Boiling Heat Transfer of Refrigerant R-113 in a Small-Diameter, Horizontal Tube

by

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<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Channel flow area (m²)</td>
</tr>
<tr>
<td>B₀</td>
<td>Boiling number, ( \frac{q''}{G_i} )</td>
</tr>
<tr>
<td>( c_p )</td>
<td>Specific heat of fluid (W·s/kg·°C)</td>
</tr>
<tr>
<td>D</td>
<td>Tube inside diameter (m)</td>
</tr>
<tr>
<td>( D_o )</td>
<td>Tube outside diameter (m)</td>
</tr>
<tr>
<td>Dp</td>
<td>Differential pressure (kPa)</td>
</tr>
<tr>
<td>E</td>
<td>Electric power input (V)</td>
</tr>
<tr>
<td>G</td>
<td>Mass flux (kg/m²s)</td>
</tr>
<tr>
<td>h</td>
<td>Local heat transfer coefficient (W/m²·°C); Eq. 1</td>
</tr>
<tr>
<td>I</td>
<td>Electric current through test channel (A)</td>
</tr>
<tr>
<td>( i_{fg} )</td>
<td>Latent heat of evaporation (W·s/kg)</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity (W/m·°C)</td>
</tr>
<tr>
<td>( k_{ss} )</td>
<td>Thermal conductivity of channel material (W/m·°C)</td>
</tr>
<tr>
<td>( L_H )</td>
<td>Heated length (m)</td>
</tr>
<tr>
<td>( L_{sc} )</td>
<td>Subcooled length (m); Eq. 7</td>
</tr>
<tr>
<td>M</td>
<td>Molecular weight (Steiner and Taborek correlation, Table 2)</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number (= ( h/Dk ))</td>
</tr>
<tr>
<td>P</td>
<td>Pressure</td>
</tr>
<tr>
<td>( P_{cr} )</td>
<td>Critical pressure</td>
</tr>
<tr>
<td>( P_r )</td>
<td>Pressure ratio (= ( P/P_{cr} ))</td>
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<tr>
<td>Pr</td>
<td>Prandtl number (= ( c_p \mu/k ))</td>
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<tr>
<td>( p_{in} )</td>
<td>Pressure at inlet (kPa)</td>
</tr>
<tr>
<td>( p_{sat-sc} )</td>
<td>Saturation pressure at the start of bulk boiling (kPa)</td>
</tr>
<tr>
<td>( p_{sat-out} )</td>
<td>Saturation pressure at outlet (kPa); Eq. 9</td>
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<td>q''</td>
<td>Input heat flux (W/m²)</td>
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\( q_E \) \hspace{1em} \text{Input heat flux calculated from electric power input (W/m}^2\); Eq. 5

\( q_T \) \hspace{1em} \text{Heat transfer rate (W)}

\( q_f \) \hspace{1em} \text{Input heat flux calculated from enthalpy increase of fluid over heated length (W/m}^2\); Eq. 2

\( R_a \) \hspace{1em} \text{Surface roughness (Steiner and Taborek Correlation, Table 2)}

\( \text{Re} \) \hspace{1em} \text{Reynolds number (= GD/} \mu \text{)}

\( T_f \) \hspace{1em} \text{Fluid bulk temperature (°C)}

\( T_{in} \) \hspace{1em} \text{Fluid inlet temperature (°C)}

\( T_{out} \) \hspace{1em} \text{Fluid outlet temperature (°C)}

\( T_{sat} \) \hspace{1em} \text{Saturation temperature at start of bulk boiling (°C); Eq. 6}

