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ABSTRACT
This report presents local convective boiling measurements in a micro-fin tube\(^1\) for four pure refrigerants: R22, R32, R125, and R134a; and four refrigerant mixtures: R410B (R32/125, 45/55 % mass), R32/R134a (27/73 % and 30/70 % mass) and, R407C (R32/125/134a, 25/23/52 % mass). Flow boiling heat transfer coefficients for the mixtures' pure components and R22 were measured to establish a baseline for the heat transfer degradation calculations. The measured convective boiling Nusselt numbers for all of the test refrigerants were correlated to a single expression consisting of a product of dimensionless properties. The correlation was shown to predict some existing data from the literature within 20 %. The degradation in heat transfer performance of the mixtures was found to range from 1 % to 50 % for all refrigerants tested.

Keywords: Enhanced heat transfer, micro-fin, refrigerant mixtures, fluid heating, boiling

\(^1\) Certain 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.
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INTRODUCTION
Most evaporators and condensers of new unitary refrigeration and air-conditioning equipment are manufactured with micro-fin tubes. The micro-fin tube dominates unitary equipment design because it provides the highest heat transfer with the lowest pressure drop of the commercially available internal enhancements (Webb, 1994). Together, R134a, R22, and R22 replacements constitute by mass nearly all the refrigerants used in unitary products (Muir, 1989). Consequently, two phase heat transfer data for the micro-fin tube with R134a, R22, and R22 replacements are essential for the design of evaporators and condensers for unitary applications.

Experimental flow boiling studies of pure and mixed refrigerant have been conducted in recent years and are well documented in the state of the art review by Thome (1996). Pure fluids in smooth tubes have been well studied and their heat transfer characteristics are reliably predicted. However, Thome (1996) calls for an improvement in the prediction methodologies for mixed refrigerant flow boiling heat transfer.

Jung et al. (2003) conducted a study of mixtures composed of R32, R125 and R134a in a smooth horizontal tube using a cartridge heater to generate uniform heat flux. For non-azeotropic mixtures, they found significant reductions in heat transfer coefficient compared to ideal values. Additionally, they developed satisfactory correlations to predict heat transfer performance. Later, Jung et al. (2004) studied three different microfin configurations but used pure refrigerants. This was done primarily to contrast the enhancement techniques rather than to develop correlations useful in predicting heat transfer performance. In both studies, global rather than local heat transfer coefficients were calculated.

Yun et al. (2002) examined existing experimental data and developed a model, which included a ratio of film thickness to fin height, to predict micro-fin heat transfer enhancement. This model assumed that smooth tube heat transfer performance could be accurately predicted and needs to be validated over a wide range of refrigerants and micro-fin geometries.

The present study provides measured local flow boiling heat transfer for both pure and mixed refrigerants in a micro-fin tube. The overall objective of the effort was to produce a reliable flow boiling correlation for pure refrigerants and mixtures in commercially available micro-fin tubes. A single heat transfer correlation based on the hydraulic diameter concept was developed to predict both mixed and pure refrigerants. A single correlation can be conveniently used to compare mixed refrigerant and pure refrigerant heat transfer performances for design evaluation.

EXPERIMENTAL APPARATUS
Figure 1 shows a sketch of the experimental apparatus used to establish and measure the convective boiling. 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 with turbine 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 counterflow to the refrigerant in the annulus that surrounded the micro-fin tube. 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 18 degree helix angle. For this geometry, the cross sectional flow area was 60.8 mm$^2$ giving an equivalent smooth diameter ($D_e$) of 8.8 mm. The root diameter of the micro-fin tube was 8.91 mm. The inside-surface area per unit length of the tube was estimated to be 44.6 mm. The hydraulic diameter ($D_h$) of the micro-fin tube was estimated to be 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.

Figure 3 provides a detailed description 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 pass between 12 of the gasketed flange pairs to measure the refrigerant-tube wall temperature at ten 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 to these, 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 was sanded to a thickness of 0.5 mm. The leads were strapped to a thin non-electrically-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 ten thermocouples in series, with the ten 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 enter 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 high water Reynolds number (Kattan et al. 1995).

As shown in Fig. 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.

