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A Comparison Between Recent Experimental Results and Existing Correlations for Microfin Tubes for Refrigerant and Nanolubricants Mixtures Two Phase Flow Boiling

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A Comparison Between Recent Experimental Results and Existing Correlations for Microfin Tubes for Refrigerant and Nanolubricants Mixtures Two Phase Flow Boiling

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ABSTRACT

Driven by higher energy efficiency targets, there is critical need for major heat transfer enhancements in heat exchangers. Nanolubricants, that is, nanoparticles dispersed in the non-volatile component of a mixture, have the potential to increase the heat transfer coefficient by 20% or more for two-phase flow boiling with small or no penalization on the two-phase flow pressure drop. The present work builds upon these intriguing yet unexplained findings, which were documented in the experiments of the present study for one type of nanolubricant, but for which a theory still does not exist. This paper presents a comparison between existing models in the literature and recent new experimental data for two-phase flow boiling in a microfin tube of refrigerant R410A and nanolubricants mixtures. Alumina Oxide (γ-Al_2O_3) based nanolubricants with 40 nominal particle diameter of approximately spherical shape were investigated. The nanoparticles concentration in the lubricant varied from 10 to about 20 in mass percentage, and the lubricant concentration varied from 0 up to 3% in mass percentage. The models available in the open domain literature were not able to capture the effects of the nanoparticles on the two-phase flow heat transfer coefficient. The augmented thermal conductivity of the lubricant due to the addition of highly conductive nanoparticles was not the main mechanism responsible for the heat transfer enhancements. The discrepancy between the simulation results and the experimental data was postulated to be due to non-Newtonian behaviors due to the presence of nanoparticles and surfactants. The flow development of the liquid phase of the mixture and the localized thickening and thinning of the liquid film thickness around the inner walls of the tube can alter the film local convective thermal resistance.

1. INTRODUCTION

The need for high energy conversion efficiencies is driving recent research towards new technologies, such as nanofluids. These fluids consist of liquids with fine nano-sized particles homogenously dispersed in them and they are used in several applications for performance improvements. Vapor compression cycles of air conditioning and refrigeration systems often use refrigerants with small traces of lubricant in circulation as the working fluids. The lubricant is needed to guarantee the compressor safe operation but it is generally detrimental for the other system components because it increases the pressure drops and affects the heat exchangers thermal performance (Shen & Groll, 2005a, 2005b). Stable dispersions of nano-sized particles in lubricants are defined to as nanolubricants. The nano-scale interactions were responsible for the heat transfer augmentation observed in previous works on pool boiling (Kedzierski, 2009, 2011, Peng *et al*., 2010, 2011, Wen & Ding, 2005) and for one flow boiling study (Bartelt *et al*., 2008). From these investigations it appeared that nanolubricants had the potential to counter act the negative effects on pressure drops and heat transfer when lubricants were present in the heat exchangers. The behavior and properties of a nanofluid can vary greatly depending on several factors, such as the type of the base fluid (water, refrigerant, lubricant…), the particles characteristics (material, shape, dimension and concentration), the particle stabilization process (particle polarization, use of surfactant, type of surfactant) (Lin *et al*., 2015), and the particles dispersion process (sonication, homogenization, stirring). These factors not only affect the thermophysical properties of the fluid (Buongiorno *et al*., 2009; Venerus *et al*., 2010; Bigi *et al*., 2015), but also affect the mechanism of the heat transfer process. Because of the complexity of the real case processes and the large amount of variables involved with the heat transfer processes, several approaches were proposed in the literature that describe the behavior of nanofluids two-phase flow (Xuan & Roetzel, 2000). However, it is still not clear how the particles affect the transport properties of the refrigerant and lubricant liquid phase mixtures for two-phase flow heat transfer and pressure drop. Few studies in the existing literature offered models that described the variation of thermophysical properties of the base fluids based on the nanoparticles type, size, and concentration. It was sometimes observed that, under particular flow regimes, the presence of nanoparticles dispersed in high-viscosity suspensions yielded to non-Newtonian behavior of the fluid and shear-thinning or thickening behaviors were proposed as a possible explanation for the observed heat transfer enhancements and sometimes degradations. Similar phenomenon were postulated for refrigerant and nanolubricants mixtures. The experimental work reported in this work was conducted on an evaporative mixture of refrigerant R410A and Alumina Oxide (γ-Al_2O_3) based nanolubricant flowing inside a micro-finned tube. The data indicated that the effect of the nanoparticles was dependent on the flow regime and, in some cases, variations of the nanoparticle concentration in the mixture did not produce measurable variations of the heat transfer coefficients. For the cases of high-viscosity suspensions, under particular flow regimes, the presence of nanoparticles can induce a non-Newtonian behavior of the fluid (Mahbubul *et al*., 2012) and the shear-thinning or thickening phenomenon can alter the nanofluid local convective thermal resistance.

