while on the contrary, smaller oil concentrations (<3%) at low vapour qualities ($x < 0.45$) moderately enhance the heat transfer coefficient (about 5% to 10%) (shown in Fig. 3). The moderate augmentation of the heat transfer coefficient may be attributed by the presence of oil (i) promoting an earlier onset of annular flow or without the presence of foaming effect; and (ii) enhancing the nucleate boiling [11].

Koyama et al. [39] 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 in smooth tube (shown in Fig. 4). Micro-fin tubes can activate annular flow or semi-annular flow automatically, and suppress the foaming effect. Therefore, micro-fin tubes experience a gradual decrease of h_{tp} with oil concentration, as compared with smooth tubes.

Table 2
Experimental studies and database on flow boiling heat transfer of CO_2–lubricant mixtures in horizontal tubes

References	Channel configuration	Tube diameter (mm)	Mass flux (kg/m^2 s)	Heat flux (kW/m^2)	T_{sat} (°C)	Vapour quality	Oil type	Miscibility	Oil concentration (%)	Number of data points
Dang et al. [32]	Stainless steel Smooth tube	2, 4, 6	170–320	4.6–36	15	0.1–0.9	PAG	Partially Miscible	0–5	48
Gao et al. [33]	Smooth steel Micro-fin copper	3 3.04	190–1300	5–30	10	0.2–1.0	PAG	Partially Miscible	0.01–0.72	45
Gao and Honda [34]	Stainless steel Smooth tube	3	200–1300	10–30	10	0.2–1.0	PAG	Partially Miscible	0.01–0.57	124
Tanaka et al. [35]	Stainless steel Smooth tube	1	360–1440	36	15	N/A	PAG	Partially Miscible	0–1	–
Katsuta et al. [36]	Stainless steel Smooth tube	4.59	400–800	5–15	10	0.1–1.0	PAG	Miscible	1–5	51
Hassan [37]	Stainless steel Smooth tube	10.06	90–125	5.0–16.5	−30 to −10	0.12–0.9	N/A[a]	Miscible	0–7	56
Zhao et al. [38]	Micro-channel	0.86 (hydraulic)	100–700	11	0–15	0.1–0.9	N/A[b]	Miscible	0–7	69
Koyama et al. [39]	Smooth copper tube Micro-fin copper tube	4.42 4.90	360–650	N/A	5.3	0.15–1.0	PAG	N/A	0.04–7	–

[a] Lubricant $\nu = 5.25 \times 10^{-5}$ m^2/s at 40 °C.

[b] The oil type is not given, but the oil viscosity is $\nu = 6.5 \times 10^{-5}$ m^2/s at 40 °C.

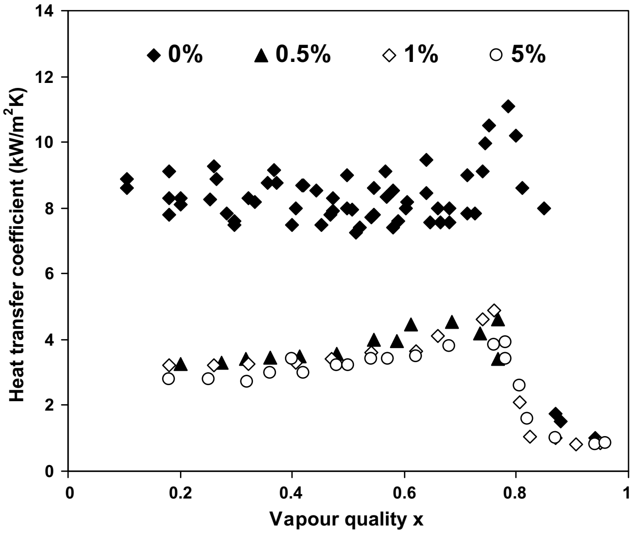

Fig. 1. Effect of oil concentration on h_{tp} of CO_2–PAG mixture (Dang et al. [32]) at $G = 360\ kg/m^2\ s$, $T_{sat} = 10\ °C$ and $q = 9\ kW/m^2$.

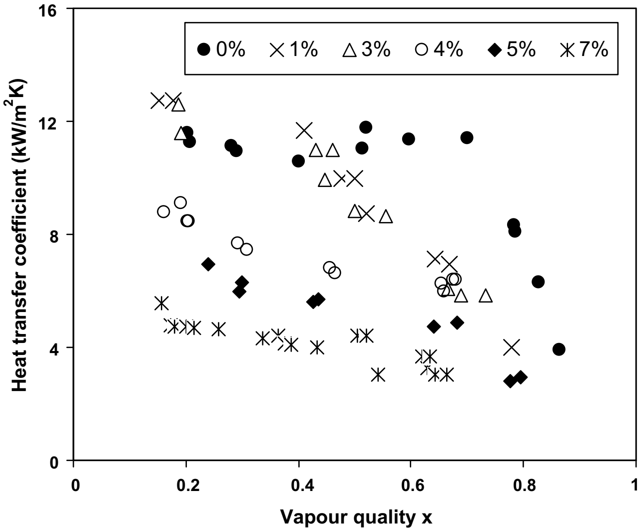

Fig. 3. Effect of oil concentration on h_{tp} (Zhao et al. [38]) at $G = 300\ kg/m^2\ s$, $T_{sat} = 10\ °C$ and $q = 11\ kW/m^2$.