HEAT TRANSFER COEFFICIENTS
Table 1 shows the expanded measurement uncertainty (U) of the various measurements along with the range of each 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 1 are median values of U for the correlated data. All saturated refrigerant properties were evaluated at the measured saturation pressure with the REFPROP (Lemmon et al. 2003) equation of state.

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

\[ h_{2φ} = \frac{q''}{T_w - T_r} \]  

(1)

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

\[ T_w = A_0 + A_1 z + A_2 z^2 \]  

(2)

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 as explained in the Results section. The uncertainty of most 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 1 was 0.25 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 was regressed to the axial location of the thermopiles along the z-coordinate (see Fig. 3). The water temperatures were fitted to:

\[
T_f = A_0 + A_1 z + A_2 z^2 + A_3 z^3
\]  

(1)

The water temperature fits, the measured water mass flow rate \( (m_f) \), 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:

\[
q'' = \frac{\dot{m}_f}{p} \left( c_{pf} \frac{dT_f}{dz} + v_f \frac{dP_f}{dz} \right)
\]  

(2)

where \( p \) is the wetted perimeter of the inside of the micro-fin tube. The specific heat \( (c_{pf}) \) and the specific volume \( (v_f) \) of the water were calculated locally as a function of the water temperature. The local, axial water temperature gradient \( (dT_f/dz) \) was calculated from a derivative of eq. (1). 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. For thermodynamic qualities greater than 10 %, the uncertainty of the heat flux was approximately ± 5 % of the measured value.

Figure 5 shows an example plot of the local heat flux as calculated from eq. (2) versus thermodynamic quality. This heat flux profile is for R22 at a Reynolds number of 7800 and a refrigerant pressure of 600 kPa.

Figure 6 plots the relative uncertainty in the water temperature gradient (which is roughly equal to the uncertainty in the heat flux) versus thermodynamic quality. As shown in Fig. 6, the larger values of heat flux exhibit smaller relative uncertainties than the lower heat fluxes.

The equilibrium refrigerant temperature \( (T_r) \) and all other thermodynamic and transport properties were calculated with version 7.1 of REFPROP (Lemmon et al., 2003) while using enthalpy and pressure as inputs. The enthalpy of the refrigerant vapor at the inlet of the test section was calculated from its measured temperature and pressure. The subsequent drop 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 average \( T_r \) was varied between 1 °C and 3 °C with approximately 5 K of subcooling at the test section inlet.

Figure 7 illustrates the results of the above-described measurement procedure. The appearances of
the profiles for water and wall temperature illustrate the need for separate regression forms.

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

\[
\text{Nu} = \frac{h_{2p}D_h}{k_i}
\]  

(3)

The hydraulic diameter was measured with a polar planimeter from a scaled drawing of the tube cross section but it can be approximated for other tube geometries with fin parameters by expanding on the expression that was given for \( D_h \) in Kedzierski and Goncalves (1999):

\[
D_h = \frac{4A_c \cos \alpha}{N_f S} = \frac{\left( \pi D_r^2 - 2N_f t_b e \right) \cos \alpha}{s + \frac{2e}{\cos(\beta / 2)}}
\]

(4)

Figure 2 shows the fin parameters that are used in eq. (6) where \( S \) is the perimeter of one fin and channel taken perpendicular to the axis of the fin, \( s \) is the spacing between the fins, \( \beta \) is the fin-tip angle, \( e \) is the fin height, \( \alpha \) is the twist angle of the fins, \( t_b \) is the thickness of the fin at its base, \( N_f \) is the total number of fins, and \( D_r \) is the diameter of the tube at the fin root, i.e., fin base. The hydraulic diameter of the present tube geometry of this study as estimated from eq. (6) is 5.2 mm, while that obtained from the planimeter and used in the data reduction was 5.45 mm.

The internal surface area of the fin per unit length \( (A_i/L) \) can be estimated from:

\[
\frac{A_i}{L} = N_f \left( \frac{s}{\cos \alpha} + \frac{2e}{\cos \alpha \cos(\beta / 2)} \right)
\]

(5)

The \( A_i/L \) estimated from eq. (7) is 46.8 mm, while that obtained from the planimeter and used in the data reduction was 44.6 mm.

