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ABSTRACT

To meet the terms of the Montreal Protocol, CFCs and HCFCs have been gradually phased out and they have been replaced by refrigerants that have zero ozone depletion potential. However, some of these fluids, such as R-410A, have global warming potential (GWP) that might still be of concern from an environmental perspective in case of leakage or improper charge management. Few studies of refrigerants that have zero ozone depletion potential and GWP less than 500 are available in the literature. Preliminary findings from these studies suggested that new development refrigerants were viable options. System COP and capacity were promising but there is not much information on the heat transfer and pressure drop characteristics of these low GWP refrigerants.

This paper contributes to address this gap and provides new data for the two phase flow boiling heat transfer coefficient (HTC) and pressure drop of two new low GWP developmental refrigerant alternatives for R-410A. Heat transfer measurements were conducted for a copper tube commonly used in direct expansion evaporators of air conditioning systems with a 9.5 mm (0.375 in.) outside diameter and internally enhanced micro-finned surface. Data of local two phase flow HTC and pressure drop are presented for refrigerants R-410A, R-32, R-1234yf and the two new developmental refrigerants referred to as DR-5 and DR-5A.

The experimental findings from this work indicated that the refrigerant R-32 had similar and slightly higher heat transfer coefficient than that of R-410A at same refrigerant mass flux and similar heat flux conditions on the outer surface of the tube. Refrigerant R-1234yf had about 15 to 20% lower heat transfer coefficient than R410-A at 4°C saturation temperature. For this saturation temperature the developmental refrigerants DR-5 and DR-5A had heat transfer coefficients between R-32 and R-1234yf when the vapor quality ranged from 0.2 to 0.7. An increase of the saturation temperature from 4°C to 9°C decreased the heat transfer coefficients for all of the refrigerants tested. The two phase flow boiling pressure drops increased monotonically if the vapor quality of the refrigerant increased. The pressure drops of refrigerant R-410A were the lowest while the pressure drop for refrigerant R-1234yf were the highest measured among the fluids investigated. The developmental refrigerants DR-5 and DR-5A showed identical characteristics in terms of pressure drop at both saturation temperatures of 4°C and 9°C.

1. INTRODUCTION

To meet the terms of the Montreal Protocol, CFCs and HCFCs have been gradually phased out and they have been replaced by new fluids that have zero ozone depletion potential. HFCs have provided an option with zero ozone depletion but their high global warming potential (GWP) might still be of some concern. A new effort started in the last decade, toward lowering the GWP value of refrigerants that will be used in the next generation of heating, ventilating, air conditioning, and refrigeration (HVAC&R) equipment for stationary applications. New refrigerants and new blends have been developed to meet this goal. For example, R-1234yf is a new HFO refrigerant designed to replace R-134a, and it has a GWP of only 4 (Forster, 2007). This refrigerant is used in medium pressure systems, but it seems that it is not a viable alternative for high pressure refrigeration cycle applications. Refrigerant R-32 has been proposed as alternative to R-410A in mini-split systems and is used primarily in China and Japan (Pham, 2010). R-32 has GWP value of about 675 (Forster, 2007) but unfortunately it is classified as mildly flammable, which might be of some concern to the HVAC&R industry as well as the end-users of the HVAC&R equipment. In order to develop new refrigerants for high pressure refrigeration cycles and mitigate the effect of GWP and flammability, blends of R-32 and R-1234yf have been investigated in the literature (Minor and Spatz, 2008, McLinden, 2011, Li et al., 2012). New developmental refrigerants were studied to retrofit R-410A in AC systems for stationary applications (Leck, 2010, Yana Motta et al., 2010). These new blends had GWP of less than 500 and provided good efficiency when they are used in AC systems as drop-in refrigerants or by adopting a soft optimization for the TXV (Barve and Cremaschi, 2012, Biswas and Cremaschi, 2012). System COP and capacity were promising but there is not much information on the heat transfer and pressure drop characteristics of these new low GWP refrigerants during two phase flow boiling processes inside the tubes of the evaporators. Research efforts are ongoing in order to investigate the effect of temperature glide on new low GWP developmental refrigerants in direct expansion equipment. This paper contributes to such research efforts and focuses on the flow boiling heat transfer coefficient and pressure drop in a horizontal tube with internally enhanced heat transfer surfaces. The expansion tube studied was made of copper, had a nominal diameter of 9.5 mm (0.375 in), and internal microgrooves. This tube is commercially available and commonly used in direct expansion evaporators. The two new development refrigerants investigated in this work are referred to as DR-5. and DR-5A (DR- stands for Developmental Refrigerant since these refrigerant do not have an official ASHRAE designation at the time of the writing of this paper). The first development refrigerant DR-5 is a mixture of R-32 and R-1234yf (Schultz and Kujak, 2012). It has a GWP of about 500 and a temperature glide of 1°C (1.8°F) (Leck, 2010). It is chemically stable, not corrosive, and has flammability characteristics of class 2L refrigerants (Leck and Yamaguchi, 2010). The second new developmental refrigerant, DR-5A, has similar properties to that of DR-5, GWP of 460, and it is expected to have better compatibility with POE oils.

