Flow Boiling of R134a in Circular Microtubes—Part II: Study of Critical Heat Flux Condition

A detailed experimental study was carried out on the critical heat flux (CHF) condition for flow boiling of R134a in single circular microtubes. The test sections had inner diameters (ID) of 0.50 mm, 0.96 mm, and 1.60 mm. Experiments were conducted over a large range of mass flux, inlet subcooling, saturation pressure, and vapor quality. CHF occurred under saturated conditions at high qualities and increased with increasing mass fluxes, tube diameters, and inlet subcoolings. CHF generally, but not always, decreases with increasing saturation pressures and vapor qualities. The experimental data were mapped to the flow pattern maps developed by Hasan [2005, “Two-Phase Flow Regime Transitions in Microchannels: A Comparative Experimental Study,” Nanoscale Microscale Thermophys. Eng., 9, pp. 165–182] and Revellin and Thome [2007, “A New Type of Diabatic Flow Pattern Map for Boiling Heat Transfer in Microchannels,” J. Micro-mech. Microeng., 17, pp. 788–796]. Based on these maps, CHF mainly occurred in the annular flow regime in the larger tubes. The flow pattern for the 0.50 mm ID tube was not conclusively identified. Four correlations—the Bowring correlation, the Katto-Ohno correlation, the Thome correlation, and the Zhang correlation—were used to predict the experimental data. The correlations predicted the correct experimental trend, but the mean absolute error (MAE) was high (>15%). A new correlation was developed to fit the experimental data with a MAE of 10%. [DOI: 10.1115/1.4003160]

Keywords: critical heat flux, R134a, boiling, microscale

1Introduction

Two-phase flow boiling at the microscale has attracted much interest over the last decade because of its high heat dissipation capabilities and potential applications in compact heat exchangers. Two-phase flow exhibits high heat transfer coefficients, uniform surface, and fluid temperatures, which have made it an attractive heat transfer mechanism for use with electronics cooling in the near future. Increasing integrated circuit (IC) packaging density has resulted in very high power dissipation rates, and the present air-cooled systems may not be able to meet the high heat flux demands. There is a need to develop alternative liquid cooling technologies, which could involve a heat flux controlled flow boiling setup. Such a system can suffer from flow instabilities, and its operation would be limited by the critical heat flux (CHF) condition. The CHF condition results in a deteriorating heat transfer process, which for imposed heat flux situation can lead to very high surface temperatures and catastrophic system failure.

A considerable number of studies on the CHF condition for flow boiling at the macroscale have been conducted but much less at the microscale. Most of these microscale studies used water or phased-out CFC and HCFC refrigerants as the test fluid and, hence, data for new refrigerants such as R134a are scarce. A detailed review of CHF studies for flow boiling of water in circular tubes is available in Ref. [1] based on over 30,000 CHF data points.

Roday and Jensen [2] studied the CHF condition for flow boiling of water and R123 in circular microtubes. Departure from nucleate boiling (DNB) was observed at the highly subcooled regime and dryout was reported to occur at the saturated regime. CHF increased with increasing mass fluxes and decreased slightly with a decrease in inlet subcooling. A significant increase in CHF was observed in the saturated region compared with the subcooled region. They speculated that this is due to the change in CHF mechanism from the subcooled DNB type behavior to the dryout mechanism at higher qualities. CHF first decreased with an increase in quality in the subcooled region but CHF showed an increase with increasing qualities at the saturation point. In the case of water, CHF was found to increase with an increase in saturation pressure while it did not affect the CHF condition in the case of R123. CHF was found to decrease with an increase in heated length. In the case of saturated R123 data, CHF decreased with increasing tube diameters. Similar to Ref. [3], wall temperature oscillations were observed just before CHF occurred. The authors also developed a subcooled CHF correlation based on their experimental data.

Roach et al. [4] studied the CHF condition for the flow of subcooled water through uniformly heated circular tubes of diameters 1.17 mm and 1.45 mm for mass fluxes of 250–1000 kg/m² s, exit pressures of 344–1043 kPa, and inlet temperatures of 49–72°C. CHF was found to occur at high qualities (greater than 0.36). Because of the high critical qualities, it was argued that dryout was the CHF governing mechanism. CHF was found to increase with increasing mass fluxes and increasing pressures. However, CHF increased with increasing channel diameters.