2. MODEL DEVELOPMENT

The authors developed a simulation model to describe and investigate the behavior of refrigerants and nanolubricants mixtures during two-phase flow boiling. The simulation code was written in C++ programming language and properties of refrigerants were calculated using the CoolProp 5.1.2 thermophysical open-source library (Bell *et al*., 2014). An input file was provided as a user interface to define both the geometry of the evaporator tube and the fluid inlet conditions, that is, the type of refrigerant, lubricant, nanoparticle mass fraction, and mass flow rates. Additional inputs to the present model were the evaporator tube inlet pressure and inlet enthalpy of the refrigerant and nanolubricant mixtures. The heat capacity of the evaporator tube was used for setting the heat flux boundary conditions. The simulation solved the mass and energy balances in the evaporator tube. Using existing two-phase flow heat transfer, pressure drop, and void fraction correlations from the open domain literature, the tube heat transfer coefficient and pressure drop were calculated. For the calculation of the pressure drop, an estimate of the outlet conditions was first made based on the inlet conditions and then an iterative loop was implemented to calculate the actual outlet pressure until convergence was achieved. During the convergence process, the local thermophysical properties of the refrigerant and nanolubricant mixtures were updated at each step in order to account for the local concentration of nanolubricant. To calculate the thermophysical properties of the refrigerant and lubricant and refrigerant and nanolubricant mixtures, five sets of correlations were implemented in the present model. These sets are the lubricant properties, refrigerant and lubricant mixture properties, nanoparticle properties, nanolubricants properties, and refrigerant and nanolubricants properties and they are summarized in Table 1 to 5. The thermophysical properties were calculated at the beginning of the analysis of the evaporator tube and then used in the correlations for the local two-phase flow heat transfer coefficient and pressure drop as function of the local quality; these three quantities were the three output of the present model. The lubricant used in this work was ester oil Emkarate RL 32-3MAF with density of 0.981 g/ml at 20ºC and kinematic viscosity of 31.2 and 5.6 cSt, respectively at 40 and 100ºC.

Table 1: POE Lubricant properties

Density (g/cm^3)	Viscosity (mm^2/s)	Conductivity* (W/m-C)	Specific Heat (kJ/kg-C)	Enthalpy (kJ/kg)	Surface Tension (mN/m)
POE oil manufacturer confidential correlation	POE oil manufacturer confidential correlation	$\mathbf{k_o = 6 \cdot 10^{-6} \cdot T^2 - 0.0006 \cdot T + 0.1513}$ $\mathbf{5 < T < 40^\circ\ C}$	(Lottin *et al*., 2003)	(Lottin *et al*., 2003)	(Hu *et al*., 2008a)

*Empirical correlation from authors' in-house experimental measurements

Table 2: Refrigerant and lubricant mixture properties*

Density (g/cm^3)	Viscosity (mm^2/s)	Conductivity (W/m-C)	Specific Heat (kJ/kg-C)	Enthalpy (kJ/kg)	Bubble Temp. (K)	Surface Tension (mN/m)
(Jensen & Jackman, 1984)	(Yokozeki, 1994)	(Filippov & Novoselova, 1955)	(Jensen & Jackman, 1984)	(Thome, 1995)	(Thome, 1995)	(Jensen & Jackman, 1984)

* Liquid mixture properties were calculated as a function of the local oil mass fraction: $m_o/\left(m_o + m_{ref,L}\right)$

Table 3: Al_2O_3 nanoparticle properties

Density (g/cm^3)	Conductivity (W/m-C)	Specific Heat (kJ/kg-C)
3.6 (Sarkas, 2014)	(Morrell, 1987)	(Touloukian, 1970)

Table 4: Nanolubricant properties*

Density (g/cm^3)	Viscosity (mm^2/s)	Conductivity (W/m-C)	Specific Heat (kJ/kg-C)
(Pak & Cho, 1998)	(Batchelor, 1977) $k_1 = 2.5$, $k_2 = 6.2$	(Maxwell, 1881)	(Murshed, 2011)

* Nanolubricant properties were calculated as a function of the nanoparticle volume fraction.