\( T_w \) \hspace{1em} \text{Inside wall temperature (°C); Eq. 3}

\( T_w' \) \hspace{1em} \text{Outside (measured) wall temperature (°C)}

\( v_G \) \hspace{1em} \text{Specific volume of vapor (m}^3/\text{kg)}

\( X_{tt} \) \hspace{1em} \text{Martinelli parameter}

\( \xi \) \hspace{1em} \text{Equilibrium mass quality; Eq. 10}

\( z \) \hspace{1em} \text{Distance along channel (m)}

\( \eta \) \hspace{1em} \text{Heat loss factor; Eq. 4}

\( \mu \) \hspace{1em} \text{Liquid viscosity (kg/m} \cdot \text{s)}

\( \rho \) \hspace{1em} \text{Liquid mass density (kg/m}^3\)

\( \sigma \) \hspace{1em} \text{Surface tension (N/m)}

\text{Subscripts}

\( g \) \hspace{1em} \text{Gas}

\( l, L \) \hspace{1em} \text{Liquid}

\( ss \) \hspace{1em} \text{Stainless steel}

\( TP \) \hspace{1em} \text{Two-phase condition}

\( z \) \hspace{1em} \text{At location} \ z \ \text{along channel length}
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M. W. Wambsganss, D. M. France, J. A. Jendrzejczyk, and T. N. Tran

Abstract

Results of a study of boiling heat transfer from refrigerant R-113 in a small-diameter (2.92-mm) tube are reported. Local heat transfer coefficients over a range of heat fluxes, mass fluxes, and equilibrium mass qualities were measured. The measured coefficients were used to evaluate eight different heat transfer correlations, some of which have been developed specifically for refrigerants. High heat fluxes and low flow rates are inherent in small channels, and this combination results in high boiling numbers. The high boiling number of the collected data shows that the nucleation mechanism was dominant. As a result, the two-phase correlations that predicted this dominance also predicted the data best if they also properly modeled the physical parameters. The correlations of Lazarek and Black (1982) and of Shah (1976), as modified in this study, predicted the data very well. It is also shown that a simple form, suggested by Stephan and Abdelsalam (1980) for nucleate boiling, correlates the data equally well. This study is part of a research program in multiphase flow and heat transfer, with the overall objective of developing validated design correlations and predictive methods that will facilitate the design and optimization of compact heat exchangers for use with environmentally acceptable alternatives for chlorofluorocarbon (CFC) refrigerants and refrigerant mixtures.

1 Introduction

Fundamental multiphase-flow and heat transfer issues, associated with the boiling and condensing of pure refrigerants and refrigerant mixtures in channels representative of compact heat exchanger flow passages, are being addressed as a part of the U.S. Department of Energy’s Thermal Sciences Research Program. Research is focused on flow patterns, pressure drop, and phase-change heat transfer. The overall objective is to develop validated design correlations and predictive methods that will facilitate the design and optimization of compact heat exchangers for use with environmentally acceptable alternatives for chlorofluorocarbon (CFC) refrigerants and refrigerant mixtures.
Several features make compact heat exchangers attractive and suggest their use in applications where energy conservation, space saving, and cost are important considerations. These features include high thermal effectiveness (ratio of actual heat transferred to the theoretical maximum that can be transferred), large heat transfer surface per volume ratio (surface area density, typically >700 m²/m³), low weight per heat-transfer duty, opportunity for true counterflow operation, close temperature approach (as a result of the ability to design for true counterflow), design flexibility, and reduced fluid inventory. Flow maldistribution and the design of headers to minimize maldistribution remain inherent problems in the application of such heat exchangers, especially in the case of phase-change heat transfer. The potential for fouling of the small flow passages represents a major disadvantage in the use of compact heat exchangers. However, fouling should not be a problem in applications that involve clean fluids, such as refrigerants.

The focus of this research is on compact heat exchangers of the plate-fin type. A plate-fin heat exchanger consists of alternate rows of small, typically noncircular, flow passages. The flow passages are usually formed of sheets of corrugated finning, which are sandwiched between a stack of flat plates referred to as parting plates. The assembly is typically brazed in a furnace to form a matrix or core. Heat is exchanged between fluid streams flowing through the flow passages between the parting plates. The corrugations provide the "fins," or secondary heat transfer surfaces.

