Water-Heated Pool Boiling of Different Refrigerants on the Outside Surface of a Horizontal Smooth Tube

Pool boiling heat transfer has been extensively studied over decades, but the effect of boundary heating conditions on boiling received little attention. In this work, heat transfer coefficients during pool boiling of five different refrigerants (R123, R245fa, R236fa, R134a, and R22) on the outside surface of a smooth copper tube were measured at the saturation temperature of 6.7°C; water flows inside the tube and provides heat to the refrigerants to boil (thus, water-heated boiling). Measurements showed that the refrigerant of a higher vapor pressure has a higher heat transfer coefficient, with the exception that R22 performs nearly the same as R134a. A correlation previously developed for electrically-heated pool boiling on cylindrical tubes underpredicts by 30%–46% the heat transfer coefficients during water-heated boiling of the five refrigerants. Among the pool boiling correlations reviewed in this work, the Cooper correlation (for pool boiling on cylindrical tubes) predicts the boiling heat transfer coefficients of R22 and R245fa reasonably well (within ±8.5%), but not as well those of the other three refrigerants (underpredicts by nearly 30% for R134a and R236fa and overpredicts by nearly 40% for R123). It is found that the predicted boiling heat transfer coefficients of the five refrigerants by the modified Gorenflo correlation (simply adding a constant multiplier of 1.47 to the Gorenflo correlation) are in excellent agreement with their respective measurements. [DOI: 10.1115/1.4004902]

Keywords: pool boiling, cylindrical tubes, refrigerants, boiling correlations, chiller efficiency, energy conservation
Table 1 A list of reviewed pool boiling heat transfer correlations; “General” denotes the correlations are not specified for a particular fluid or geometry

<table>
<thead>
<tr>
<th>Correlations</th>
<th>Fluids</th>
<th>Geometries</th>
</tr>
</thead>
<tbody>
<tr>
<td>Based on reduced pressure</td>
<td>Cooper correlation [3]</td>
<td>General Flat surfaces and cylindrical surfaces</td>
</tr>
<tr>
<td></td>
<td>Gorenflo correlation [6]</td>
<td>Halogenated refrigerants (Table 2) Cylindrical surfaces</td>
</tr>
<tr>
<td>Based on thermo physical properties</td>
<td>Rohsenow correlation [7]</td>
<td>Water, carbon tetrachloride, Isopropyl alcohol, n-butyl alcohol General</td>
</tr>
<tr>
<td></td>
<td>Stephan and Abdelsalam correlation [8]</td>
<td>Refrigerants; for other fluid, this correlation has different forms General</td>
</tr>
<tr>
<td></td>
<td>Forster and Zuber correlation [9]</td>
<td>General General</td>
</tr>
<tr>
<td></td>
<td>Jung et al. [12]</td>
<td>Flammable refrigerants Cylindrical electrically-heated surfaces</td>
</tr>
</tbody>
</table>

Based on the data of Borishansky et al. [14]

\[
q^{\prime \prime} = \frac{1}{\mu_0 b_{sv}} \left[ \frac{b}{(\rho_0 - \rho_v)} \right]^{0.825} \left( \frac{D}{\mu_0 b_{sv}} \right)^{0.55} \left( \frac{D}{\mu_0 b_{sv}} \right)^{1.22} \left( \frac{D}{\mu_0 b_{sv}} \right) \left( \frac{D}{\mu_0 b_{sv}} \right)^{0.72} \left( \frac{D}{\mu_0 b_{sv}} \right)^{0.55}
\]
horizontal smooth tube heated by water flowing inside the tube, i.e., it is water-heated boiling in comparison to electrically-heated boiling. Most empirical boiling correlations were developed on the basis of electrically-heated boiling data; however, fluid-heated boiling is closer to the operating conditions in actual boiling applications where the heat is usually provided from another heat transfer medium, such as water in chillers, to the boiling refrigerant. Providing a boiling correlation applicable to the actual boiling operating conditions for accurate prediction of boiling heat transfer is of paramount significance to industry engineers.