Figure 8 shows the relative uncertainty of the Nu versus thermodynamic quality. The average uncertainty of Nu for qualities greater than 10 % was approximately ± 15 %. For qualities less than 10 %, the average uncertainty of Nu was approximately ± 30 %. Measurements of Nu with uncertainties greater than 50 % were omitted from the regression.

RESULTS

The 6360 data points generated in this study for R22, R134a, R32, R125, R410B, R407C and R32/R134a are tabulated in two separate appendices. Appendix A contains the Nusselt and Reynolds numbers and other reduced data that were used in the correlation of the data. Appendix B
contains the raw data measurements including the heat flux and the wall and water temperatures and locations. The column entitled, "flow," provides a "C" to show that all the data was for counterflow. All the parameters are defined in the nomenclature.

Heat Transfer
The convective boiling Nusselt numbers (Nu) were correlated following the law of Corresponding States philosophy presented by Cooper (1984). Cooper (1984) suggested that the fluid properties that govern nucleate pool boiling can be well represented by a product of the reduced pressure \( \left( \frac{P_r}{P_c} \right) \), the acentric factor \((-\log_{10}\left( \frac{P_r}{P_c} \right))\), and other dimensionless variables to various powers. The above reduced pressure terms and several other locally evaluated terms were used to correlate the local Nu measurements for all boiling flow conditions and refrigerants in this study to:

\[
Nu = 482.18 \text{Re}^{0.3} \text{Pr}^{C_1} \left( \frac{P_r}{P_c} \right)^{C_2} \text{Bo}^{C_1} (-\log_{10}\left( \frac{P_r}{P_c} \right))^{C_4} M_w^{C_3} 1.1^{C_6}
\]

where

\[
C_1 = 0.51x_q
\]
\[
C_2 = 5.57x_q - 5.21x_q^2
\]
\[
C_3 = 0.54 - 1.56x_q + 1.42x_q^2
\]
\[
C_4 = -0.81 + 12.56x_q - 11.00x_q^2
\]
\[
C_5 = 0.25 - 0.035x_q
\]
\[
C_6 = \frac{(T_{LV} - T_{MV})\left(279.8(x_q - x_t) - 4298(T_d - T_b)/T_s\right)}{T_s}
\]

The limits of applicability for the correlation are:

\[
70 \leq G_r \leq 370 \text{ kg/m}^2\text{s}
\]

and

\[
0 \leq x_q \leq 0.7
\]

In this correlation, the all-liquid Reynolds number (Re), the Boiling number (Bo), the liquid Prandtl number (Pr), the reduced pressure \( \left( \frac{P_r}{P_c} \right) \), and the quality \( (x_q) \) are all evaluated locally at the saturation temperature. The all-liquid Reynolds number and the Nusselt number are based on hydraulic diameter. The Nusselt number is also based on the actual inner surface area of the tube. The \( T_d \) and the \( T_b \) are the dew point temperature and the bubble point temperature of the mixture, respectively, evaluated at the local saturated pressure and overall composition. The \( T_{LV} \) and the
$T_{MV}$ are the temperatures of the least volatile component and the most volatile component evaluated at the saturated pressure of the mixture, respectively. The mass fraction of the vapor ($x_v$) and that of the liquid ($x_l$) are evaluated at the saturation pressure and the local thermodynamic quality, while the overall composition is the all liquid or all vapor value. The constant $C_6$ is zero for pure refrigerants.

The search for the above form of the correlation began with quadratic exponents in quality for each dimensionless variable. The quadratic exponent form was used with good results by Kedzierski and Kim (1997) to correlate several other pure refrigerants and mixtures for a wide range of qualities for both evaporative and condensing flows. The number of dimensionless variables and constants in the exponents were reduced to only those with significant influence on the residual standard deviation of the fit. For example, because R410B is a near-azeotrope, the composition difference between vapor and liquid phases had a negligible influence on the fit of the correlation. Consequently, the composition difference was not used in the fit of the R410B data.

Not all of the 6360 data points were used in the regression of eq. (6). Table 2 shows the number of data points for each refrigerant that were not used in the fit. Measurements with large uncertainties, and measurements that had high influence, or high leverage on the model were all candidates for exclusion from the regression (Belsley et al., 1980). Most of the data that satisfied the last two criteria were data at or near the inlet and exit of the test section. There were nearly as many outliers at the inlet as there were for the outlet of the test section. Outliers at the inlet and outlet of the test section are consistent with the largest uncertainties in the water and wall temperature fits being located at the inlet and outlet. Also, the $\Delta T_s$ can be less than the uncertainty of the measurement at the refrigerant inlet of the test section.

