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Horizontal convective boiling of R448A, R449A, and R452B within a micro-fin tube

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Abstract

This article presents local convective boiling measurements in a micro-fin tube for three low global warming potential refrigerants: R448A, R449A, and R452B¹. An existing correlation was modified to predict multi-component mixtures, which predicted 98% of the measurements to within $\pm 20\%$. The new correlation was used to compare the heat transfer coefficient of the three test fluids at the same heat flux, saturated refrigerant temperature, and refrigerant mass flux. The resulting comparison showed that refrigerant R452B exhibited the highest heat transfer, in large part due to its approximately 28% larger liquid thermal conductivity and smaller temperature glide as compared to the tested low-global warming potential refrigerants. For the example case, the heat transfer coefficient for R449A was approximately 8% larger than that for R448A, while the heat transfer coefficient for R452B was more than 59% larger than either R448A or R449A. The heat transfer coefficients for R448A and R449A were roughly between 26 and 48% less than that of R404A for the example case. In contrast, the model predicts that the R452B heat transfer coefficient was approximately 13% larger than that of R404A for the same conditions.

Introduction²

Internally enhanced tubes, like the micro-fin tube, are used by most manufacturers in the construction of evaporators and condensers for new unitary refrigeration and air-conditioning equipment. The reason for the micro-fin tube's hold on unitary equipment is that it provides the highest heat transfer with the lowest pressure drop of the commercially available internal enhancements (Webb and Kim 2005). Most of the experimental measurements for evaporative heat transfer coefficients in the micro-fin tube have been done for traditional refrigerants like R134a. Pressure from the policies set by the Montreal Protocol (1987), the Kyoto Protocol (1997), and the European Mobile Directive (2006) have caused a recent shift to refrigerants with both zero ozone depletion potential (ODP) and low global warming potential (GWP). Although two current workhorse refrigerants, R410A and R404A, for air-conditioning and refrigeration, respectively, have zero ODP, they both have rather large GWP: 1725 and 3300, respectively (IPCC 2013). Three new refrigerants,

¹R452B is the pending ASHRAE designation for the mixture R32/R125/R1234yf (67/7/26), which has the developmental designation DR-55.

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²Certain trade names and company products are mentioned in the text or identified in an illustration in order to adequately specify the experimental procedure and equipment used. In no case does such an identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the products are necessarily the best available for the purpose.

R448A, R449A, and R452B, are potential low GWP replacements for R404A (R125/143A/134A [44/52/4]).³ In addition, R452B is also a proposed replacement for R410A (R32/125 [50/50]). R448A, R449A, and R452B are of interest as replacements because they have zero ODP and 100 year GWP of approximately 1274 (IPCC 2013), 1283 (Minor et al. 2015), and 676 (Minor 2016), respectively. Consequently, flow boiling heat transfer data for the micro-fin tube with R448A (R32/125/134a/1234ze/1234yf [26/26/21/7/20]), R449A (R32/125/134a/1234yf [24.3/24.7/25.7/25.3]), and R452B⁴ (R32/125/1234yf [67/7/26]) are essential for the evaluation of their use for unitary applications.

The bulk of the research on flow boiling in micro-fin tubes has been for well-established alternative refrigerants with GWP greater than 1200. For example, Targanski and Cieslinski (2007) measured the evaporation heat transfer characteristics of R407C inside a micro-fin tube in the presence of oil. Zhang et al. (2007, 2015) measured the evaporation heat transfer coefficients of R417A and R22 inside a micro-fin tube and introduced a new heat transfer correlation to predict their values. Yun et al. (2002) examined existing experimental data and developed a model, which was validated for R22, R113, R123, R134a, and R410A and a variety of micro-fin tube geometries. Seo and Kim (2000), Yu et al. (2002), and Kim et al. (2002) measured the flow boiling heat transfer coefficient in micro-fin tubes for R22, R134a, and R410A, respectively. Wellsandt and Vamling (2005) have investigated in-tube evaporation of R134a in a special type of micro-fin tube where the fin rifling, instead of being continuous, is arranged into V-grooves that resemble herringbones; hence, it is called the herringbone micro-fin tube. Olivier et al. (2004) have also studied the two-phase heat transfer performance of a herringbone and a standard 18-deg helical micro-fin tube with R22, R134a, and R407C.

Recent research on alternative refrigerants with GWP less than 1300 have been done for smooth tubes. For example, Grauso et al. (2013), and Hossain et al. (2013) have provided flow boiling studies with R1234ze(E) in horizontal smooth tubes. Three studies used electrically heated test rigs to measure the local flow boiling heat transfer coefficient in smooth tubes. The first, Meng et al. (2013), tested R141b in vertical and inclined serpentine tubes. The second, Nasr et al. (2015), tested R600a in a horizontal plain tube. The third, Zou et al. (2010), tested two different R290/R152a mixtures in a horizontal tube.

Most if not all of the relatively recent research on flow boiling in micro-fin tubes for refrigerants having GWP less than 1300 has been done for R134a and R410A replacements rather than for those intended to replace R404A. For example, Mendoza-Miranda et al. (2015) and Diani et al. (2015) have experimentally investigated the heat transfer performance of R1234yf in micro-fin tubes. Diani et al. (2015) made measurements on a single tube and found that they compared well with heat transfer correlations from the literature. Mendoza-Miranda et al. (2015) measured the overall heat transfer performance of a shell-and-tube heat exchanger in an operating refrigerant cycle. They used micro-fin tube correlations from the literature and found that the heat transfer performance of a shell-and-

³All compositions are by mass.