2. BRIEF LITERATURE SUMMARY OF THE STATE-OF-THE-ART

Studies have been conducted on the performance of various refrigerants in micro-fin tubes (Balcilar et al., 2012). Jung et al. (2004) focused on a comparison of condensation heat transfer coefficients between smooth and micro fin tubes for several HFC refrigerants used today. It was pointed out that with micro-fin tubes all four different refrigerants tested showed a heat transfer coefficient enhancement of 2 to 3 times with respect to the smooth tube coefficient. Kim et al. (2002) investigated the evaporative heat transfer coefficients of R-410A in 7 and 9.52 mm smooth and microfin tubes. This work showed similar enhancements for R-410A in the 9.52 mm tube with respect to the plain tube. However the author noted the pressure drop penalty factor in the 7 mm tube relative to its smooth tube counterpart was as high as 160%. Yun et al. (2002) proposed a dimensionless correlation to predict the heat transfer augmentation of a microfin tube relative to the smooth tube performance. The resulting correlation yielded a 12% deviation from experimental data compared to the 21% deviation achieved using the previous smooth tube correlation. While the above studies focused on the enhancement of current HFC refrigerants (R-410A, R-134a, etc…), few studies explored the performance of new developmental refrigerants in microfin tubes. Li et al. (2012) studied the performance of refrigerants R-32, R-1234yf, and some blends of the two refrigerants during in-tube flow evaporation. Li et al. noticed that if the mass fraction of R-32 in the R-1234yf/R32 mixture was 20% then the heat transfer coefficient of the mixture of R-1234yf/R-32 was less than that of pure refrigerant R-1234yf. The results of their work seemed to suggest that for 50/50 blends the heat transfer coefficient was consistent for low to mid qualities, where at high qualities a steep drop in the heat transfer coefficient was observed. Li et al. explained that this phenomenon might be possibly associated with the local liquid film dry out. In Wu and Cremaschi (2013), the heat transfer coefficient and pressure drop for two phase flow boiling of R-32 and DR-5 were investigated for a smooth horizontal copper tube. The results showed similar trends to the ones of the Li et al. work but there was some scattering evident in their data. They attributed this behavior to the variation of heat flux during the laboratory tests for the various vapor quality of the refrigerant. From the brief review of the current state-of-the-art literature it appears that some studies provide data and correlations for predicting the two phase flow heat transfer and pressure drop of tubes with internal microfins with various refrigerants. However, to the authors' best knowledge, there is not any data available for the new DR refrigerants in microfin tubes. In addition the data of R-32 and R-1234yf in this tube geometry are sporadic and the range of operating conditions is quite limited. Thus, new data on these two low GWP refrigerants can expand the range of operating conditions for the correlations and provided further verification of existing two phase flow boiling heat transfer and pressure drop correlations in the literature.

3. EXPERIMENTAL FACILITY

Figure 1 shows a schematic of the experimental apparatus used for the heat transfer and pressure drop measurements of the present work. Subcooled refrigerant was circulated to a Coriolis-type flow meter with an accuracy of ±0.1% of the reading value. The temperature was measured with an in-line T-type thermocouple that had an accuracy of ±0.15K (0.3°F). For pressure measurements, an absolute piezo-transducer was used and its accuracy was ±0.13% of the full scale (full scale was 3550 kPa (500 psi)). Using the temperature and pressure measurements at the inlet of the preheater the enthalpy of the refrigerant at the inlet of the preheater was determined. The refrigerant then entered the preheater, which consisted of a counter flow tube in tube heat exchanger. Hot water was circulated in the outer tube of this heat exchanger. Using a Coriolis flow meter and thermocouples at the inlet and outlet of the water jacket, the total heat transfer rate from the water side to the refrigerant side was measured. Thus, the enthalpy and quality of the refrigerant exiting the preheater, which were identical to the enthalpy and quality at the inlet of the test section, were controlled. The mass flow of the refrigerant entering the test section was regulated by using a variable speed gear pump and a bypass loop as shown in the Figure 1. From the test section the refrigerant was circulated to a post-cooler where it was brought to subcooled liquid before entering the gear pump and starting the cycle again.