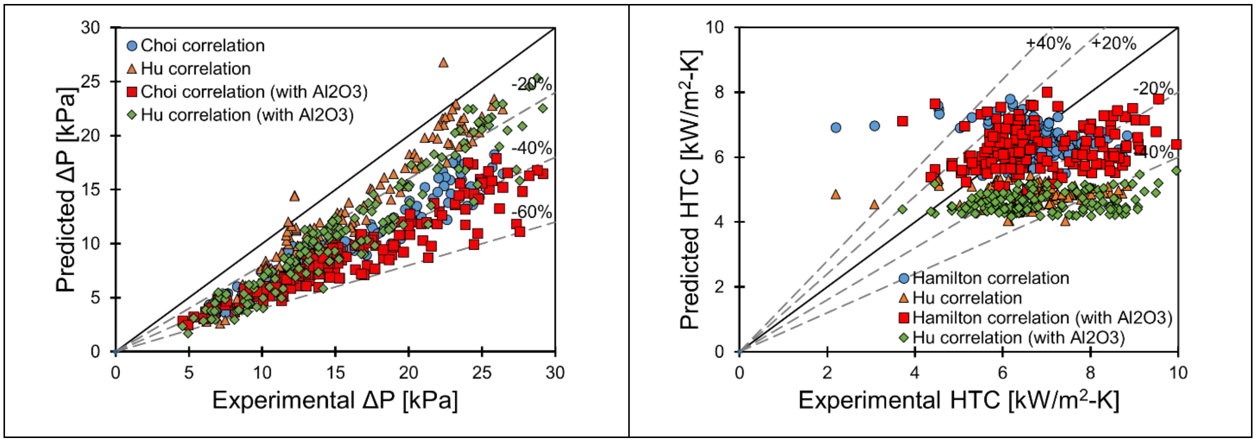
Table 5: Refrigerant and nanolubricant mixture properties*

Density (g/cm^3)	Viscosity (mm^2/s)	Conductivity (W/m-C)	Specific Heat (kJ/kg-C)	Enthalpy (kJ/kg)	Bubble Temp. (K)	Surface Tension (mN/m)
(Jensen & Jackman, 1984)	(Kedzierski & Kaul, 1998)	(Filippov & Novoselova, 1955)	(Jensen & Jackman, 1984)	Assumed same correlation as for refrigerant/lubricant mixture (see Table 2)		

* Liquid mixture properties were calculated as a function of the local nanolubricant mass fraction: $\left(m_o + m_n\right)/\left(m_o + m_n + m_{ref,L}\right)$

3. MODEL EXPERIMENTAL VALIDATION

The model developed in the present work was validated against the experimental data presented by Deokar *et al.* (2016) for two-phase flow boiling (i) of refrigerant R410A, (ii) of refrigerant and POE oil mixtures at 3% oil mass fraction, and (iii) of refrigerant and Al_2O_3 nanolubricant mixtures with oil mass fraction of 3% and nanoparticle mass concentration in the lubricant of 10 and 20% (that corresponds to a nanoparticle volume concentration in oil of about 2.6 and 5.8%). Data were for a horizontal 9.5 mm micro-fin tube evaporator with hydraulic diameter of 5.45mm. The refrigerant saturation temperature varied from 3.1 to 4.0°C, the mixture mass flux was 250 and 350 kg/m^2-s and tube heat flux ranged from 12 to 15 kW/m^2. The experimental results can be found in a companion paper (Deokar *et al.*, 2016) to this conference and they will not be repeated in this paper for conciseness. However, it is important to point out that the experimental uncertainty on the data of heat transfer coefficient ranged from ±4 to ±11% and the uncertainty on the pressure drop data ranged from ±9 to ±16%. This uncertainty should be considered when comparing the predicted pressure drops and heat transfer coefficients against the experimental data, as shown in Figure 1.



(a) (b)

Figure 1: Comparison of predicted (a) pressure drops (ΔP) and (b) heat transfer coefficients (HTC) vs. experimental data (data were presented in a companion paper by Deokar *et al*. (2016))

