DRAFT: STUDY OF CHF CONDITION FOR FLOW BOILING OF R134A IN CIRCULAR MICROCHANNELS


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ABSTRACT
An experimental study was carried out to investigate the CHF condition for flow boiling of R134a in single circular microtubes with inner diameters of 0.50 mm and 0.96 mm. The effects of mass flux, saturation pressure, inlet subcooling, tube diameter, and vapor quality on CHF were studied. The flow parameters investigated were as follows: exit pressures of 670, 890, and 1160 kPa; mass fluxes of 300-1500 kg/m²s; and inlet subcooling of 5, 20, and 40 °C. CHF occurred under saturated conditions. CHF was found to increase with an increase in mass flux, tube diameter, and inlet subcooling. CHF decreased with increasing saturation pressures and increasing vapor qualities. The experimental data were compared to three CHF correlations — the Bowring correlation, the Katto-Ohno correlation, and the Thome correlation. The correlations predicted the correct experimental trend.

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<table>
<thead>
<tr>
<th>Symbol</th>
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<tbody>
<tr>
<td>C_p</td>
<td>Specific heat, J/kgK</td>
</tr>
<tr>
<td>D, d</td>
<td>Diameter, m</td>
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<td>G</td>
<td>Mass flux, kg/m²s</td>
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<td>H</td>
<td>Specific enthalpy, J/kg</td>
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<td>I</td>
<td>Current, amp</td>
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<td>V̇</td>
<td>Volumetric flow rate, m³/s</td>
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<td>f</td>
<td>Friction factor</td>
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<tr>
<td>m</td>
<td>Mass flow rate, kg/s</td>
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<td>q''</td>
<td>Heat flux, W/m²</td>
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<tr>
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<td>Quality</td>
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<td>Temperature difference, K</td>
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<td>Dynamic viscosity, Ns/m²</td>
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<td>ν</td>
<td>Kinematic viscosity, m²/s</td>
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<tr>
<td>ρ</td>
<td>Density, kg/m³</td>
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<td>σ</td>
<td>Surface tension, N/m</td>
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<table>
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<th>Subscripts</th>
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<td>CHF</td>
<td>Critical heat flux</td>
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<td>Gas</td>
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<td>Fluid-Gas</td>
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<td>in</td>
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<td>Liquid</td>
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Flow boiling heat transfer at small scales exhibit very high heat transfer coefficients, low pressure drops, and uniform fluid and wall temperatures. Thus, forced convection boiling in microchannels can be effectively used to dissipate high heat fluxes in densely packed microelectronic devices. However, the flow boiling process suffers from two serious disadvantages—flow instabilities and the critical heat flux (CHF) condition. The critical heat flux condition limits the maximum heat dissipation capability of a system. The onset of CHF condition is accompanied by a deterioration of the heat transfer process resulting in low heat transfer coefficients and for a constant heat flux boundary condition, high surface temperatures. These conditions are not desirable in a heat flux controlled system. Therefore, it is important to accurately predict the CHF condition and identify the correct parametric trends.

Exact mechanisms that lead to the CHF condition are not fully understood. It is believed that the CHF mechanism is linked to the prevalent flow regime. Generally, at higher mass fluxes and high qualities, the annular flow regime is observed. Under these conditions, CHF occurred mainly due to liquid dryout. On the other hand, during subcooled boiling or nucleate boiling, CHF is triggered mainly due to the Departure from Nucleate Boiling (DNB) mechanism. This occurs when liquid is unable to reach the heated surface due to intensive bubble nucleation. CHF can occur under both subcooled and saturated flow conditions.