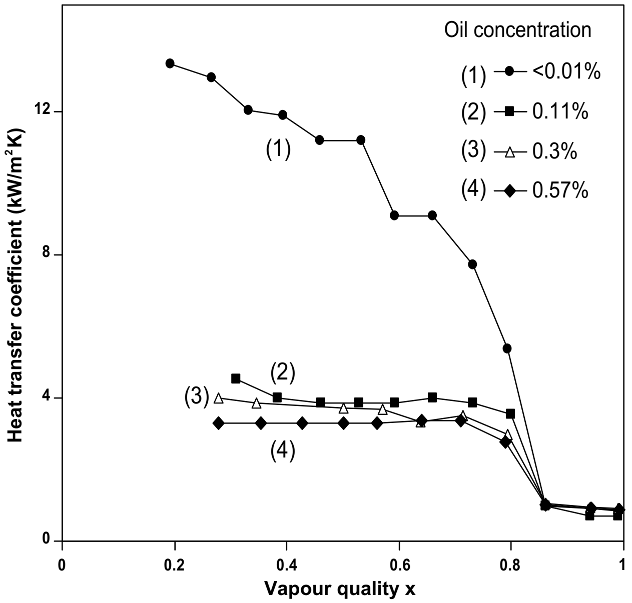

Fig. 2. Effect of oil concentration on h_{tp} of CO_2–PAG mixture (Gao and Honda [34]) at $G = 380\ kg/m^2\ s$, $T_{sat} = 10\ °C$ and $q = 20\ kW/m^2$.

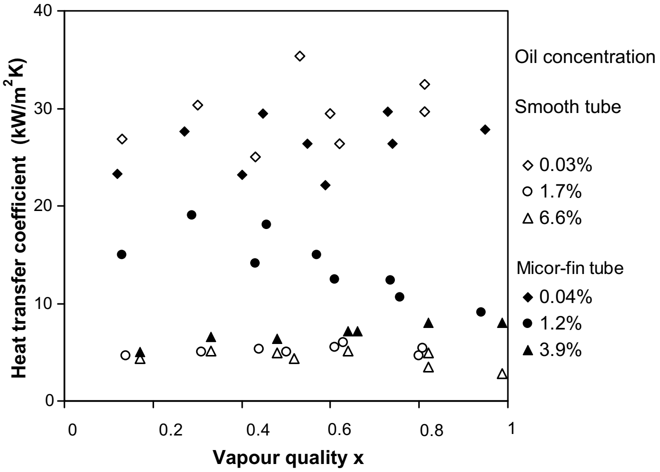

Fig. 4. Effect of oil concentration on h_{tp} of CO_2–PAG mixture (Koyama et al. [39]) at $G = 360\ kg/m^2\ s$ and $T_{sat} = 5.3\ °C$.

3.2. Effect of vapour quality

The lubricant influence on the flow boiling heat transfer is a local behaviour, and is specific to the vapour quality. The local oil concentration increases with vapour quality due to the nonvolatile behaviour of the lubricant. Its partial pressure in the vapour phase is usually negligible. In low vapour quality region, the lubricant may increase the wetted surface due to its high surface tension and viscosity or due to the foaming effect. But in high vapour 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 may also introduce additional thermal resistance to the heat transfer process [11].

Zhao et al. [38] reported considerable decrease of the boiling heat transfer coefficient of CO_2–lubricant mixture in high vapour quality region, especially for low oil concentrations less than 3%. Gao and Honda [34] and Dang et al. [32] data also confirmed that the influence of oil concentration on h_{tp} was higher at high vapour 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%, Hassan [37] and Zhao et al. [38] found that h_{tp} was nearly independent of vapour 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. Therefore, the overall h_{tp} decreases, while dryout becomes a secondary phenomenon.

3.3. 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 vapour quality prior to dryout [3,4,40,41]. However, addition of lubricant in CO_2 results in higher convective boiling contribution in the low vapour quality region.

Zhao et al. [38] found that h_{tp} of CO_2–oil mixture increased with mass flux, and this trend was more apparent at higher oil concentrations. Small oil concentrations enhanced h_{tp} more significantly at large mass fluxes. This phenomenon is similar to Cho and Tae [42] data on the flow boiling heat transfer of R22 and R407C with lubricant mixtures. They found that refrigerant and oil can mix more uniformly at high mass fluxes. This reduces the detrimental effects of oil mass transfer resistance and local oil accumulation.

Dang et al. [32] also reported the effect of heat flux with 1% oil concentration at two heat flux conditions. At low heat flux of 18 kW/m^2, h_{tp} 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. Gao and Honda [33,34] also found that h_{tp} increased remarkably with mass flux, but there was no increase with heat flux. Their experimental data are shown in Figs. 5 and 6. This may lead to the conclusion that the heat transfer has changed from nucleate boiling dominant regime to convective boiling. Hassan [37] data at low temperatures down to −30 °C showed that h_{tp} increases with both heat and mass fluxes.