Test Facility

Figure 1 is a schematic of the test rig that includes a chilled water main, a water steam main, a closed water loop, an open chilled water loop, a boiling chamber, and a condensation chamber. The closed water loop maintains water through the test section in the boiling chamber at desired conditions (flow rate and temperature). The inlet temperature of water at a given flow rate is controlled by adjusting the amount of heat transferred to water in the water/steam heat exchanger included in the loop; each inlet temperature corresponds to a heat transfer rate from the water to the refrigerant. The chilled water loop provides chilled water to a tube bundle in the condensation chamber. The condensation chamber and the boiling chamber are open to each other both at the top and at the bottom. By adjusting the flow rate of the chilled water through the tube bundle in the condensation chamber, a desired pressure in the boiling chamber and thus the saturation temperature of the refrigerant is obtained.

In this work, experiments were performed at the pressures corresponding to the saturation temperature of 6.7 °C (44 °F) for all the five refrigerants. Table 3 lists the refrigerant properties at the saturation temperature of 6.7 °C in the order of increasing vapor pressure: R-123, R-245fa, R-236fa, R-134a, and R-22. For each refrigerant, experiments were performed at three flow rates of water inside the tube in the range of 1.70 to 2.50 m/s (typical in chiller applications) that corresponds to Reynolds number in the range of 40,000 to 65,000 depending on the water inlet temperature. For each refrigerant at a given flow rate, the test section average heat flux depends on the water inlet temperature; a fixed maximum water inlet temperature about 26.7 °C (80 °F) at each flow rate results in the heat flux ranging from 10,000 to 100,000 W/m² covered in this work.

Test Section

The test section consists of two 10-ft-long smooth copper tubes installed horizontally side-by-side in the boiling chamber. In addition to the 8-ft-long boiling segment, each test tube also includes two 1-ft-long segments for sealing, plumbing, and instrumentation at the inlet and the outlet of each tube. Figure 2 is a schematic of the test section; the two tubes were positioned on a horizontal plane (with a center-to-center separation distance of 3.5 in. and thus a tube-to-tube clearance of 2.5 in.) and boiling on one tube is not influenced by the other. A thermally insulated flexible hose connects the two tubes at one end of the boiling chamber while

<table>
<thead>
<tr>
<th>Fluids</th>
<th>$P_c$</th>
<th>$M$</th>
<th>$h_0$ (W/m² K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>R11</td>
<td>44.0</td>
<td>137.4</td>
<td>2800</td>
</tr>
<tr>
<td>R12</td>
<td>41.6</td>
<td>120.9</td>
<td>4000</td>
</tr>
<tr>
<td>R13</td>
<td>38.6</td>
<td>104.5</td>
<td>3900</td>
</tr>
<tr>
<td>R13B1</td>
<td>39.8</td>
<td>148.9</td>
<td>3500</td>
</tr>
<tr>
<td>R22</td>
<td>49.9</td>
<td>86.5</td>
<td>3900</td>
</tr>
<tr>
<td>R23</td>
<td>48.7</td>
<td>70.0</td>
<td>4400</td>
</tr>
<tr>
<td>R113</td>
<td>34.1</td>
<td>187.4</td>
<td>2650</td>
</tr>
<tr>
<td>R114</td>
<td>32.6</td>
<td>170.9</td>
<td>2800</td>
</tr>
<tr>
<td>R115</td>
<td>31.3</td>
<td>154.5</td>
<td>4200</td>
</tr>
<tr>
<td>R123</td>
<td>36.7</td>
<td>152.9</td>
<td>2600</td>
</tr>
<tr>
<td>R134a</td>
<td>40.2</td>
<td>102.0</td>
<td>4500</td>
</tr>
<tr>
<td>R152a</td>
<td>45.2</td>
<td>66.1</td>
<td>4000</td>
</tr>
<tr>
<td>R226</td>
<td>30.6</td>
<td>186.5</td>
<td>3700</td>
</tr>
<tr>
<td>R227</td>
<td>29.3</td>
<td>170.0</td>
<td>3900</td>
</tr>
<tr>
<td>RC318</td>
<td>28.0</td>
<td>200.0</td>
<td>4200</td>
</tr>
<tr>
<td>R502</td>
<td>40.8</td>
<td>111.6</td>
<td>3300</td>
</tr>
<tr>
<td>R245fa</td>
<td>36.4</td>
<td>134.0</td>
<td>3641</td>
</tr>
<tr>
<td>R236fa</td>
<td>32.0</td>
<td>152.0</td>
<td>3641</td>
</tr>
</tbody>
</table>