The state of the art of boiling heat transfer in plate-fin heat exchangers was summarized most recently by Thome (1990), who notes that more widespread application of plate-fin heat exchangers is inhibited by the limited access to well-proven design technology for boiling in these geometries. He calls attention to the fundamental importance of flow patterns in understanding the physics of the heat transfer process, and the concurrent need for more research on two-phase flow patterns and the corresponding two-phase pressure drops. He notes that there have been surprisingly few experimental studies of flow boiling and that the available predictive methods have not been widely tested against independent sets of data. Finally, he suggests that the effects of mixtures on heat transfer must be studied to expand the range of applications of plate-fin heat exchanger designs.

Based on a review of the state of the art, the identified research needs are in the areas of flow patterns, pressure drop, and phase-change (boiling and condensing) heat transfer of single-component fluids and mixtures. A research plan to address these needs was developed and is being implemented. The research plan includes adiabatic two-phase flow experiments with gas/liquid
mixtures, phase-change heat transfer experiments, and experiments on prototypical heat exchanger cores.

In planning a research program that addresses the fundamental issues associated with two-phase fluid dynamics and heat transfer, one is immediately faced with the fact that the multichannel arrangements that are characteristic of plate-fin heat exchangers are known to result in flow maldistribution among the channels. Moreover, the flow passages can vary greatly in geometry and size, with equivalent hydraulic diameters ranging from <1 mm and up to 5 mm, and not only triangular and rectangular geometries, but other more complex variations of these basic shapes as well. Both size and shape can be expected to affect the two-phase flow and heat transfer processes.

To circumvent potential flow maldistribution problems that are inherent in multichannel arrangements, it was decided to first study individual channels. Geometries with both rectangular and circular cross sections and with channel hydraulic diameters of <5.5 mm were studied.

Initial efforts focused on adiabatic-flow experiments with gas/liquid mixtures in horizontal, rectangular channels. Experiments with air/water mixtures in rectangular channels with dimensions of 19.1 x 3.18 mm and 9.52 x 1.59 mm have been completed. The results from the 19.1 x 3.18-mm channel, including data analysis and the development of flow pattern maps and a pressure drop correlation, have been reported by Wambsganss et al. (1990a, 1990b, 1991, 1992a); results from the 9.52 x 1.59-mm channel are reported by Wambsganss et al. (1992b). In addition, Obot et al. (1991) have used the experimental data from the 19.1 x 3.18-mm channel to develop a generalized method for correlating the frictional pressure drop data for adiabatic two-phase flow. Future plans call for experiments with mixtures of air and the refrigerant R-113 to investigate the effects of fluid properties.

This report discusses the preliminary heat transfer experiments that involved the evaporation of R-113 in a small-diameter, horizontal tube. R-113 is a useful research fluid, and, as such, was selected as the fluid for use in the initial flow boiling experiments. It is a useful research fluid because it has a relatively high saturation temperature (47.6°C) at atmospheric pressure, compared to that of many other refrigerants. This allows testing at relatively low system pressures with a fluid that still exhibits properties similar to those of many other refrigerants. Future plans call for studies that involve CFC-alternative refrigerants, e.g., R-134a and refrigerant mixtures.
A review of the literature revealed that evaporation of R-113 flowing inside horizontal, smooth tubes, with inside diameters of 8.71 and 10.92 mm, was recently studied and reported by Reid et al. (1987). The objectives of their study were (1) to obtain a carefully controlled set of flow boiling data for R-113, and (2) to evaluate state-of-the-art correlations for predicting evaporative heat transfer coefficients. They identified six flow boiling heat transfer correlations for comparison with their data. These commonly used correlations include five that were developed from refrigerant data bases and the Chen (1966) correlation whose format was followed by many of the others. In evaluating their correlation against their data, Reid et al. (1987) concluded that no one correlation predicted the data over the entire range of experimental parameters, although several were in reasonable agreement.

