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Literature Review of Condensation and Evaporation of R290

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ABSTRACT

Nowadays, an increasing attention in environmental issues including the global warming effect and ozone layer depletion has been attracted. R22 is urged to be ruled out until 2020 and 2030 for developed and developing countries. R290 (propane) has been regarded as a promising alternative refrigerant for the air-conditionings. However, R290 has different thermo-physical properties than conventional refrigerants and it would influence the application in tubes such as shear forces, gravity and surface tension. The condensation and evaporation behavior of R290 would have a great effect in designing suitable heat exchanger for R290. In recent years, researchers have made great efforts in studying different working fluids behaviors and multiple semi-empirical correlations have been established to predict the heat transfer and pressure drop while the study about R290 behavior is very limited. According to the past experiences in condensation and evaporation studies, the semi-empirical models may be not very accurate in the working conditions outside the range where they are established. This paper presents a detailed review of research work done on the condensation and evaporation of R290. This paper is a starting point for future R290 studies and R290 applications in air conditioning systems.

1. INTRODUCTION

To reduce the Global Warming and Ozone Layer Depletion Effects, the refrigerants experience great changes, CFCs、HCFCs、HFCs、HCs and other natural refrigerants. The Montreal protocol has proposed a regulation in 1987 to regulate the substances that deplete the Ozone Layer. The Kyoto Protocol was signed by 142 countries to limit the green gas discharge. R290, as a natural refrigerant, has zero ozone depletion potential and negligible GWP. It has been considered as a promising sustainable candidate of refrigerants in heat pump and refrigeration system. Excluded the flammability, R290 is a very suitable and simple method for refrigerant replacement due to its similar thermo-physical properties compared to R22 and its miscibility with multiple mineral oil.

Although the thermo-physical properties of R290 are very similar to R22, due to its differences in density, latent heat, heat capacity and surface tension, R290 would have different performances in refrigeration systems. Thus it is important to study R290 two-phase heat transfer and pressure drop behaviors. There are multiple researches studying different working fluids behaviors and multiple semi-empirical correlations have been established to predict the heat transfer and pressure drop of different refrigerants (Shah, 1990; Kattan *et al*., 2003; Cavallini *et al*., 2001; Jung *et al*., 2003; Cavallini *et al*., 2006; Milkie, 2014). However, the studies related to R290 behaviors are very limited. This paper reviews the works on condensation and evaporation of R290 including heat transfer and pressure drop behaviors since it is a starting point for studying R290 applications.

2. CONDENSATION

Chang *et al*. (2000) have studied the in-tube condensation heat transfer characteristics for R290, R600a and a 50/50 mixture of R290 and R600a and compared the results against R22. The results suggested that at the same mass flux condition, hydrocarbons can contribute a much higher average condensation heat transfer coefficient. However, from the application point of view, for similar capacity condition, hydrocarbons show slightly lower values of heat transfer coefficients than R22. They modified Shah (1990) equation for hydrocarbons to predict their condensation heat transfer coefficient and the agreement are within ±20% and a mean deviation of 8.6% for R290.

$$h/h_l = 1 + 2.5/X_c^{0.912} \tag{1}$$

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4} k_l/D \tag{2}$$

$$X_c = ((1-x)/x)^{0.8}(p_r)^{0.5} \tag{3}$$

El Hajal *et al*. (2003) developed a new flow pattern map for condensation originally from an existing flow pattern map developed by Kattan *et al*. (2003) based on Cavallini *et al*. (1999, 2001) condensation heat transfer database. The new flow pattern map includes the flow region of annular flow, intermittent flow, stratified-wavy flow, fully stratified flow and mist flow. They also verified their new flow pattern map against Kim and Chang (1996) test data and the results show a good agreement. They suggested the range of the new flow pattern map application is $16<G<1532$ kg/m^{2}s, $3.14< d_i <21.4$ mm, $0.02<p_r<0.8$, $76< (We/Fr)_l<884$ for various refrigerants including propane.