⁴R452B is the pending ASHRAE designation for the mixture R32/R125/R1234yf (67/7/26), which has the developmental designation DR-55.

tube heat exchanger with micro-fin tubes could be successfully modeled. Kondou et al. (2013) measured the flow boiling heat transfer coefficient of R32/R1234ze(E) and its components in a horizontal micro-fin tube and, they developed a new correlation to predict their zeotropic mixture heat transfer coefficients.

Because of the relatively recent introduction of R448A, R449A, and R452B, measured heat transfer data in a micro-fin tube are not available in the literature for these refrigerants. Consequently, the present study provides measured local flow boiling heat transfer for these three low-GWP refrigerants in a micro-fin tube for test conditions that are applicable for air-conditioning applications.

Experimental apparatus

Figure 1 shows a sketch of the experimental apparatus used to establish and measure convective boiling heat transfer coefficients. The experimental test facility consisted of two main systems: the refrigerant loop and water loop. The refrigerant flow rate, pressure, and superheat were fixed at the inlet to the test section. The water flow rate and the inlet temperature were fixed to establish the overall refrigerant quality change in the test section. The water temperature drop, the tube wall temperature, the refrigerant temperatures, pressures, and pressure drops were measured at several axial locations along the test section. These measurements were used to calculate the local heat-transfer coefficient for the micro-fin tube.

The test section consisted of a pair of 3.34 m long, horizontal tubes connected by a U-bend. A fixed test pressure was maintained by balancing the refrigerant duty between the subcooler, the test section, and the evaporator. A magnetically coupled gear pump delivered the test refrigerant to the entrance of the test section with a few degrees of vapor superheat. Another magnetically coupled gear pump supplied a steady flow of water to the annulus of the test section. The inlet temperature of the water loop was held constant for each test with a water chilled heat exchanger and variable electric heaters. The refrigerant and water flow rates were controlled by varying the pump speeds using frequency inverters. Redundant flow rate measurements were made with Coriolis flowmeters and acoustic flowmeters for both the refrigerant and water sides.

Figure 2 shows a cross-section of the test section with a detail of the micro-fin tube geometry. The test refrigerant flowed inside a micro-fin tube, while distilled water flowed either in parallel flow or counterflow to the refrigerant in the annulus that surrounded the micro-fin tube. Having some tests in parallel flow and others in counterflow produced a broad range of heat fluxes at both low and high flow qualities. The annulus gap was 2.2 mm, and the micro-fin tube wall thickness was 0.3 mm. The micro fin tube had 60, 0.2 mm high fins with an 18 degree helix angle. For this geometry, the cross-sectional flow area was 60.8 mm², giving an equivalent smooth diameter (D_e) of 8.8 mm. The root diameter of the micro-fin tube was 8.9 mm. The inside-surface area per unit length of the tube was estimated to be 44.6 mm. The hydraulic diameter (D_h) was measured with a polar planimeter from a scaled drawing of the tube cross-section and determined to be approximately 5.45 mm. The ratio of the inner surface area of the micro fin tube to the surface area of a smooth tube of the same

D_e was 1.6. The fins rifled down the axis of the tube at a helix angle of 18° with respect to the tube axis. Table 1 summarizes the geometric parameters of the micro-fin tube.

Figure 3 provides a detailed schematic of the test section. The annulus was constructed by connecting a series of tubes with 14 pairs of stainless steel flanges. This construction permitted the measurement of both the outer micro-fin wall temperature and the water temperature drop as discussed in the following two paragraphs. The design also avoided abrupt discontinuities such as unheated portions of the test section and tube-wall “fins” between thermopile ends.

Figure 3 shows that thermocouple wires passed between 12 of the gasketed flange pairs to measure the refrigerant-tube wall temperature at 10 locations on the top, side, and bottom of the tube wall. These locations were separated by 0.6 m on average, and they were located near the intersection of the shell flanges. In addition, thermocouples were also mounted next to the pressure taps near the middle of each test section length. The thermocouple junction was soldered to the outside surface and sanded to a thickness of approximately 0.5 mm. The leads were strapped to a thin nonelectrically conducting epoxy layer on the wall for a distance of 14.3 mm before they passed between a pair of the shell flanges. The wall temperature was corrected for a heat flux dependent fin effect. The correction was typically 0.05 K. Figure 3 also shows that a chain of thermopiles was used to measure the water temperature drop between each flange location. Each thermopile consisted of 10 thermocouples in series, with the 10 junctions at each end evenly spaced around the circumference of the annulus. Because the upstream junctions of one thermopile and the downstream junctions of another entered the annulus at the same axial location (except at the water inlet and outlet), the junctions of the adjacent piles were alternated around the circumference. A series of Teflon half-rings attached to the inner refrigerant tube centered the tube in the annulus. The half-rings were circumferentially baffled to mix the water flow. Mixing was further ensured by a turbulent water Reynolds number (Kattan et al. 1995).

As shown in Figure 3, six refrigerant pressure taps along the test section allowed the measurement of the upstream absolute pressure and five pressure drops along the test section. Two sets of two water pressure taps were used to measure the water pressure drop along each tube. Also, a sheathed thermocouple measured the refrigerant temperature at each end of the two refrigerant tubes, with the junction of each centered radially. Only the thermocouple at the inlet of the first tube was used in the calculations. The entire test section was wrapped with 5 cm of foam insulation to minimize heat transfer between the water and the ambient.

Measurements

Table 2 shows the expanded measurement uncertainty (U) of the various measurements along with the range of each test parameter in this study. The U was estimated with the law of propagation of uncertainty. All expanded measurement uncertainties are reported at the 95% confidence level. The estimates shown in Table 2 are median values of U for the correlated data. Saturated refrigerant properties were evaluated at the measured saturation pressure with the REFPROP (Lemmon et al. 2013) equation of state while using refrigerant-vendor proprietary fluid files. Table 3 shows representative properties that were obtained

from REFPROP for the fluids germane to this study. The liquid dynamic viscosity as calculated with REFPROP for R449A did not agree with that provided by the fluid vendor.⁵ The left side of Table 3 shows properties that were evaluated for the average test conditions of each fluid, while the right side of the table provides properties that were evaluated at 277.6 K.