Traditionally, most two-phase micro scale studies have dealt with flow boiling heat transfer in microchannels while very few experimental studies were conducted to study the CHF condition. Most CHF studies have used water as the test fluid. For applications in electronics cooling, however, dielectric fluids like refrigerants are more practical. Refrigerants generally have inferior and much different thermophysical properties compared to water. A summary of CHF studies with water in circular tubes is given in [1]. Roday and Jensen [2] also studied the CHF condition for flow boiling of water and R123 in circular microtubes. CHF occurred mostly under subcooled conditions. Wojtan et al. [3] studied the saturated CHF condition for flow boiling of R134a and R245fa in single circular microtubes with an internal diameter of 0.5 mm and 0.8 mm. The tests were conducted at saturation temperatures of 30 °C and 35 °C while inlet subcooling varied from 2-15 °C. Several studies [4–8] experimentally studied the CHF condition for flow boiling of water and refrigerants in parallel microchannel heat sinks.

R134a has been used as the test fluid as it is a dielectric fluid and has low ozone-depleting potential. However, there are a very limited number of experimental studies investigating the CHF condition for flow boiling of R134a at the micro scale. The main focus of the present study is to investigate the physical processes controlling the CHF mechanism, and a single circular microtube was chosen as the test section. A circular microtube is free of extraneous influences like parallel channel flow instabilities, flow maldistribution, conjugate heat transfer effects, etc., which adversely affected CHF. Wojtan et al. [3] was the only study to investigate the CHF phenomenon in circular tubes for flow boiling of R134a at diameters below 1.0 mm. The experiments were conducted over a very small range of saturation pressures and inlet subcoolings. The present experimental program was designed to address the lack of experimental data on CHF condition for flow boiling of R134a at the micro scale.

EXPERIMENTAL APPARATUS AND PROCEDURES

Experimental Facility

To study the CHF condition for flow boiling of R134a, a closed flow loop (Fig. 1) was constructed, which comprised of the test section, inlet and outlet plenums, the associated instrumentation, tube-in-tube counterflow condenser, bladder accumulator to control the system pressure, liquid pump, filter, flow meter, second subcooler, and an inlet preheater. Several needle and shut off valves at different locations were used for safety, easy charging and discharging and additional control of the flow rate. The needle valve before the test section was installed to throttle the flow before entering the microchannels to reduce the probability of back flow and flow oscillations. The preheater was constructed by wrapping high resistance Nichrome wire around a tube and power was supplied using a variable transformer.

The flow loop was insulated using vinyl-backed fiberglass and semi-rigid foam insulations to prevent heat loss to the ambient. The complete system was placed inside an enclosure made of Styrofoam. The air trapped inside the enclosure was then cooled by a heat exchanger, which distributed cold air using a fan. This ensured that the ambient temperature was sufficiently low to prevent significant heat loss or gain and allowed operation at lower saturation temperatures and higher degrees of subcooling. The cold fluid to this heat exchanger, the condenser, and the second subcooler was supplied by a chiller unit with 40% ethylene glycol-water solution as the operating fluid.

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Test Section Apparatus

The test sections were made of 120 and 127 mm long stainless steel hypodermic round tubes, with an inner diameter of 0.50 and 0.96 mm, respectively. The test section ends were sanded and polished on a sander machine using 400- and 600- grit sandpaper, lapped using a 1µm diamond solution, and then inspected under a microscope for burrs or other structural non-uniformities. The test section was connected to two brass plates that served as electrical buss bars, which were then fastened using bolts to two Delrin blocks on either side. The Delrin blocks served as the inlet and outlet plena, which were used to house the thermocouple probes and the two pressure transducers. This allowed the measurement of temperatures and pressures at the inlet and outlet of the test section. The test section was heated using a DC power supply. For safety reasons, the electric circuit also included a relay to shut off the power at the onset of CHF condition.

In addition to the thermocouple probes at the inlet and exit, an array of T-type ungrounded thermocouples were placed at regular intervals on the test section to map the wall temperature. The test section was heavily insulated and single-phase studies indicated that heat loss was negligible.

Instrumentation

Temperature measurements involved measuring the test section wall temperatures and the fluid temperatures at different locations. Several ungrounded insulated T-type thermocouples (AWG no.30) were epoxied to the test section wall at fixed intervals. Five thermocouple probes were placed inline with the fluid flow to reduce fin effect at different locations in the loop for monitoring the fluid temperatures. Two probes were placed at the inlet and exit to measure the single-phase enthalpy gain across the test section. All thermocouples and thermocouple probes were calibrated using a water/ethylene glycol heater-chiller from -15°C to 100°C. The accuracy of temperature measurements using these thermocouple probes was ±0.4°C.