Table 2 Fluid-specific reference heat transfer coefficients $h_0$ for halogenated fluids in the Gorenflo correlation [6]; R245fa and R236fa are not included in the correlation.
water enters the first tube and exits the second tube on the other end, which forms a two-pass water flow. The outside diameter of the test tubes is 1 in. and the wall thickness is 0.044 in. The inside diameter of the boiling chamber is 6 in. and its length is 8 feet. The test tubes run through the boiling chamber and are sealed at the two end-flanges. The boiling chamber is nearly 2/3 filled with a pool of refrigerant and the two 8-ft boiling segments are immersed in the refrigerant pool.

A computer-based data acquisition system was used to record data including water temperatures, water flow rate, and refrigerant temperature and pressure in the boil chamber; data were recorded only when the thermal and the hydrodynamic steady states were reached. Calibration was performed on all temperature sensors, pressure transducers, and flow meters prior to experiments. The refrigerant temperature measured in the boiling chamber matches well (±0.5%) the measured pressure in the chamber.

Heat transfer calculations can be readily performed for each of the two tubes

\[ q'' = \frac{m_w C_p (T_{in} - T_{out})}{A_o} \]

(11)

\[ U_o = \frac{q''}{LMTD} \]

(12)

\[ LMTD = \frac{T_{in} - T_{out}}{\ln [(T_{in} - T_{sat})/(T_{sat} - T_{out})]} \]

(13)

\[ \frac{1}{h_o} = \frac{1}{U_o} - \frac{1}{A_o} R_{wall} = \frac{1}{A_o} \frac{A_i}{h_i} \]

(14)

\[ R_{wall} = \frac{1}{2\pi k_{wall} L} \]

(15)

in which \( q'' \) is the heat flux with respect to the outside surface area \( A_o \) of each 8-ft-long boiling segment, \( m_w \) is the water flow rate, \( T_{in} \) and \( T_{out} \) are the water inlet and outlet temperatures of each tube, and \( C_p \) is the specific heat of water at its average temperature over the boiling segment, \( T_{ave} = (T_{in} + T_{out})/2 \). LMTD is the log mean temperature difference between the water and the refrigerant. \( h_o \) is the overall heat transfer coefficient with respect to the outside surface area, \( h_n \) is the average outside heat transfer coefficient, \( h_i \) is the inside heat transfer coefficient, \( R_{wall} \) is the wall thermal resistance, and \( k_{wall} \) is the wall thermal conductivity. In Eq. (14), the fouling factor is neglected since the test tubes were new tubes from raw stock and each tube was well cleaned and dried before being put into the test chamber. In Eq. (15), \( D_o \), \( D_i \), and \( L \) are respectively the outside diameter, the inside diameter, and the length of each tube. For turbulent flow inside the tubes \[ \frac{h D_i}{k_{wall}} = 0.027 Re^{0.8} Pr^{1/3} \left( \frac{\mu}{\mu_{wall}} \right)^{0.14} \].

(16)

In Eq. (16), the viscosity, \( \mu \), and the Prandtl number, \( Pr \), are at the average water temperature, \( T_{ave} \), and the Reynolds number is given by \( Re = 4 m_w / (\pi D_i) \). \( \mu_{wall} \) is at the average inside wall temperature \( T_{wall,i} \) over the tube length obtained by

\[ T_{wall,i} = T_{sat} + q''/h_o \]

(17)

\[ T_{wall,o} = T_{wall,i} + \left( q''/t \right)/k_{wall} \].