Yu et al. [5] also studied the CHF condition for the flow boiling of water in small horizontal tubes with a diameter of 2.98 mm and a length of 0.91 m at a pressure of 200 kPa, mass fluxes of 50–200 kg/m² s, and inlet temperatures from ambient to 80°C. The test loop did not have an upstream throttle valve. CHF was found to occur at qualities greater than 0.5, and CHF increased with increasing mass fluxes. CHF was also found to increase with increasing qualities.

Wojtan et al. [6] undertook an experimental study of the CHF condition for the saturated flow boiling of R134a and R245fa in
single circular microtubes with internal diameters of 0.50 mm and 0.80 mm. Results indicated a very strong dependence of CHF on heated length, diameter, and mass flux. However, inlet subcoolings and saturation pressures were found to have a negligible effect on saturated CHF. CHF was found to decrease with increasing exit qualities and increase with increasing tube diameters. The authors also proposed a modified version of the Katto–Ohno correlation [7] for microscale flow boiling. This was the only study undertaken to investigate the CHF condition for flow boiling of R134a in circular microchannels at diameters less than 1.0 mm. However, the experiments were carried out over limited ranges of saturation pressures and inlet subcoolings.

Several researchers studied the CHF phenomenon in rectangular parallel microchannels [8–14] using a variety of fluids (e.g., water or refrigerants). There are several discrepancies in the literature regarding the CHF data at the microscale. Researchers have observed different parametric trends, which might be due to the variety of fluids, geometries, and operating conditions used in the experiments. Typically, CHF increases with increasing mass flux, but the effects of saturation pressure, inlet subcooling, tube diameters, tube geometry, and exit qualities were not clearly defined. Most of the experimental CHF data were on water and CFC/HCFC refrigerants. Unfortunately, due to environmental restrictions and safety considerations, these fluids cannot be used for electronics cooling.

The present study uses R134a, a HFC refrigerant with low ozone-depleting potential, as the test fluid. A single circular microchannel was chosen as the test section as it is free of extraneous effects such as parallel channel flow instabilities, flow redistribution, conjugate heat transfer effects, etc. To the best of the author’s knowledge, there has been only one study [6], which studied the CHF condition for the flow boiling of R134a in circular microtubes with diameters below 1.0 mm. However, the experiments were conducted over a limited range of inlet subcooling (2–15°C) and saturation temperatures (30 and 35°C). The present study was designed to address the lack of experimental data and identify the correct parametric trends by conducting extensive experiments with R134a over a wide range of mass fluxes, saturation pressures, exit qualities, inlet subcooling, and tube diameters. The existing CHF correlations were compared with the experimental data, and a modified CHF correlation was proposed.

2 Experimental Apparatus and Procedures

To study the CHF condition for the flow boiling of R134a, a closed flow loop was constructed, which was composed of the test section, inlet and outlet plenums, the associated instrumentation, tube-in-tube countercflow condenser, bladder accumulator to control the system pressure, liquid pump, filter, flowmeter, second subcooler, and an inlet preheater. A needle valve before the test section throttled the flow before the test section to reduce flow oscillations. The schematic of the flow loop is shown in part I of this study [15].

The test sections were made of 120 mm, 127 mm, and 128 mm long stainless steel hypodermic round tubes, with inner diameters of 0.50 mm, 0.96 mm, and 1.60 mm, respectively. The test section was heated using a direct current (dc) power supply. For safety reasons, the electric circuit also included a relay to shut off the power at the onset of the CHF condition. Details regarding the experimental apparatus, test section setup, and experimental procedures are available in Ref. [16] and in part I of this paper [15].

The CHF study involved increasing the heat flux until substantial temperature increases (>40°C) were observed at the last thermocouple location. This phenomenon indicated a deteriorating heat transfer process and the initiation of CHF. Hence, the power input was increased at minute levels until CHF was reached. This process was repeated at least twice to accurately determine the critical heat flux. The solid-state relay switched off the power if the temperature increased beyond 80°C to prevent device burnout or refrigerant breakdown.

A computerized data acquisition system recorded the temperature, pressure, and voltage measurements after each experimental run. The flowmeter and the pressure gage readings were recorded manually. The experimental system was controlled and the data were stored using National Instruments LabView software. The solid-state shutoff relay was controlled using the LabView program.