The convective boiling heat transfer coefficient based on the actual inner surface area ($h_{2\phi}$) was calculated as:

$$h_{2\phi} = \frac{q''}{T_w - T_s} \quad (1)$$

where the measured wall temperatures (T_w) were fitted to their axial position to reduce the uncertainty in the measurement.

Figure 4 shows the estimated expanded uncertainty of the wall temperature fit for all the measurements as a function of thermodynamic quality. Figure 4 includes some data that was omitted from the correlation, explained in the Results section. The uncertainty of roughly 90% of the fitted wall temperatures was less than 0.5 K at the 95% confidence level. The median of the uncertainty in T_w as shown in Table 2 was approximately 0.3 K.

The water temperature (T_f) was determined from the measured temperature change obtained from each thermopile and the inlet water temperature measurement. The water temperature gradient (dT_f/dz) was calculated with second-order finite difference equations using the measured water temperature and their locations along the tube length z . The water temperature gradients were then fitted with respect to the tube length. As a check on the water temperature gradient calculation, Figure 5 shows that the measured water temperatures (open circles) typically agreed with the integrated fit of the water temperature gradient (solid line) to within 0.2 K.

The fitted, local, axial water temperature gradient (dT_f/dz) the measured water mass flow rate ($\dot{m}_f$), and the properties of the water were used to calculate the local heat flux (q'') to the micro-fin tube based on the actual inner surface area:

$$q'' = \frac{\dot{m}_f}{p} \left(c_{pf} \frac{dT_f}{dz} + \nu_f \frac{dP_f}{dz} \right) \quad (2)$$

where p is the wetted perimeter of the inside of the micro-fin tube. The specific heat (c_{pf}) and the specific volume (ν_f) of the water were calculated locally as a function of the water temperature. The water pressure gradient (dP_f/dz) was linearly interpolated between the pressure taps to the location of the wall thermocouples. The pressure gradient term was typically less than 3% of the temperature gradient term. The heat flux obtained by Equation

⁵For this case, the following equation was used, which was obtained from a fit the vendor data for the temperature range of this study: $\mu_r = 4548.61 - 38.4351 T_r + 0.114838 T_r^2 - 0.119168 \times 10^{-3} T_r^3$ where μ_r and T_r have units of $\mu\text{Pa s}$ and K, respectively.

2 was reduced by the amount of heat lost to the surroundings. The heat loss to the surroundings was obtained by calibration of single phase heat transfer tests and it was based on the temperature difference between the room and the test fluid. Typically, the heat loss correction was less than a 0.1% correction of Equation 2. Kedzierski and Kang (2016) plot the relative uncertainty of the heat flux measurement versus thermodynamic quality for this study. In general, the uncertainty of the heat flux remains less than 40% of the measured value, while the average uncertainty for the counterflow and the parallel flow data is approximately 8 and 19% of the measured value, respectively.

Figure 6 shows example plots of the local heat flux as calculated from Equation 2 versus thermodynamic quality for both cases when the water and the refrigerant are in counter-flow and parallel flow, respectively. Both heat flux profiles are for R449A at an all-liquid Reynolds number (Re) of roughly 6800 and a refrigerant reduced pressure of approximately 0.16, which was evaluated at the exit of the test section. The discontinuity exhibited in the heat flux profiles is due to the change in refrigerant saturation temperature as caused by the adiabatic pressure drop in the bend that is used to transition from the first leg of the test section to the second leg. The decrease in the refrigerant saturation temperature causes an increase in the difference between the water and the refrigerant temperature, which leads to an increase in the local heat flux. For the counterflow case, the heat flux increases from approximately $2 \text{ kW}\cdot\text{m}^{-2}$ at a quality near zero to approximately $12 \text{ kW}\cdot\text{m}^{-2}$ at a quality slightly greater than 0.7. The parallel flow case has the opposite slope and a slightly different range of that for counterflow, for which the heat flux decreases from approximately $25 \text{ kW}\cdot\text{m}^{-2}$ at a quality near 0.11 to approximately $3 \text{ kW}\cdot\text{m}^{-2}$ at a quality of approximately 0.9.

The thermodynamic and transport properties were calculated with version 9.1 of REFPROP (Lemmon et al. 2013) using enthalpy and pressure as inputs. The enthalpy of the refrigerant liquid at the inlet of the test section was calculated from its measured temperature and pressure. The subsequent increase in refrigerant enthalpy along the test section was calculated from the local heat flux and the measured refrigerant mass flow rate. The refrigerant pressures were measured at six pressure taps along the test section. The pressure was linearly interpolated between the taps. The refrigerant entering the test section was near saturation and held to a saturation temperature of approximately 277.6 K for all of the tests. Considering that the tests were done for quality ranges between near zero and slightly greater than 0.7, the saturation temperature of R448A and R449A both increased from roughly 277.6 to 282.0 K for most tests. By comparison, the variation of the saturation temperature of R452B from the inlet temperature, as caused by the pressure drop and the small temperature glide, was generally less than 1 K.

The local Nusselt number (Nu) was calculated using the hydraulic diameter and the heat transfer coefficient based on the actual inner surface area of the tube as:

$$\text{Nu} = \frac{h_{2\phi} D_h}{k_l} \quad (3)$$

Figure 7 shows the relative uncertainty of the Nu versus thermodynamic quality was between roughly 10 and 40%. Measurements of Nu with uncertainties greater than 40% were discarded. The average uncertainty of Nu for presented data was approximately 25% for all qualities.