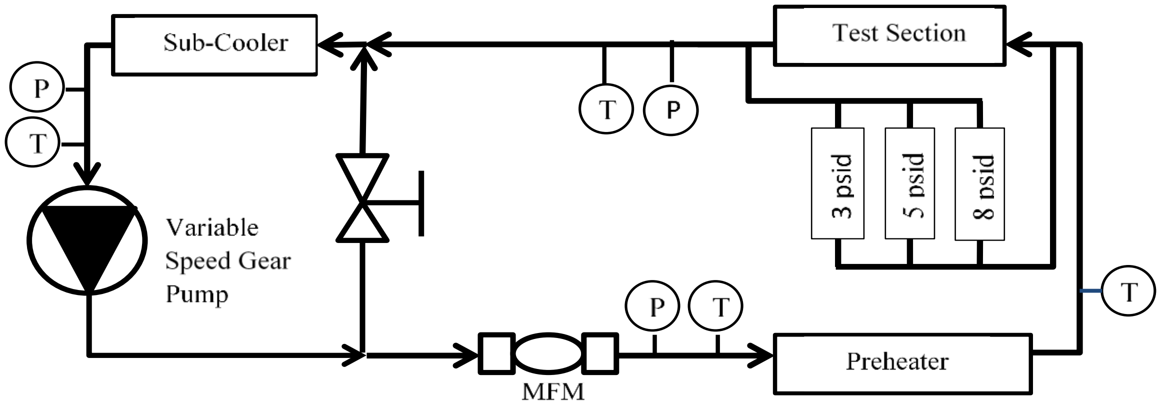

Figure 1: Schematic of the test apparatus and location of the instrumentation

The test section was a counter flow tube in tube heat exchanger with three tubes: the inner most tube, shown as tube 1 in Figure 2, is the internally enhanced heat transfer surface tube for the refrigerant, an intermediate smooth copper tube (tube 2), that housed the inner tube 1, and an outer most tube, tube 3, that served for the water jacket. Seven thermocouples were embedded in the gap between tube 1 and tube 2 and they measured the outer surface temperature of tube 1. The gap between the two tubes was filled with silver thermal paste that had a nominal thermal conductivity from 1 to 8.7 W/m-K (0.6 to 5 Btu/hr-ft-F). The thermal paste promoted heat conduction from tube 2 to tube 1 while the embedded thermocouples read the local copper tube temperature. Table 1 provides the geometry of the internally enhanced heat transfer surface tube (i.e. Tube 1 in Figure 2) used in the present work. The letter and symbols in Table 1 are described in the Nomenclature section.

Table 1: Geometry of the microfin copper tube used in the present work (i.e. Tube 1 in Figure 2 below)

Outer diameter, d_o	=	9.53 mm (0.375 in)	No. of microfins, N	=	60
Equivalent diameter, d_e	=	8.8 mm (0.35 in)	Wall thickness, t_w	=	0.3 mm (0.012 in)
Length, L	=	2.4 m (7.83 ft)	$A_{Sectional}$	=	60.8 mm^2 (0.094 in^2)
Helical Angle, β	=	18°	$A_{Surface}$	=	107,040 mm^2 (165.9 in^2)

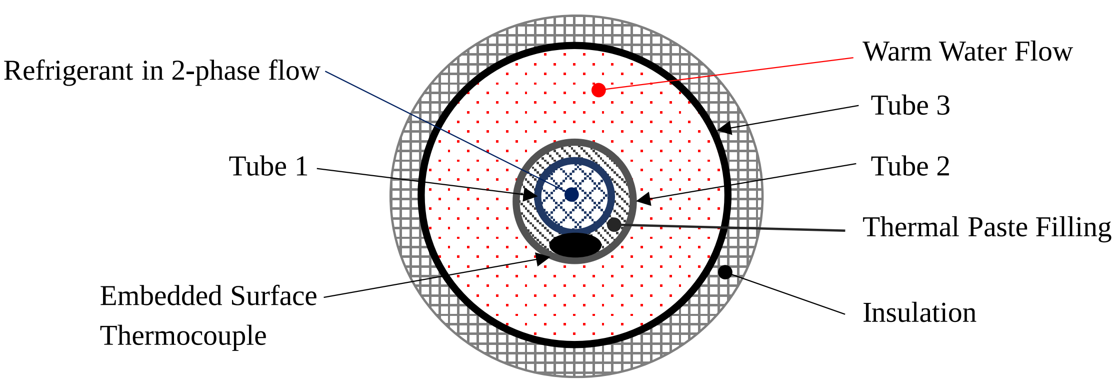

Figure 2: Schematic of the cross section of the three tubes of the test section with the location of the embedded surface thermocouples on the exterior surface of the refrigerant tube evaporator (that is, Tube 1)

The heat flux and wall temperature in the test section were controlled based on a thermal amplification technique shown in Figure 3 and previously presented in Garimella (2004). In the test section there were two loops connected in cascade by a plate heat exchanger. In the top loop water was circulated at very high velocity and the temperature difference across the test section, that is, $\Delta T=T_1-T_2$ in Figure 3, was kept below 1°C. The inlet and outlet temperatures were measured by in stream RTDs with an accuracy of ±0.05°C (0.1 °F). The heat transfer rate to the test section was calculated from the summation of the heat transfer rate across the plate heat exchanger plus the pump work. Both sources were directly measured as follows. In the bottom loop, water was circulated at low velocity and the temperature difference $\Delta T=T_5-T_6$ across the plate was fairly large, that is, in the range of 2 to 7°C (~3 to 13°F). The temperature measurements from this side of the brazed plate heat exchanger were used to determine the heat transfer exchange in the brazed plate heat exchanger from the bottom flow stream (between T_5 and T_6) to the top water loop. The pump work in the top loop, which was approximately constant, was measured at all time with a precision Watt meter transducer. All measurements were monitored using a LabView and National Instruments data logger. The data were sampled every two seconds and recorded for at least 30 minutes after steady state conditions were reached.