Figures 9 and 10 provide a comparison between the boiling Nusselt numbers predicted with eq. (6) for the micro-fin tube to those measured for pure and mixed refrigerants, respectively. Equation (6) correlates 95% of the pure component and near-azeotropic convective boiling Nusselt numbers to within approximately ± 9%. The mean of the correlation has an average uncertainty of ± 3% over the entire range of Nusselt numbers. Only random trends were observed in the residual plots against each of the parameters of eq. (6). The residual standard deviation of eq. (6) and that for the separate fits for each fluid were nearly the same. This suggests that the scatter in the data is not caused by the different fluids.

Figure 11 shows the heat transfer coefficient versus quality for each of the seven test fluids at $T_i = 278$ K, $q'' = \left(48x_v + 3\right)$ kW/m$^2$ and $G_i = 250$ kg/(m$^2$·s). The heat flux relationship was based on a linear fit of experimental data for one test run, but represents most of the data. The solid lines are predictions for the present micro-fin tube geometry, which were obtained from the correlation of the data given as eq. (8). In general, the measured boiling heat-transfer coefficient rapidly increases for increasing qualities within the ranges tested. This is expected as the refrigerants absorb larger quantities of heat during vaporization. No fall off in heat transfer coefficient is seen indicating partial dryout was likely not achieved. The level of refrigerant performance appears to be loosely tied to the latent heat of vaporization for each fluid particularly at higher quality. The refrigerant
R32 exhibits the highest heat transfer performance of the eight test fluids due to its high thermal conductivity and latent heat of vaporization. As expected, the performance of the near-azeotropic mixture R410B is between that of its pure components R32 and R125. The predicted performance of R22 is actually lower than that of its proposed replacement R410B. The flow boiling performance of R125 is greater than that of R134a for lower qualities but drops off as quality exceeds 0.3 likely due to its lower latent heat of vaporization.

Over 140 figures would be required to depict the Nusselt number versus thermodynamic quality relationships for each test. Consequently, only representative plots of Nu versus $x_q$ are given in Figs. 12 through 18. The solid lines are predictions for the present micro-fin tube geometry, which were obtained from the correlation of the data given as eq. (6). The symbols are the measured data points. In general, the predictions slightly under-predict the measured counterflow Nusselt numbers.

Figure 19 compares the predictions of eq. (6) to local boiling heat transfer data from the literature. The studies used for comparison are listed in Table 3. Most of the heat transfer coefficients provided in the literature were based on the micro-fin tube equivalent diameter or the root-diameter. In this case, the provided area ratio or eq. (7) was used to adjusted the heat transfer coefficients from the literature so that they were based on the actual inner surface area of the tube and thus consistent with eq. (8). Additionally, Nusselt numbers were calculated using the hydraulic diameter of the micro-fin tube, which was estimated from eq. (6). As shown in Fig. 19, most of the measured Nusselt numbers lie with ± 20% of the predictions. In general, the correlation over predicted Nusselt number for the studies using cartridge heating and under predicted Nusselt number when fluid heating was used. No trend was seen for pure refrigerants versus mixtures.

The data from the literature are all within the mass velocity limits for which the correlation was developed. Also, the $e/D_i$ ratios for the tubes from the literature are approximately equivalent to the present tube (0.02). However, considering the wide range of tube diameters, helix angles, fluids, fin shapes, and estimates made for the literature data, the agreement between eq. (6) and Nusselt numbers from the literature is remarkably good.

Surprisingly, even the data of Chamra and Webb (1995) were predicted fairly well by the correlation despite the additional heat transfer enhancement created by their cross-grooved micro-fin configuration. This may be due to the fact that the correlation includes parameters based on hydraulic diameter, which may account for the three dimensional configuration features.