Experimental Validation of the Pressure Drop Models Used in the Present Work

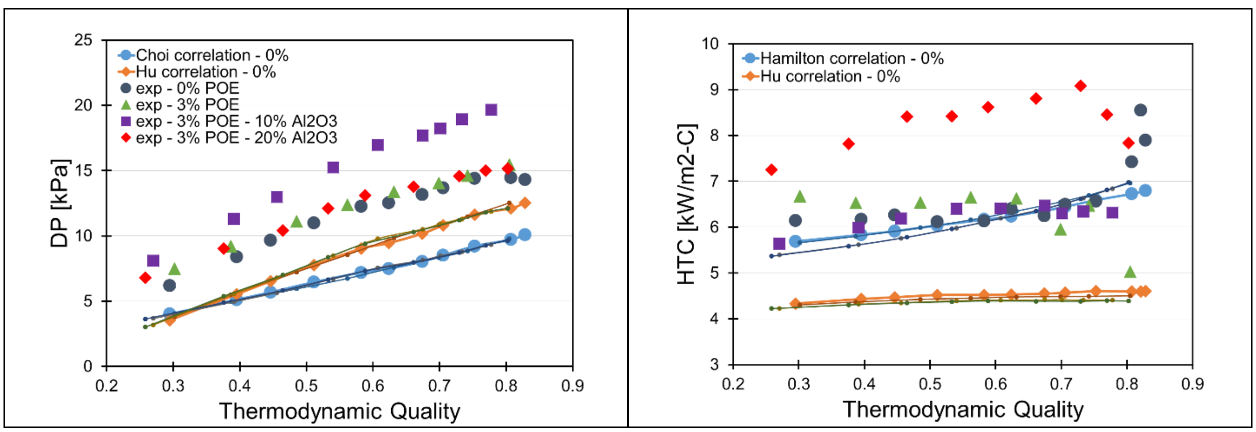
The simulation predictions for two-phase flow pressure drop in a microfin tube are reported in Figure 1(a). For refrigerant R410A and for refrigerant and lubricant mixture, Figure 1(a) reports the simulation results obtained with the correlation by Choi *et al*. (1999) (blue solid circles) and with the correlation by Hu *et al*. (2008a) (orange solid triangles). Figure 1(a) also reports the results of the application of the two correlations to the cases with refrigerant and nanolubricant mixture (red solid squares and green solid diamonds). The comparison with the experimental data showed that both the correlations from the literature underpredicted the experimental data. The refrigerant R410A was underpredicted by up to -40%. For the refrigerant and POE oil, the correlation by Choi *et al*. (1999) calculated the total pressure drop and was designed for blends of refrigerants and refrigerant and lubricant mixtures flowing through a microfin tube with outside diameter of 9.52 mm. Their tube geometry was similar to the one used by Deokar *et al*. (2016) and in the present work. However, there was a lack of specific information about the specific properties of the particular POE lubricant used in the work of Choi *et al*. and these properties were estimated in the present work considering a general ISO VG 32 POE lubricant. Additives and surfactants used in the specific POE lubricant might change some of its properties and could lead to significant variation of the predicted pressure drops from the present model. For this reason, a sensitivity study was performed and will be presented later in this paper. In the sensitivity study, the viscosity of the base lubricant was purposely varied to up to 25% higher than what is generally estimated for ISO VG 32 POE lubricant in order to investigate the impact of lubricant viscosity on the predicted pressure drop and heat transfer coefficient of the refrigerant and oil mixture during flow boiling. As shown in Figure 1(a), and as also pointed out by the original authors of the correlation of Choi *et al*. (1999), the Choi *et al*. correlation seemed to underpredict the two-phase flow boiling pressure drop of refrigerant and lubricant mixtures, and the error was up to -50%. Similar findings were observed in the work by Hu *et al*. (2008a) who proposed a new vapor-phase multiplier correlation of frictional pressure drop for boiling mixture of R410A/lubricant flowing inside a microfin tube with a 7 mm outside diameter. They observed higher pressure drops with increasing oil mass fractions and mass fluxes. The oil used was slightly more viscous than the one used in the present work. In their work, Hu *et al.* reported a maximum deviation of their correlation of 15% and their correlation provided better predictions, that is, within -20%, of the experimental data for refrigerant and refrigerant and oil mixture reported in Figure 1(a). It should be noted that in the present model, the momentum pressure drop was calculated by using the void fraction correlation by Rouhani & Axelsson (1970) and the pressure drop correlations were implemented using the thermodynamic properties of refrigerant and lubricant and refrigerant and nanolubricant mixtures described in Table 1 to 5.