(18)

\( T_{sat} \) in Eq. (17) is the saturation temperature obtained by averaging the measurements from four temperature sensors in the refrigerant pool, and \( T_{wall,o} \) is the average outside wall temperature. In Eq. (18), \( t \) is the wall thickness of the tube. The outside heat transfer coefficient \( h_o \) is obtained by an iteration of Eqs. (14) through (18). Following the method of Kline and McClintock [21], the uncertainties in boiling heat transfer coefficients are estimated to be in the range of 3.8%–6.1%; a larger uncertainty occurs at a lower heat flux.

**Results and Discussion**

Figure 3 gives the average boiling heat transfer coefficients of R134a on the first and the second tubes (Fig. 2) to compare the
boiling performances measured on the two individual tubes. For each tube, there are a total of 27 data points corresponding to three water flow rates (1.70, 2.16, and 2.50 m/s) and nine water inlet temperatures at each flow rate. It is apparent that, also as expected, the boiling heat transfer coefficients measured on the two tubes are essentially the same, but the first tube covers a range of larger heat fluxes than the second tube. This holds true for all the five refrigerants tested in this work. At a given condition (water flow rate and inlet temperature), the average water temperature in the first tube is higher than in the second tube as the water temperature decreases downstream, which results in a larger range of heat flux for the first tube. The same heat transfer coefficients measured on the two tubes indirectly confirmed that boiling on one tube was not influenced by the other.

Figure 4 presents the measured boiling heat transfer coefficients of the five refrigerants; the data points for each refrigerant include those from both the first and the second tubes. For instance, there are a total of 54 data points for R134a (27 data points for each tube). Throughout the tests, the maximum water inlet temperature at each flow rate for all the five refrigerants was fixed at nearly the same temperature (26.7°C); the maximum heat flux for a refrigerant depends on its boiling performance. The maximum heat flux for R123 is the lowest as it performs lowest among the five refrigerants. For each refrigerant, the boiling heat transfer coefficient increases with increasing heat flux within the heat flux range for each refrigerant. Figure 4 also shows that the refrigerant of a higher vapor pressure has a higher heat transfer coefficient, with the exception that R22 performs nearly the same as R134a although the vapor pressure of R22 is almost 60% higher than that of R134a. This seemingly “abnormal” phenomenon for R22 and R134a was also observed in a recent work by Park et al. [22] who performed nucleate boiling of different refrigerants (R32, R22, R134a, and R123, in the order of decreasing vapor pressure at a given temperature) on an electrically-heated horizontal flat smooth copper surface. Their results showed that, although the heat transfer coefficients of different refrigerants generally
increase with increasing vapor pressure (i.e., R32 performs highest and R123 performs lowest), the performances of R22 and R134a differ very little. Similar results have been seen in the work by Webb and Pais [23]. This “abnormal” behavior reiterates that boiling heat transfer is a complicated process involving compound effects of thermophysical properties of the boiling fluids. Table 3 lists the thermal and transport properties of the five refrigerants. An examination of these properties indicates that the heat transfer coefficients measured in this work inversely follow the trend of their surface tension values. In particular, R134a and R22 have the same surface tension values, as with nearly the same heat transfer coefficients of the two refrigerants. The importance of surface tension to boiling heat transfer is widely recognized; however, the degree of its importance relative to those of other properties (including reduced pressure) awaits further investigation.

The correlation proposed by Jung et al. [11] was based on electrically-heated pool boiling of pure halogenated refrigerants on the smooth tubes; comparison of its predicted results with the measurements in this work would provide insights into the effect of boundary heating conditions on boiling heat transfer. Toward this end, the boiling heat transfer coefficients of the five refrigerants are predicted at the saturation temperature (6.7 °C) by the correlation (Eq. (8) in Table 1) and the predictions are plotted in Fig. 5 with measurements. Figure 5 shows that the electrically-heated boiling correlation underpredicts the water-heated boiling performances by 30 to 45% depending on the refrigerant and the heat flux. As noted above, better heat transfer performance during the fluid-heated boiling relative to the electrically-heated boiling has also been observed by Kedzierski [18] and Darabi et al. [19]. Kedzierski postulated that the interactions between the fluctuating wall temperature and the fixed electrical heat flux induced a higher degree of superheated liquid on the electrically-heated surface than on the fluid-heated surface. For the same time-averaged heat flux, a larger fraction of it is used to superheat liquid for electrically-heated boiling than for fluid-heated boiling, which leads to a higher heat transfer coefficient for fluid-heated boiling. Darabi et al. [19] extended an explanation based on the fact that the temperature profiles along the tube (axially) for the water-heated and the electrically-heated cases are quite different. In spite of these attempted explanations, it is reasoned here that the difference in the performances is attributed, at least