Thome *et al*. (2003) have developed a condensation heat transfer model based the flow pattern map mentioned above. They categorized the in-tube condensation has two types of heat transfer mechanisms: convective condensation and film condensation and both of them needed to be considered.

$$h = [h_{film}\,\theta + (2\pi - \theta)h_{conv}\,]/2\pi \tag{4}$$

$$h_{conv} = cRe_l^n Pr_l^m (k_l/\delta) f_i \tag{5}$$

$$h_{film} = 0.728\left[\rho_l(\rho_l - \rho_v) g h_{lv} k_l^3/\mu_l D(T_{sat} - T_w)\right]^{1/4} \text{ or } h_{film} = 0.655\left[\rho_l(\rho_l - \rho_v) g h_{lv} k_l^3/\mu_l Dq\right]^{1/3} \tag{6}$$

By verifying their new heat transfer model against hydrocarbon heat transfer database derived from Kim and Chang (1996), the results suggested the new model works well for hydrocarbons which the majority are predicted within 20%. However, they pointed out in Kim's database, there exists an experimental problem in low mass range and high vapor quality range. They suggested their model is applicable for the following working condition: $24<G<1022$ kg/m^{2}s, $0.03<x<0.97$, $0.02<p_r<0.8$ and $3.1<d_i<21.4$.

Wen *et al*. (2006) tested in-tube condensation heat transfer and pressure drop behavior of R600, R290 and a 50/50 mixture of R600 and R290. The test is conducted in a three-line serpentine tube bank with a diameter of 2.46 mm under a constant heat flux of 5.2 kW/m^2, mass flux varied from 205 -510 kg/m^{2}s and refrigerant quality varied from 0.15 to 0.84. According to their results, R290 shows a similar trend as other refrigerants. The heat transfer coefficient increases with the increase of mass flux and vapor quality. And the heat transfer coefficient of R290 is 89% higher than R134a at the same conditions. However, it is lower than the coefficients of R290/R600 mixture and R600. Besides heat transfer, they have also studied the pressure drop through the tube during condensation. The pressure drop of R290 is also increases with mass flux and refrigerant quality. It is 36% higher than R134a and it is smaller than that of R290/R600 mixture and R600 and the variance is explained by the density differences between fluids. They have also carried out a comparison between their experimental data and the prediction models. According to the results, Dobson and Chato (1998) and Friedel (1980) models show good agreements with the experimental results, 100% and 92.5% of data in ±20% range respectively.

Lee *et al*. (2006) carried out researches on condensing heat transfer coefficients and pressure gradients of hydrocarbons including R1270, R290 and R600a in horizontal double pipe heat exchangers with diameters of 12.7mm and 9.52mm under the test condition of mass flux varied from 50-350 kg/m^{2}s and vapor quality varied from 0-0.9. The results showed a same trend as Mao's tests. Compared to R22, the heat transfer coefficient of R290 is over 20% higher and the pressure drop is over 50% higher. Besides, they suggest Shah (1979), Traviss *et al*. (1972) and Cavallini and Zecchin (1974) correlations all show good agreement with their test results and Cavallini-Zecchin's model has the best accuracy.

Park *et al*. (2008) studied the flow condensation heat transfer coefficients and pressure drop of various refrigerants including propane. Their experiments were carried out on plain copper tube of 8.8mm inner diameter and 530mm length under 100-300 kg/m^{2}s mass flux, 7.3-7.7 kW/m^2 heat flux and 40 ℃ condensation temperature. They reported the increasing trend of R290 with mass flux is much faster than that of R22. At 100, 200 and 300 kg/m2s， the heat transfer coefficients of R290 are 4.7%, 33.6% and 53.3% higher than those of R22 respectively. They are similar to that of propylene, slightly lower than isobutene and much lower than DME. By comparing the test results with the existing correlations, Jung *et al*. (2003) and Solliman *et al*. (1968) models show the best accuracy with R290 results. The pressure drop of R290 is lower than R600a but higher than R22 and Propylene. The authors suggested the pressure drop is inversely proportional to the vapor pressures of the refrigerants. They have also proposed Jung and Radermacher (1989) pressure drop correlation may be appropriate for predicting the annular flow pressure drop.