3 Data Reduction

The quantities measured experimentally were pressure at the exit of the test section \(P_{exit}\), pressure drop across the test section \((ΔP)\), volumetric flow rate \((V)\), test section internal diameter \((D)\) and heated length \((L)\), temperature of fluid at the inlet to the test section \((T)_i\), temperature of fluid at the exit to the test section \((T)_{exit}\), test section outer wall temperatures \((T)_{out}\), voltage across the test section \((V)\), and voltage across the shunt \((V)_{shunt}\). The derived quantities included heat loss, electrical power to the test section, heat flux, mass flow rate, mass flux, test section inner wall temperatures, local axial pressures, and qualities. The local inner wall temperature was obtained from the outer wall temperature with a 1D heat conduction model assuming steady-state radial conduction through the wall with uniform heat generation, constant material properties, and no heat losses. Heat loss \(\dot{Q}_{loss}\) was estimated by carrying out single-phase experiments and comparing the applied heat to the sensible heat gain of the fluid. The heat loss was calculated as a function of the difference between wall temperature and ambient temperature. The heat loss was taken into account while calculating the effective heat flux transferred to the fluid. The Two-phase flow generally involved much higher heat fluxes compared with single-phase flow; the heat loss percentage was less in boiling experiments (2–15%). The pressure distribution was modeled using the Lockhart–Martinelli correlation adapted for the microscale flow [17]; this distribution was adjusted so that the calculated total pressure drop matched the experimental pressure drop. The exit pressure was experimentally measured using an absolute pressure transducer. The local pressures and local enthalpy at different axial locations were used to determine the local fluid temperatures. The enthalpy at different axial locations was determined by a simple energy balance given by

\[
H_i = H_{in} + \frac{\dot{Q}_z}{m} L_i
\]

\(H_i\) is the local enthalpy at a given axial location, \(H_{in}\) is the inlet enthalpy, and \(z\) is the axial distance from the inlet. The fluid temperature was determined at different locations with the local enthalpy and pressure as the independent variables. The equilibrium vapor quality was determined as follows:

\[
x_{eq} = \frac{H_z - H_{f, sat}}{H_f}
\]

where \(H_{f, sat}\) is the saturated fluid enthalpy and \(H_f\) is the enthalpy of vaporization at the saturation temperature corresponding to the local fluid pressure \(P_z\). The qualities at the inlet and exit of the test section were also calculated in a similar fashion by substituting the appropriate value of \(z\) in Eq. (2).

An uncertainty analysis was conducted using the propagation-of-uncertainty method developed by Kline and McClintock [18]. The typical uncertainties were estimated as (depending on the test section) mass flux: ±5.0–7.0%; heat flux: ±1.0–8.0%; and quality: ±2.0–4.0%. Details of data reduction and uncertainty analysis are described by Basu [19].

4 Experimental Results

Extensive experiments were conducted to determine the parametric effects on CHF for the flow boiling of R134a in horizontal circular microchannels with internal diameters of 0.50 mm, 0.96

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mm, and 1.60 mm. The operating conditions were mass fluxes, 300–1500 kg/m² s; saturation pressures, 490–1160 kPa; inlet subcooling, 5–40°C; and exit qualities of 0.3 to 1.0. CHF occurred under saturated conditions. The experimental CHF data (in tabular form), detailed results, and plots are provided in Basu [19].

4.1 Single-Phase Studies. A single-phase heat transfer study was carried out to validate the experimental setup and data reduction procedures. The experimental Nusselt number is compared with the Gnielinski correlation [20] and showed excellent agreement with the correlation with a MAE of 9% [15].

4.2 CHF Results. The flow was throttled using a needle valve before the test section; the upstream pressure drop (100–200 kPa) reduced flow oscillations and backflow. The pressure drop showed small fluctuations (<1%) around the mean pressure drop with time.

In the experiments, boiling occurred first near the exit of the tube as it had the highest temperature along the tube. At the onset of nucleate boiling, the wall temperature dropped because of the high two-phase heat transfer coefficients. Boiling inception gradually moved upstream and the axial wall temperature distribution gradually became more uniform. Further increases in heat flux did not appreciably change the wall temperature. However, at a significantly higher heat flux, there was a rapid increase in the wall temperature near the channel exit. This indicated a deteriorating heat transfer process, which signaled the initiation of the CHF condition.