Three commercial pressure gauges were placed at the inlet to the pump and before and after the preheater. An absolute pressure transducer was placed at the exit of the test section, and a differential pressure transducer measured the pressure drop across the test section. The absolute pressure transducer was calibrated using a deadweight tester while the differential one was factory calibrated. The measurements of the pressure transducers were read on a National Instruments LabVIEW Data Acquisition System using a computer. Both pressure transducers have an accuracy of ±0.25% of full scale. The absolute pressure transducer has a pressure range of 0 to 1400 kPa while the differential pressure transducer can measure pressure drops from 0 to 170 kPa.

Flow rate was measured using a three-tube rotameter, which was calibrated for R134a using the weigh tank approach and a pressurized flow loop at different temperatures and pressures. The accuracy of the flow measurement was ±4% of the reading.

The diameter of the test section was measured using an optical microscope with a 40X magnification and a measuring probe attachment. The accuracy of the microscope was ±0.0054 mm. The length of the test section was measured by vernier calipers and the accuracy was ±0.03 mm.

Power supplied to the test section was measured by taking voltage measurements across the test section using LabVIEW and current measurement across the current shunt of desired rating. The accuracy of the shunts was ±0.25%.

A computerized data acquisition system was designed to record 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 thermocouple calibration was stored in the LabVIEW software. The voltage measurements included the output voltages from the two pressure transducers, test section voltage, and the shunt voltage. The solid-state shutoff relay was controlled using the LabVIEW program. In the event of CHF, the relay cut-off power to the test section to prevent overheating. The temperature of the last thermocouple was monitored for determining the CHF condition.

The raw data were then analyzed and reduced to useful thermal data using computer programs developed in Lab View, MS Excel, and Engineering Equation Solver (EES).

Experimental Procedures

A step-by-step procedure was followed to safely make fast and precise measurements. The water chiller’s temperature was set based on the desired test section inlet temperature. The pump was then switched on, and the preheater was used to achieve a desired test section inlet temperature.
Steady state was reached when the temperatures of the test section wall thermocouples and the thermocouple probes at the inlet and exit of the test section did not vary more than 0.1 °C for five minutes, which took approximately two hours from the time the preheater was switched on. Once steady state was reached, the test section power was switched on and adjusted to the desired power.

The temperature and pressure transducer readings were automatically recorded using LabVIEW. After a particular set of data had been recorded, the system condition was changed and the entire process was repeated. The experiments were conducted for various power inputs (heat flux), mass flow rates, inlet conditions, and system pressure.

The CHF study involved increasing the heat flux until substantial temperature increases were observed in the last thermocouple readings. 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. CHF involved a sudden increase in the temperature of around 40 °C in about twenty seconds. 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.

DATA REDUCTION

The quantities measured experimentally were as follows: pressure at the exit of the test section \( (P_{\text{exit}}) \), pressure drop across the test section \( (\Delta P) \), volumetric flow rate \( (V) \), temperature of fluid at inlet \( (T_{\text{in}}) \), and heated length \( (L_h) \), temperature of fluid at exit to the test section \( (T_{\text{out}}) \), test section outer wall temperature \( (T_w) \), and voltage across the shunt \( (V_{\text{shunt}}) \), and voltage across the test section \( (V_{ts}) \), and voltage across the shunt \( (V_{\text{shunt}}) \).

The Engineering Equation Solver (EES) was used for data reduction and the relevant equations are discussed below.

