DRAFT: HEAT TRANSFER CHARACTERISTICS OF FLOW BOILING OF R134A IN UNIFORMLY HEATED HORIZONTAL CIRCULAR MICROTUBES

Saptarshi Basu, Sidy Ndao, Gregory J. Michna, Yoav Peles, Michael K. Jensen

ABSTRACT

An experimental study of two-phase heat transfer coefficients was carried out using R134a in uniformly heated horizontal circular microtubes with diameters of 0.50 mm and 1.60 mm. The effects of mass flux, heat flux, saturation pressure, and vapor quality on heat transfer coefficients were studied. The flow parameters investigated were as follows: exit pressures of 490, 670, 890, and 1160 kPa; mass fluxes of 300-1500 kg/m²s; heat fluxes of 0-350 kW/m²; inlet subcooling of 5, 20, and 40 °C; and exit qualities of 0.3 to 1.0. The parametric trends presented in the study are consistent with published literature. Heat transfer coefficients increased with increasing heat flux and saturation pressure while they were independent of variations in mass flux. Vapor quality had a negligible influence on heat transfer coefficients. For the conditions studied, the trends indicated that the dominant heat transfer mechanism was nucleate boiling. The experimental data was compared to three microchannel correlations — the Lazarek-Black, the Kandlikar, and the Tran Correlations. None of the correlations predicted the experimental data very well, although they all predicted the correct trend within limits of experimental error.

NOMENCLATURE

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<th>Symbol</th>
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<td>Diameter, m</td>
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<td>$G$</td>
<td>Mass flux, kg/m²s</td>
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<td>$\sigma$</td>
<td>Surface tension, N/m</td>
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Greek Letters

- $\Delta T$: Temperature difference, K
- $\Delta P$: Pressure difference, kPa
- $\mu$: Dynamic viscosity, Ns/m²
- $\nu$: Kinematic viscosity, m²/s
- $\rho$: Density, kg/m³
- $\sigma$: Surface tension, N/m

Subscripts

- $exit$: Exit
- $eq$: Equilibrium
- $expt$: Experimental
- $f$: Fluid
- $g$: Gas

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*Address all correspondence to this author
†Email address: jensem@rpi.edu
inside diameters. Experimental trends showed that heat transfer coefficients increased linearly with increasing qualities except at low mass fluxes and high heat fluxes. Heat transfer coefficients also increased with increasing mass fluxes, heat fluxes, and saturation pressures. Heat transfer coefficients also increased with increasing inlet qualities, and this increase might be due to decreasing liquid film thickness at higher qualities resulting in lower resistance. At low mass fluxes and high heat fluxes, heat transfer coefficients for R134a in the 0.83 mm tube might have decreased with increasing qualities because of partial dryout prevalent at high qualities in small tubes. The authors concluded that annular flow was prevalent in the microtubes. Under similar conditions, R407C had a higher heat transfer coefficient than R134a.

Saitoh et al. [3] studied evaporation heat transfer and pressure drop for flow boiling of R134a in microtubes with internal diameters of 0.51, 1.12, and 3.1 mm. The principle objective of the study was to investigate the effects of diameter on heat transfer coefficients and pressure drops. The parameters studied were heat fluxes ranging from 5 to 39 kW/m², mass fluxes from 150 to 450 kg/m²s, evaporating temperatures of 278 and 288 K, and inlet vapor qualities from 0 to 0.2. Local heat transfer coefficients at lower qualities decreased with decreasing tube diameters. The contribution of convective evaporation to boiling heat transfer rate decreased with decreasing tube diameters. In the 3.1 mm diameter tube, heat transfer coefficients increased with increasing heat fluxes and mass fluxes, but in the smallest tube, heat transfer coefficients were not a function of mass flux. Unstable flow was observed in the 3.1 mm tube but was not present in the 0.51 mm tube. Dryout occurred at lower qualities with decreasing diameters. Heat transfer coefficients increased with increasing saturation pressures, and this effect was more pronounced in the smaller diameter tubes.

Huo at al. [4] undertook a heat transfer and flow visualization study to experimentally investigate flow boiling of R134a in horizontal small circular tubes with internal diameters of 4.26 and 2.01 mm. Nucleate boiling was found to be the dominant heat transfer mechanism for qualities below 40-50% for the 4.26 mm tube and below 20-30% for the 2.01 mm tube. In the 4.26 mm ID tube, dispersed bubble, bubbly, slug, churn, annular, and mist flow patterns were observed. A flow pattern map was also developed. Heat transfer coefficients were found to increase with increasing heat fluxes and saturation pressures. However, heat transfer coefficients were independent of changes in mass flux. Higher heat transfer rates were obtained in the smaller tube.