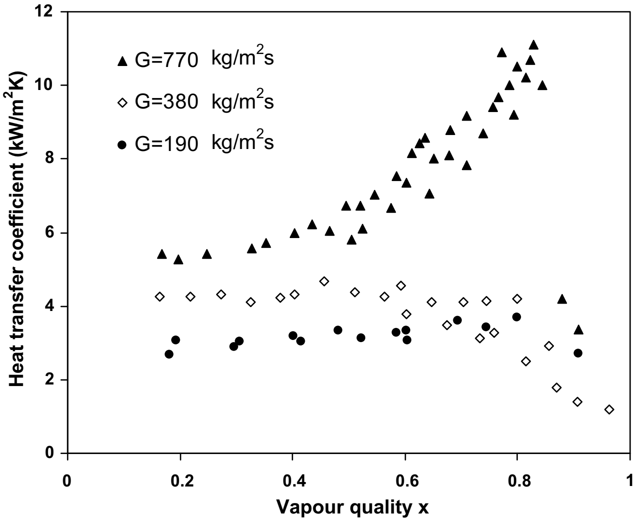

Fig. 5. Effect of mass flux on h_{tp} of CO_2–PAG mixture (Gao et al. [33]) at $w_o = 0.72$ wt%, $T_{sat} = 10$ °C and $q = 20$ kW/m^2.

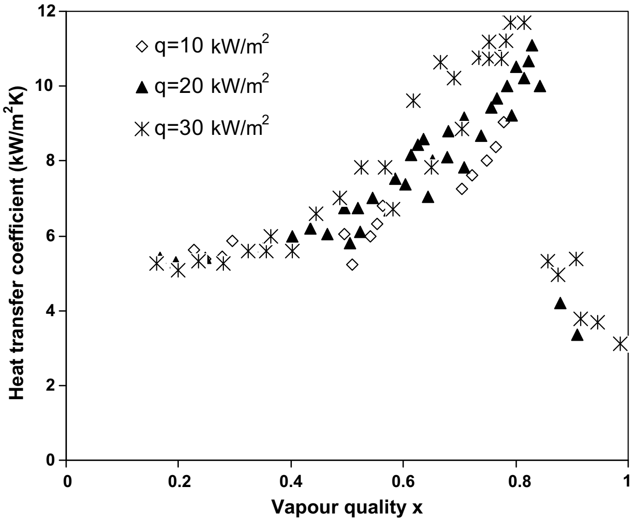

Fig. 6. Effect of heat flux on h_{tp} of CO_2–PAG mixture (Gao et al. [33]) at $w_o = 0.72$ wt%, $T_{sat} = 10$ °C and $G = 770$ kg/m^2 s.

3.4. Effect of saturation temperature

Zhao et al. [38] 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 high concentration of oil (>3%) caused a larger heat transfer degradation at high saturation temperatures. For example, at 0 °C and vapour quality of 0.05, the difference of h_{tp} between 1% mixture and 7% mixture was only 0.5 kW/m^2 K, but it increased to about 4 kW/m^2 K at 10 °C and 7 kW/m^2 K at 15 °C. In Hassan [37] experimental data at low saturation temperatures ranging from −10 to −30 °C, the oil effect on h_{tp} also decreased with decreasing saturation temperature. They explained that during the boiling process, as CO_2 boils and moves away from the surface, an oil rich sub-layer covers the heat transfer surface. As CO_2 boiling continues, bubbles form and penetrate the oil-rich sub-layer, causing turbulence in the liquid layer, significantly enhancing the heat transfer coefficient. The more the oil-rich sub-layer disturbed, the better the heat transfer will be. As saturation temperature increases, the difference in thermal properties between the liquid and vapour phase becomes smaller, and the disturbing effect on the oil-rich layer decreases. This leads to the deterioration of h_{tp} increases with increasing saturation temperature.

4. Comparison with empirical correlations for refrigerant–lubricant mixtures

4.1. Empirical correlations for refrigerant–lubricant mixtures

Two typical methods are suggested to study the lubricant influence on the flow boiling heat transfer of refrigerants [43].

The first method is the “thermodynamically right approach” [43], which treats the refrigerant–lubricant mixture to behave like a zeotropic mixture. Its performance is based on the refrigerant–lubricant mixture’s properties. In particular, this method associates the flow boiling heat transfer coefficient with the mixture bubble point temperature, as calculated in the following equation:

$$h_{tp} = \frac{q}{(T_w - T_{bub})} \tag{1}$$

Since lubricant has a much higher boiling temperature than the common refrigerants, the bubble point temperature of refrigerant–lubricant mixture is higher than the saturation temperature of pure refrigerant, and increases with vapour quality at a fixed pressure like a zeotropic refrigerant mixture. But this method can only be used for completely miscible refrigerant–lubricant and with a good knowledge of mixture properties.