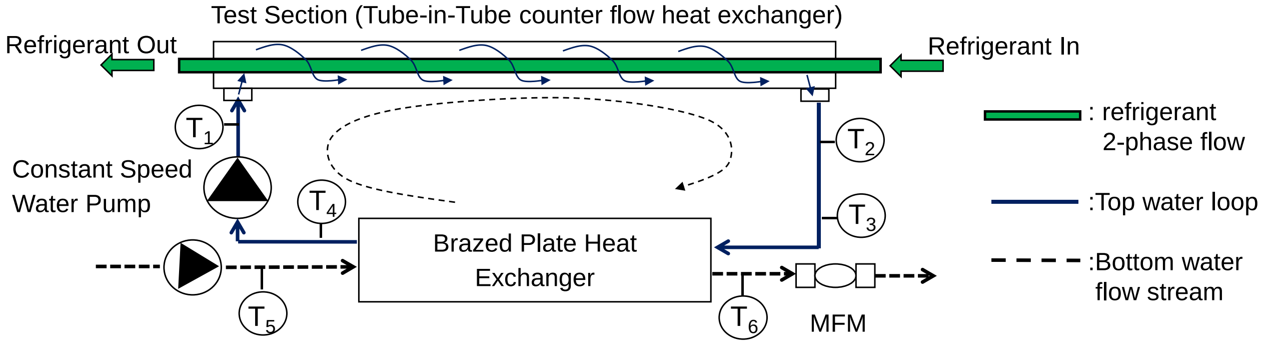

Figure 3: Schematic of water loops used for the thermal amplification technique on the test section

4. EXPERIMENTAL METHODOLOGY

An energy balance was conducted on the test section at the beginning of the experiments and at regular intervals during the experimental campaign. The heat transfer rate was measured from the water jacket side, Q_w, which was the primary method shown in eq. 1, and from the refrigerant side, Q_r, shown in eq. 2. The heat balance error on the preheater was within 2%.

$$Q_w = Q_{plate} + W_{pump} \tag{1}$$

$$Q_r = m_r * (h_{out} - h_{in}) \tag{2}$$

During the actual heat transfer experiments, the outlet of the preheater was in the two phase region and eq. 2.was used to solve for the outlet enthalpy of the refrigerant. With the measured pressure at the inlet of the test section and the enthalpy obtained from the preheater heat balance, the inlet quality of the refrigerant to the test section was computed according to eq. 3. The heat flux was calculated by using the actual inner surface heat transfer area of the internally enhanced tube, $A_{surface}$, (see eq. 4). Once the outlet quality of the refrigerant exiting the test section was obtained, the heat transfer coefficient and pressure drop were reported with respect to an average quality in the test section.

$$x_{in} = \frac{h_{out} - h_{in}}{h_{fg}} \tag{3}$$

$$q" = \frac{Q_w}{A_{surface}} \tag{4}$$

The internal wall temperature, $T_{wall,in}$, was determined from the radial heat conduction eq. 5, where d_o an d_i are the outer and inner diameter of the internally enhanced copper tube. The heat transfer coefficient, α, was calculated by using eq. 6, where the refrigerant saturation temperature, $T_{r,sat}$, was determined from the average temperature inside the internally enhanced tube (i.e. tube 1 in Figure 2).

$$T_{wall,in} = T_{wall,out} - \frac{q" * \ln(^{d_o}/_{d_i})}{2 * \pi * k} \tag{5}$$

$$\alpha = \frac{q"}{T_{r,sat} - T_{wall,in}} \quad (6)$$

Pressure drop in the test section was measured using one of three differential pressure transducers installed across the test section. The differential pressure transducers covered a range of 0-55 kPa (0-8 psi) but one transducer was used for the small range from 0 to 20 kPa (0 to 3 psi), one for the medium range from 0 to 34 kPa (0 to 5 psi) and one for the high range from 0 to 55 kPa (0 to 8 psi). The accuracy of each pressure transducer was ± 0.1% of the full scale.

The measured pressure drop was divided by the length between the two pressure taps in the test section in order to obtain an average pressure gradient along the test section for each saturation temperature and quality. By varying the saturation temperature in a parametric fashion the heat transfer coefficient and pressure drop of DR-5 and DR-5a were compared with those of R-410A, R-32, and R-1234yf when operating at similar conditions. The conditions tested are described in Table 2.