Kuwahara et al. [5] undertook heat transfer and flow visualization studies on R134a flowing in capillary tubes with internal diameters of 0.84 and 2.0 mm. A glass tube at the exit of the test section was used for flow visualization. Heat transfer coefficients were of similar order for both tubes and, hence, smaller diameters did not result in an enhancement of heat transfer rate. Heat transfer coefficients increased with an increase in quality for the 0.83 mm tube. The authors reported good agreement of

INTRODUCTION

Moore’s Law predicted that the number of transistors that can be placed inexpensively in an integrated circuit will double every two years [1]. This law has held true for the last half century and is expected to be valid in the near future. However, due to increasing packaging density, heat dissipation has also increased significantly. New cooling technologies need to be developed to meet the high heat flux demands. Electronics cooling also places added constraints on the heat transfer fluid and the system size and weight.

Compact heat exchangers utilizing flow boiling of refrigerants show technological promise in dissipating very high heat fluxes. Flow boiling in small size channels result in high heat transfer coefficients. Compared to single-phase flow, there is less variation in surface temperatures. All the above factors make flow boiling at the micro scale a very attractive and viable electronics cooling option.

Heat transfer and fluid flow at the micro scale is considerably different than at conventional scales. The reduced size results in increased surface tension and surface roughness effects. Small conduit size also affects bubble formation and bubble growth, which in turn determines flow morphology and heat transfer mechanisms. These effects are not completely understood.

Most experimental studies have used water or ozone depleting refrigerants as the test fluid. Since water is an electrically conductive fluid, it is not suitable for electronics cooling and the Montreal protocol prohibited the use of refrigerants such as R12, R11, R113, R22, R123, etc. So new refrigerants like R134a were considered. However, few experimental studies have been conducted with R134a and experimental data were limited.

Lie et al. [2] studied flow boiling heat transfer of R134a and R407C in horizontal circular microtubes with 0.83 and 2.0 mm diameters.
their experimental data with the predictions of conventional scale correlations.

Details of experimental studies dealing with flow boiling of water and other fluids in horizontal circular microtubes are available in [6–11]. A few studies [12,13] experimentally investigated the flow boiling characteristics of R134a in MEMS based parallel microchannel heat sinks.

Based on the literature review, it can be concluded that the observed parametric trends are contradictory. Furthermore, experiments with R134a were few and conducted over a limited range of parameters. There is thus a need to conduct experiments over a wide range of parameters to develop an extensive database. The focus of the present study is more on fundamental understanding of flow boiling of R134a. For this reason, a circular microtube was chosen as it is free of extraneous effects such as parallel flow instabilities and conjugate heat transfer.

**EXPERIMENTAL APPARATUS AND PROCEDURES**

**Experimental Facility**

To study the heat transfer process for flow boiling of R134a, a closed flow loop (Fig. 1) was constructed, which comprised of the test section, inlet and outlet plenums and 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 enclosed within 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.

**Test Section Apparatus**

The test sections were made of 120 and 128 mm long stainless steel hypodermic round tubes, with an inner diameter of 0.50 and 1.60 mm respectively.

The test section ends were sanded and polished on a sander machine using 400- and 600- grit sand paper, 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 abso-
lute 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 fixed resistance. 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 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 set 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.

**Data Reduction**

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

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

The electrical power to the test section, $\dot{Q}_{ts}$, is given by

$$\dot{Q}_{ts} = I_{ts}V_{ts}$$ (1)

where $V_{ts}$ is the voltage across the test section and $I_{ts}$ is the current in the circuit, which was determined by measuring the voltage $V_{shunt}$ across the current shunt of fixed resistance.

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 then plotted as a function of the difference between wall temperature and ambient temperature. Two-phase flow generally involved much higher heat fluxes compared to single-phase flow; the heat loss percentage was less in boiling experiments.