The second method is the “oil contamination approach”, where the performance of refrigerant–lubricant mixture is based on the pure refrigerant properties. The heat transfer coefficients are calculated from the pure refrigerant saturation temperature

$$h_{tp} = \frac{q}{(T_w - T_{sat})} \tag{2}$$

The lubricant influence is regarded as a correction factor, which is generally a function of the oil concentration. Most of the currently available refrigerant–lubricant empirical correlations follow this method. Some correlations are reviewed and shown in Table 3, which can be divided into following three categories:

(a) The first group suggested enhanced factor, EF, which is the ratio of heat transfer coefficient of refrigerant–lubricant mixture to that of pure refrigerant. EF is calculated as the following:

Table 3

Comparison between empirical correlations of refrigerant–lubricant mixtures and the experimental database

Reference	Fluid/oil	Correlations	Average deviation[a] (%)	Mean deviation[b] (%)
Schlager et al. [19]	R22/150-SUS	Exponential: $\mathrm{EF} = 1.03 \cdot \exp(17.7w_o - 286w_o^2 - 0.0496 \cdot G/300)$	52.7	62.5
		Polynomial: $\mathrm{EF} = 1.03 + 20.4w_o - 332w_o^2 - 0.061 \cdot G/300$	59.9	68.5
	R22/300-SUS	Exponential: $\mathrm{EF} = 1.03 \cdot \exp(w_o \cdot (4.98 \cdot G/300) - 8.77)$	39.0	61.7
		Polynomial: $\mathrm{EF} = 1.03 + w_o \cdot (4.16 \cdot G/300 - 7.62)$	38.0	60.5
Eckels et al. [14,18]		Smooth tube		
	R134a/169-SUS	$\mathrm{EF} = 1.0 + 6.92w_o - 572.10w_o^2 \cdot G/250 + 304.90(w_o \cdot G/250)^2$	80.7	108.5
	R134a/369-SUS	$\mathrm{EF} = 1.0 - 7.05w_o - 31.11w_o^2 \cdot G/250 + 63.55w_o^2$	18.6	48.3
	R134a/150-SUS	$\mathrm{EF} = 1.0 - 3.96w_o - 411w_o^2 \cdot G/250 + 202.1w_o^2(G/250)^2$	29.8	78.4
		Micro-fin tube		
	R134a/169-SUS	$\mathrm{EF} = 2.69 + 17.27w_o - 1.43G/250 + 3.73w_o(G/250)^2 - 508.2w_o^2 + 0.39(G/250)^2$	114.9	120.6
	R134a/369-SUS	$\mathrm{EF} = 2.29 - 1.23w_o - 0.99G/250 + 0.76w_o(G/250) + 0.26(G/250)^2$	117.2	119.8
	R134a/150-SUS	$\mathrm{EF} = 2.45 + 29.95w_o + 1.01 \cdot G/250 + 14.87w_o(G/250) + 50.28w_o^2(G/250)^2 + 194.7w_o^2 + 0.24(G/250)^2$	824.2	824.2
Thome [16]		$\mathrm{EF} = \left[\frac{\mu_{\mathrm{ref}}}{\mu_{\mathrm{oil}}}\right]^{0.26w_{\mathrm{local}}}$ for $\dot{m} \geqslant 200\ \mathrm{kg\ m^{-2}\ s^{-1}}$ and $w_o \leqslant 0.05$	119.6	139.6
Tichy et al. [20]	R12/300-SUS	$E_{\mathrm{L.mix}} = 10^{K_1(\log_{10}X_{tt} - K_2)}$ $\quad K_1 = K_{1a}(w_o) + K_{1b}(w_o)\log_{10}(Ja)$ $\quad K_2 = K_{2a}(w_o) + 2.0\log_{10}(Ja)$	−7.0	42.1
		K_{1a}, K_{1b} and K_{2a} are empirical constants specific to oil concentration		
Cawte et al. [17]	R12, R22/130-SUS	$E_{\mathrm{L.mix}} = Co^{-0.8} \times A \quad A = 2.5 - 3.94w_o - \frac{2.73}{6.33 + 1000w_o}$	59.2	78.8
Wei et al. [12]	R22/NM56	Tube $d_i = 6.34$ mm		
		$E_{l,\mathrm{mix}} = 18.3077 + (0.9728Co^{-0.9} - 90174.5990Bo^{0.97}) \cdot Fr_l^{0.5} + 314.5576w_{lo}[x(1-x)]^{0.5}/G^{-0.1}$	5.2	624.8
		Tube $d_i = 2.50$ mm		
		$E_{l,\mathrm{mix}} = 11.1258 + (0.9216Co^{-0.9} - 80830.65Bo^{0.96}) \cdot Fr_l^{0.5} + 150.0294w_{lo}[x(1-x)]^{0.5}/G^{-0.1}$		
		$h_{l,\mathrm{mix}} = \max\left(\frac{4.364 \cdot \lambda_{l,\mathrm{mix}}}{d_i}, \frac{0.023 \cdot \lambda_{l,\mathrm{mix}} \cdot Re_{l,\mathrm{mix}}^{0.8} \cdot Pr_{l,\mathrm{mix}}^{0.4}}{d_i}\right)$	−179.8	618.5
Zurcher et al. [13]	R407C/POE	$h_{tp} = \frac{\mathrm{d_i}\theta_{\mathrm{dry}} \times h_v + d_i(2\pi - \theta_{\mathrm{dry}}) \times h_{\mathrm{wet}}}{2\pi \mathrm{d_i}}$ $\quad h_{\mathrm{wet}} = [(h_{nb})^3 + (\mathrm{F_{oil}}h_{cb})^3]^{1/3}$		
		$h_{nb} = 55p_r^{0.12}q^{2/3}(-\log_{10}p_r)^{-0.55}M^{-0.5}$ $\quad F_{\mathrm{oil}} = \left[\frac{\mu_{\mathrm{ref}}}{\mu_{\mathrm{oil}}}\right]^{0.26w_{\mathrm{local}}}$	145.4	161.9
		$h_{cb} = 0.0133\left(\frac{4G(1-x)\delta}{(1-\varepsilon)\mu_l}\right)^{0.69}\left(\frac{Cp_l\mu_l}{\lambda_l}\right)^{0.4}\frac{\lambda_l}{\delta}$ $\quad h_v = 0.023\left(\frac{mxd}{\varepsilon\mu_v}\right)^{0.8}\left(\frac{C_{p,v}\mu_v}{\lambda_v}\right)^{0.4}\frac{\lambda_v}{d}$		
Kattan et al. [44]	Pure refrigerant	$h_{tp} = \frac{\theta_{\mathrm{dry}} \times h_v + (2\pi - \theta_{\mathrm{dry}}) \times h_{\mathrm{wet}}}{2\pi}$ $\quad h_{\mathrm{wet}} = [h_{nb}^3 + h_{cb}^3]^{1/3}$ $\quad h_v = 0.023\left(\frac{mxd}{\varepsilon\mu_v}\right)^{0.8}\left(\frac{C_{p,v}\mu_v}{\lambda_v}\right)^{0.4}\frac{\lambda_v}{d}$	145.0	161.1
		$h_{cb} = 0.0133\left(\frac{4G(1-x)\delta}{(1-\varepsilon)\mu_l}\right)^{0.69}\left(\frac{Cp_l\mu_l}{\lambda_l}\right)^{0.4}\frac{\lambda_l}{\delta}$ $\quad h_{nb} = 55p_r^{0.12}q^{2/3}(-\log_{10}p_r)^{-0.55}M^{-0.5}$		
Liu and Winterton [9]	Pure refrigerant	$h_{TP} = \sqrt{(Fh_l)^2 + (Sh_{nb})^2}$ $\quad F = \left[1 + xPr_1\left(\frac{\rho_1}{\rho_G} - 1\right)\right]^{0.35}$ $\quad S = \left(1 + 0.055F^{0.1}Re_l^{0.16}\right)^{-1}$	149.8	152.0
Kandlikar [8]	Pure refrigerant	$h_{CBD} = (1.1360Co^{-0.9} + 667.2Bo^{0.7}F_{fl}) \cdot h_l$ $\quad h_{NBD} = (0.6683Co^{-0.2} + 1058Bo^{0.7}F_{fl}) \cdot h_l$	9.0	43.4
		h_{tp} is the maximum of h_{NBD} and h_{CBD}		