Table 2: Test Matrix and Legend for the Experiments in the present paper

Test Matrix		
Saturation Temperature (°C)	Mass Flux (kg/m^2-s)	Heat Flux (kW/m^2)
4	250	11 - 12
9	250	11 - 12

5. EXPERIMENTAL RESULTS AND DISCUSSION

5.1 Calibration of the Test Apparatus and Verification of the Experimental Procedure

Refrigerant R-410A was tested first in the newly developed test apparatus for calibration of the test apparatus and for establishing a series of data that were used as reference data. Preliminary tests were conducted prior to the actual heat transfer measurements in which the refrigerant R-410A exited the test section in superheated conditions. During these preliminary tests the heat balance in the preheater and in the test section combined resulted within less than 2%. Thus, the results verified that the test section was properly insulated and that the instrumentation was working properly. Then heat transfer experiments were conducted with R-410A in the two phase region across the test section and the heat transfer coefficient and pressure drop data of R-410A were compared with data from the literature. The data of heat transfer coefficient from the present work were similar to those of Li et al. (2012), in which they reported a similar tube geometry as the one of the present work. The pressure drop data of the present work were validated by using the semi-empirical correlation developed by Choi et al. (1999). They developed a two phase flow pressure drop correlation for a tube geometry that was similar to the one of the present work and the agreement between their predicted values of pressure drop and the present data for R-410A was within 20%.

5.2 Uncertainty Analysis

The uncertainty in the heat transfer and pressure drop measurements was estimated according to uncertainty propagation analysis (Taylor and Kuyatt, 1994) and the uncertainties calculated for each measurement type are given in Table 3. Uncertainty bars are reported in each figure of the result and discussion section for representative data points.

Table 3: Experimental Uncertainty of the main measured and calculated variables

Measured or Calculated Variable	Range	Uncertainty
Refrigerant Dew Point Temperature or Refrigerant Saturation Temperature	4°C - 9°C	±0.2 °C
Surface Temperature	4°C - 21°C	± 0.1 °C
Refrigerant Mass Flux	250 kg/m^{2}s	± 0.1 %
Heat Flux	11 to 12 kW/m^2	± 0.5 %
Heat Transfer Coefficient (R410-A)	3.3 - 5.5 kW/m^2-°C	± 6 %
Pressure Drop	2.2 – 8 kPa/m	± 0.5 %

5.3 Heat Transfer Coefficient of R-410A and of Low GWP Refrigerants

The heat transfer coefficients (αs) of the low GWP refrigerants investigated in the present work were normalized with respect to the maximum value of the heat transfer coefficient measured for R-410A, referred in this paper as to α_0. This heat transfer coefficient was α_0 = 5.15 kW/(m^2-°C) and it was measured for the low saturation temperature of 4°C and at refrigerant quality of about, x = 0.75. The pressure drop data were normalized with respect to the pressure drop Δp_0 associated with α_0, and it was Δp_0 = 5.34 kPa/m. Figure 4a shows the normalized two phase flow boiling heat transfer coefficients, α/α_0, for the various refrigerants tested in the present work. Refrigerant R-410A, which was selected as the baseline refrigerant, had normalized heat transfer coefficient that ranged from 0.93 to 1 when the refrigerant vapor quality increased from 0.34 to 0.74. The R-410A data, which are represented by the black circles and dashed line in Figure 4, indicated that the heat transfer coefficient monotonically increased with vapor quality in the range of vapor quality of the present work. Refrigerant R-32 showed similar trends for the heat transfer coefficients as that of R-410A and its α/α_0 ranged from 0.94 to 1.07. This means that R-32 had similar and up to 7% higher two phase flow heat transfer coefficient than that of R-410A at a saturation temperature of 4°C, similar mass flux, and similar heat flux conditions. Refrigerant R-1234yf had the normalized heat transfer coefficient α/α_0 that ranged from 0.84 to 0.88, shown by the square data points in Figure 4. The heat transfer coefficient of R-1234yf seemed to increase until the quality was about 0.6 and then they started to decrease at higher vapor quality. Although only one data point in Figure 4 supports this observation and further measurements are required to confirm this trend.