A second study, which is closely related to the present investigation, was reported by Lazarek and Black (1982). Their study, motivated by the need to develop improved methods for electronic cooling, was conducted with R-113 as the boiling fluid and focused on evaporation of R-113 in a vertical tube with a diameter of 3.15 mm. The flow direction was both up and down, which is different from the present case, but, of all the studies found in the literature, the channel diameter was the closest to that in this study.

The R-113 flow boiling data of the present investigation, from a 2.92-mm circular tube, were compared with the R-113 results of Reid et al. (1987) and Lazarek and Black (1982). The use of the same refrigerant in all three studies eliminated the fluid variable from the comparisons. The present data extend the results of Reid et al. (1987) from the 9-11-mm range of tube diameters to the 3-mm range, with similar parameters of heat flux and mass flux and quality, and a horizontal tube orientation. The work of Lazarek and Black (1982) used approximately the same tube diameter as the present study, and the present data serve to extend those results from vertical to horizontal orientation.

Data on saturated flow boiling upstream of the critical heat flux (CHF) have been correlated according to a superposition principle first introduced by Chen (1966). The physical mechanisms of bubble nucleation in a liquid on the heated surface and forced convection of the liquid have been modeled by many researchers. Either mechanism can dominate the heat transfer process, depending on flow parameters (e.g., mass flux and quality) and heat flux. An objective of this investigation was to determine the characteristics of the boiling data on small horizontal channels with respect to these mechanisms and to evaluate the correlations available in the literature for the prediction of heat transfer coefficients. To this end, several correlations were chosen for comparison with the present data. Most of these correlations were developed from refrigerant data bases; some have appeared frequently in the engineering
literature; some are relatively new. The original Chen (1966) correlation was included with some form of the other five considered by Reid et al. (1987), as was the correlation proposed by Lazarek and Black (1982), which contained a data base closest to that of the present study. The recent correlation of Jung and Radermacher (1991) was included because it was developed from a data base of numerous refrigerants. In addition, the recent correlation of Steiner and Taborek (1992) was evaluated because of its asymptotic behavior and its extensive data base.

2 Experimental Apparatus and Test Channel

The heat transfer test apparatus is shown schematically in Fig. 1. It is a closed-loop system that includes a gear pump with variable-speed drive, a set of rotameters, a preheater, condenser, accumulator, and sight-glasses. The bladder-type accumulator allows for stable control of system pressure. The rotameters were sized to cover the range of flow rates expected in testing small channel flows, and calibrated by a weighing-with-stop-watch technique. Equations were developed to fit the calibration data. The estimated uncertainty in flow rate measurement is ±3%. Provisions were made to measure temperature and pressure at various locations, as indicated in Fig. 1. One sight glass was placed before the test channel, another after the condenser, to verify single-phase flow at these locations.

The test channel is shown schematically in Fig. 2. It is a Type 304 stainless steel tube, with a heated length of 368 mm, and outside and inside diameters of 4.75 and 2.92 mm, respectively. Inlet pressure (p_in) is measured with a piezoresistive-type transducer (Endevco Model 8510B-50). Differential pressure across the channel (Dp) is measured with a strain-gauge-type transducer (Viatran Model 209). The pressure transducers were calibrated against a known standard. The estimated uncertainty in the pressure measurements is ±5%. The temperature of the liquid at the inlet to the test channel (T_in) is measured with a chromel-constantan wall thermocouple, located 38.1 mm upstream of the start of the heated length, and mounted with electrically insulating, thermally conductive epoxy. The bulk temperature of the fluid at the outlet (T_out) is measured with an in-stream, chromel-constantan sheathed thermocouple probe, positioned at the exit from the heated length. Temperatures at the thermocouples on the test channel wall were compared to those at the sheathed stream thermocouple under isothermal conditions. The stream thermocouple and the system pressure port were located together, and pressure and temperature were compared at saturation conditions. From these comparison tests and with the low amount of electronic noise after signal averaging, it is estimated that the temperature measurements are accurate to ±0.2°C.
As illustrated in Fig. 2, the test channel was uniformly heated by passing direct electrical current through the channel wall. Wall temperature was measured at four locations along the channel length (98, 175, 261, and 337 mm, measured from the start of the heated length at the inlet, corresponding to L/D = 34, 60, 89, and 115, respectively) with four pairs of chromel-constantan thermocouples, each pair oriented to measure temperature at the top and bottom of the channel and attached to the surface with electrically insulating, thermally conductive epoxy. Total electrical power supplied to the test channel was determined from a measurement of the overall voltage (E) across the channel and the current (I), as determined from the measured voltage across a calibrated resistance installed in parallel with the test channel.
Fig. 2. Test channel, showing thermocouple and pressure tap locations