Results

The 816 data points generated in this study for R448A, R449A, and R452B are tabulated in Kedzierski and Kang (2016), which contains the Nusselt and all-liquid Reynolds numbers and other reduced data that are typically used to characterize flow boiling. Kedzierski and Kang (2016) also provide tabulated raw data measurements including the heat flux and the wall and water temperatures and locations. The column entitled, “flow,” provides a “C” or a “P” to indicate that the measurements were made for either counterflow or parallel flow between the refrigerant and water, respectively.

The measured local convective boiling Nus were compared to the pure-refrigerant (single component) version of the Hamilton et al. (2008) correlation:

$$\text{Nu}_p = 482.18 \text{Re}^{0.3} \text{Pr}^{C_1} \left(\frac{P_s}{P_c} \right)^{C_2} \text{Bo}^{C_3} \left(-\log_{10} \frac{P_s}{P_c} \right)^{C_4} M_w^{C_5} \quad (4)$$

where

$$C_1 = 0.51x_q$$

$$C_2 = 5.57x_q - 5.21x_q^2$$

$$C_3 = 0.54 - 1.56x_q + 1.42x_q^2$$

$$C_4 = -0.81 + 12.56x_q - 11.00x_q^2$$

$$C_5 = 0.25 - 0.035x_q^2$$

Here, the all-liquid Reynolds number (Re), the Boiling number (Bo), the liquid Prandtl number (Pr), the reduced pressure (P_s/P_c), and the quality (x_q) are all evaluated locally at the saturation temperature. The all-liquid Reynolds number and the Nu are based on the hydraulic diameter (D_h). The Nu is also based on the actual inner surface area of the tube.

Hamilton et al. (2008) also provide a correction factor for Equation 4 to predict the heat transfer coefficient for binary mixtures, which cannot be used for mixtures of more than two components. Consequently, a new correction for Equation 4 was developed to predict the flow boiling Nu for mixtures of any number of refrigerants. This was done by multiplying the single-component Nusselt Number (Nu_p) by a modifier to predict multi-component mixtures:

$$\text{Nu} = \text{Nu}_p \left(1 - 36.23 \left[\frac{T_d - T_b}{T_b} \right] e^{-0.007 \text{ReBo}^{0.47}} \right) \quad (5)$$

where T_d and T_b are the dew point and bubble point temperatures, respectively, evaluated at the local saturation pressure and overall composition of the mixture. The $T_d - T_b$ difference is commonly called the temperature glide of the mixture. Typically, large temperature glides cause concentration gradients that lead to heat transfer degradations as compared to what would be expected from a single component prediction model (Kedzierski et al. 1992). Consequently, the bracketed term in Equation 5 that multiplies Nu_p describes the mixture degradation effect, which is a function of temperature glide, Bo and Re. A single component refrigerant would have zero temperature glide, which would result in the mixture degradation effect, represented by the bracketed term, being equal to one.

The flow map of Yu et al. (2002) for micro-fin tubes was used to determine that approximately 87% of the measurements were in annular or semi-annular flow. Manwell and Bergles (1990) suggest that the reason annular-like flow is a strong characteristic of micro-fin tubes is that the spiraling fins along the tube axis encourage wetting of the upper tube wall.

Figure 8 shows a comparison between the boiling Nus predicted with Equation 5 for the micro-fin tube to those measured here for R448A, R449A, and R452B. Equation 5 predicts approximately 98% of the measured convective boiling Nus for R448A, R449A, and R452B in the micro-fin tube to within approximately $\pm 20\%$. The measurements for each fluid are roughly centered about the mean of the correlation suggesting a lack of bias in the prediction due to the different fluids or some other cause. The gray dashed lines are multi-use 95% confidence intervals on the mean prediction, which vary from $\pm 16\%$ of the prediction at a Nu of approximately 50 to less than $\pm 5\%$ for Nus larger than 100. If the uncertainty in the fluid properties are neglected and the model is assumed to be correct, then the source for the uncertainty of the mean resides entirely in the uncertainty of the fitted constants. If this is true, then the uncertainty in the ratio of the heat transfer coefficients as predicted with Equation 5 for two different fluids is no greater than the uncertainty of the mean for Equation 5 as shown in Figure 8 for the 95% confidence level, which is less than $\pm 5\%$ for Nus larger than 100.

Representative plots of the heat transfer coefficient ($h_{2\phi}$) versus thermodynamic quality (x_q) are given in Figures 9 and 10. The solid lines are predictions for the present micro-fin tube geometry, which were obtained from the modified Hamilton et al. (2008) correlation given in Equation 5. The symbols are the measured data points, while the shaded regions between the dashed lines provide the measurement uncertainty for a 95% confidence level. The uncertainty in the R448A and R449A heat transfer coefficients is shown to be roughly $500 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$ for most of data for qualities greater than 20%. For these measurements, the uncertainty in the heat flux is the greatest contributor to the uncertainty in the heat transfer coefficient. The uncertainty in the R452B heat transfer coefficient is approximately three times larger than that of the other refrigerants for this particular example and is shown to be roughly $1500 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$ for most of data for qualities greater than 20%. For these measurements, the uncertainty in the heat flux is the greatest contributor to the uncertainty in the heat transfer coefficient. This is primarily due to the larger heat transfer coefficients measurements. More specifically, because of the larger heat transfer coefficients, and with it, the associated smaller driving temperature difference, the uncertainty in the heat flux and the

uncertainty in the driving temperature difference contribute nearly equally to the uncertainty in the heat transfer coefficient, resulting in a larger total uncertainty.