Figure 20 shows the heat transfer degradation ($\Delta Nu_d$) as a function of the thermodynamic quality for four different refrigerant mixtures at a mass flux of 250 kg/m$^2$s. The $\Delta Nu_d$ was calculated from the correlation using the following definition for a given composition:

$$\Delta Nu_d = 100(Nu_p - Nu_n) / Nu_p = 100\left(1 - 1.1^{\Gamma_o}\right)$$  \hspace{1cm} (7)
where $\text{Nu}_p$ is the Nusselt number obtained from the single component correlation using the mixture properties, while $\text{Nu}_m$ is the Nusselt number obtained from the mixture correlation. Equation 9 quantifies the degradation due to mass transfer resistance and concentration gradients. That is to say, no effects of mixture properties are predicted by eq. 9 given that the mixture properties are used in the single component correlation (Kedzierski et al., 1992). As a result, R410B exhibited essentially no degradation due to concentration gradients given that it is a near azeotrope. In contrast, the zoetrope, R407C, gave a heat transfer degradation of nearly 50 %, which was independent of quality. The remaining two zeotropic mixtures of R32 and R134a showed a slight increase in degradation (from 35 % to 40 %) with respect to quality.

CONCLUSIONS
Local convective boiling measurements for four pure refrigerants and four refrigerant mixtures in a fluid heated micro-fin tube were presented. The measured convective boiling Nusselt numbers for all of the test refrigerants were correlated to a single expression consisting of a product of dimensionless properties. The correlation was shown to predict measured boiling Nusselt numbers for various micro-fin configurations within 20 %.

In general, the measured boiling heat-transfer coefficient increased with increasing qualities. The refrigerant R32 exhibited the highest heat transfer performance due to its high thermal conductivity and latent heat of vaporization. As expected, the performance of the near-azeotropic mixture R410B was between that of its pure components R32 and R125.

Finally, the heat transfer degradation of the refrigerant mixtures was evaluated. The heat transfer degradation associated with R410B was shown to be relatively small while the degradation of R32/134a and R407C approached 40 % and 50 % respectively.

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NOMENCLATURE

English symbols

- $A_c$: cross sectional flow area inside tube (m$^2$)
- $A_o$: coefficients given in eq. (2)
- $A_i$: actual inner surface area of tube (m$^2$)
- $A_r$: ratio of actual inner surface area of micro-fin tube to that of a smooth tube of inner diameter $D_t$
- $Bo$: local boiling number, $\frac{q''}{G_i i_f}$
- $c_p$: specific heat (J/kg·K)
- $C$: coefficients given in eq. (8)
- $D$: tube diameter (m)
- $D_e$: 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 (m)
- $E_h$: heat-transfer enhancement ratio given in Appendix: $E_h = \frac{h_{2\theta} A_i}{h_i \pi D_h L}$
- $G$: total mass velocity (kg/m$^2$·s)
- $h_{2\theta}$: local two-phase heat-transfer coefficient (W/m$^2$·K)
- $i_f$: latent heat of vaporization (J/kg)
- $k$: refrigerant thermal conductivity (W/m·K)
- $L$: tube length (m)
- $Nu$: local Nusselt number based on $D_h$
- $N_f$: number of fins
- $m$: mass flow rate (kg/s)
- $M_w$: molecular weight (g/mole)
- $p$: wetted perimeter (m)
- $P$: local fluid pressure (Pa)
- $Pr$: liquid refrigerant Prandtl number $\frac{c_p \mu}{k}$
- $q''$: local heat flux based on $A_i$ (W/m$^2$)
- $Re$: all liquid, refrigerant Reynolds number based on $D_h$ $= \frac{G_i D_h}{\mu_{r,f}}$
- $s$: spacing between the fins (m)
- $S$: perimeter of one fin and channel (m)
- $S_v$: non-dimensional refrigerant specific volume given in Appendix: $V_v - \frac{V_f}{V}$
- $t_b$: thickness of the fin at its base (m)
- $t_w$: thickness of the tube wall (m)
\( T \)  temperature (K)
\( T_b \)  bubble point temperature of mixture, eq. (6) (K)
\( T_d \)  dew point temperature of mixture, eq. (6) (K)
\( U \)  expanded relative uncertainty
\( x_q \)  thermodynamic mass quality
\( z \)  axial distance (m)