[a] Average_deviation $= \frac{1}{N}\left(\sum_{i=1}^{N}\frac{h_{tp,\mathrm{predicted}} - h_{tp,\mathrm{experiment}}}{h_{tp,\mathrm{experiment}}}\right) \times 100\%$.

[b] Mean_deviation $= \frac{1}{N}\left(\sum_{i=1}^{N}\frac{\lvert h_{tp,\mathrm{predicted}} - h_{tp,\mathrm{experiment}}\rvert}{h_{tp,\mathrm{experiment}}}\right) \times 100\%$.

$$\mathrm{EF} = h_{\mathrm{tp,mix}}/h_{\mathrm{tp,ref}} \tag{3}$$

where $h_{\mathrm{tp,ref}}$ is the flow boiling heat transfer coefficient of pure refrigerant. Correlations of Eckels et al. [14,18] and Schlager et al. [19] suggested different EF to correct the pure refrigerant heat transfer directly based on their experimental data. Kandlikar [8] correlation was suggested by them to calculate $h_{\mathrm{tp,ref}}$. In their equations for enhanced factor EF, only nominal oil concentration was included, while the oil properties were not considered. Thome [16] correlation introduced oil viscosity and local oil concentration for evaluating the refrigerant–lubricant mixture boiling heat transfer coefficient.

(b) The second group introduced two-phase heat transfer multiplier, $E_{\mathrm{l,mix}}$, which is the ratio of two-phase heat transfer coefficient of refrigerant–lubricant mixture to liquid-phase heat transfer coefficient of refrigerant–lubricant mixture. $E_{\mathrm{l,mix}}$ is calculated as below:

$$E_{\mathrm{l,mix}} = h_{\mathrm{tp,mix}}/h_{\mathrm{l,mix}} \tag{4}$$

Tichy et al. [20], Cawte [17] and Wei et al. [12] proposed different $E_{\mathrm{l,mix}}$ to satisfy their experimental data. Liquid-phase boiling heat transfer coefficient was calculated using Dittus–Boelter equation and the refrigerant–lubricant mixture properties. Zurcher et al. [13] correlation suggested F_{oil} to correct only the convective boiling equation of Kattan et al. [44] (Zurcher et al. [13] in Table 3).