![Graphs showing heat transfer coefficients for different refrigerants](image-url)
that there is a wide scatter of the disparities for the Rohsenow, Stephan, Abdelsalam [8], Gorenflo [2], and Mostinski [5] correlations underpredict the boiling heat transfer coefficients (Table 2). A comparison with the measurements indicates that all these four correlations underpredict the boiling performances of the five refrigerants by nearly 30% and overpredict those of R123 and R245fa by nearly 30% and 47%, respectively. In particular, the Gorenflo correlation captures the ‘abnormal’ behavior of the same heat transfer coefficients of R22 and R134a. It is noted in Table 4 that Stephan and Abdelsalam correlation predicts the same performances of R22 and R134a, but their disparities (~73%) and the disparities for the other refrigerants (~65%) differ considerably.

In spite of the disparities by the Gorenflo correlation in Table 4, its capability of excellently capturing the trend of water-heated boiling heat transfer coefficients of the five refrigerants sets it apart from the other correlations, which may be attributed to the fact that the Gorenflo correlation utilizes an experimentally-determined heat transfer coefficient $h_0$ (at reduced pressure of 0.1, heat flux of 20,000 W/m², and surface roughness of 0.4 μm) for each fluid as the reference value. With the effect of thermophysical properties of the boiling liquid being contained in the fluid-specific reference heat transfer coefficient $h_0$, the Gorenflo correlation predicts the heat transfer coefficients at other conditions by accounting for the effects of heat flux, reduced pressure, and surface roughness. Motivated by this observation, the predicted heat transfer coefficients of the five refrigerants are nearly 30% lower than their respective measurements. In this work, the refrigerants were tested at a fixed temperature of 6.7 °C (44 °F), the standard temperature of chiller operation. However, the variation in the cooling load during the actual chiller operation may partially, to the circumferential nonuniformity of heating during water-heated boiling in comparison to the uniform heat flux during electrically-heated boiling. For boiling on cylindrical surfaces, bubbles (except those on the very top portion) slide along the tube circumference to the top portion before they depart into the bulk liquid. As a consequence, the boiling characteristics (forms of bubbles and their activities) vary along the tube circumference due to accumulation and possible agglomeration of bubbles on the top portion. Compared with the uniform heat flux during the electrically-heated boiling, the heat flux over the tube circumference for the water-heated boiling is most likely nonuniform due to heat redistribution and strong turbulence within the fluid, and it is suspected that the circumferentially-uniform heat flux during the electrically-heated boiling results in boiling characteristics on the tube surface not as effective a heat transfer process.

It is desirable to examine the applicability of commonly-used nucleate pool boiling correlations for predicting the boiling heat transfer during water-heated boiling on smooth tubes. Compared with boiling on flat surfaces, the gravity effect during boiling on cylindrical tubes is quite different. Nishikawa et al. [24] showed that at heat fluxes below a threshold value, the boiling heat transfer increases as the inclination angle of the boiling surface increases from 0 deg (facing upward) to 180 deg (facing downward). Carey [25] attributed the increase in boiling heat transfer on an inclined surface to two effects: (1) the thicker natural convection boundary layer on an inclined surface favorable for bubble nucleation and (2) the promoted heat convection and film evaporation during sweeping of bubbles over an inclined surface since bubbles on an inclined surface must travel along the surface to its lateral edge before escaping to the ambient. The Cooper correlation [3] has been recommended as one of the most accurate correlations in predicting nucleate boiling heat transfer [26]; for boiling on cylindrical tubes, Cooper recommended adding a multiplier of 1.7 to the Cooper correlation (for boiling on upward-facing flat surfaces) to account for the gravity effect [3]. Figure 6 compares the measurements obtained in this work with those predicted by the Cooper correlation for the five refrigerants, which shows that the Cooper correlation predicts very well the performances of R22 and R245fa (within ±8.5%), but underpredicts those of R134a and R236fa by nearly 30% and overpredicts those of R123 by nearly 40%.