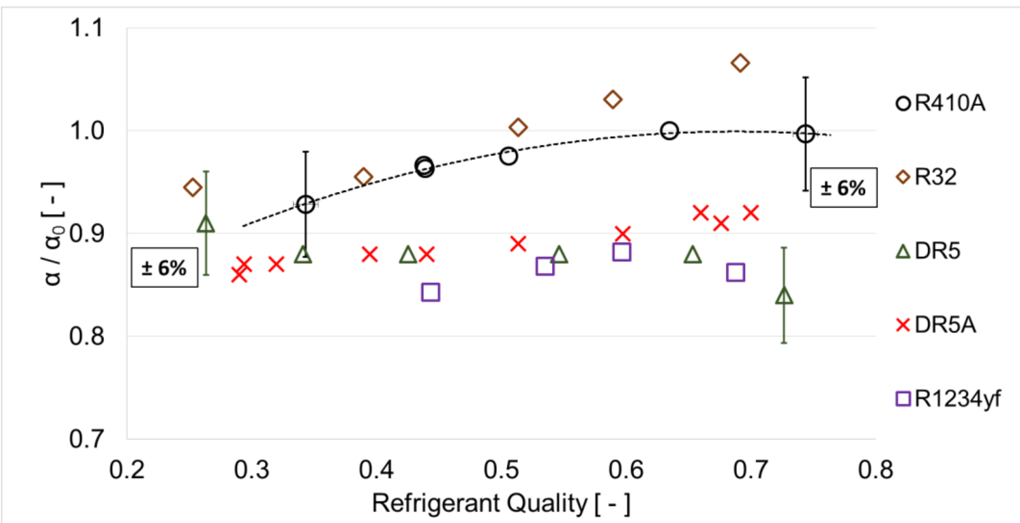

Figure 4: Refrigerant two phase flow boiling heat transfer coefficients for saturation temperature of 4°C (data were normalized with respect to R-410A heat transfer coefficient of α_0 = 5.15 kW/(m^2-°C)).

At 4°C, the new developmental refrigerants DR-5 and DR-5A had similar heat transfer coefficients and they varied between 0.84 and 0.92. The results in Figure 4 indicate that DR-5 had slightly higher heat transfer coefficient but the two measurements were actually within the uncertainty range for the heat transfer measurements of these developmental refrigerants. In addition, it could be noticed that while the heat transfer coefficient of DR-5 seemed to decrease with increasing vapor quality (see triangular data points in Figure 4), DR-5A heat transfer coefficients increased gradually if the vapor quality increased from 0.3 to 0.7. In other words, DR-5A followed the R410A and R32 trends while DR-5 seemed to follow the R-1234yf trend.

At higher dew point temperature of 9°C, the two phase flow boiling heat transfer coefficients of all refrigerants decreased but with different magnitude for each refrigerant, as shown in Figure 5. At this saturation temperature, it seems that DR-5 and R-32 have similar heat transfer coefficients within a vapor quality range of 0.3 to 0.6. At lower vapor quality DR-5 had higher heat transfer coefficient than R-32 while the opposite occurred if the vapor quality was higher than 0.6. Similar to the data at saturation temperature of 4°C, the heat transfer coefficients of DR-5A were lower than that of R-32 and close to that of R-1234yf. The normalized heat transfer coefficients for DR-5A were about 0.87 at low vapor quality, decreased to about 0.8 in the vapor quality range of 0.3 to 0.55, and increased to 0.84 if the vapor quality increased to 0.7. This trend was also observed for the heat transfer coefficients of DR-5, in which the α/α_0 were slightly higher than that of DR-5A. Because of the experimental uncertainty, careful attention should be paid to the generalization of the trends observed in the heat transfer coefficients. Uncertainty bars are reported in Figure 5 for four representative data points.

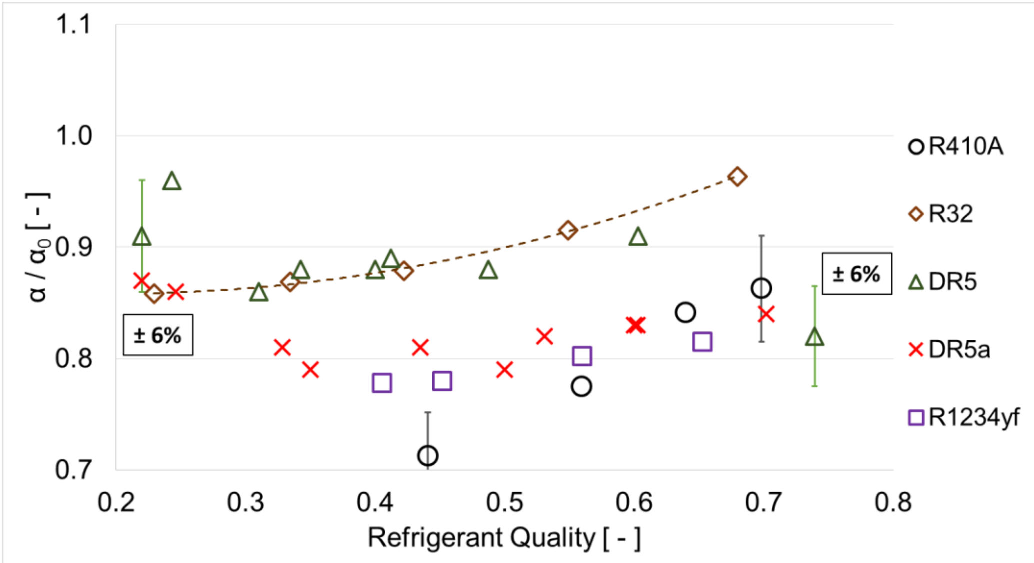

Figure 5: Refrigerant two phase flow boiling heat transfer coefficient for saturation temperature of 9°C (data were normalized with respect to R-410A heat transfer coefficient of α_0 = 5.15 kW/(m^2-°C)).