(c) In the third group, several researchers tried to use a pure refrigerant empirical correlation together with the refrigerant–lubricant mixture viscosity to predict the refrigerant–lubricant mixture heat transfer coefficient. For example, Zurcher et al. [13] suggested to use refrigerant–lubricant mixture viscosity to replace the refrigerant viscosity of Kattan et al. [44] correlation.

4.2. Evaluation of existing empirical correlations

4.2.1. Empirical correlations for flow boiling heat transfer of pure CO_2

Several researchers have studied the flow boiling heat transfer of pure CO_2, along with empirical correlations. Thome and Ribatski [4] evaluated the empirical correlations with pure CO_2 experimental database, and found that Thome and Hajal [7] correlation fits the best. Zhao and Bansal [1,2,45] found that Liu and Winterton [9] and Kandlikar [8] correlations agree reasonably well with pure CO_2 experimental data at low temperatures down to −40 °C.

In this paper, flow boiling heat transfer data of pure CO_2 as well as CO_2–lubricant mixture {from Dang et al. [32], Zhao et al. [38], Hassan [37] and Gao et al. [33,34]} was collected and compared with widely used empirical correlations for pure CO_2, e.g. Gungor and Winterton [10], Thome and Hajal [7], Kattan et al. [44], Liu and Winterton [9], and Kandlikar [8] correlations. It is found that these correlations predict the current pure CO_2 experimental data with average deviations of 50.6%, −40.8%, −24.0%, 15.4% and −13.7%, respectively. Thus the last three correlations are chosen here to compare their predictions with the CO_2–lubricant mixture data by replacing pure CO_2 viscosity with the mixture viscosity in the correlations.

4.2.2. Comparisons of empirical correlations with CO_2–lubricant mixtures data

The open literature search reveals that there is no study that compares the predictions of any refrigerant–lubricant mixture empirical correlations (shown in Table 3) with the experimental data of CO_2–lubricant mixtures. Therefore, this study attempts to compare the existing empirical correlations predictions against a broad experimental database of CO_2–lubricant mixtures. An experimental database has been created by collecting experimental data on CO_2–lubricant mixtures from various publications (in Table 2), along with the number of data points from each paper. Due to the absence of the lubricant properties in [35,39], their experimental data were not included in the database. The current database has 393 points, and covers tube diameters from 2 to 10.06 mm, mass fluxes from 90 to 770 kg/m^2 s, heat fluxes from 5 to 30 kW/m^2 and saturation temperatures from −30 to −10 °C. The nominal oil concentration ranges from 0.01% to 7%.

A simulation model, (using all correlations as shown in Table 3), was developed in EES [21] software package. Pure refrigerant liquid viscosity was replaced by CO_2–lubricant mixture viscosity in empirical correlations that were developed for pure refrigerants. The average and mean deviations of each correlation with the experimental database are shown in Table 3. The simulation results illustrate that all empirical correlations have major disagreements with the CO_2 experimental data. For example, in the first group, Schlager et al. [19] correlation for 300-SUS (both exponential and polynomial) and Eckels et al. [14,18] correlation for 369-SUS and 150-SUS of smooth tube can predict h_{tp} with average deviations of 39.0%, 38.0%, 18.6% and 29.8%, respectively. For the second group, Tichy et al. [20] correlation predicts with average deviation of −7%. In the third group, Kandlikar correlation [8] is better with average deviation of 9%. But all these correlations have larger mean deviations around ±50%.

The comparisons of Eckels et al. [14,18] correlation for 369-SUS, Tichy et al. [20] correlation and Kandlikar [8] correlation with experimental data are shown in Figs. 7–9, where the average deviations are low but larger disagreements are visible in the low vapour quality region. This disagreement may be due to the stronger effect of oil concentration on h_{tp} of CO_2.

Eckels et al. [14,18] correlation for 369-SUS always overpredicted the heat transfer coefficient in the low vapour quality region for low oil concentrations. This empirical correlation was developed for R134a–lubricant mixtures, where the flow boiling heat transfer coefficient remained higher at low oil concentrations. So the predicted EF in their correlation was normally around 1.0 at low vapour quality, which was much higher than the current

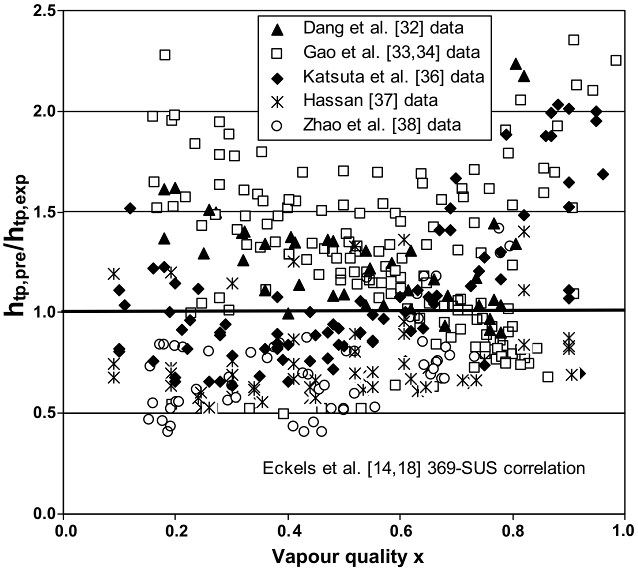

Fig. 7. Comparison of h_{tp} predicted from Eckels et al. [14,18] correlation with the experimental database shown in Table 2.