A representative wall temperature distribution plot is shown in Fig. 1. The wall temperatures were fairly uniform when most of the test section was undergoing boiling. However, at a sufficiently high heat flux, the wall temperature at the downstream location (z = 0.12 m) increased drastically. Similar trends were observed in other test sections under different operating conditions.

The effects of mass flux on CHF for the three test sections are shown in Fig. 2. CHF increased with increasing mass flux. This trend was observed for a variety of operating conditions and concurred with previous studies [2,6–9]. CHF occurred at high qualities in the present set of experiments (ζ ≥ 0.3) and, hence, dryout appears to be the dominant CHF mechanism. As discussed in Refs. [8,21], increasing flow velocity in the churn or annular regime results in increased liquid entrainment in the vapor core. This reduces the available liquid near the wall and, thus, dry patches are more likely to be formed. The CHF data were plotted on the Hasan flow pattern map [22] and most of the data fell in the churn-annular regime. The plot is shown later in this paper.

As shown in Fig. 2, CHF increased significantly when the tube diameter changed from 0.50 mm to 0.96 mm but showed little variation when the diameter changed from 0.96 mm to 1.60 mm. The tube diameter also affected the slope of the CHF-G curve in Fig. 2. CHF-G curve shows the effect of mass flux (G) on CHF. A similar trend was observed by Roach et al. [4] with water and Wojtan et al. [6] with R134a and R245fa. Small tube diameters restricted bubble growth perpendicular to the tube wall, which generally resulted in axial bubble expansion. This seemed to influence the flow patterns prevalent in the test sections. The flow regime analysis later in the paper showed that different flow patterns were prevalent in the smallest tube compared with the larger tubes. Flow patterns (e.g., bubbly, annular, intermittent, and churn) play a significant role in the CHF mechanism. Generally, at low qualities and the bubbly regime, DNB is the dominant CHF mechanism. At higher qualities and the annular flow regime, CHF occurs mainly due to dryout.

Saturation pressure had a significant and complex effect on CHF; and it was manifested in different forms in the three test sections. As has been pointed out in Ref. [8], it is difficult to isolate independently the effects of saturation pressure as it affected several key variables including enthalpy of vaporization, liquid-to-vapor density ratio (ρl/ρv), surface tension, and inlet subcooled conditions. Decreasing (ρl/ρv) and increasing inlet subcooled temperature tend to increase CHF with increasing saturation pressures, while decreasing surface tension and enthalpy of vaporization tend to reduce CHF. The final effect of saturation pressure on CHF depends on the individual contributions of each of the variables. The effects of saturation pressure on CHF are shown on Fig. 3 for the 0.50 mm ID tube. Similar results were obtained for the other two test sections. The inlet subcooling was kept constant in all the experiments. The (ρl/ρv) ratio decreased from 52 to 20 corresponding to a saturation pressure increase from 490 kPa to 1160 kPa while the enthalpy of vaporization decreased from 187 kJ/kg to 158 kJ/kg for the same pressure rise. The surface tension decreased from 0.009442 N/m (Tsat=15°C) to 0.005502 N/m (Tsat=45°C). A combination of the above factors generally resulted in decreasing CHF with increasing saturation pressures. However, as seen in Fig. 3, the opposite trend was also observed over limited pressure ranges.

CHF generally increased with an increase in the inlet subcool-
ing, as the bulk enthalpy of the fluid decreases as the subcooling increases. For a fixed mass flux, this resulted in boiling inception at higher heat fluxes and a higher CHF in all three test sections. Figure 4 shows a representative plot for the 1.60 mm ID test section. This contradicts to an extent the studies of Wojtan et al. [6] and Qu and Mudawar [9], who found CHF to be independent of inlet subcooling but concur with Roday and Jensen [2], who observed a significant dependency of inlet subcooling on CHF in their study. However, in Ref. [6], the experiments were conducted over a limited range of subcooling, while a significant amount of vapor backflow was observed in Ref. [9].