Experimental Validation of the Heat Transfer Coefficient Models Used in the Present Work

The comparison between the experimental data of heat transfer coefficients taken from Deokar *et al*. (2016) paper and the predicted two-phase flow heat transfer coefficients from the model developed in the present work are summarized in Figure 1(b). For refrigerant R410A and refrigerant and lubricant mixture, Figure 1(b) reports the simulation results obtained with the correlation by Hamilton *et al*. (2008) (blue solid circles) and with the correlation by Hu *et al.* (2008b) (orange solid triangles). Figure 1(b) also reports the simulation results of the application of the two correlations to the cases with refrigerant and nanolubricant mixture (red solid squares and green solid diamonds). The simulation results were able to predict most of the experimental data within ±40%. If refrigerant R410A was modeled, then the heat transfer coefficients were predicted with an uncertainty of ±20% for Hamilton *et al.* correlation and of ±30% for Hu *et al.* correlation. These uncertainties were consistent with the ones reported in the original studies from which the correlations were developed. The correlation by Hamilton *et al*. (2008) described flow boiling of refrigerants and refrigerants blends inside an horizontal microfin tube. This correlation was built upon the theory of the law of corresponding states and it is only applicable for mass fluxes between 70 and 370 kg/m^2-s and for a quality range of 0 to 0.7. The work by Sawant *et al*. (2007) proved the applicability of Hamilton *et al.* correlation to mixtures of R410A and POE oil with ±20% error. The oil used in their work had about same viscosity as the one used for the present work. The same authors also stated that the relative heat transfer coefficient of the R410A and POE mixture ranged from –20% up to +42% compared to that of refrigerant R410A only heat transfer coefficient. Hu *et al*. (2008b) developed another correlation to describe the flow boiling of R410A and lubricant mixtures in a microfin tube with a 7 mm outside diameter. Their correlation accounted for both convective and nucleate boiling contributions to the heat transfer and was validated with a deviation from experimental data of ±30%. The oil used in their experiments was slightly more viscous than the one used for the present work. Although the correlation for kinematic viscosity used in their heat transfer correlation provided values of kinematic viscosity that are almost one order of magnitude higher with respect to other sources. Hu *et al*. (2008b) observed that for qualities lower than 0.4, the heat transfer was enhanced in presence of oil, while for qualities higher than 0.65, the heat transfer decreased drastically.

4. SIMULATION RESULTS AND DISCUSSION

Figure 2 shows the pressure drop and heat transfer coefficients for the case of 250 kg/m^2-s mass flux and 0 to 3% oil mass fraction, with and without nanoparticles. The plots are given with refrigerant thermodynamic quality on the x-axis and for both heat transfer and pressure drop correlations used in the present work. Different series of experimental data are also reported, showing the behavior of the mixtures when the quality increases. The experimental series reported are for the following refrigerant mixtures: at 0% POE -oil-free case- (in blue solid circles); at 3% POE (in green solid triangles); at 10 and 20% nanoparticle mass concentration in 3% POE oil (respectively, in purple solid squares and red solid diamonds). The predicted results are summarized by blue and orange solid lines for the oil-free cases. For the cases with POE oil and nanolubricants the predicted results are on the top of the oil-free case solid lines, that is, the predicted pressure drops and heat transfer coefficients when oil and nanolubricants were present did not vary appreciably to be distinguished in Figure 2 as separated individual lines.



(a) (b)

Figure 2: Experimental and simulation trends of different refrigerant/lubricant mixtures for (a) pressure drop and (b) heat transfer coefficient

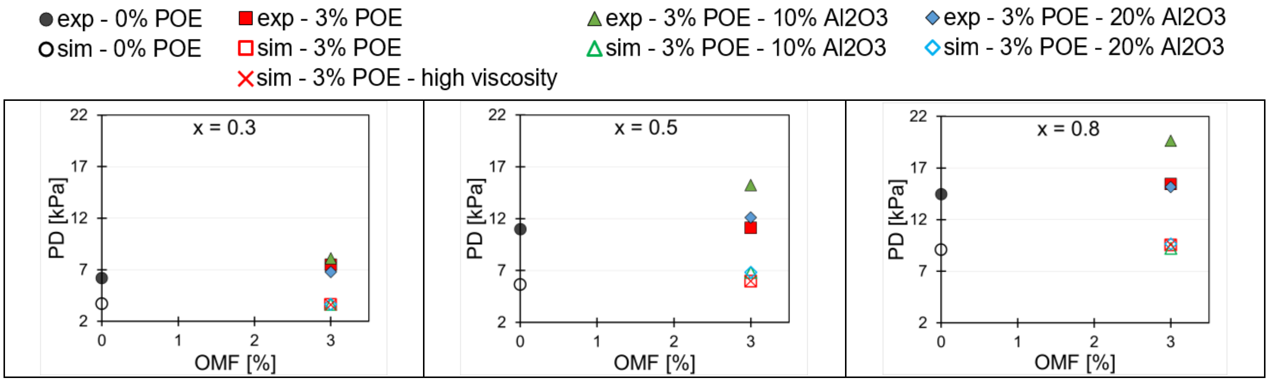
Table 6 shows the results of a sensitivity analysis of the existing correlations used to estimate kinematic viscosity and thermal conductivity, whose estimated values were varied by ±25% in a parametric fashion. The error was calculated as difference of the simulation results minus the experimental data, in percentage, and for two representative qualities. The comparison was conducted for the case of refrigerant-nanolubricant mixture at 3% POE oil OMF and 20% nanoparticles concentration in oil (see row 1 in the Table 6). The variation of the nanolubricant kinematic viscosity and thermal conductivity by ±25% did not decrease the error, as shown in rows 2 and 3. A slight reduction of few percentages was observed for the predicted heat transfer coefficients at quality of 0.75, as indicated in the last column of row 3. While a variation of the refrigerant R410A and nanolubricant mixture kinematic viscosity had small effects, an increase of thermal conductivity of the refrigerant R410A and nanolubricant liquid phase mixture of +25% increased the predicted heat transfer coefficient significantly, and reduced the error to 6 and 15%, as shown in row 5 of Table 6. However, none of the existing correlations resulted in such increase of thermal conductivity of the liquid phase of the refrigerant R410A and nanolubricant mixture.