To further examine the applicability of commonly-used pool boiling correlations for predicting water-heated boiling heat transfer on cylindrical tubes, four correlations (Rohsenow correlation, Stephan and Abdelsalam correlation, Gorenflo correlation, and Mostinski correlation; details of each are given in Table 1) are selected and their predictions are presented in Fig. 7 for the five refrigerants corresponding to the test conditions in this work. In order to use the Gorenflo correlation to predict the boiling heat transfer coefficients of R245fa and R236fa that are not included in the correlation, the averaged value of the given reference heat transfer coefficients for all the halogenated fluids is assigned as the reference value ($h_0 = 3641$ W/m²K) for the two refrigerants (Table 2). A comparison with the measurements indicates that all these four correlations underpredict the boiling heat transfer coefficients during water-heated boiling on smooth tubes; the disparities are given in Table 4. A close examination of Table 4 reveals that there is a wide scatter of the disparities for the Rohsenow, Stephan and Abdelsalam, and Mostinski correlations. However, the disparities by the Gorenflo correlation are almost the same, i.e., the predictions for all the five refrigerants by the Gorenflo correlation are all nearly 30% lower than their respective measurements over the entire heat flux range covered by each refrigerant. In particular, the Gorenflo correlation captures the ‘abnormal’ behavior of the same heat transfer coefficients of R22 and R134a. It is noted in Table 4 that Stephan and Abdelsalam correlation predicts the same performances of R22 and R134a, but their disparities (~73%) and the disparities for the other refrigerants (~65%) differ considerably.

In spite of the disparities by the Gorenflo correlation in Table 4, its capability of excellently capturing the trend of water-heated boiling heat transfer coefficients of the five refrigerants sets it apart from the other correlations, which may be attributed to the fact that the Gorenflo correlation utilizes an experimentally-determined heat transfer coefficient $h_0$ (at reduced pressure of 0.1, heat flux of 20,000 W/m², and surface roughness of 0.4 μm) for each fluid as the reference value. With the effect of thermophysical properties of the boiling liquid being contained in the fluid-specific reference heat transfer coefficient $h_0$, the Gorenflo correlation predicts the heat transfer coefficients at other conditions by accounting for the effects of heat flux, reduced pressure, and surface roughness. Motivated by this observation, the predicted heat transfer coefficients of the five refrigerants are nearly 30% lower than their respective measurements. In this work, the refrigerants were tested at a fixed temperature of 6.7 °C (44 °F), the standard temperature of chiller operation. However, the variation in the cooling load during the actual chiller operation may

### Table 4 The disparity between the predictions given in Fig. 7 and the measurements obtained in this work; ‘-’ indicates that the correlations underpredict the boiling performances

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>R123</td>
<td>-20 to -50%</td>
<td>-51 to -66%</td>
<td>-33 to -37%</td>
<td>-13 to -47%</td>
</tr>
<tr>
<td>R245fa</td>
<td>-47 to -65%</td>
<td>-59 to -66%</td>
<td>-30%</td>
<td>-43% to -59%</td>
</tr>
<tr>
<td>R236fa</td>
<td>-29 to -61%</td>
<td>-51 to -66%</td>
<td>-30%</td>
<td>-43% to -59%</td>
</tr>
<tr>
<td>R134a</td>
<td>-47 to -57%</td>
<td>-73%</td>
<td>-28 to -30%</td>
<td>-47% to -69%</td>
</tr>
<tr>
<td>R22</td>
<td>-24 to -39%</td>
<td>-72%</td>
<td>-27 to -28%</td>
<td>-66 to -71%</td>
</tr>
</tbody>
</table>

![Fig. 8 The predicted boiling heat transfer coefficients by the modified Gorenflo correlation in good agreement (3.3%–9.5%) with those measured in this work](image)
lead to a maximum variation of 10 °C in the refrigerant saturation temperature. The follow-up of this work is to determine whether or not the modified Gorenflo correlation predicts the boiling heat transfer coefficients at various refrigerant saturation temperatures.