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

$$\dot{Q} = \dot{Q}_{ts} - \dot{Q}_{loss}$$ (2)

$$q'' = \frac{\dot{Q}}{(\pi D_i L_h)}$$ (3)

while the mass flow rate, $m$, and the mass flux, $G$, is given by

$$m = \rho V$$ (4)

$$G = \frac{4m}{\pi D_i^2}$$ (5)
The wall superheat, \( \Delta T_{\text{wall}} \), is given by

\[
\Delta T_{\text{wall}} = T_{\text{in},i} - T_{\text{sat}}
\]

where \( T_{\text{sat}} \) is the fluid saturation temperature corresponding to the local pressure \( P_z \) and \( T_{\text{in},i} \) 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 [14]. The model is given by the following equations

\[
P_{\text{in}} = P_{\text{exit}} + \Delta P_{\text{fs}}
\]

\[
\frac{dP}{dz} = \left( \frac{dP_f}{dz} \right)_{t_p} \phi_f^2 + \frac{d}{dz} \left( \frac{G^2(1-x)^2}{\rho f(1-\alpha)} + \frac{G^2x^2}{\rho \alpha} \right) ;
\]  

where \( P_{\text{in}} \) is the pressure at the inlet, \( P_{\text{exit}} \) is the pressure at the exit of the test section, \( \Delta P_{\text{fs}} \) is the pressure drop due to friction and acceleration, respectively.

A uniform heat flux \( q'' \) was applied to the test section and 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, \( H_{\text{in}} \), was determined from the measured inlet temperature and pressure. The enthalpy at different locations was determined by

\[
H_z = H_{\text{in}} + \frac{q''}{m} L_n
\]

The fluid temperature, \( T_{f,z} \), at any \( z \) is similarly calculated using EES with the pressure \( P_z \) and enthalpy \( H_z \) as the independent variables. \( T_{f,z} \) is equal to the fluid saturation temperature when it undergoes saturated flow boiling.

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

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

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 \( z \).

The local heat transfer coefficient, \( h \), at any position \( z \) is given by

\[
h = \frac{q''}{T_{\text{in},i} - T_{f,z}}
\]

Here \( T_{f,z} \) equals \( T_{\text{sat}} \) if the fluid undergoes boiling at that location.

**Uncertainty Analysis**

An uncertainty analysis was conducted using the propagation of error method developed by Kline and McClintock [15]. 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: \( \pm 2.0 - 4.0\% \); heat transfer coefficient: \( \approx \pm 10\% \). Details of data reduction and uncertainty analysis are described in Basu [16].

**EXPERIMENTAL RESULTS**

Extensive experiments were conducted to determine the parametric effects on heat transfer coefficients for flow boiling of R134a in horizontal circular microchannels. The tests were carried out in stainless steel circular microtubes of diameters 0.50 mm and 1.60 mm. The operating conditions were: mass fluxes, 300-1500 kg/m²s; saturation pressures, 490-1160 kPa; inlet subcooling, 5-40 °C; and heat fluxes, 0-350 kW/m². The results obtained are discussed below.

**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 performed. As shown in Fig. 2, the Nusselt number increased with increasing Reynolds number. For Reynolds numbers considered in the study \( (Re \geq 3,000) \), flow
was turbulent. Experimental results showed excellent agreement with the Gnielinski correlation (Fig. 2), which is given by

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

(12)

where, the friction factor \( f \) is given by

\[ f = (0.79ln(Re) - 1.64)^{-2} \]  

(13)

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 [17] 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 [17].

Two-Phase Studies

Boiling curves are plotted for a variety of operating conditions for flow boiling of R134a in two different test sections. Figures 3 and 4 depict the boiling curves for a wide range of mass fluxes and heat fluxes. Higher mass fluxes generally supported higher heat fluxes for the same wall temperature than lower mass fluxes in the single-phase region. Mass flux also affected the onset of nucleate boiling (ONB). ONB occurred at a higher heat flux for higher mass fluxes. However, the wall superheat at ONB remained fairly independent of the mass flux. This indicated that nucleate boiling may be the dominant heat transfer mechanism. The wall temperature was measured using the thermocouple located at an axial location of approximately 60 mm from the inlet to the test section. A similar trend was observed by Bertsch et al. [13] in their study of flow boiling of R134a in microchannels with a hydraulic diameter of 0.54 mm at a saturation pressure of 750 kPa, low mass fluxes of 42-334 kg/m²s, and heat fluxes of 0-30 W/cm².