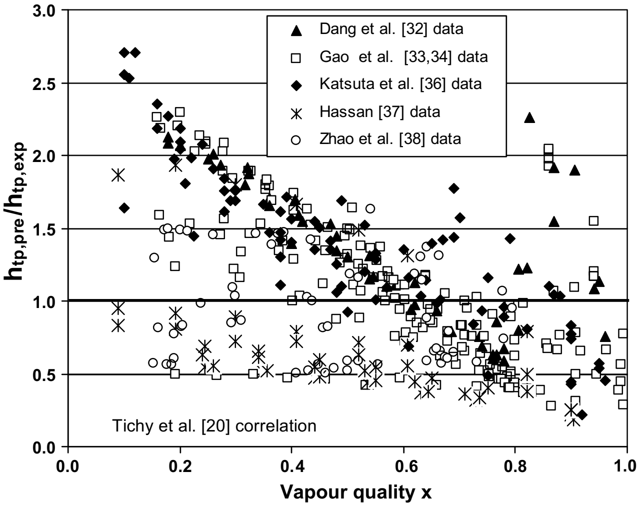

Fig. 8. Comparison of h_{tp} predicted from Tichy et al. [20] correlation with the experimental database shown in Table 2.

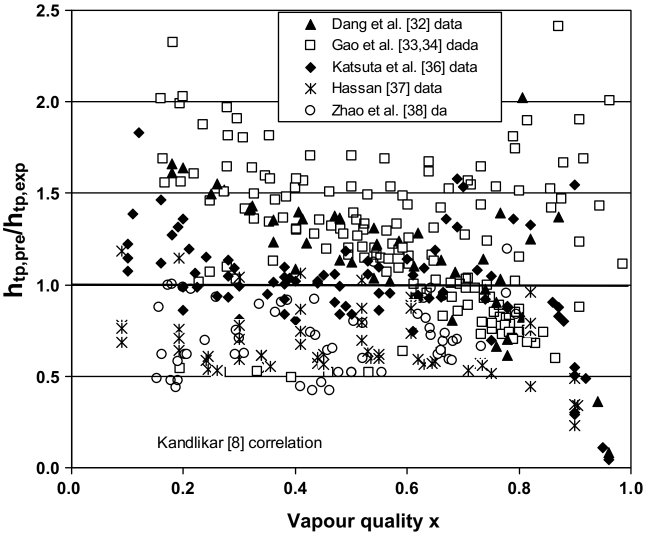

Fig. 9. Comparison of h_{tp} predicted from Kandlikar [8] correlation with the experimental database shown in Table 2.

CO_2–lubricant experimental data, and hence a larger disagreement. It may be observed from Fig. 7 that at vapour quality between 0.5 and 0.8, Eckels et al. [14,18] correlation predicts h_{tp} well. The similar trend can be found in Tichy et al. [20] correlation (shown in Fig. 8), at low vapour quality, the average deviation is around 100%, and it decreases to 50% when vapour quality is higher than 0.4.

For Kandlikar [8] correlation, CO_2–lubricant mixture viscosity is used, which increases with the oil concentration, while the predicted h_{tp} decreases continually. At low oil concentration, there is a small difference between predicted h_{tp} of pure CO_2 and CO_2–lubricant mixture. The sharp drop of experimental h_{tp} with very low oil concentration can not be predicted by the empirical correlation. One comparison of Dang et al. [32] data with Kandlikar [8] correlation's prediction is shown in Fig. 10. When h_{tp} decrease sharply at oil concentration of 0.5% in the experimental data, the predicted flow boiling heat transfer coefficients from Kandlikar [8] are almost the same with the predictions of pure CO_2. Kandlikar

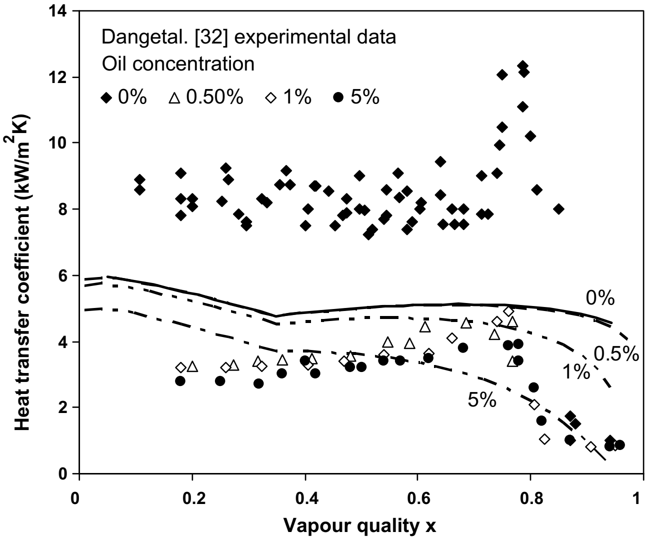

Fig. 10. Comparison of h_{tp} predicted from Kandlikar [8] correlation with the experimental database of Dang et al. [26] at $T_{sat} = 15$ °C.