Measurements were recorded with a Hewlett-Packard (HP) data acquisition system (DAS) consisting of an HP Vectra micro-computer and a Model 3421 multiplexor. After steady state was achieved, as indicated by analog recordings of stream and wall temperatures, all sensor-output voltages were read by the DAS 30 times and averaged. The process took ~3 min. As an additional check of steady state, the data were averaged in three groups of 10 data scans each. Consistency was checked before all 30 scans were averaged together. The final results of 50 averages were changed to engineering units and stored on computer disk for future processing.

3 Test Procedure and Data Reduction

In the data reduction discussed below, the pertinent fluid properties were obtained from the ASHRAE Handbook (1989). Curve-fitting techniques were used to obtain equations for the fluid properties of interest as a function of temperature or pressure over the range of interest for the experiments. In particular, equations were developed for the following fluid properties: liquid thermal conductivity $k_l$, liquid specific heat $c_{pl}$, liquid viscosity $\mu_l$, density of liquid $\rho_l$, specific volume of vapor $v_G$, saturation pressure $p_{sat}$, and latent heat of
evaporation $i_{fg}$. These equations were used in the computer analysis of the data subsequent to the experiments.

3.1 Single-Phase Heat Transfer

Single-phase tests were performed at mass fluxes ranging from 50 to 2200 kg/m$^2$s, selected to cover a Reynolds number range (236-8700) from laminar to turbulent flow. Once a desired mass flux was established in the test section, electric power to the section was increased to a value at which the temperature increase of the liquid across the test section was $\approx15^\circ$C. This temperature range was large enough to minimize the magnitude of any thermocouple error in the data. The temperature of the heated wall was maintained below saturation to ensure avoidance of boiling. The test section was allowed to reach equilibrium and data were recorded, each data point consisting of 30 averaged measurements.

The local heat transfer coefficient, at position $z$ along the length of the tube, is defined as

$$h_z = \frac{q_T}{(T_{wz} - T_{fz})},$$

(1)

where the input heat flux $q_T$ was obtained from the enthalpy increase of the fluid over the heated length from

$$q_T = \frac{(GAC_p)(T_{out} - T_{in})}{\pi DL_H}.$$  

(2)

For a given mass velocity, wall temperature data were obtained at four axial positions along the length of the test channel as indicated in Fig. 2. Temperatures from the top and bottom of the channel were then averaged to give an average outside-wall temperature $T_{wz}'$ at each axial position $z$. Inside-wall temperatures were calculated, accounting for heat generation within and conduction through the wall, from

$$T_{wz} = T_{wz}' + \left(\frac{q_T}{4\pi k_{ss}L_H}\right)\left[\frac{\xi(1-\ell n \xi)}{1-\xi}\right],$$

(3)

where $\xi = (D_0/D)^2$. The corresponding bulk fluid temperature $T_{fz}$ at each axial position $z$ was determined by interpolation, assuming a linear temperature gradient in the bulk fluid over the heated length.
The single-phase heat transfer results were compared with standard correlation equations to establish the validity of the measurements and the data reduction method. For this purpose, log-log plots of Nusselt number versus Reynolds number were compared to the correlations. The results are presented in the Single-Phase Heat Transfer part of the Results and Analysis section.