Table 6: Sensitivity analysis for kinematic viscosity and thermal conductivity

		x = 0.5		x = 0.75	
		DP error [%]	HTC error [%]	DP error [%]	HTC error [%]
1	R410A - 3% POE - 20% Al2O3	-45.1	-29.1	-38.1	-19.5
2	Nanolubricant ν of +25% / -25%	-45.0 / -45.1	-29.2 / -29.0	-38.0 / -38.2	-19.5 / -19.5
3	Nanolubricant k_{th} of +25% / -25%	-45.1 / -45.1	-28.8 / -29.5	-38.1 / -38.1	-18.8 / -20.1
4	R410A-nanolub. ν of +25% / -25%	-43.9 / -46.4	-30.8 / -26.9	-36.9 / -39.7	-19.3 /-19.7
5	R410A-nanolub. k_{th} of +25% / -25%	-45.1 / -45.1	**-15.2** / -43.8	-38.1 / -38.1	**-6.0** / -34.0

Discussion of the simulation results for two-phase flow pressure drop of nanolubricants

For the case of 250 kg/m^2-s mass flux and 0 to 3% oil mass fraction, Figure 3 shows that the pressure drop tended to increase if the quality increases. The lubricant had over 10 times higher viscosity than liquid refrigerant and thus it significantly increased the viscosity of the refrigerant/lubricant mixture liquid phase. This generally resulted in higher frictional pressure drops of the refrigerant and lubricant mixture compared to refrigerant only. However, Figure 3 shows that for both the simulation results of the present model and the experimental data used to verify the model, the pressure drop penalization due to the presence of oil was small. The simulations results indicated that at 3% OMF, both POE lubricant and Al_2O_3 nanolubricant had estimated pressure losses that were just slightly higher than that of refrigerant R410A. The data showed similar trends for POE, while higher pressure drop were measured for the Al_2O_3 nanolubricant at medium quality (see Figure 3(b)) and high quality (see Figure 3(c)). An increase of the frictional losses became evident only at higher qualities, as shown in Figure 3(c). Similar findings were also observed in the literature (Nidegger *et al*., 1997, Zürcher *et al*., 1998). According to the aforementioned correlations for mixtures of nanofluids and assuming that the nanoparticles remained well dispersed in the POE and refrigerant mixture liquid phase, the nanolubricants must have higher viscosity than that of liquid refrigerant and POE oil mixture. Thus, the highest pressure losses were expected for the 3% POE oil OMF and 10% and 20% Al_2O_3 nanoparticle concentration case in Figure 3. This was more or less the case in the experimental data. The work of Deokar *et al*. (2016) confirmed that at low quality the pressure losses of lubricant and nanolubricant were very close to each other while nanolubricants tended to have slightly higher pressure losses at medium and high qualities, as shown by the solid red square data points for POE at 3% OMF experimental data with respect to the solid green triangles data points for the Al_2O_3 based nanolubricant at 3% OMF and 10% nanoparticle mass concentration. The model predicted this trend well at medium quality while at both low and high qualities the difference of the pressure drop between POE oil at 3% OMF case (see void red square simulation results points with the “sim – 3% POE” legend in Figure 3) and Al_2O_3 based nanolubricant at 3% OMF and 10% nanoparticle mass concentration case (see void green triangles simulation results with the “sim 3% POE - 10% Al_2O_3” legend) were very small. Similar observations could be made for the case of Al_2O_3 based nanolubricant at 3% OMF and 20% nanoparticle mass concentration, at medium quality, where the solid blue diamond showed a slightly higher pressured drop than the solid red square of the 3% POE. The simulation pressure drop of the 20% nanolubricant case was slightly higher than the 10% nanolubricant case both at medium and high qualities. The model seemed to capture trends similar to the experimental data, and closer to the 3% POE (comparison between void blue diamonds and void red squares). In order to investigate these results, a sensitivity analysis of the viscosity was conducted by increasing the viscosity value up to 10 times (reported in Figure 3 as a red cross) for the case of 3% POE. Interestingly, the model did not seem to be affected by a higher viscosity as the new pressure drop indicated by the red cross did not move from the void red square of the base 3% POE case. More recent investigations on pool boiling of non-Newtonian fluids and Al_2O_3 nanolubricants (Soltani *et al*., 2010, Kedzierski, 2011) showed how even a 1.4 to 1.6% nanoparticle volume fraction can drastically enhance the heat transfer of the base fluid, thank to the interaction of the nanoparticles with the bubbles formation process. However, other works on nanofluids also observed a share-rate dependency of the viscosity, arguing the possibility of a transition from a Newtonian to a non-Newtonian behavior (Venerus *et al*., 2010). Aladag *et al*. (2012) studied nanofluids with nanoparticles of different shapes and reported a shear-thickening behavior for Al_2O_3-water nanofluid over a wide range of shear rates and for temperatures between 2 and 10ºC. The pressure drop correlations used in this work lack of information on the change of the fluid behavior when nanoparticles are added, as well as a dependency from the flow rate and the nanoparticles material, shape, size and dispersion. It might be possible that a similar situation to the one described by Aladag *et al*. is occurring for the flow regime of the present work and this aspect requires further investigation in future follow up research of this work.