Figure 9 shows the local heat transfer coefficient for R448A, R449A, and R452B for $G_t \sim 100 \text{ kg s}^{-1}\text{m}^{-2}$ and $P_s/P_c \sim 0.16$ ($T_s/T_c \sim 0.79$) with counterflow between the refrigerant and the water. The P_s/P_c ratio is evaluated at the exit of the test section. As shown in Figure 6, the counterflow condition provided increasing heat flux with increasing thermodynamic quality. As the convective boiling heat transfer coefficient is moderately dependent upon the heat flux, the increasing heat flux and the thinning liquid films on the wall cause the heat transfer coefficient to increase with respect to quality. Equation 5 is shown to predict both the R452B and the R449A measurements to within approximately $100 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$, for example, approximately ± 3 and $\pm 5\%$, respectively. The maximum deviation between the Equation 5 predictions and the measurements for R448A is larger for this case being approximately $200 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$, or approximately $\pm 12\%$. Overall, the average difference between the measurements and the predictions for all of the counterflow measurements for all the test fluids is approximately $\pm 7\%$.

Figure 10 shows the local heat transfer coefficient for R448A, R449A, and R452B for $G_t \sim 200 \text{ kg s}^{-1}\text{m}^{-2}$ and $P_s/P_c \sim 0.16$ ($T_s/T_c \sim 0.79$) with parallel flow between the refrigerant and the water. As shown in Figure 6, the parallel flow condition provided decreasing heat flux with increasing thermodynamic quality. Because of the high heat flux for qualities less than 20%, it is likely that nucleate boiling may be a larger contribution to the total heat transfer than it is for the counterflow condition, thus contributing to the large heat transfer coefficient the low quality region. For qualities larger than 20%, the effects of decreasing heat flux and the thinning liquid films on the wall cause the heat transfer coefficient to moderately decrease with respect to quality. Equation 5 is shown to predict both the R449A and the R449B measurements to within approximately $200 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$, for example, approximately $\pm 5\%$, for qualities less than 0.7. The maximum deviation between the Equation 5 predictions and the measurements for R452B is larger for this case being approximately $500 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$, or approximately $\pm 9\%$. Overall, the average difference between the measurements and the predictions for all of the parallel flow measurements for all the test fluids is approximately $\pm 10\%$.

The main purpose of Figures 9 and 10 was to compare the Equation 5 predictions to the measurements for each fluid at as similar as conditions as the present data set would allow. Even though the measurements were compared at nearly the same mass velocity, $100 \text{ kg s}^{-1}\text{m}^{-2}$ for Figure 9 and $200 \text{ kg s}^{-1}\text{m}^{-2}$ for Figure 10, the local heat flux could vary significantly between fluids. For the Figure 9 example, the heat flux for R449A varied from near zero to approximately $4.5 \text{ kW}\cdot\text{m}^{-2}$, while that for R448A varied from approximately $1.5 \text{ kW}\cdot\text{m}^{-2}$ to $3.2 \text{ kW}\cdot\text{m}^{-2}$. This illustrates that the maximum heat flux for the R449A counterflow data set example was approximately 41% larger than that for R448A, which accentuated the difference in the measured heat transfer coefficients between the two fluids. For this reason, it is important to use a validated model to compare the performance of the fluids at identical conditions in order to establish a fair comparison of relative fluid heat transfer performance.

Figure 11 uses Equation 5 to illustrate the relative heat transfer performance of R448A, R449A, and R452B versus quality for the same saturated refrigerant inlet temperature ($T_{r,i} = 277.6$ K), and the same refrigerant mass flux ($G_r = 100$ kg·m⁻²·s⁻¹) for the present micro-fin tube geometry. Both counterflow and parallel flow conditions are shown. The counterflow heat flux is approximated with $q'' = 15.1x_q^{0.51}$ kW·m⁻², while the parallel flow is approximated with $q'' = (29.6 - 29.8x_q)$ kW·m⁻². The heat flux profiles with respect to quality that were used to calculate the heat transfer coefficient are approximately equivalent to those shown in Figure 6. Five different line styles for each flow condition are used to represent the predictions for the three different test fluids and two baseline refrigerants, R410A and R404A, as labeled.

In general, for counterflow, Figure 11 predictions show that the boiling heat-transfer coefficient rapidly increases with increasing quality for qualities less than 20%. For quality ranges between 20 and 70%, the rate of increase in the heat transfer coefficient with respect to increasing quality is roughly a fourth of that for qualities less than 20%. For parallel flow, the heat transfer coefficient for all the fluids decreases with increasing quality in response to the decreasing heat flux with respect to increasing quality. For the example case presented here, the heat transfer coefficient for R449A is approximately 8% larger than the heat transfer coefficient for R448A for qualities between 10 and 70% for both counterflow and parallel flow conditions. For both counterflow and parallel flow, the heat transfer coefficients for R449A and R448A are roughly 1.5 kW·K⁻¹·m⁻² less than that of R404A for qualities between 10 and 70%. R449A has a larger heat transfer coefficient than R448A due partly to its 2% larger thermal conductivity and its 6% smaller temperature glide as illustrated in Table 3. As a percentage, the heat transfer coefficients for R449A and R448A are 26 to 43% and 31 to 48%, respectively, less than that of R404A. The smaller heat transfer coefficient of the alternative refrigerants as compared to that of R404A is primarily due to the larger mixture degradation effect as calculated by the bracketed right-side of Equation 5 being on average 0.61, 0.63, and 0.97 for R449A, R448A, and R404A, respectively. This degradation factor reduces the single component prediction by an amount that is sufficient to account for the mixture degradation effects. A value of 0.97 for R404A indicates that the flow boiling performance of R404A closely matches (within 3%) that of a single component refrigerant because of its negligible temperature glide. In addition, R404A has a 21% larger Prandtl number than either R449A or R448A, which also contributes to the favorable performance of R404A as compared to the two low-GWP refrigerants examined here.