Heat loss from the test channel, including end losses, under flow conditions was determined as the ratio of the heat flux found from the liquid enthalpy change (Eq. 2) divided by the heat flux calculated from the electric power input, i.e., from

\[ \eta = \frac{q^*}{q_E} \]  

(4)

where

\[ q_E = \frac{EI}{\pi DL_H} \]  

(5)

The percentage heat loss \( \eta \) was determined as a function of temperature from these single-phase tests, and it was subsequently used in determining the input heat flux for the flow boiling experiments.

3.2 Flow Boiling Heat Transfer

As with the single-phase heat transfer tests, the flow boiling tests were performed at selected values of mass velocity (in this case, 50, 100, 150, 200, 242, and 300 kg/m$^2$s). The preheater was used to vary the inlet temperature between \( \pm 20 \) and 50°C. The fluid entered the test section as subcooled liquid in all tests. System pressure was approximately constant in a given test, ranging from 140 to 180 kPa. The electric power to the test section was set for a particular test to achieve a desired outlet quality or to maintain a prescribed heat flux. The experimental heat flux ranged from 8.8 to 90.75 kW/m$^2$, which was calculated by multiplying the measured electrical input \( q_E \) by the heat loss factor \( \eta \).

The other data included averaged wall temperatures \( T_{wz} \) at four axial locations, inlet (wall) temperature \( T_{in} \), fluid pressure \( p_{in} \) just upstream of the start of the heated length, and bulk fluid temperature at the exit of the heated length \( T_{out} \). Equation 3 was used to calculate inside-wall temperatures \( T_{wz} \).
In the data analysis, thermal equilibrium of the vapor and liquid phases was assumed along the entire length of the channel. The length of the subcooled inlet region was determined by iteration from

\[ L_{sc} = \frac{GAC_p(T_{sat} - T_{in})}{\pi D_q T} \]  

with the initial guess that the pressure drop over the subcooled length was negligible, so that \( T_{sat} \) was based on inlet pressure \( p_{in} \). Subsequent guesses accounted for the liquid pressure drop by using the subcooled length determined from the previous calculation in the iterative process. The fluid temperature at the start of flow boiling \( T_{sat} \) was then calculated from the pressure at the end of the subcooled region, \( p_{sat-sc} \), assuming no subcooled boiling. The fluid exited the test section with a quality \(<1\) in all tests, so that the saturation pressure at the exit from the heated length was calculated from the bulk fluid temperature measured there. It was assumed that the pressure gradient over the two-phase region was linear. This allowed the use of linear interpolation for estimation of the fluid saturation pressure at the four measurement locations. Assuming thermodynamic equilibrium, the bulk fluid temperatures at the measurement locations could then be readily calculated from the saturated vapor/liquid relationship. The total pressure drop in the test section was small (\(<40\, \text{kPa}\) in all tests), which produced typical test section saturation temperature changes of 5-6°C. A deviation of 10% from the linear assumption typically translates to \(<5\%\) error in the heat transfer coefficient.

With knowledge of the input heat flux, inside-wall temperatures, and corresponding bulk fluid temperatures, local heat transfer coefficients were calculated from Eq. 1. The mass quality \( x \) at measurement location \( z \) was calculated as

\[ x(z) = \frac{\pi D(z - L_{sc})q_T}{AG_{ifg}} \]  

Local heat transfer coefficients were then presented as a function of mass quality for given values of mass flux (velocity) and heat flux. Results are presented in the Flow Boiling Heat Transfer part of the Results and Analysis section.
4 Results and Analysis

4.1 Single-Phase Heat Transfer

Results from the single-phase heat transfer tests are given in Fig. 3, as a plot of Nusselt number versus Reynolds number. A laminar, transition, and turbulent regime are identifiable. As determined from application of Eq. 4, total heat loss was \( \approx 11\% \), for the higher values of heat flux \( q_T > 16,000 \text{ W/m}^2 \).