The electrical power to the test section, \( Q_{ts} \), is given by

\[
Q_{ts} = I_{ts}V_{ts} \tag{1}
\]

The heat flux, \( q'' \), was calculated from the power, \( \dot{Q} \), as follows

\[
\dot{Q} = Q_{ts} - Q_{\text{loss}} \tag{2}
\]

\[
q'' = \frac{\dot{Q}}{\pi D_i L_h} \tag{3}
\]

while the mass flow rate, \( \dot{m} \), and the mass flux, \( G \), is given by

\[
\dot{m} = \rho \dot{V} \tag{4}
\]

\[
G = \frac{4\dot{m}}{\pi D_i^2} \tag{5}
\]

The wall superheat, \( \Delta T_{wall} \), is given by

\[
\Delta T_{wall} = T_{w,i} - T_{sat} \tag{6}
\]

where \( T_{sat} \) is the fluid saturation temperature corresponding to the local pressure \( P_l \) and \( T_{w,i} \) is the local inside wall temperature at the thermocouple locations. The inside wall temperature was calculated from the outside wall temperature using a 1D heat conduction model assuming steady-state radial conduction through the wall with uniform heat generation and no heat loss.

The pressure drop was modeled using the well known Lockhart-Martinelli correlation [9]. The model is given by the following equations

\[
P_{\text{in}} = P_{\text{exit}} + \Delta P_{ts} \tag{7}
\]

\[
\left( \frac{dP}{dz} \right)_{tp} = \left( \frac{dP}{dz} \right)_{tp} \left( \phi_1^2 + \frac{d}{dz} \left( \frac{G^2(1-x)^2}{\rho(1-\alpha)} + \frac{G^2x^2}{\rho\alpha} \right) \right) ; \tag{8}
\]

\[
\phi_1^2 = \frac{(dP/dz)_tp}{(dP/dz)_ts}. \]

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where $P_m$ is the pressure at the inlet to the test section. The local pressure at different thermocouple locations, $P_z$, was obtained by integrating Eq. (8) and $z$ is the distance of the thermocouples from the test section inlet. The RHS of Eq. (8) accounts for pressure drop due to friction and acceleration, respectively.

A uniform heat flux $q$ was applied to the test section and hence, the power supplied to a particular section of the tube was a linear function of the length of tube considered. Since the incoming fluid was always subcooled, the inlet enthalpy, $H_{in}$, was determined from the measured inlet temperature and pressure. The enthalpy at different locations is determined by

$$H_z = H_{in} + \frac{Q}{m} z$$

(9)

The fluid temperature, $T_f$, at any $z$ is similarly calculated using EES with the pressure $P_z$ and enthalpy $H_z$ as the independent variables. $T_f$ is equal to the fluid saturation temperature when the fluid undergoes saturated flow boiling at that location.

The average equilibrium quality at different thermocouple locations is determined as follows:

$$x_{eq} = \frac{H_z - H_{f,sat}}{H_{fg}}$$

(10)

where $H_{f,sat}$ is the saturated fluid enthalpy and $H_{fg}$ is the enthalpy of vaporization at the saturation temperature corresponding to the local fluid pressure $P_z$. The quality at the inlet and exit of the test section was also calculated in a similar fashion by substituting the corresponding value for $z$.

**Uncertainty Analysis**

An uncertainty analysis was conducted using the propagation of error method developed by Kline and McClintock [10]. The typical uncertainties were estimated as (depending on the test section) mass flux: $\pm 5.0 - 7.0\%$; heat flux: $\pm 1.0 - 8.0\%$; quality: $\pm 2.0 - 4.0\%$. Details of data reduction and uncertainty analysis are described in Basu [11].

**EXPERIMENTAL RESULTS**

Extensive experiments were conducted to determine the parametric effects on CHF for flow boiling of R134a in horizontal circular microchannels. The operating conditions were: mass fluxes: 300-1500 kg/m$^2$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 results are discussed as follows.