**Greek symbols**
\( \alpha \)  helix angle between micro fin and tube axis
\( \beta \)  fin-tip angle (rad)
\( \Delta h_{2\beta} \)  heat transfer degradation (W/m\(^2\)·K)
\( \Delta T_s \)  \( T_s - T_w \) (K)
\( \Delta Nu_d \)  defined in eq. (7)
\( \nu \)  specific volume, \( x_q \nu + (1-x_q)\mu \) (m\(^3\)/kg)

**Subscripts**
\( b \)  bulk condition, fin base
\( c \)  critical condition
\( f \)  water
\( i \)  inlet, inner
\( l \)  liquid
\( LV \)  least volatile component
\( m \)  measured
\( MV \)  more volatile component
\( p \)  plain or smooth tube, predicted
\( r \)  refrigerant
\( s \)  saturated state
\( w \)  heat transfer surface
\( v \)  vapor
REFERENCES


Table 1  Median estimated 95 % relative expanded uncertainties for measurements

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<th>Maximum</th>
<th>U %</th>
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<td>552</td>
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<td>$T_r$ [K]</td>
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<td>323.0</td>
<td>0.1 (0.3 K)</td>
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<td>$T_w$ [K]</td>
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<td>318.0</td>
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<td>0.0450</td>
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<td>$T_i$ [K]</td>
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<td>$P_i$ [kPa]</td>
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<td>$q''$ [kW/m$^2$]</td>
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<td>16.4</td>
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<tr>
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<td>16500</td>
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<td>Bo</td>
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<td>.00055</td>
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<tr>
<td>Pr</td>
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<tr>
<td>$\Delta T_s$ [K]</td>
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Table 2  Data distribution

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<th>R410B</th>
<th>R134a</th>
<th>R125</th>
<th>R32</th>
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<td>70</td>
<td>64</td>
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<td>124</td>
<td>530</td>
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<tr>
<td># points</td>
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<td>696</td>
<td>840</td>
<td>768</td>
<td>1164</td>
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<td>6360</td>
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<td>Fluid</td>
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<td>$N_f$</td>
<td>helix angle, $\alpha$ (deg)</td>
<td>Heat source</td>
<td>$A_r$</td>
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<td></td>
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<tr>
<td>Nidegger et al (1997)</td>
<td>R134a</td>
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<td>water</td>
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Figure 1  Schematic of test rig
Figure 2 Test section cross section
Figure 3  Detailed schematic of test section
Figure 4  Relative uncertainty of inner wall temperature
Figure 5  Heat Flux distribution for R22
Figure 6  Relative uncertainty of water temperature gradient with respect to quality
Figure 7  Counterflow temperature profiles for a R22 test
Figure 8  Relative uncertainty of the Nusselt number with respect to the quality
Figure 9  Comparison of measured and predicted convective boiling Nusselt numbers for pure refrigerants
Figure 10  Comparison of measured and predicted convective boiling Nusselt numbers for mixed refrigerants
Figure 11 Comparison of the heat transfer coefficient for different fluids versus thermodynamic quality
Figure 12  Flow boiling Nusselt numbers versus thermodynamic quality for R22

Micro-fin tube, R22 evaporation, counterflow

$\alpha = 18^\circ$, 60 fins, $D_h = 5.45$ mm, $Re = 8400$, $P_i = 0.12$

0 measured
--- predicted

Figure 12  Flow boiling Nusselt numbers versus thermodynamic quality for R22
Figure 13  Flow boiling Nusselt numbers versus thermodynamic quality for R32
Figure 14  Flow boiling Nusselt numbers versus thermodynamic quality for R125
Figure 15  Flow boiling Nusselt numbers versus thermodynamic quality for R134a

Micro-fin tube, R134a evaporation, counterflow
\[ \alpha = 18^\circ, 60 \text{ fins}, D_n = 5.45 \text{ mm}, Re = 7800, P_f = 0.08 \]
Figure 16  Flow boiling Nusselt numbers versus thermodynamic quality for R410B
Figure 17  Flow boiling Nusselt numbers versus thermodynamic quality for R32/134a
Figure 18  Flow boiling Nusselt numbers versus thermodynamic quality for R407C
Figure 19  Comparison of eq. (6) to measured flow boiling Nu from the literature (see Table 3)
Figure 20  Heat transfer degradation for mixture data
### APPENDIX A1