Macdonald and Garmella (2016a) have studied the propane condensing inside horizontal tubes with internal diameters of 7.75mm & 14.45mm at mass flux from 150 kg/m^{2}s to 450kg/m^{2}s, condensing temperature from 30 to 94 ℃. The results indicated the two-phase frictional pressure drop decreases with the increase of saturation temperature, the decrease of mass flux and vapor quality and the increase in tube diameter which can be explained by density change of all phases and velocity gradient change at the tube wall. The condensing heat transfer coefficient is decreasing with the increase in saturation temperature, the decrease in vapor quality and mass flux and the decrease of tube diameter which can be explained by the liquid film change around the tube wall. Besides, according to their observations, the tube radius has an effect in forming the liquid entrainment. Increased liquid entrainment can be observed in larger tube which fits Milkie's (2014) finding. They proposed comparisons between their experimental results with existing correlations and the results suggested Cavallini *et al*. (2006) has the best accuracy in predicting heat transfer of propane while Garimella *et al*. (2005) has the best accuracy in predicting the pressure drop of propane. They have also proposed condensing heat transfer coefficient and frictional pressure drop correlations in literature Macdonald and Garmella (2016b). The pressure drop model is established based on Chrisholm (1967) model by redefining constant C. The proposed pressure drop model presents the interactions between liquid and vapor phases and suspends the application range to lower pressure drop cases.

$$\mathrm{C} = 20\mathrm{Re}^{-0.15}\mathrm{S_r^{1.15}}\mathrm{Bo}^{-0.2} \tag{7}$$

$$\mathrm{Re_l} = \mathrm{G}(1-\mathrm{x})\mathrm{D_h}/\mu_\mathrm{l} \tag{8}$$

$$\mathrm{S_r} = \rho_\mathrm{l}/\rho_\mathrm{v}\cdot x/1-x\cdot 1-\varepsilon/\varepsilon \equiv u_v/u_l \tag{9}$$

$$\mathrm{Bo} = \left(\rho_\mathrm{l}-\rho_\mathrm{v}\right)\mathrm{g}\mathrm{D_h^2}/\sigma \tag{10}$$

They have also developed a condensing heat transfer model based on a model presenting the evolution of liquid film formation with vapor quality which is decided by $\mathrm{Fr_{So}}$ number.

$$\mathrm{h_{condensation}} = \mathrm{h_{film}}\,\theta + \mathrm{h_{pool}}\,(2\pi-\theta)/2\pi \tag{11}$$

$$\begin{gathered}\mathrm{h_{film}} = \mathrm{Ah_{falling\ film}} + \mathrm{Bh_{annular}}\\ \text{where } \mathrm{A} = \tau_\mathrm{verticle}/\tau_\mathrm{resultant} \;\; \& \;\mathrm{B} = \tau_\mathrm{horizontal}/\tau_\mathrm{resultant}\end{gathered} \tag{12}$$

$$\tau_\mathrm{verticle} = \Delta\rho g\delta_\mathrm{stratified}\,;\;\; \tau_\mathrm{horizontal} = \mathrm{dp/dz}\,|_\mathrm{frictional}\;\pi\mathrm{D^2}/4/\pi\mathrm{D}\,;\;\; \tau_\mathrm{resultant} = \sqrt{\tau_\mathrm{verticle}^2+\tau_\mathrm{horizontal}^2} \tag{13}$$

$$\begin{gathered}\mathrm{Nu} = \delta_\mathrm{film}\,\mathrm{h_{annular}}/\mathrm{k_l} = 0.0039Re^{0.775}Pr_l^{0.3}\kappa_i\kappa_E\\ \text{where } \kappa_\mathrm{i} = 1+\left(\Delta\rho g\delta_\mathrm{film}^2/\sigma\right)^{0.1}(\mathrm{u_v/u_l})^{0.5} \;\text{ and }\; \kappa_\mathrm{E} = \delta_\mathrm{film}/\delta_\mathrm{effective}\end{gathered} \tag{14}$$

$$\begin{gathered}\mathrm{Nu} = \mathrm{l_{fallingfilm}}\;\mathrm{h_{fallingfilm}}/\mathrm{k_l} = 0.2\mathrm{Re}^{-0.08}\kappa_\mathrm{i}\\ \text{where } \mathrm{l_{fallingfilm}} = \left[\left(\mu_\mathrm{l}/\rho_\mathrm{l}\right)^2/\mathrm{g}\right]^{\frac{1}{3}}\; and\; Re = 4G(1-x)\delta_{film}/(1-\varepsilon)\mu_l\cdot\xi\end{gathered} \tag{15}$$

$$\eta(1-\varepsilon)\cdot\pi\mathrm{D^2}/4 = \theta/2\pi\cdot\pi\cdot(\mathrm{D^2}/4-(\mathrm{D}/2-\delta_\mathrm{stratified})^2) \tag{16}$$