Heat transfer coefficients are primarily a function of the mass flux, heat flux, saturation pressure, and vapor quality. However, the dependence is highly non-linear, and the interactions between the parameters significantly alter the heat transfer mechanism. Hence, great care needs to be exercised to interpret the results. Heat transfer coefficients were found to increase with increasing heat flux in the saturated region. This indicated that high heat flux augmented the heat transfer process at the liquid-vapor interface, which resulted in more intensive bubble formation and enhanced mixing. This trend was evident in both test sections and at high as well as low mass fluxes. Figures 5 and 6 depict this trend for test sections with internal diameters of 0.50 mm and 1.60 mm, respectively. High heat fluxes also generated a large quantity of vapor for the same mass flux as evident from the high exit vapor qualities.

In the present study, heat transfer coefficients were independent of the mass flux effect. This could be due to the stronger
effect of surface tension at micro scale which made phase separation difficult [18]. This trend was observed in both test sections and is shown in Figs. 7 and 8. For a constant heat flux, lower mass fluxes resulted in higher exit vapor qualities. As has been observed in the boiling curves, onset of boiling occurred much earlier at low mass fluxes than at high mass fluxes. CHF also decreased with decreasing mass flux. For this reason, intermediate heat fluxes were considered for the above figures (Figs. 7 and 8), such that the major portion of the test section was undergoing boiling. At low heat fluxes and very high heat fluxes, ONB and CHF tend to affect heat transfer characteristics.

A close look at the literature indicated that a variety of trends have been observed by previous researchers. Lie et al. [2] found heat transfer coefficients to increase with increasing heat flux in their study with R134a and R407C in horizontal small tubes with diameters of 0.83 mm and 2.0 mm. However, they found heat transfer coefficients to increase with increasing mass flux. Bertsch et al. [13] studied flow boiling of R134a in copper microchannel arrays with hydraulic diameters of 1.09 and 0.54 mm. Heat transfer coefficients were a strong function of the heat flux while mass flux had no effect on the heat transfer rate. Yen et al. [11] also observed no effect of mass flux in their study but observed a decreasing trend in heat transfer coefficients with increasing heat flux. They attributed this to be due to the size of the nucleate bubble being limited by the confined space at high heat fluxes and mass fluxes. Several studies [4, 6, 7, 9, 10] with water and other refrigerants reported negligible effects of mass flux and significant heat flux effects on heat transfer coefficients in the saturated region.

Saturation pressure varied from 490 kPa to 1160 kPa in the present study. The saturation temperature varied from 15 °C to 45 °C. In all test sections, for a fixed $G$ and $q''$, heat transfer coefficients increased with increasing saturation pressures in the saturated boiling regime. As shown in Figs 9 and 10, the effect was not apparent in the subcooled boiling regime. Similar trends were also observed by Lie et al. [2]. They explained this trend to be due to decreasing enthalpy of vaporization with increasing saturation pressures. Due to the decrease in enthalpy of vaporization, rate of evaporation increased, and bubble formation was intensified. This resulted in an increase in heat transfer coefficients with increasing saturation pressures. Saitoh et al. [3] also reported an increase in heat transfer coefficients with an increase in saturation pressure in their study with R134a. However, they found the effect to be more dominant at lower diameters. Similar trends were also observed by Huo et al. [4], Tran et al. [6], Bao et al. [9], and Bertsch et al. [13].

Quality affected the enthalpy and also the relative liquid and vapor mass fluxes. Flow regimes are also defined by the vapor quality. Thus, the effects of vapor quality on heat transfer coefficients are very complex and are dependent on a number of factors, which change with quality. For the smaller tube, vapor quality had a negligible influence on heat transfer coefficients (Fig. 11). Heat transfer coefficients were found to increase slightly with quality in the subcooled regime until $x \approx 0.05$. For
heat transfer coefficients decreased slightly with increasing quality. The initial increase at lower qualities could be due to increased vapor generation and mixing. At higher qualities, increasing volume of vapor could inhibit the heat transfer process, which could result in decreasing heat transfer coefficients. In the test section with an internal diameter of 1.60 mm, a different trend was observed (Fig. 12). Heat transfer coefficients decreased with increasing vapor qualities. In case of the larger tube, all data points were obtained under saturated conditions.

**COMPARISON WITH EXISTING CORRELATIONS**

The experimental data were compared to three microchannel correlations — the Lazarek-Black correlation [19], the Kandlikar correlation [20], and the Tran correlation [6]. The predictive capabilities of the correlations are shown in Figs. 13 through 15 and are quantitatively assessed using the concept of Mean Absolute Error (MAE) which is defined as

$$ MAE = \left( \frac{1}{M} \right) \sum \left| \frac{h_{p,pred} - h_{p,exp}}{h_{p,exp}} \right| \times 100\% $$

where $M$ is the number of data points.