[8] can predict better when oil concentration increases to 5%, the average and mean deviations are 7% and 24.1%, respectively.

By analyzing the overall statistical parameters presented in Table 3 and Figs. 7–10, it can be noted that different correlations may predict some particular experimental data well, but the overall prediction is not good, especially in low vapour quality region and low oil concentrations. Overall, Eckels et al. [14,18] for 369-SUS and Kandlikar [8] correlations are better, but still have large disagreements, and hence new correlations need to be developed to predict the sharp drop of the flow boiling heat transfer coefficient of CO_2–lubricant mixtures at very low oil concentrations.

5. Conclusions

A comprehensive review of the flow boiling heat transfer of CO_2–lubricant mixtures has been presented in this paper, by analyzing previous experimental data, the empirical correlations and comparisons of these empirical correlations with the experimental database. From this review, the following conclusions can be drawn:

1. Further investigations are needed on CO_2–lubricant mixture properties to explain the CO_2–lubricant mixture heat transfer behaviour.
2. In general, the addition of lubricant in CO_2 sharply decreases the flow boiling heat transfer coefficient, even with very low oil concentrations. At high oil concentrations, the flow boiling heat transfer coefficient is almost independent of vapour quality. The deterioration of the flow boiling heat transfer due to lubricant increases with increasing saturation temperature. However, limited information is available on the topic in the open literature, and hence an in-depth research is needed to understand the boiling heat transfer of CO_2–lubricant mixtures.
3. Widely used correlations (in the literature) can predict the flow boiling heat transfer of CO_2–lubricant experimental database with large average and mean deviations. Therefore, there is a need to develop new empirical correlations for the flow boiling heat transfer of CO_2–lubricant mixtures.

Appendix A

Table A1
Calculation methods to predict properties of pure oils

Density	$\rho(T) = \rho(T_0) - A(T - T_0)$ Conde [46] recommended $A = 0.60$ as the reasonable approximation,T_o is the normalized temperature
Kinematic viscosity	$\log(\log(v + 0.7)) = A + B\log(T)$ A and B are constants for each lubricant T is thermodynamic temperature, K
Dynamic viscosity	$\mu(T) = \frac{T-T_1}{T_2-T_1}(\mu_2 - \mu_1) + \mu_1$ μ_1 and μ_2 are the two known viscosities at two temperatures, T_1 and T_2, respectively
Specific heat	$Cp_{\text{oil}} = 4.186\frac{0.388+0.00045(1.8T+32)}{\sqrt{s}}$ by Thome [43]
Surface tension	$Q_{\text{oil}} = 0.1196\left[1 + \frac{\frac{T_b}{T_c}\ln\left(\frac{P_c}{1.01325}\right)}{1-T_b/T_C}\right]P_c^{2/3}T_c^{1/3}Q_{\text{oil}}\left(1 - \frac{T}{T_c}\right)^{11/9}$ $\sigma_{\text{oil}} = P_c^{2/3}T_c^{1/3}Q_{\text{oil}}\left(1 - \frac{T}{T_c}\right)^{11/9}$ by Brock and Bird [47]

Appendix B

Table A2
Refrigerant–lubricant mixture properties calculation methods

Nominal oil concentration	$w_o = \frac{\dot{m}_{\text{oil}}}{\dot{m}_{\text{oil}}+\dot{m}_{\text{ref}}}$
Local oil concentration	$w_{\text{oil,local}} = \frac{w_o}{1-x}$
Mixture dynamic viscosity	$\ln\mu_{\text{mix}} = w_{\text{ref}}\ln\mu_{\text{ref}} + w_{\text{oil}}\ln\mu_{\text{oil}}$ by Yokozeki [48]
Mixture density	$\frac{1}{\rho_{\text{mix}}} = \frac{w_{\text{local}}}{\rho_{\text{oil}}} + \frac{1-w_{\text{local}}}{\rho_{\text{ref,l}}}$ by Jensen and Jackman [49]
Liquid specific heat of refrigerant–oil mixture	$Cp_{\text{mix,l}} = w_{\text{oil}}Cp_{\text{oil}} + (1 - w_{\text{oil}})Cp_{\text{ref}}$ by Jensen and Jackman [49]
Mixture surface tension	$\sigma_{\text{mix}} = \sigma_{\text{ref,l}} + (\sigma_{\text{oil}} - w_{\text{ref}})\sqrt{w_{\text{local}}}$ by Jensen and Jackman [49]
Mixture bubble point temperature	$T_{\text{bub}} = \frac{A(w_{\text{oil}})}{\ln(P_{\text{sat}})-B(w_{\text{oil}})}$ by Thome [43] $A(w_{\text{oil}}) = a_0 + a_1w_{\text{oil}} + a_2w_{\text{oil}}^3 + a_3w_{\text{oil}}^5 + a_4w_{\text{oil}}^7$ $B(w_{\text{oil}}) = b_0 + b_1w_{\text{oil}} + b_2w_{\text{oil}}^3 + b_3w_{\text{oil}}^5 + b_4w_{\text{oil}}^7$

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