The quality at which CHF occurs is another important factor, influencing the CHF mechanism. CHF occurred under saturated conditions at qualities greater than 0.30. Increasing vapor quality at a given mass flux resulted in a lower CHF as it was difficult to keep the wall wetted. This trend was observed in the 0.96 mm and 1.60 mm ID tubes. Results for the 0.96 mm ID test section are shown in Fig. 5. Similar trends have been observed by Wojtan et al. [6] and Košar and Peles [8]. However, a different trend was observed for the 0.50 mm ID test section (Fig. 6). This could be mainly due to the different flow regimes being prevalent in the smallest tube or the higher uncertainties associated with flow in the miniscale channel size. A detailed discussion is presented in Sec. 5.

5 Flow Regimes During CHF Condition

Flow regimes play an important role in determining the CHF mechanism. Flow characteristics are markedly different in microscale than in macroscale tubes because of the increased surface tension effects at diminishing length scales. Flow regimes depend upon several factors such as fluid properties, channel geometry and size, and flow rates. In a microchannel, the flow regimes are primarily dependent on the interactions between the surface tension forces and the inertia forces. Surface tension dominated flow regimes include bubbly, plug, and slug flow, while inertia dominated flow includes annular flow and churn flow [22]. To obtain better insight to the dominant flow regimes, the experimental data were mapped using the universal flow map of Hasan et al. [22] (Fig. 7) using the superficial liquid and the superficial vapor velocities, respectively.
length 20–70 mm, mass fluxes of 210–2094 kg/m²s, saturation temperatures of 26°C, 30°C, and 35°C, and inlet subcooling of 2–15°C. It was recommended that the flow pattern map be used within these operating parameters. Furthermore, the flow pattern map is not universal and a new map is required for different diameters, heat flux, and saturation pressures. It should also be noted that the flow map was developed with the fluid properties defined at the inlet pressure; however, in the present study, the exit pressure was used to determine the fluid properties. The transition line for the isolated bubble and coalesced bubble regime is given by

\[ \frac{j_f}{P_f} = \frac{G(1-x_s)}{P_f} \]  

(3)

\[ \frac{j_g}{P_g} = \frac{G_{x_c}}{P_g} \]  

(4)

Most of the CHF data points fell in the intermittent zone and the annular regime. The points that fell in the annular regime were close to the churn-annular transition line or the boundary of the annular-intermittent zone. The intermittent flow regime regroups all the slug and plug flows, as well as the transition flow occurring in the vicinity of the intermittent region [2]. It is clearly seen that CHF in the 0.5 mm ID tube mainly occurred in the intermittent region or at the transition of annular to intermittent or churn flow. This could be linked to the unique CHF characteristics with respect to vapor quality in the smallest tube. Most of the data points for the 0.96 mm and 1.60 mm tube fall in the annular regime or very close to the intermittent-annular boundary. CHF decreases with increasing vapor qualities at this regime and the same has been observed in the experiments. High mass fluxes generally result in an annular flow regime. Higher mass fluxes also result in liquid entrainment in the vapor core. This resulted in less liquid in the film near the wall, leading to dryout.

For completeness, the experimental CHF data were also plotted on the Revellin and Thome [23] flow pattern map. The diabatic flow pattern map was developed for boiling heat transfer flow of refrigerants R134a and R245fa in circular microtubes. Three distinct flow regimes—isolated bubble regime, coalesced bubble regime, and the annular flow regime—were identified. Empirical correlations were given to identify the transition boundaries. The flow pattern map was developed for flow boiling of R134a and R245fa in circular microtubes of 0.509 mm and 0.790 mm ID, length 20–70 mm, mass fluxes of 210–2094 kg/m²s, heat fluxes of 3.1–597 kW/m², saturation temperatures of 26°C, 30°C, and 35°C, and inlet subcooling of 2–15°C. It was recommended that the flow pattern map be used within these operating parameters. Furthermore, the flow pattern map is not universal and a new map is required for different diameters, heat flux, and saturation pressures. It should also be noted that the flow map was developed with the fluid properties defined at the inlet pressure; however, in the present study, the exit pressure was used to determine the fluid properties. The transition line for the isolated bubble and coalesced bubble regime is given by

\[ x_{CB} = 0.763 \left( \frac{\text{Re}_{c} \text{Bo}^{0.41}}{\text{We}_{c}^{0.34}} \right) \]  

(5)

Unfortunately, this requires a flow map to be developed for every CHF data point as the transition line is influenced by the heat flux (present in the nondimensional Boiling number). Therefore, in the present study, only the transition from the coalesced bubble (CB) to the annular flow regime is studied for a few select cases. The transition boundary is defined as

\[ x_{CB/4} = 0.00014(\text{Re}_{c}^{0.47} \text{We}_{c}^{1.23}) \]  

(6)

A few representative plots are shown in Fig. 8 for the three test sections for a constant saturation pressure of \( P_{sat} = 890 \) kPa. The flow regime was annular in all the three tubes.