The 8% difference in heat transfer performance between R449A and R448A may not significantly affect the cycle performance of an air conditioner because the air-side heat transfer resistance typically dominates the overall resistance of a refrigerant-to-air, air-conditioning evaporator. As a result, the change in the overall conductance of the evaporator would be limited to some fraction of the 8% change in refrigerant-side heat transfer. In addition, the uncertainty of the heat transfer ratio prediction for this case is approximately $\pm 5\%$, which puts the range of possible overall difference roughly between 3 and 13% for 95% confidence. Finally, the heat transfer performance shown in Figure 11 is only a representative performance, which is likely to change for differing conditions such as different mass velocities. However, because R449A has a 2% larger thermal conductivity

and a 6% smaller temperature glide as compared to R448A, with other salient fluid properties being nearly the same, it is not surprising that the present study has shown R449A to have heat transfer coefficients that are marginally larger than those of R448A for the example case.

Figure 11 also shows the predicted relative performance of R452B to the other new replacements and also to the existing refrigerants R404A and R410A. Because of its large liquid thermal conductivity (29, 27, and 54% larger than R448A, R449A, and R404A, respectively) and its negligible temperature glide, which results in nearly single component heat transfer performance (0.92 mixture degradation factor), the heat transfer performance of R452B is significantly larger than R404A, R448A, and R449A. Averaged for qualities between 10 and 70% as presented in Figure 11 for counterflow, the heat transfer coefficient is approximately 90, 70, and 13% larger than that of R448A, R449A, and R404A, respectively. A similar comparison results for parallel flow: the average heat transfer coefficient for R452B is approximately 73, 59, and 14% larger than that of R448A, R449A, and R404A, respectively. As a R410A replacement, R452B closely matches the performance of R410A being approximately within 10% of the R410A heat transfer coefficient for qualities between 10 and 70%. On average, the heat transfer coefficient for R452B is predicted to be approximately 5% less than that of R410A for qualities between 10 and 70% for both flow conditions.

According to Minor (2016), R407A and R407F are replacements currently used for R404A. Considering this, additional predictive calculations were made using Equation 5 for R407A and R407F for the same conditions as were used for the above-discussed predictions and are presented in Figure 12. The results showed that the heat transfer coefficients for R407A and R407F reside between that of R448A and R449A, with the heat transfer coefficient of R407F being no more than $100 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$ greater than that for R407A. The difference between the R449A and the R448A heat transfer coefficients remains within $300 \text{ W}\cdot\text{K}^{-1}\cdot\text{m}^{-2}$ for both counterflow and parallel flow for all qualities for the example shown in Figure 12. This difference, on average, is approximately 10% of the predicted heat transfer coefficient. By comparison, the heat transfer coefficient of R407A and R407F are roughly, on average, within 5% of that of either R449A or R448A.

Conclusions

Local convective boiling heat transfer measurements for three low-GWP refrigerants in a fluid heated micro-fin tube were presented. The measured convective boiling Nus for all of the test refrigerants were compared to an existing correlation from the literature that was modified to predict multi-component mixtures. Approximately, 98% of the measurements were predicted to within $\pm 20\%$ and centered about the mean prediction.

In general, the measured boiling heat-transfer coefficient increased with increasing qualities for counterflow (low heat flux at low quality) between the refrigerant and the water. In contrast, for parallel flow (high heat flux at low quality), the measured heat transfer coefficient was relatively constant with respect to quality. The heat transfer coefficient of the three test fluids were compared at the same heat flux, saturated refrigerant temperature, and

refrigerant mass flux by using the modified correlation from the literature. The resulting comparison showed that refrigerant R452B exhibited the highest heat transfer performance of the three low-GWP refrigerants in large part due to its higher thermal conductivity and small temperature glide. For the example case presented here, the heat transfer coefficient for R449A was approximately 8% larger than the heat transfer coefficient for R448A for qualities between 10 and 70%. The heat transfer coefficient for R452B was predicted to be, on average, approximately 73 to 90% larger than R448A and approximately 59 to 70% larger than R449A. The heat transfer coefficients for R449A and R448A are 26 to 43% and 31 to 48%, respectively, less than that of R404A for qualities between 10 and 70%. The smaller heat transfer coefficient of R448A and R449A, as compared to that of R404A, results despite having an approximate 20% larger liquid thermal conductivity than R404A. In contrast, the model predicts that the R452B heat transfer coefficient is approximately 13% larger than that of R404A for qualities between 10 and 70%. On average, the heat transfer coefficient for R452B is predicted to be approximately 5% less than that of R410A for qualities between 10 and 70% for both flow conditions.

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Nomenclature

A_c	cross-sectional area (m^2)
Bo	local boiling number, $\frac{q}{Gr^{i_{fg}}}$
c_p	specific heat ($J\ kg^{-1}\ K^{-1}$)
C	coefficients given in Equation 5
D_e	equivalent inner diameter of smooth tube, $\sqrt{\frac{4 A_c}{\pi}}$ (m)
D_h	hydraulic diameter of micro-fin tube (m)
e	fin height (mm)
G	total mass velocity ($kg\ m^{-2}\ s^{-1}$)
h_{2f}	local two-phase heat-transfer coefficient ($W\ m^{-2}\ K^{-1}$)
i_{fg}	latent heat of vaporization ($J\ kg^{-1}$)
k	refrigerant thermal conductivity ($W\ m^{-1}\ K^{-1}$)
Nu	local Nusselt number based on D_h