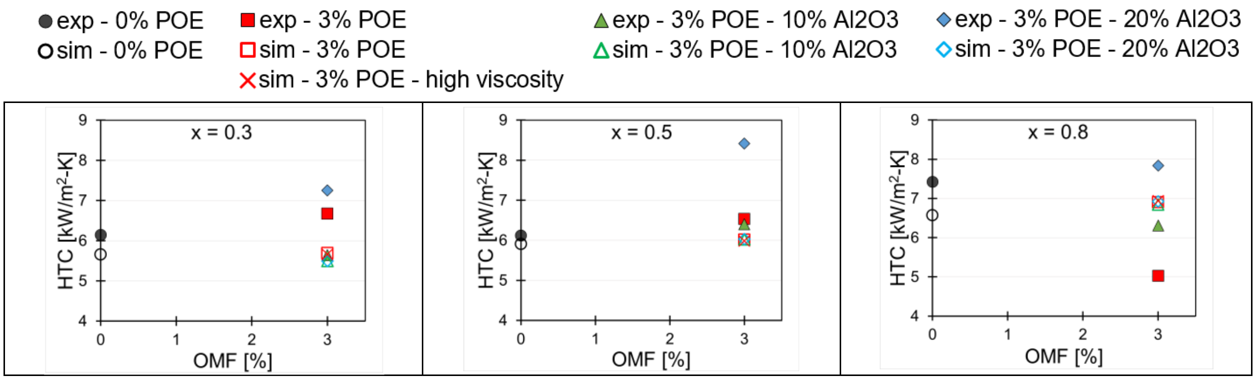
(a) (b) (c)

Figure 3: Pressure drop for (a) low quality, (b) medium quality and (c) high quality of different refrigerant/lubricant mixtures at test conditions of 250 kg/m^2-s and 12 W/m^2 (the simulation data in this figure were obtained from application of Choi *et al.* (1999) correlation)

Discussion of the simulation results for two-phase flow heat transfer coefficients of nanolubricants

For the case of 250 kg/m^2-s mass flux and 0 (refrigerant only) to 3% oil mass fraction, the experimental results by Deokar *et al*. (2016) in Figure 4 show that the oil-free case slightly increased heat transfer coefficient if the quality increase from 0.3 (low quality in Figure 4(a)) up to 0.8 (high quality in Figure 4(c)). For 3% oil mass fraction, the heat transfer coefficient was higher than the oil-free case at lower and medium qualities, but it dropped at higher qualities. This behavior was unexpected but similar to what observed in the experimental work of Hu *et al*. (2008b). The review paper by Bandarra *et al*. (2009) on flow boiling of refrigerant/lubricant mixtures reported other literature studies for microfin tubes where the presence of oil increased the heat transfer coefficient. Compared to the liquid phase of most refrigerants, generally lubricants have lower density (that, at constant mass flux, can increase the fluid velocity and promote more uniform mixture), higher thermal conductivity, higher specific heat, higher surface tension (increasing the wettability), higher bubble temperature and higher viscosity, which greatly affects both pressure drop and heat transfer, especially at higher qualities. Oil might induce some foaming at the liquid-vapor interface. The internal geometry of a microfin tube also affects the flow patterns, promoting annual type flow regime. The effect of these phenomena on the heat transfer coefficient are not properly captured by the heat transfer correlations used in the present model, as shown by the discrepancy between simulation results for POE (void red squares) and nanolubricant (void green triangles) mixtures and experimental data (solid red square and solid green triangles) in Figure 4(a) and 3(b).