Rodriguez and Jensen [2] also undertook a similar flow map analysis in their water and R123 study. Based on the Hasan flow pattern map [22], the dominant flow patterns for water were churn and annular, while for R123, it was in the intermittent and annular flow regimes. According to the Thome flow pattern map [23], the dominant flow pattern was confined bubble for both fluids. However, for the flow boiling of R123 in the 0.286 mm ID tube, data were obtained in the annular-confined bubble boundary. In comparison, the present study with R134a showed that the dominant flow pattern was annular using both flow pattern maps. It is interesting to note the differences in the flow patterns between a medium-pressure (R134a) and low pressure (R123) refrigerant. It is possible that fluid properties such as surface tension and viscosity affect the flow patterns at the microscale.

### 6 Correlations for Saturated CHF

A total of 113 CHF data points were obtained in the present experimental study. CHF occurred under saturated conditions. The experimental data were compared with four saturated CHF correlations—the macroscale Katto–Ohno correlation [7], the Bowring correlation [24], the Wojtan–Thome correlation [6], and the Zhang correlation [25]. Details of the correlation are given in Table 1. The predictive capabilities of these correlations are quantitatively assessed using the MAE defined as

\[ \text{MAE} = \frac{1}{M} \sum_{i=1}^{M} \left( \frac{q_{\text{crit, pred}} - q_{\text{crit, exp}}}{q_{\text{crit, exp}}} \right) \times 100\% \]  

(7)

where \( M \) is the number of data points.

The Katto–Ohno correlation generally overpredicted the data slightly for all three tubes with a MAE of 26%; 64% of the data were predicted within the ±30% error band. Although the correlation was developed for macroscale flow, it provided better results at lower diameters.

The Bowring correlation predicted the experimental data with a MAE of 27%; 59% of the data fell within the ±30% error band. The correlation best fit the data for the 1.60 mm ID tube, and the accuracy was lower at smaller diameters.

The Thome correlation predicted the experimental data with a MAE of 25%; 61% of the data fell within the ±30% error band. The correlation generally underpredicted the experimental data. The accuracy of the correlation was the highest for the 1.60 mm ID tube.

The Zhang correlation predicted the experimental data with a MAE of 19%; 85% of the data fell within the ±30% error band. The correlation generally overpredicted the experimental data. However, all four correlations predicted the correct trend. This is illustrated in Fig. 9 for the 1.60 mm ID tube.
Table 1 List of correlations used in the present study

<table>
<thead>
<tr>
<th>Author(s)</th>
<th>Correlation equation</th>
<th>Fluid, geometry</th>
<th>Parameter range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wojtan–Thome [6]</td>
<td>( q_{\text{crit}} = 0.437 \left( \frac{\rho}{\rho_v} \right)^{0.073} \text{We}^{-0.24} \left( \frac{L}{d} \right)^{-0.72} \text{Gh} )</td>
<td>R134a, circular microtubes ( d = 0.50, 0.80 ) mm</td>
<td>( G = 400–1600 ) kg/m² s ( T_{\text{sat}} = 30, 35^\circ \text{C} ) ( x = 0.35–0.95 )</td>
</tr>
<tr>
<td>Katto and Ohno [7]</td>
<td>( q_{\text{crit}} = q_{\text{co}} + K (h_f - h_t) ) ( q_{\text{co}} = f(\text{Gh}, \text{We}, \text{L}, \text{v}) )</td>
<td>uniformly heated, vertical tubes, ( 5 \leq (L/d) \leq 880 )</td>
<td>( 0.00003 \leq (\rho/\rho_v) \leq 0.41 ) ( 3 \times 10^{-3} \leq \frac{L}{\text{We}} \leq 2 \times 10^{-2} )</td>
</tr>
<tr>
<td>Bowring [24]</td>
<td>( q_{\text{crit}} = \frac{A + 0.25 \text{Gh} \rho \rho_v}{c_e} ), where ( A ) and ( c_e ) are empirical constants</td>
<td>2 \leq D \leq 45 mm ( 0.15 \leq \tau \leq 3.7 ) m</td>
<td>136 \leq G \leq 18,600 kg/m² s ( 2 \leq P_{\text{sat}} \leq 190 ) bar</td>
</tr>
<tr>
<td>Zhang et al. [25]</td>
<td>( \frac{q_{\text{crit}}}{\text{Gh}} = 0.0352 \left[ \text{We}^{0.0119} \left( \frac{\text{L}}{\text{d}} \right)^{0.31} \left( \frac{\rho}{\rho_v} \right)^{0.3617} \left( \frac{\text{v}}{\text{L}} \right)^{-0.295} \right] )</td>
<td>Water, small diameter tubes ( D_h = 0.33–6.22 ) mm</td>
<td>( P_{\text{sat}} = 0.101–19.0 ) MPa ( G = 5.33–1.34 \times 10^5 ) kg/m³ s ( x_{\text{crit}} = -1.75–1.00 ) ( x_{\text{in}} = -2.35–0.0 )</td>
</tr>
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</table>