$\dot{m}$	mass flow rate (kg s^{-1})
M_w	molar mass (g mole^{-1})
p	wetted perimeter (m)
P	local fluid pressure (Pa)
Pr	liquid refrigerant Prandtl number $\frac{c_p \mu}{k} \Big _{r,l}$
q''	local heat flux based on A_i (W m^{-2})
Re	all liquid, refrigerant Reynolds number based on $D_h = \frac{G_T D_h}{\mu_{r,l}}$
s	distance between fins (mm)
T	temperature (K)
t_b	bottom thickness of fin (mm)
t_w	tube wall thickness (mm)
U	expanded relative uncertainty (%)
x_q	thermodynamic mass quality (–)
z	axial distance (m)
α	helix angle ($^\circ$)
β	fin angle ($^\circ$)
T_s	$T_s - T_w$ (K)
μ	viscosity (Pa·s)
ν	specific volume, $x_q \nu_v + (1-x_q) \nu_l$ ($\text{m}^3 \text{kg}^{-1}$)
ρ	density (kg m^{-3})
σ	surface tension (kg s^{-2})
b	bubble point
c	critical condition
d	dew point
f	water
i	inside, inlet
l	liquid
p	prediction

r	refrigerant
s	saturated state
v	vapor
w	heat transfer surface

References

- Diani A, Mancin S, Rossetto L. Flow boiling heat transfer of R1234yf inside a 3.4 mm ID microfin tube. *Experimental Thermal Fluid Science*. 2015; 66:127–36.
- Directive 2006/40/EC of The European Parliament and of the Council. Relating to emissions from air-conditioning systems in motor vehicles and amending council directive 70/156/EC. *Official Journal of the European Union*. 2006; 49(L 161):12–8.
- Grauso S, Mastrullo R, Mauro AW, Thome JR, Vanoli GP. Flow pattern map, heat transfer and pressure drops during evaporation of R1234ze(e) and R134a in a horizontal, circular smooth tube: experiments and assessment of predictive methods. *International Journal of Refrigeration*. 2013; 36(2):478–91.
- Hamilton LJ, Kedzierski MA, Kaul MP. Horizontal convective boiling of pure and mixed refrigerants within a micro-fin tube. *Journal of Enhanced Heat Transfer*. 2008; 15(3):211–26.
- Hossain, MdA, Onaka, Y., Afroz, HMM., Miyara, A. Heat transfer during evaporation of R1234ze(E), R32, R410A, and a mixture of R1234ze(E) and R32 inside a horizontal smooth tube. *International Journal of Refrigeration*. 2013; 36(2):465–77.
- Stocker, TF, Qin, D, Plattner, G-K, Tignor, M, Allen, SK, Boschung, J, Nauels, A, Xia, Y, Bex, V., Midgley, PM., editors. IPCC. *Climate Change 2013: The Physical Science Basis*. Contribution of Working Group I to the Fifth Assessment Report of the Intergovernmental Panel on Climate Change. United Kingdom/New York, NY: Cambridge University Press, Cambridge; 2013. p. 1535
- Kattan N, Favret D, Thome JR. R-502 and two near-azeotropic alternatives: part 1—in tube flow-boiling tests. *ASHRAE Transactions*. 1995; 101(1):491–508.
- Kedzierski, MA., Kang, DY. Horizontal convective boiling of R448A, R449A, and R452B within a micro-fin tube with extensive measurement and analysis details. NIST Technical Note 1915. U.S. Department of Commerce; Washington, D.C: 2016.
- Kedzierski, MA., Kim, JH., Didion, DA. Causes of the apparent heat transfer degradation for refrigerant mixtures. In: Kim, JH, Nelson, RA., Hashemi, A., editors. *Two-Phase Flow and Heat Transfer, HTD*. Vol. 197. New York, NY: ASME; 1992. p. 149–58.
- Kim Y, Seo K, Chung J. Evaporation heat transfer characteristics of R-410A in 7 and 9.52 mm smooth/micro-fin tubes. *International Journal of Refrigeration*. 2002; 25(6):716–30.
- Kondou C, BaBa D, Mishima F, Koyama S. Flow boiling of non-azeotropic mixture R32/R1234ze(E) in horizontal microfin tubes. *International Journal of Refrigeration*. 2013; 36(8):2366–78.
- Kyoto Protocol. United Nations Framework Convention on Climate Change. New York, NY: United Nations (UN); 1997.
- Lemmon, EW., Huber, ML., McLinden, MO. NIST Standard Reference Database 23, Version 9.1. Private Communications with McLinden, National Institute of Standards and Technology; Boulder, CO: 2013.
- Manwell SP, Bergles AE. Gas-liquid flow patterns in refrigerant-oil mixtures. *ASHRAE Transactions*. 1990; 96(2):456–64.
- Mendoza-Miranda JM, Ramirez-Minguela JJ, Munoz-Carpio VD, Navarro-Esbri J. Development and validation of a micro-fin tubes evaporator model using R134a and R1234yf as working fluids. *International Journal of Refrigeration*. 2015; 50:32–43.
- Meng M, Yang Z, Duan YY, Chen Y. Boiling flow of R141b in vertical and inclined serpentine tubes. *International Journal of Heat and Mass Transfer*. 2013; 57:312–20.
- Minor, B. Private communications. Chemours; Wilmington, DE: 2016 Jan 13.