Summary

Heat transfer coefficients during boiling of five different refrigerants on an 8-ft-long smooth tube were measured at the saturation temperature of 6.7 °C (44 °F). Water flows inside the test section and provides heat to the refrigerants to boil. The refrigerant of a higher vapor pressure was measured to have a higher boiling heat transfer coefficient, with the exception that R22 has essentially the same performance as R134a although the vapor pressure of the former is almost 66% higher. A correlation previously developed for electrically-heated pool boiling on cylindrical tubes underpredicts by 30%–46% the boiling heat transfer coefficients of the five refrigerants heated by flowing water inside the tubes. Compared with the uniform heat flux along the tube circumference during electrically-heated boiling, nonuniform heat flux along the tube circumference during the water-heated boiling is suspected to be, at least partially, responsible for the more effective boiling heat transfer. The Cooper correlation (for pool boiling on cylindrical tubes) predicts the boiling heat transfer coefficients of R22 and R245fa reasonably well (within ±8.5%), but not as well for the other three refrigerants (underpredicts by nearly 30% for R134a and R236fa and overpredicts by nearly 40% for R123). It is found that the modified Gorenflo correlation (simply adding a constant multiplier of 1.47 to the Gorenflo correlation) predicts very well the boiling heat transfer coefficients during water-heated pool boiling of the five refrigerants on the outside surface of a smooth tube (within the range of 3.3–9.5%).

Nomenclature

\[ A_t = \text{tube inside area, m}^2 \]
\[ A_o = \text{tube outside area, m}^2 \]
\[ C_p = \text{specific heat, J/kg·K} \]
\[ D_t = \text{tube inside diameter, m} \]
\[ D_o = \text{tube outside diameter, m} \]
\[ g = \text{gravity acceleration, m/s}^2 \]
\[ h = \text{heat transfer coefficient, W/m}^2\text{K} \]
\[ \dot{h}_f = \text{latent heat, J/kg} \]
\[ h_o = \text{boiling heat transfer coefficient, W/m}^2\text{K} \]
\[ k_{wall} = \text{wall thermal conductivity, W/m·K} \]
\[ L = \text{tube length, m} \]
\[ M = \text{molecular weight, g/mol} \]
\[ \dot{m}_w = \text{water flow rate, kg/s} \]
\[ P = \text{pressure, kPa} \]
\[ P_c = \text{critical pressure, kPa} \]
\[ P_r = \text{reduced pressure, P/P_c} \]
\[ Pr = \text{Prandtl number} \]
\[ q'' = \text{heat flux, W/m}^2 \]
\[ Re = \text{Reynolds number} \]
\[ R_{wall} = \text{wall thermal resistance, K/W} \]
\[ t = \text{tube wall thickness, m} \]
\[ T_{ave} = \text{water average temperature, °C} \]
\[ T_{in} = \text{water temperature at the inlet, °C} \]
\[ T_{out} = \text{water temperature at the outlet, °C} \]
\[ T_{sat} = \text{refrigerant saturation temperature, °C} \]
\[ T_{wall,i} = \text{average tube inside wall temperature, °C} \]
\[ T_{wall,o} = \text{average tube outside wall temperature, °C} \]
\[ U_o = \text{overall heat transfer coefficient, W/m}^2\text{K} \]

Greek Letters

\[ \rho = \text{density, kg/m}^3 \]
\[ \Delta = \text{difference} \]

Subscripts

\[ l = \text{liquid} \]
\[ v = \text{vapor} \]
\[ w = \text{water} \]
\[ wall = \text{tube wall} \]

References

[27] NIST Thermochemical and Transport Properties of Refrigerants and Refrigerant Mixtures—Refprop Version 8.0.