As shown in Fig.13, the Lazarek-Black correlation [19] underpredicted the experimental data for the tube with internal diameter of 1.60 mm and overpredicted the data for the 0.50 mm ID tube. There was considerable scatter in the data for the 1.60 mm ID tube but the correlation predicted the experimental data well, specially at high mass fluxes. On the other hand, there was less scatter in the data for the 0.50 mm ID tube although the correlation did not predict the experimental data accurately. MAE was 56%.

The Kandlikar correlation [20] fitted the test data best with an MAE of 40% (Fig.14). Kandlikar correlation predicted the data for the 0.50 mm ID tube best and there was less scatter. It was not as effective in case of the larger test section. Generally, the Kandlikar correlation predicted the high mass flux data very well but the accuracy deteriorated at low mass fluxes. The correlation failed to capture the high heat transfer coefficient at the onset of boiling and the deteriorating heat transfer rate near dry-out. The experimental data set included a significant amount of data at the onset of boiling (low qualities) and near the CHF condition (high qualities), and this decreased the agreement between the data and predicted values.

The Tran correlation [6] (Fig.15) consistently underpredicted the experimental data for both test sections. There was less scatter in the data. MAE was 44%.

Fig. 16 shows that all the correlations underpredicted the experimental data for the 1.60 mm ID test section. The Lazarek-Black and Tran correlations failed to capture the decrease in heat transfer coefficients with increase in equilibrium vapor quality. Kandlikar correlation did qualitatively predict the decreasing
trend.

It can be concluded that none of the existing correlations considered in the present study predicted the experimental data accurately. MAE was very high and there was considerable scatter. This does not point to the weakness of the correlations as such as some of the correlations were applied beyond the recommended range. The experimental data included conditions near onset of boiling and CHF. None of the correlations captured the unique behavior under those conditions. Furthermore, the correlations were developed for different fluids and operating conditions, which resulted in large errors. However, this study does point out the need to develop new correlations to predict the heat transfer rate for flow boiling of R134a at micro scale.

CONCLUSIONS

An extensive set of experiments were conducted to study the effect of different parameters on heat transfer coefficients for flow boiling of R134a. Based on the experiments, the following conclusions can be drawn:

-Heat transfer coefficients increased almost linearly with an increase in heat flux in the two-phase regime. They also increased with an increase in saturation pressure. However, they were independent of variations in mass flux. Vapor quality had a negligible influence on heat transfer coefficients.

-The above parametric trends indicated that nucleate boiling was the dominant heat transfer mechanism. Nucleate boiling is intensified by increasing heat flux as it enhances bubble formation. The lack of mass flux effect indicated that the contribution of forced convective heat transfer was minimal.

-Three correlations (Lazarek-Black, Kandlikar, and Tran) were compared with the experimental heat transfer coefficient data. None of the correlations predicted the heat transfer data with a high degree of accuracy. Within limits of experimental error, the correlations captured the correct heat transfer coefficient versus quality trend. There is thus a need to develop better correlations to predict heat transfer coefficients for flow boiling of R134a at micro scale.

ACKNOWLEDGMENT

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Figure 10. Effect of saturation pressure on heat transfer coefficients for flow boiling of R134a for $d_{in} = 1.60$ mm, $G = 1000$ $kg/m^2s$, $q'' = 150$ $kW/m^2$, and $\Delta T_{subcooling} = 20^\circ C$.

Figure 11. Effect of equilibrium vapor quality on heat transfer coefficients for flow boiling of R134a for $d_{in} = 0.50$ mm, $P_{sat} = 1160$ kPa, and $\Delta T_{subcooling} = 5^\circ C$.

Figure 12. Effect of equilibrium vapor quality on heat transfer coefficients for flow boiling of R134a for $d_{in} = 1.60$ mm, $P_{sat} = 1160$ kPa, and $\Delta T_{subcooling} = 5^\circ C$.

Figure 13. Comparison of the experimental data to the Lazarek-Black correlation.
Figure 14. Comparison of the experimental data to the Kandlikar correlation

Figure 15. Comparison of the experimental data to the Tran correlation

Figure 16. Comparison of R134a heat transfer coefficient data with the correlations for $d = 1.60$ mm, $q'' = 120$ kW/m$^2$, $P_{sat} = 1160$ kPa, $G = 1000$ kg/m$^2$s, and $\Delta T_{subcooling} = 5$ °C
REFERENCES