$$\delta_\mathrm{annular} = \mathrm{D}/2\,(1-\sqrt{\varepsilon}) \tag{17}$$

$$\eta = 1-\left(1+\mathrm{Fr_{So}^{2.5}}\right)^{-0.05} \tag{18}$$

$$\varepsilon_\mathrm{stratified} = ((\mathrm{D}/2-\delta_\mathrm{stratified})/(\mathrm{D}/2))^2 \tag{19}$$

$$u_l = (G(1-x))/(\rho_l(1-\varepsilon)) \cdot \xi \ \ \& \ u_v = Gq/\rho_v \varepsilon_{Baroczy} \tag{20}$$

$$\delta_{entrain} = (D/2) \cdot \left(1 - \sqrt{E(1-\varepsilon)+\varepsilon}\right) \ \text{ where } E = 0.0003\left(u_v/\sqrt{gD}\right)^{0.5}(\Delta\rho g D^2/\sigma)^{0.75} \tag{21}$$

$$Nu = h_{pool}\, D/k_l = 0.023 Re_l^{0.8} Pr_l^{0.3} \text{ where } Re_l = G(1-x)D/\mu_l \tag{22}$$

$$\theta = 2\pi - 2[\pi(1-\varepsilon) + 3\pi/2 \cdot \left(1 - 2(1-\varepsilon) + (1-\varepsilon)^{1/3} - \varepsilon^{1/3}\right) - 1/200 \cdot (1-\varepsilon)\varepsilon(1-2(1-\varepsilon))(1+4((1-\varepsilon)^2+\varepsilon^2))] \tag{23}$$

$$h = h_{condensation}\ \chi_{LM} \ \text{ where } \chi_{LM} = \left(\left(k_{l,wall-subcool}/k_{l,sat}\right)^2 - 0.3\right) \cdot (1/p_r^{0.1}) \tag{24}$$

$$\text{If } Fr_{So} \leq 7, \delta = \delta_{stratified}, \delta_{effective} = \delta_{stratified}, \xi = \eta, \varepsilon = \varepsilon_{stratified}$$

$$\text{If } Fr_{So} > 7, \delta_{film} = \delta_{annular}, \delta_{effective} = \delta_{entrain}, \xi = 1, \varepsilon = \varepsilon_{Baroczy}$$

Lee *et al*. (2010) have conducted the experiments on R290, R600a, R22 and R134a in a horizontal double-pipe heat exchanger having pipe inner diameter of 10.07, 7.73, 6.54 and 5.80mm under mass flux from 35.5-210.4 kg/m^2s and condensation temperature of 40 ℃. Their observed the heat transfer coefficient of R290 has a sharper decreasing trend with quality comparing to R22 and R134a and slightly smoother than R600a which can be explained by their thermal conductivity differences. However, for the effect of the tube diameter in heat transfer coefficient, the experiments suggested a totally opposite result as Macdonald and Garmella (2016). They proposed the heat transfer coefficient is increasing with the decrease in tube diameter. And they suggested the HC refrigerants have much better performance for smaller tubes comparing to Freon refrigerants. For pressure drops, the results indicate for different refrigerants, pressure drops become larger with vapor quality and the deviations between refrigerants become larger at higher vapor qualities and larger mass flux. The four refrigerants pressure drop were in the order of R600a>R290>R22>R134a. And the authors ascribe the results to the vapor density differences between refrigerants. They compared their heat transfer results against five famous correlations, the results suggested Haraguchi *et al*. (1972) reach the best accuracy while Cavallini and Zecchin (1974) can predict the large tube diameter condensation heat transfer well and Dobson *et al*. (1994) can predict small tube diameter heat transfer accurately.

Cavallini *et al*. (2012) have proposed a new heat transfer model for predicting the refrigerants condensing inside horizontal tube. They proposed there is a transition line existing to divide the tube condensation into two parts: ΔT-independent part and ΔT-dependent part and they proposed heat transfer correlations for each part. Besides, they have also categorized flow regimes by these two parts. They suggested annular flow can be covered by ΔT independent part and stratified-wavy and stratified-smooth flow can be covered by ΔT dependent part while slug flow can contain in both parts. They also compared the new model to 4471 data points including R290 data points from Kim *et al*. (1996) and Nan and Infante (2000) and the predictions can be considered satisfactory. They indicated their model is applicable when internal diameter of tube D≥3mm.
Thonon (2005) have carried out propane condensation experiments on enhanced tubes. Wieland PB-8 and PB-9 have been used as test tubes. According to the results, at low vapor quality, a 100% heat transfer enhancement can be reached. By average, the heat transfer can be enhanced by 60-70%. While for pressure drop, the increase is about 90-100% compared to smooth tubes.