Fig. 8 Experimental R134a data mapped on the flow map developed by Revellin and Thome [23]
To better predict the present experimental data, a new saturated CHF correlation was developed using a nonlinear regression model developed by DataFit9. Five data points corresponding to \( G = 1000 \text{ kg/m}^2\text{s} \) for the \( d = 0.50 \text{ mm} \) ID tube were discarded in the analysis as the experiments suffered from significant heat loss under those conditions. The correlation was developed using the experimental R134a CHF data obtained in the present experiment and the saturated R123 data of Roday and Jensen \(^1\) with a MAE of 12% (Fig. 11); 92% of the data points fell in the ±30% error band. The performance of the different correlations was quantitatively measured in terms of MAE and the percentage of data points within the ±30% error band (Table 2).

The parametric effects of mass flux, vapor quality, tube diameter, saturation pressure, and inlet subcooling on CHF were studied in the present study. CHF was found to be a local phenomenon that was mainly defined by the vapor quality. Thus, quality was used as one of the independent variables. CHF was found to increase with mass flux, while vapor quality decreased with increasing mass flux. Mass flux and vapor quality had a combined influence on CHF, which is evident from the correlation. There was a complex relationship between saturation pressure and CHF. Increasing saturation pressure resulted in increasing vapor-to-liquid density \( (p_v/p_l) \) ratio and decreasing enthalpy of vaporization. The effect of saturation pressure was, therefore, dependent on the relative contribution of two contradicting influences. Increasing \( (p_v/p_l) \) ratio and increasing enthalphy of vaporization resulted in an increase in CHF. This effect is also correctly captured in the correlation. The test section geometry effect is manifested in the \( (L_d/d) \) ratio. CHF increased with increasing tube diameter and decreased with increasing heated length. The negative exponent to

\[
\frac{q_{\text{crit}}}{Gh_{lv}} = 0.3784 \left( \frac{p_v}{p_l} \right)^{0.051} \left( \frac{L_d}{d} \right)^{-1.03} x^{0.8} \tag{8}
\]

The new correlation fit the test data well with a MAE of 10% (Fig. 10); 100% of the data points were within the ±30% error limits. The correlation predicted the R123 data of Roday and Jensen \(^2\) with a MAE of 12% (Fig. 11); 92% of the data points fell in the ±30% error band. The performance of the different correlations was quantitatively measured in terms of MAE and the percentage of data points within the ±30% error band (Table 2).