5.4 Pressure Drop of R-410A and of Low GWP Refrigerants

The two phase flow boiling pressure drop measurements for the various refrigerants at a saturation temperature of 4°C are shown in Figure 6. The uncertainty error bars were smaller than the symbols. For all refrigerants investigated the pressure drop increased monotonically if the vapor quality of the refrigerant increased. It should be noted that the pressure drop measurements reported in figures 6 and 7 were normalized with respect to the measured pressure drop Δp_0 associated with α_0. In Figure 6, refrigerant R-410A, which was the baseline refrigerant, had a normalized pressure drop that increased from about 0.6 to 1 when the vapor quality varied 0.3 to 0.75.

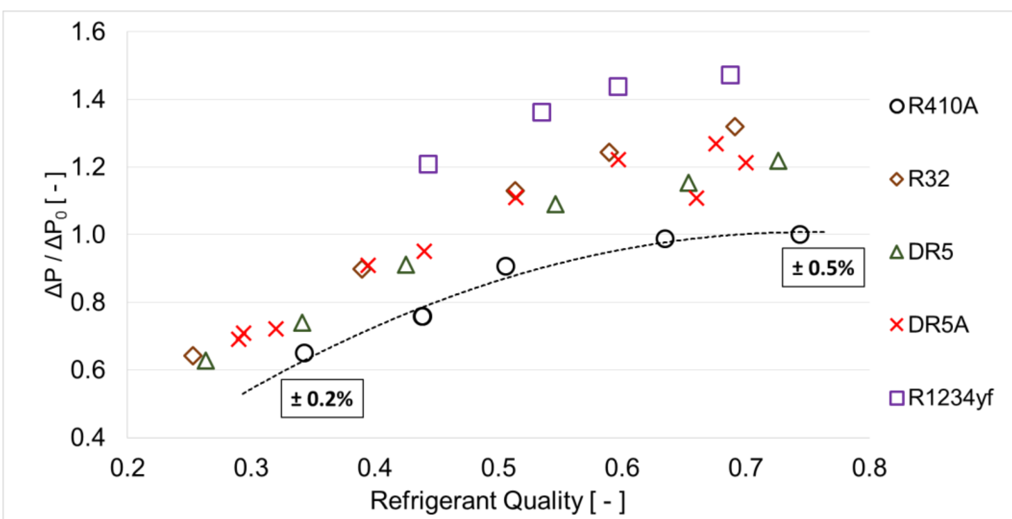

Figure 6: Refrigerant two phase flow pressure drop for saturation temperature of 4°C (data were normalized with respect to R-410A pressure drop of Δp_0 = 5.34 kPa/m).

At a 4°C saturation temperature, the pressure drop of the refrigerants R-32, DR-5, and DR-5A were approximately 20% higher than that of refrigerant R-410A at the same mass flux, similar heat flux, and at similar vapor quality. These are indicated by the diamond data points (R-32), triangle data points (DR-5), and cross data points (DR-5A) in Figure 6. The two phase flow pressure drop for the refrigerant R-1234yf were about 40% higher than that of R-410A at the same refrigerant mass flux, similar heat flux, and similar vapor quality.

Figure 7 shows the two phase flow boiling pressure drop for the refrigerants tested in the present work at saturation temperature of 9°C. The pressure drop had similar trends as the ones observed at a lower saturation temperature of 4°C, but at saturation temperature of 9°C they were lower in magnitude. For example, refrigerant R-410A had a normalized two phase flow boiling pressure drop that increased from about 0.4 to 0.85 when the vapor quality varied from 0.2 to 0.75. Refrigerants R-32, DR-5, and DR-5A had higher pressure drop at the same mass flux, similar heat flux, and similar vapor quality. In particular their pressure drop was significantly higher than that of R-410A when the vapor quality was above 0.6. This could be seen in Figure 7, in which the data of R-410A and the data of R-32, DR-5, and DR-5A become far apart toward the high vapor quality region.