Lee *et al*. (2006) have tested R290, R600a, R1270 and R22 using 12.70 mm tube at 32 ℃ condensation temperature and 50-200 kg/m^2s mass flux. They compared their results against Shah (1979) correlation, Traviss *et al*. (1972) correlation and Cavallini and Zecchin (1974) correlation and they suggested all of them can fit the experimental results within 20% error and Cavallini-Zecchin (1974) correlation has the best accuracy.

Davide *et al*. (2014) have analyzed the thermal performance of propane in enhanced mini-channels with an internal diameter of 0.96 mm under mass flux ranges from 100 to 1000 kg/m^2s and condensation temperature between 40 and 41 ℃. They concluded propane is very promising inside mini-channels with limited pressure drop but highly enhanced heat transfer. They observed at high mass velocities and vapor quality between 0.5-0.6, the pressure drop gradient trend has an inflection point and the slope become steeper at high vapor qualities which is not similar as R134a and R1234ze. They proposed this phenonmena is due to the liquid entrainment occurrence of low density propane. By comparing their models against Cavallini *et al*. (2006) and Moser *et al*. (1998), they claimed among their test conditions, all the data points are in ΔT independent region and the predicted results are in good agreement with experimental results. Zhang and Webb (2001) modified Moser *et al*. (1998) equation to extend it for mini-channels and the modified correlation is able to predict most data points except for the low quality and low mass flow rate region.

Dongsoo *et al*. (2004) have tested external condensation heat transfer coefficients of flammable refrigerants including R290. The test condition is vapor temperature of 39 ℃ on a plain tube of 19.0 mm outside diameter with a wall sub-cooling of 3-8 ℃ under a heat flux of 7-23 kW/m^2. The test results show a typical trend of tested refrigerants external condensation heat transfer coefficients. They decrease as the wall sub-cooling increases. Besides, they compared the results with R22 and R134a and the results suggest the external condensation heat transfer of R290 is 9% lower than R22 and similar compared to R134a. They modified Nusselt's equation to generate a new correlation for predicting external condensation transfer coefficients of flammable refrigerants.

$$h = 0.79\left[\rho_l(\rho_l - \rho_v) g k_l^3 h_{lv} / \mu_l \Delta TD\right]^{1/4} \tag{25}$$

Thomas G. *et al*. (2013) have evaluated the condensation heat transfer of R134a and R290 on coated and uncoated smooth, finned and high performance tubes of an outer diameter of 18.8mm by experiments and CFD simulation.The test condition is 37 ℃ saturation temperature under 4-102 kW/m^2 heat flux. They simulated single tube condensation via CFD and compare their results against experiments and the comparison shows good agreement. The heat transfer coefficients decrease with the increase of heat flux. They have also compared their results with Honda *et al*. (1987) and Murata and Hashizume (1992) and the results suggest both models predict propane heat transfer coefficients larger than the experimental results.

3. BOILING AND EVAPORATION

Chang *et al*. (2000) studied the in-tube evaporation heat transfer characteristics for R290, R600a and a 50/50 mixture of R290 and R600a and compared the results against R22. The results suggested that the evaporation heat transfer coefficients of hydrocarbons are slightly higher than that of R22. They modified Wattelet *et al*. (1994) and Cooper *et al*. (1984) correlations to develop evaporation heat transfer coefficient correlation for hydrocarbons.

$$h = \left(h_{NB}^{2.5} + h_{CB}^{2.5}\right)^{0.4} \tag{26}$$

$$h_{NB} = 55 M^{-0.5} q^{0.67} p_r^{0.12} [-\log p_r]^{-0.55} \tag{27}$$

$$h_{CB} = F h_l R \tag{28}$$

$$F = 1 + 1.97/X^{1.46} \quad \text{where } X = ((1-x)/x)^{0.8} \left(\rho_v/\rho_l\right)^{0.2} \tag{29}$$