### Table 2 Comparison of the experimental data with the correlations

<table>
<thead>
<tr>
<th>Correlations</th>
<th>( d = 0.50 \text{ mm} )</th>
<th>( d = 0.96 \text{ mm} )</th>
<th>( d = 1.60 \text{ mm} )</th>
<th>Average</th>
<th>( d = 0.50 \text{ mm} )</th>
<th>( d = 0.96 \text{ mm} )</th>
<th>( d = 1.60 \text{ mm} )</th>
<th>Average</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wojtan–Thome (^6)</td>
<td>28</td>
<td>32</td>
<td>17</td>
<td>25</td>
<td>44</td>
<td>41</td>
<td>87</td>
<td>61</td>
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<tr>
<td>Katto and Ohno (^7)</td>
<td>30</td>
<td>16</td>
<td>31</td>
<td>26</td>
<td>56</td>
<td>84</td>
<td>51</td>
<td>64</td>
</tr>
<tr>
<td>Bowring (^24)</td>
<td>31</td>
<td>34</td>
<td>19</td>
<td>27</td>
<td>52</td>
<td>32</td>
<td>84</td>
<td>59</td>
</tr>
<tr>
<td>Zhang (^25)</td>
<td>28</td>
<td>12</td>
<td>15</td>
<td>19</td>
<td>63</td>
<td>97</td>
<td>91</td>
<td>85</td>
</tr>
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<td>New correlation</td>
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\(^1\) Roday and Jensen, \(^2\) Roday and Jensen, \(^6\) Wojtan–Thome, \(^7\) Katto and Ohno.
the \((L_v/d)\) term in the correlation captured the correct parametric trend. We recommend the correlation to be used for flow boiling of R134a and R123 within the limits of the operating conditions prescribed in the present paper and the study of Roday and Jensen [2].

7 Conclusions

In the present study, extensive experiments were conducted to investigate the CHF condition for flow boiling of R134a in circular microtubes with internal diameters of 0.50 mm, 0.96 mm, and 1.60 mm. Based on this study, the following conclusions can be drawn.

- CHF generally increased with increasing mass flux. Similar trends were observed by other studies. Higher mass flux generally resulted in CHF occurring at a lower quality. CHF was found to decrease with decreasing diameters and increasing saturation pressures. CHF also increased with increasing inlet subcooling. CHF generally decreased with increasing quality in the 0.96 mm and 1.60 mm ID tubes. The effect of quality on CHF for the 0.50 mm ID tube was not clearly understood.

- The experimental CHF data were mapped using the universal Haslan flow map [22] and the Revellin and Thome diabatic flow pattern map [23]. The dominant flow regime was found to be annular flow in the larger test sections according to the Haslan flow map while the flow regime for the 0.50 mm ID tube during CHF was a combination of intermittent, churn, and annular flows. The Revellin and Thome flow map, on the other hand, identified the flow regime as annular in all three test sections.

- The experimental data were compared with four correlations—the Katto–Ohno correlation, the Bowring correlation, the Thome correlation, and the Zang correlation. Bowring, Thome, and Katto–Ohno predicted the data with a MAE of approximately 25%. The Zang correlation predicted the data with a MAE of 19%. A new correlation was developed to better fit the experimental data, which predicted the data with a MAE of 10%.

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Nomenclature

- \(Bo\) = boiling number, \(q^* / \Delta H_{fg}\)
- \(D\) = diameter, m
- \(G\) = mass flux, kg/m² s
- \(H\) = specific enthalpy, J/kg
- \(ID\) = inner diameter, mm
- \(L\) = length, m
- \(P\) = pressure, kPa
- \(\dot{Q}\) = power, W
- \(q^*\) = heat flux, W/m²
- \(Re\) = Reynolds number
- \(T\) = temperature, K
- \(V\) = voltage, V
- \(V\) = volumetric flow rate, m³/s
- \(We\) = Weber number, \(G^2L/\rho \sigma\)
- \(x\) = quality
- \(z\) = axial location along the test section, m

Greek Letters

- \(\Delta T\) = temperature difference, K
- \(\mu\) = dynamic viscosity, Ns/m²
- \(\rho\) = density, kg/m³
- \(\sigma\) = surface tension, N/m

Subscripts

- CHF = critical heat flux
- crit = critical
- eq = equilibrium
- exit = exit
- expt = experimental
- \(f\) = fluid
- \(fg\) = fluid-gas
- \(g\) = gas
- \(go\) = gas only
- \(h\) = heated
- \(i\) = internal
- \(in\) = inlet
- \(l\) = liquid
- \(lo\) = liquid only
- \(loss\) = loss
- \(lv\) = liquid-vapor
- \(pred\) = predicted
- \(tp\) = two-phase
- \(ts\) = test section
- \(sat\) = saturation
- subcooling = subcooling
- \(w\), \(wall\) = wall
- \(v\) = vapor
- \(z\) = axial location along the test section, m

References

Proceedings of the International Heat Transfer Conference, IHTC14, Washing-
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