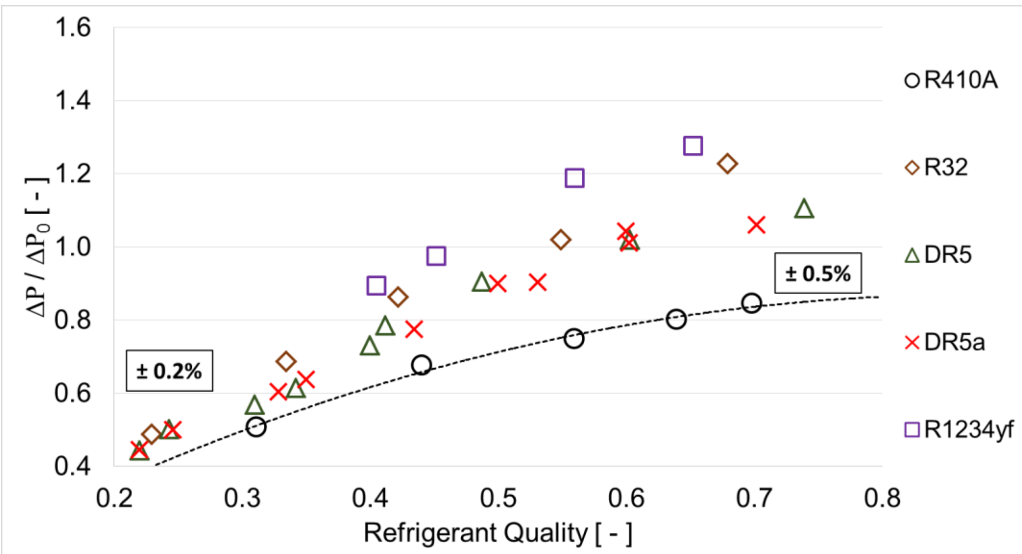

Figure 7: Two phase flow pressure drop for saturation temperature of 9°C (data were normalized with respect to R-410A pressure drop of Δp_0 = 5.34 kPa/m).

6. CONCLUSIONS

This paper presented new data for the two phase flow boiling heat transfer coefficient and pressure drop of two new LGWP developmental refrigerant alternatives for R-410A. Heat transfer measurements were conducted for a copper tube commonly used in direct expansion evaporators of air conditioning systems of 9.5 mm (0.375 in.) outside diameter and with an internally enhanced micro-finned surface. Data of local two phase flow heat transfer coefficient and pressure drop were presented for refrigerants R-410A, R-32, R-1234yf and the two new developmental refrigerants referred to as DR-5 and DR-5A.

The experimental findings from this work indicated that the refrigerant R-32 had similar and slightly higher heat transfer coefficient than that of R-410A at same refrigerant mass flux and similar heat flux conditions on the outer surface of the tube. Refrigerant R-1234yf had about 15 to 20% lower heat transfer coefficient than R410-A at 4°C saturation temperature. For this saturation temperature the developmental refrigerants DR-5 and DR-5A had heat transfer coefficients between R-32 and R-1234yf when the vapor quality ranged from 0.2 to 0.7. An increase of the saturation temperature from 4°C to 9°C decreased the heat transfer coefficients for all of the refrigerants tested.

For all refrigerants included in the present study, the two phase flow boiling pressure drops increased monotonically if the vapor quality of the refrigerant increased. The pressure drops of refrigerant R-410A were the lowest while the pressure drop for refrigerant R-1234yf were the highest measured among the fluids investigated. An increase of saturation temperature decreased slightly the pressure drop in two phase flow at the same mass flux and heat flux conditions. The developmental refrigerants DR-5 and DR-5A showed identical characteristics in terms of pressure drop at both saturation temperatures of 4°C and 9°C. Thus, from a heat transfer and pressure drop perspectives, DR-5A and DR-5 were similar with DR-5 possibly showing a slightly higher heat transfer coefficient than DR-5A. Further research is needed to expand this investigation to broader ranges of mass flux and heat flux so that existing heat transfer and pressure drop correlations for two phase flow boiling in internally enhanced horizontal tubes can be verified, and eventually improved, for the new low GWP refrigerants and the new developmental refrigerants discussed.

NOMENCLATURE

$A_{surface}$: Internally Grooved Tube Inner Surface Area, (m^2)
d_i: Inner Diameter of Evaporator Tube, (m)
d_o: Outer Diameter of Evaporator Tube, (m)
h: Enthalpy of Refrigerant, (kJ/kg)
h_{fg}: Latent Heat of Vaporization, (kJ/kg)
k: Thermal Conductivity of Evaporator Tube, (W/m°C)
m_r: Mass Flow rate of Refrigerant, (kg/s)
N: Number of Microfins in Cross-Section, (-)
ΔP: Pressure Drop per unit Length, (kPa/m)
ΔP_0: Reference Pressure Drop per unit Length, (kPa/m)
Q_{plate}: Heat Transfer Rate from Lower Water Loop, (kW)
Q_r: Refrigerant Side Heat Transfer Rate, (kW)
Q_w: Water Side Heat Transfer Rate, (kW)
q": Heat Flux to the Tube Test Section, (kW/m^2)
$T_{r,sat}$: Saturation Temperature of Refrigerant, (°C)
T_{wall}: Internally Grooved Tube Wall Surface Temperature, (°C)
t_w: Thickness of Internally Grooved Tube Wall, (mm)
W: Pump Work in to the Water Flow, (kW)
x: Vapor Quality of the Two-Phase Refrigerant, (-)
α: Heat Transfer Coefficient (HTC), (kW/m^2°C)
α_0: Reference Heat Transfer Coefficient (HTC), (kW/m^2°C)
β: Helical Angle of the Microfins, (°)

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