**Single-Phase Studies**

A detailed single-phase heat transfer study was carried out to validate the experimental setup and data reduction procedures. Due to limitations of the differential pressure transducers used and the low pressure drops observed, a single-phase pressure drop study was not carried out. The heat transfer results are shown in a non-dimensional form in Fig. 2. Nusselt number increased with increasing Reynolds number. Since, Reynolds number was larger than 3000, flow was turbulent. The data were compared to the Gnielinski correlation. Experimental results showed excellent agreement with the Gnielinski correlation (Fig. 2), which is given by

$$Nu_d = \frac{(f/8)(Re_d - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$$

(11)

where, the friction factor $f$ is given by

$$f = (0.79ln(Re) - 1.64)^{-2}$$

(12)

The Gnielinski correlation is valid for $3000 < Re_d < 5 \times 10^6$ and $0.5 < Pr < 2000$. A similar single-phase study was carried out by Owhaib and Palm [12] with R134a. Reynolds number varied over a wide range from 1,000 to 17,000. Experimental results obtained in the present study show a strong agreement with the results published in [12].

**CHF Results**

Boiling occurred first near the exit of the tube as it had the highest temperature along the tube. At the onset of boiling, the wall temperature dropped because of high two-phase heat transfer coefficients. As boiling inception moved upstream, the wall temperature distribution became uniform and remained fairly constant as the heat flux was increased. However, when the heat flux increased above a certain threshold value, the heat transfer process deteriorated significantly. This occurred first near the exit of the test section, and the wall temperature at the last thermocouple location rose significantly above the rest of the test section, thus indicating the onset of CHF. To prevent device burnout, a solid-state shut-off relay was used to shutdown the power supply when this wall temperature rose above 80 °C.

A representative wall temperature distribution plot is shown in Fig. 3. The wall temperatures were fairly uniform when the entire test section was undergoing boiling. However, at high heat fluxes, wall temperature at the downstream locations increased drastically indicating the onset of CHF.

Figures 4 and 5 describe the effect of mass flux on CHF for both the 0.50 and 0.96 mm ID tubes. CHF increased generally linearly with an increase in mass flux. A similar effect was
Figure 2. Nu vs. Re for single-phase flow of R134a

Figure 3. Wall temperature variations at different heat fluxes for \( d = 0.96 \text{ mm} \) at \( P_{\text{sat}} = 890 \text{ kPa} \), \( G = 1000 \text{ kg/m}^2\text{s} \), and \( \Delta T_{\text{subcooling}} = 5 \text{ °C} \)

Figure 4. Effect of mass flux on CHF for \( d = 0.50 \text{ mm} \) and \( \Delta T_{\text{subcooling}} = 5 \text{ °C} \)

A similar effect of mass flux on CHF has been observed in other studies for a variety of fluids, geometry, and operating conditions [2–6]. Generally, increases in mass flux were accompanied by a decrease in the quality for a given inlet subcooling at CHF conditions. As has been mentioned in [4, 12], increases in flow velocity in the churn or annular regime are generally accompanied by increased liquid entrainment in the vapor core. This reduces the amount of liquid available near the wall and, thus, it becomes more difficult to keep the walls fully wetted. CHF occurred at high qualities in the present set of experiments and, hence, dryout could be regarded as the dominant CHF mechanism.

Test section geometry (e.g., diameter and length) has a significant effect on flow boiling characteristics and CHF condition. The heated length was kept approximately constant in the present set of experiments. CHF increased significantly with increasing tube diameters (Fig. 6). The tube diameter also affected the slope of the CHF-G curve. A similar trend was observed by Wojtan et al. [3] with R134a and Roach et al. [13] with water.

The CHF condition was also affected by the saturation pressure. However, the effect of saturation pressure on CHF is very complex. Fig. 7 depicts the effect of saturation pressure on CHF for the 0.50 \text{ mm} ID tube for different mass fluxes. CHF generally decreased with increasing saturation pressures. The degree of inlet subcooling was kept constant. Similar results were obtained with the 0.96 \text{ mm} ID tube. This is contrary to the trend generally observed in the literature. As has been pointed out in [4], it is difficult to isolate independently the effects of saturation pressure as it affects several key variables including enthalpy of vaporization, liquid to vapor density ratio \( (\rho_L/\rho_G) \), surface tension, and inlet subcooled conditions. Decreasing \( (\rho_L/\rho_G) \) and increasing inlet subcooled temperature tend to increase CHF with increas-
was present in all the tubes for different saturation pressures and mass fluxes (e.g., Fig. 8). This trend is similar to those observed in conventional sized tubes. Higher degrees of inlet subcooling reduced the bulk enthalpy of the fluid, which leads to an increase in CHF. However, some micro scale studies [3, 5] reported a negligible influence of inlet subcooling on CHF, while others [2] have reported a significant impact.