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4} k_l / D \tag{30}$$

$$R = 1.32\, Fr_l^{0.2} \text{ for } Fr_l < 0.25; \;\; R = 1 \; for \; Fr_l \geq 0.25 \tag{31}$$

Chen *et al*. (2005) have carried out pool boiling heat transfer experiments with smooth and four enhanced tubes with four refrigerants including propane. From their observations, the heat transfer coefficients increase with heat fluxes and saturation temperatures. They mentioned at a heat flux from 20-50 kg/m^{2}s, the increase slope become smaller. Different enhanced geometry would bring different enhanced effects and it does not always enhance the heat transfer especially at high heat fluxes.

Pamiran *et al*. (2010) have conducted experiments on five refrigerants including propane with three different small diameter tubes under a heat flux of 5-40kW/m^2, mass flux of 50-600 kg/m^{2}s and saturation temperature from 0-15 ℃. They have mapped their experimental data on flow pattern maps of Wang *et al*. (1997) and Wojtan *et al*. (2005) and the results indicate overall Wang *et al*. (1997) map predicted better than Wojtan *et al*. (2005). They have also tested the pressure drop performance of these six refrigerants. The pressure drop is increase with the increase of mass fluxes and heat fluxes, but decrease with the increase of saturation temperature and inner tube diameters. They have also compared the pressure drop among six refrigerants: R134a>R22>R290>R410a>R744. They compared their results with existing correlations and the results indicate Beattie and Whalley (1982) correlation has the best accuracy. Besides, they developed a new pressure drop correlation based on Lockhart and Martinelli (1949) correlation.

$$C = 3000 We_{tp}^{-0.433} Re_{tp}^{1.23} \quad \text{where } We_{tp} = G^2 D/\bar{\rho}\sigma \quad \text{and} \quad Re_{tp} = GD/\bar{\mu} \tag{32}$$

Lee *et al*. (2006) have tested R290, R600a, R1270 and R22 using 12.70mm tube 50-200 kg/m^{2}s mass flux. They indicated the evaporative heat transfer coefficient increase with the increase in vapor quality and after it decreases rapidly after 0.85 because of the dry-out. And compared to R22, R290 shows 67.6% higher performance in average evaporative heat transfer coefficient. They compared their results against correlations from existing literatures (Shah, 1979; Traviss *et al*., 1972; Cavallini and Zecchin, 1974; Shah, 1982; Gungor and Winterton, 1987; Kandlikar, 1990) and they suggested all of them can fit the experimental results within 20% error and Kandlikar (1990) correlation has the best accuracy.

Col *et al*. (2014) have analyzed the thermal performance of propane in enhanced mini-channels with an internal diameter of 0.96mm and a rough internal surface under mass flux ranges from 100 to 600 kg/m^{2}s and evaporation temperature of 31 ℃. They suggested among the range of vapor quality from 0.05-0.6 and heat flux from 10-315 kW/m^2, the flow lies in the slug and annular flow region according to Tibirica and Ribatski (2014). Besides, among this region, the heat transfer coefficient increase with the increase of heat flux and has small relation with other operation conditions. They also indicated the heat transfer coefficients decrease with the increase of vapor quality before 0.25 and don't vary much after 0.3. In the vapor quality ranges from 0.16-0.36, the heat transfer coefficient is independent of mass fluxes. By comparing their tests with existing correlations, all the models underestimate the heat transfer coefficient. However, Sun and Mishima (2009) and Thome *et al*. (2004) models can predict the trend right.

Maqbool *et al*. (2011) have presented the experimental results of flow boiling heat transfer of propane in a smooth vertical mini-channel with an internal diameter of 1.7mm, mass flux of 100-400 kg/m^{2}s, heat flux of 5 -240 kW/m^{2}s and saturation temperature of 23 ℃. They indicated an independency of the boiling heat transfer coefficient with mass flux. They concluded the local boiling heat transfer coefficients are generally independent of vapor quality but increases with the increase of heat fluxes except at higher vapor qualities. There exists a maximum point where the heat transfer starts deterioration and they suggested that is the start of dry-out. By comparing their results with existing correlations (Cooper, 1984; Tran *et al*., 1996; Gungor and Winterton, 1986; Liu and Winterton, 1991; Kew and Cornwell, 1997; Lazarek and Black, 1982), all correlations show well predictions except Tran *et al*. (1996) correlation.