- Minor, B., Gerstel, J., Roberts, N. Evaluation of R-449A in field retrofits of R-404A supermarket systems. Proceedings of the 24th International Congress of Refrigeration; Yokohama, Japan. August 16–22; 2015.
- Montreal Protocol. Montreal Protocol on Substances that Deplete the Ozone Layer. United Nations (UN); New York, NY: 1987. (1987 with subsequent amendments)
- Nasr M, Akhavan-Behabadi MA, Momenifar MR, Hanafizadeh P. Heat transfer characteristic of R-600a during flow boiling inside horizontal plain tube. International Communications in Heat and Mass Transfer. 2015; 66:93–9.
- Olivier JA, Liebenberg L, Kedzierski MA, Meyer JP. Pressure drop during refrigerant condensation inside horizontal smooth, helical micro-fin, and herringbone micro-fin tubes. Journal of Heat Transfer. 2004; 126(5):687–96.
- Seo K, Kim Y. Evaporation heat transfer and pressure drop of R-22 in 7 and 9.52 mm smooth/micro-fin tubes. International Journal of Heat and Mass Transfer. 2000; 43(16):2869–82.
- Targanski W, Cieslinski JT. Evaporation of R407C/oil mixtures inside corrugated and micro-fin tubes. Applied Thermal Engineering. 2007; 27(13):2226–32.
- Webb, RL., Kim, N-H. Principles of Enhanced Heat Transfer. 2. New York, NY: Taylor & Francis; 2005.
- Wellsandt S, Vamling L. Evaporation of R134a in a horizontal herringbone microfin tube: heat transfer and pressure drop. International Journal of Refrigeration. 2005; 28(6):889–900.
- Yu M, Lin T, Tseng C. Heat transfer and flow pattern during two-phase flow boiling of R-134a in horizontal smooth and micro-fin tubes. International Journal of Refrigeration. 2002; 25(6):789–98.
- Yun R, Kim Y, Seo K, Kim H. A generalized correlation for evaporation heat transfer of refrigerants in micro-fin tubes. International Journal of Heat and Mass Transfer. 2002; 45(10):2003–10.
- Zhang X, Zhang J, Ji H, Zhao D. Heat transfer enhancement and pressure drop performance for R417A flow boiling in internally grooved tubes. Energy. 2015; 86:446–54.
- Zhang, X., Zhang, X., Yuan, X. Evaporation heat transfer coefficients of R417a in horizontal smooth and microfin tubes. International Congress of Refrigeration; Beijing, China. August 21–26; 2007. ICR07-B1-228
- Zou X, Gong M, Chen G, Sun Z, Wu J. Experimental study on saturated flow boiling heat transfer of R290/R152a binary mixtures in a horizontal tube. Frontiers of Energy and Power Engineering in China. 2010; 4(4):527–34.

Fig. 1.
Schematic of test rig.

Fig. 2.
Test section cross-section.

Fig. 3.
Detailed schematic of test section.

Fig. 4.
 Relative uncertainty of inner-wall temperature with respect to quality.

Fig. 5.
Counterflow temperature profiles for a 449A test.

Fig. 6.
Heat flux distribution for R449A.

Fig. 7.
 Relative uncertainty of the Nusselt number with respect to the quality.

Fig. 8.
 Comparison between measured Nusselt numbers and those predicted by the modified Hamilton et al. (2008) correlation.

Fig. 9. Flow boiling heat transfer coefficient for micro-fin tube versus quality for R452B, R449A, and R448A (counterflow).

Fig. 10.

Flow boiling heat transfer coefficient for micro-fin tube versus quality for R452B, R449A, and R448A (parallel flow).

Fig. 11.
 Flow boiling heat transfer coefficient versus quality for test refrigerants.

Fig. 12.
Flow boiling heat transfer coefficient versus quality for R404A replacements.

Table 1

Geometrical parameter of test section.

Parameter	Value
t_w	0.3 mm
e	0.2 mm
Number of fins	60
β	50°
Fin helix angle	18°
D_e	8.8 mm
D_t	8.91 mm
D_o	9.5 mm
D_h	5.45 mm
t_b	0.25 mm
s	0.22 mm
Inside-surface area per unit length	44.6 mm
Annulus gap	2.2 mm

Table 2Median estimated 95% relative expanded uncertainties for measurements (U).

Parameter	Minimum	Maximum	$U\%$
G_t [kg m ⁻² ·s ⁻¹]	91	304	2.0
T_s [K]	277.4	282.0	0.1 (0.3 K)
P [kPa]	692	917	1.5
T_w [K]	278.6	284.3	0.1 (0.25 K)
$\dot{m}_f$ [kg s ⁻¹]	0.006	0.018	2.0
T_f [K]	281.0	321.0	0.1
P_f [kPa]	200	110	1.0
q'' [kW m ⁻²]	1.2	24.2	15
$(T_d - T_b)/T_b$	0.004	0.02	3.0
Nu	41	376	20
Re	2759	11311	4.0
Bo	0.000043	0.00062	16.0
Pr	2.0	3.5	2.0
P_s/P_c	0.15	0.17	2.0
x_q	0.02	0.86	8.0
T_s [K]	1.00	5.28	0.44 K

Table 3

Representative properties from REFPROP (Lemmon et al. 2013).

Refrigerant	Evaluated at average test conditions for each fluid				Evaluated at $T_s = 277.6\text{ K}$		
	$T_d - T_b\text{ (K)}$	$k_1\text{ (W m}^{-1}\text{ K}^{-1}\text{)}$	Pr	$\sigma\text{ (mN m}^{-1}\text{)}$	$\rho\text{ (kg m}^{-3}\text{)}$	$\rho_v\text{ (kg m}^{-3}\text{)}$	$[P]_{s,kq=0}\text{ (kPa)}$
R404A	0.50	0.074	3.5	7.05	1132.8	35.8	700.6
R410A	0.11	0.100	2.3	8.38	1151.9	34.6	920.5
R448A	5.74	0.088	2.9	9.73	1180.7	28.8	717.1
R449A	5.39	0.090	2.9	9.53	1180.2	26.0	708.2
R452B	1.24	0.110	2.1	10.1	1072.5	28.1	889.7