Increasing vapor quality tends to decrease CHF as it was more difficult to keep the wall wetted. This effect is evident for the 0.96 mm ID tube as shown in Fig. 9. For the 0.5 mm ID tube, the effect of vapor quality on CHF is shown in Figure 10. The trends observed for the 0.5 mm tube is different from the larger tube. This might be due to different flow regimes prevalent in the smaller sized tube. Similar trends have been observed by various researchers [3, 4] in studies with refrigerants R134a and R123.

**COMPARISON WITH EXISTING CORRELATIONS**

The experimental data were compared to three empirical correlations: the Katto-Ohno correlation [15], the Bowring correlation [16], and the Thome correlation [3]. The predictive capabilities of the correlations are illustrated in Figs. 11 through 13 and are quantitatively assessed using the concept of Mean Absolute Error (MAE) which is defined as

\[
MAE = \left( \frac{1}{M} \right) \sum \left| \frac{q'_{\text{crit, pred}} - q'_{\text{crit, expt}}}{q'_{\text{crit, expt}}} \right| \times 100\%
\]  

(13)

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where $M$ is the number of data points.

The Bowring correlation in general underpredicted the data with a MAE of 32%; 31% of the data fell within the ±30% error band. The Katto-Ohno correlation generally slightly overpredicted the data for both tubes. MAE was 21%; 92% of the data was predicted within the ±30% error band. The Thome correlation generally underpredicted the experimental data with a MAE of 30%, while 27% of the data fell within the ±30% error band.

All three correlations failed to predict the experimental data accurately. However, the correlations predicted the correct trends for the variation of CHF with critical quality (Figs. 14 and 15).
There were considerable scatter in the data. However, the Bowring and Katto-Ohno correlation were applied beyond the recommended range. The Katto-Ohno correlation showed the best performance. The correlations are based on a small number of data, which make their accuracy unreliable.

CONCLUSIONS

Experiments were conducted to study the effects of different parameters on CHF for flow boiling of R134a. The tests were carried out in test sections with internal diameters of 0.50 and 0.96 mm over an extensive range of mass fluxes, saturation pressures, vapor qualities, and inlet subcoolings.
Based on the present set of experiments the following conclusions can be drawn:

— CHF was found to increase with increasing mass flux. This effect was universally observed in microchannels by other researchers. High mass flux generally resulted in low vapor qualities.

— CHF decreased with decreasing diameters. The observed trend indicates that CHF is a major issue at smaller diameters. CHF thus provides a limit to the degree of miniaturization possible practically.

— CHF was found to decrease with increasing saturation pressures, which is contradictory to previously observed trends. Increasing saturation pressures increased the subcooled inlet condition and decreased the surface tension, liquid to vapor density, and enthalpy of vaporization. CHF was affected by a complex interaction between the four parameters when the system pressure changed.

— CHF generally increased with increasing inlet subcooling. This was mainly due to the enhanced contribution of single-phase heat transfer. CHF decreased with increasing vapor quality for the 0.96 mm ID tubes. However, the effect of vapor quality on CHF was not very clear for the 0.50 mm ID tube. This might be due to different flow regimes in the two test sections.

— The experimental data were compared to three correlations: the Katto-Ohno correlation; the Bowring correlation; and the Thome correlation. All three correlations failed to quantitatively predict the CHF data. The Katto-Ohno correlation showed the best agreement with the experimental data. The correlations predicted the correct CHF trend with variations in vapor quality.

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REFERENCES