Wang *et al*. (2014) have conducted propane boiling experiments in a copper tube with an inner diameter of 6 mm under mass fluxes from 62 to 104 kg/m^{2}s, heat fluxes from 11.7 to 87.1 kW/m^2 and saturated temperatures from -35 ℃ to -1.9 ℃. They declared at low vapor quality, mass flux influences are subtle and at high vapor quality, the influence of mass flux is more obvious: 60% mass flux increase brings 20% heat transfer coefficient increment. However, the heat flux influences are substantial. An almost linear increasing tendency of heat transfer coefficient with increase of heat flux is observed. At small heat fluxes, the effects of saturation temperature can be negligible while with the increase of heat fluxes, the saturation temperature increase can cause an increase in the heat transfer coefficients. The vapor quality effects are influenced by boiling number and liquid to vapor density ratio (Kandlikar and Steinke, 2002), depending on the flow boiling mechanism. They have compared their data with existing correlations, among which, Liu and Winterton (1991) correlation has the best accuracy. They have also studied the pressure drop characteristics. They concluded the pressure drop gradient increase with the decrease of saturation temperature, the increase of mass flux and the increase of vapor quality, while it has a peak value at vapor quality of around 0.88 and after that a decrease trend has been observed. They have compared their data with existing correlations while the results indicate Muller-Steinhagen and Heck (1986) correlation gives the best accuracy.

Jung *et al*. (2004) have studied the nucleate boiling heat transfer coefficients of multiple refrigerants including propane on a horizontal smooth tube of 19 mm outside diameter. The test conditions are: saturation temperature of 7 ℃ and the heat flux ranges from 10-80 kW/m^2. The results indicate the heat transfer coefficient of propane is only 2.5% higher than R22. They compared their data with existing correlations and Jung *et al*. (2003) correlation showed the best accuracy, with 15% deviation. Besides, they have also developed a new correlation based on the experiments and they claimed their new correlation has a deviation of 5.3%.

$$\mathrm{h} = 41.4\mathrm{k_l}/\mathrm{D_b}\left[\frac{(q/A)D_b}{k_l T_{sat}}\right]^{C_2}(-\log_{10} p_r)^{-1.52}\cdot(1-\rho_v/\rho_l)^{0.53}\ where\ C_2 = 0.835(1-p_r)^{1.33} \tag{33}$$

Choi *et al*. (2009) have conducted the flow boiling heat transfer and pressure drop characteristics of propane in horizontal mini-channels with inner diameters of 1.5mm and 3mm under heat fluxes of 5-20 kW/m^2, mass fluxes of

50-400 kg/m^2s, saturation temperature of 10.5 and 0 ℃ and vapor quality up to 1.0. According to their results, the pressure drop increase with the increase of mass flux and heat flux, the decrease of inner tube diameter and the decrease of saturation pressure. By comparing their results against 13 existing correlations and Mishima and Hibiki (1996), Friedel (1979) and Chang *et al*. (2000) give a good prediction with a mean deviation less than 40%. They have also developed a new pressure drop prediction model based on Lockhart and Martinelli (1949) model which gives a mean deviation less than 20%.

$$C = \left(\phi_l^2 - 1 - 1/{X_{tt}}^2\right)X_{tt} = 1732.953 Re_{tp}^{-0.323} We_{tp}^{-0.24} \tag{34}$$

For heat transfer coefficient prediction, they illustrated the heat transfer coefficient is insignificantly influenced by mass flux in the low quality region while it can cause a heat transfer drop in high quality region. Unlike mass flux, heat flux has a large effect on heat transfer coefficient in low-moderate quality region while the influence becomes smaller in high quality region. Besides, the decrease of inner tube diameter and the increase of saturation temperature cause an increase in the heat transfer coefficient. Seven correlations have been used to compare with experimental data while Shah (1988) and Tran *et al*. (1996) correlations have good agreements. They have also developed a new correlation by using pool boiling correlation developed by Cooper (1984) and redefined Chen (1966) suppression factor based on the experimental data and the new correlation can predict the experimental results with a mean deviation of 14.4%.