Convective Boiling of R22 within a micro-fin tube

(file: taepure.tbl)
Convective Boiling of R32 within a micro-fin tube

(file: taepure.tbl)
| Product Code | Read Date | Read Time | Read Value | Read Count | Mean Value | Mean Count | Read Value | Read Count | Mean Value | Mean Count | Read Value | Read Count | Mean Value | Mean Count | Read Value | Read Count | Mean Value | Mean Count |
|--------------|-----------|-----------|-------------|------------|-------------|------------|-------------|------------|-------------|------------|-------------|------------|-------------|------------|-------------|-------------|------------|-------------|------------|-------------|
| 20813983055  | 3.71E-05  | 0.165      | 0.192       | 12.69      | 1.79        | 2.18       | 26.21       | 350.9       | 13858.657  | 3.72E-04  | 0.181        | 0.790        | 2.02        | 1.88       | 1.80        | 1.52        | 14.83       |
| 20913977007  | 3.06E-05  | 0.165      | 0.209       | 1.94       | 1.79        | 2.09       | 26.43       | 81.3        | 13991.007  | 4.69E-05  | 0.155        | 0.792        | 2.02        | 2.03       | 1.80        | 1.52        | 14.83       |
| 21013972014  | 3.13E-05  | 0.165      | 0.194       | 0.79       | 1.79        | 1.67       | 26.29       | 199.9       | 13987.023  | 5.66E-05  | 0.165        | 0.792        | 1.80        | 2.00       | 1.76        | 26.03       |
| 21113965017  | 3.04E-05  | 0.164      | 0.219       | 2.00       | 1.79        | 2.00       | 24.51       | 175.8       | 13985.017  | 6.66E-05  | 0.165        | 0.792        | 2.02        | 1.76       | 1.80        | 1.32        | 26.28       |
| 21213971021  | 3.09E-05  | 0.166      | 0.224       | 2.08       | 1.79        | 2.01       | 24.65       | 205.4       | 13976.053  | 7.65E-05  | 0.165        | 0.794        | 1.80        | 2.02       | 1.76        | 26.17       |
| 21313996026  | 3.07E-05  | 0.166      | 0.232       | 2.00       | 1.79        | 1.99       | 20.84       | 234.4       | 13976.092  | 1.03E-04  | 0.165        | 0.792        | 2.02        | 1.76       | 1.80        | 1.60        | 20.77       |
| 21413998031  | 3.11E-05  | 0.167      | 0.214       | 2.00       | 1.79        | 1.73       | 27.56       | 331.6       | 13973.042  | 1.75E-05  | 0.165        | 0.794        | 1.80        | 2.02       | 1.76        | 26.28       |
| 21513948032  | 3.12E-05  | 0.164      | 0.226       | 1.80       | 1.79        | 1.82       | 3.14        | 212.7       | 13933.173  | 8.55E-05  | 0.164        | 0.791        | 2.01        | 2.02       | 1.96        | 43.22       |
| 21613948031  | 3.11E-05  | 0.166      | 0.223       | 0.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 21713948031  | 3.07E-05  | 0.166      | 0.221       | 1.80       | 1.79        | 1.97       | 27.56       | 324.4       | 13976.092  | 1.03E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 21813948032  | 3.12E-05  | 0.167      | 0.226       | 1.80       | 1.79        | 1.82       | 3.14        | 212.7       | 13933.173  | 8.55E-05  | 0.164        | 0.791        | 2.01        | 2.02       | 1.96        | 43.22       |
| 21913948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22013948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22113948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22213948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22313948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22413948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
| 22513948031  | 3.11E-05  | 0.164      | 0.223       | 1.80       | 1.79        | 1.97       | 27.56       | 328.3       | 13938.918  | 1.87E-04  | 0.165        | 0.794        | 1.80        | 2.02       | 1.96        | 43.22       |
# APPENDIX A4

Convective Bolling of R134a within a micro-finn tube

(file: taepure.tbl)
## APPENDIX B1

Convective Bolling of R32/134a within a micro-fin tube

(file: taemix.tbl)
## APPENDIX B2

Convoluting Boiling of R407C within a micro-fin tube

(file: taemix.tbl)
### APPENDIX B3

Convective Boiling of R410B within a micro-fin tube

(file: taemix.tbl)
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