A sensitivity analysis of these results with respect to the mixture thermophysical properties, suggested that at higher qualities the increase in viscosity was much faster and it could affect greatly the Reynolds numbers used to estimate the heat transfer coefficients. Thus, a steeper increase of viscosity could lead to a sudden decrease of heat transfer coefficient. For the case of nanolubricants, even if nanoparticles enhance thermal conductivity, they could also further increase the viscosity by promoting a shear-thickening behavior, typical of some non-Newtonian fluids. The existing viscosity models in the literature used for nanolubricants did not include non-Newtonian behaviors, which affect the flow development of the liquid phase of the mixture. The localized thickening and thinning of the liquid film thickness around the inner walls of the tube can alter the film local convective thermal resistance. This mechanism could explain the discrepancy between the simulation results of the present and the experimental data. However, this behavior was not properly captured by the existing two-phase flow boiling heat transfer coefficient correlations that were implemented in the present heat transfer model from the state-of-the-art literature and that are commonly used for predicting heat transfer performance of refrigerant and POE oil mixtures during flow boiling in micro-fin tubes. Similarly to what was observed for pressure drops case, the heat transfer correlations were not able to predict the nanolubricant behavior. Information on the change of the fluid flow behavior in presence of nanoparticles should be added in future work to the present model.



(a) (b) (c)

Figure 4: Heat transfer coefficient for (a) low quality, (b) medium quality and (c) high quality of different refrigerant/lubricant mixtures at test conditions of 250 kg/m^2-s and 12 W/m^2 (the simulation data in this figure were obtained from application and extrapolation of Hamilton *et al.* (2008) correlation)

5. CONCLUSIONS AND FUTURE WORK

This paper presented a comparison between experimental results of two-phase flow boiling in a microfin tube of refrigerant R410A, R410A/lubricant mixture and R410A/Al_2O_3 nanolubricant mixture and models available in the literature for estimating pressure drop and heat transfer coefficient. The comparison was made by using a newly developed simulation tool that included literature correlations for predicting thermophysical properties of lubricants, nanolubricants and refrigerant/lubricant and nanolubricant mixtures. For high nanoparticle concentrations and in some flow conditions, the refrigerant and nanolubricant mixture showed higher heat transfer coefficient than that of both the refrigerant R410A/POE oil mixture and the refrigerant R410A only. However, the enhancements were dependent on quality, mass flux, and heat flux, and in some case, the data showed a degradation of heat transfer coefficient. Similar findings were documented in the literature and it was reported that some lubricants could enhance the thermophysical properties of the refrigerant liquid phase during evaporation. The findings of the present paper showed that, within their reported uncertainty, the correlations in the literature were generally able to predict the experimental data for the cases of refrigerant R410A and refrigerant/lubricant mixture of the present work but they were inadequate to describe the behavior of the refrigerant/nanolubricant mixtures. An increase of thermal conductivity of the refrigerant R410A and nanolubricant liquid phase mixture of +25% increased the predicted heat transfer coefficient significantly, and in this case, the error between the simulation results and the data was within 15%. However, none of the existing correlations resulted in such increase of thermal conductivity of the liquid phase of the refrigerant R410A and nanolubricant mixture. The authors speculated, as observed in available work on nanofluids research, that Al_2O_3 nanoparticles could induce a change in the nature of the mixture depending on the local dispersion concentration and promote a transition to non-Newtonian behavior. For this reason, different approaches to model these types of nanolubricants mixture might be future follow up work.

NOMENCLATURE

k_{th}	Thermal Conductivity	(W/m-K)
k1, k2	Batchelor (1977) correlation constants	(-)
m	mass	(kg)
OMF	Oil Mass Fraction	(%)
ν	Kinematic Viscosity	(mm^2/s)

Subscript

n	nanoparticle
o	oil
ref,L	liquid refrigerant

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