$$S = 181.458\left(\phi_l^2\right)^{0.002} Bo^{0.816} \tag{35}$$

Maqbool *et al*. (2013) have performed investigations on propane heat transfer and pressure behaviors in verticle circular mini-channel with an inner diameter of 1.7mm under saturation temperatures of 23.33 and 43 ℃, heat fluxes of 5-280 kW/m^2 and mass fluxes of 100-500 kg/m^2s. According their observation, the heat transfer coefficient has insignificant relationship with vapor quality and mass flux, but increase with the increase of heat flux and saturation temperature. The results were used to compare with seven correlations while Cooper (1984) gives the best accuracy. They have also investigated the frictional pressure drop characteristics. The frictional pressure drop increases with the increase of heat flux, mass flux and vapor quality and the decrease of saturation temperature. After the inception point of dry-out, the frictional pressure drop and the heat transfer decrease. Seven models are used to compare against the experimental data, the results indicate Muller-Steinhagen and Heck (1986) correlation have the best accuracy.

4. CONCLUSIONS

Propane is treated as a very promising refrigerant in replacing CFCs, HCFCs and HFCs. Based on the review work above, conclusion can be drawn as below:

- The condensation heat transfer of propane basically follows the same trend as other refrigerants. It decreases with the increase in saturation temperature, the decrease in vapor quality and mass flux and the decrease of tube diameter except the trend with tube diameter, Lee *et al*. (2010) suggested the opposite trend. Comparing to R22 and R134a, propane shows a much better heat transfer performance at the same mass flux. However, at the same heat flux, propane performs a litter worse than R22. Most existing correlations can be used to predict the propane trend well. Some researchers also proposed correlations based on propane/hydrocarbon experimental data which may fit propane condensing heat transfer characteristics better.
- The boiling and evaporation heat transfer of propane increase with the increase in heat flux until the inception of dry-out. However, the relationship between boiling and evaporation heat transfer with mass flux and vapor quality is hard to decide, largely depending on the operation conditions. The dry-out inception point is also largely depending on the flow mechanism. The increment in evaporation and boiling heat transfer of propane comparing to R22 is not as large as that in condensation. Same as condensation, most existing correlations can predict the propane evaporation and boiling heat transfer and some new correlation has been built specially for predicting propane and other hydrocarbons.
- The pressure drop of propane can be well predicted by existing models from Friedel (1979) concluded from most literatures.
- Propane is a very promising refrigerant in future generation refrigeration systems except for its flammability. Low-charge system could be a good option to solve the problem. The studies about propane performances in mini-channels are very limited. Davide *et al*. (2014) suggested current correlations can only predict the trend well but underestimate the boiling heat transfer inside mini-channels. Thus a necessary and desirable effort needs to be paid on the studies of propane behaviors in mini-channels.

NOMENCLATURE

A	vertical contribution to film heat transfer coefficient	--	δ	Film thickness	m
B	horizontal contribution to film heat transfer coefficient	--	ε	Void fraction	--
Bo	Bond number	--	η	Liquid fraction in upper film	--
D	diameter	m	θ	Stratified film angle	Rad
E	Entrained liquid fraction	--	κ	Heat transfer enhancement factor	--
q	Heat flux	W m^{-2}	μ	Viscosity	kg $m^{-1}s^{-1}$
Fr_{So}	Froude number	--	ξ	Liquid fraction in upper film	--
g	Gravitational constant	ms^{-2}	ρ	Density	kg m^{-3}
G	Mass flux	kg $m^{-2}s^{-1}$	σ	Surface tension	N m^{-1}
h_{lv}	Latent heat of vaporization	J kg^{-1}	τ	Shear stress	Pa
h	Heat transfer coefficient	$Wm^{-2}K^{-1}$	ϕ	Two-phase difference correction factor	--
k	Thermal conductivity	$Wm^{-1}K^{-1}$	**Subscript**		
Nu	Nusselt number	--	CB	Convective boiling	
p	pressure	kPa	E	Entrainment	
Pr	Prandtl number	--	h	Hydraulic	
Re	Reynolds number	--	i	Interfacial roughness	
Sr	Slip ratio	--	l	Liquid	
u	Phase velocity	ms^{-1}	NB	Nucleate boiling	
x	Vapor quality	--	r	Reduced	
T	Temperature	K	tp	Two-phase	
We	Weber number	--	v	Vapor	
X	Martinelli parameter	--	w	wall	
M	Molecular weight	--			

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