For laminar flow in a circular tube with fully developed velocity and temperature profiles, the analytical solution (Kays and London, 1984) for the constant heat flux boundary condition is \( \text{Nu} = 4.36 \). A comparison with the results shown in Fig. 3 reveals that the measured Nusselt values are high, averaging approximately \( \text{Nu} = 9 \) over the laminar range, compared with the theoretical value of 4.36. This deviation from the analytical result can readily be explained by the fact that the velocity and temperature profiles are not fully developed over the length of the channel. Because the channel was designed for flow boiling experiments, the channel length is not sufficiently long to allow development of the profiles and, further, no attempt was made to streamline the entrance. For such a channel, one would expect the experimental results to be higher than the analytical results, and that is the case.

![Fig. 3. Results from single-phase heat transfer tests (•) and predicted values based on the Petukhov-Popov correlation (—).](image)
In the turbulent regime, Fig. 3 shows that the experimental data are asymptotically approaching the straight-line curve representing the Petukhov-Popov correlation (Petukhov, 1970). The good agreement with the Petukhov-Popov correlation, as the Reynolds number is increased from its critical value at transition, provides confidence in the experimental apparatus, instrumentation, and data acquisition and analysis techniques.

4.2 Flow Boiling Heat Transfer

The experimental runs are summarized in Table 1. For each run, Eqs. 1 and 7 were used to calculate the evaporative heat transfer coefficients and corresponding mass qualities, respectively, at the various thermocouple measurement locations by following the procedure outlined in Section 3.1. A typical result is presented in Fig. 4, which illustrates the calculational scheme.

Results from a representative sample of the experimental runs, corresponding to a range of mass and heat fluxes, are given in Fig. 5. The data were obtained for a mass quality range from 0 to ~0.9. The heat transfer coefficients for this data set, as determined by the procedure outlined in Section 3.2, ranged from ~1,000 to 6,500 W/m²s. These results are in qualitative agreement with the measurements of Reid et al. (1987) for the evaporation of R-113 in smooth tubes, 8.71 and 10.92 mm in diameter.

<table>
<thead>
<tr>
<th>G (kg/m²s)</th>
<th>( q''_T ) (kW/m²)</th>
<th>G (kg/m²s)</th>
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<td>50</td>
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<tr>
<td></td>
<td>21.5</td>
<td></td>
<td>46.3</td>
</tr>
<tr>
<td></td>
<td>25.5</td>
<td></td>
<td>55.4</td>
</tr>
<tr>
<td></td>
<td>30.0</td>
<td></td>
<td>74.9</td>
</tr>
<tr>
<td>150</td>
<td>34.8</td>
<td>300</td>
<td>44.7</td>
</tr>
<tr>
<td></td>
<td>44.4</td>
<td></td>
<td>63.3</td>
</tr>
</tbody>
</table>

Table 1. Experimental runs
Fig. 4. Typical results illustrating calculational scheme for determining saturation temperatures and pressures; experimental run: \( G = 200 \text{kg/m}^2\text{s}, q''_T = 41,766 \text{W/m}^2 \)
The bulk boiling region of the test section can be dominated by either the nucleate or convective two-phase component, or both components can contribute to the heat transfer. All three conditions have been found by various investigators under a variety of system parameters. The present small-channel data were analyzed to determine the dominant mechanisms. All of the data are shown in Fig. 6, where mass flux is constant in each part of the figure. It is clear that the heat transfer coefficient is a function of heat flux over most of the quality region. This is indicative of the nucleate boiling mechanism. The data tend to converge to produce a heat flux independence, which is indicative of convective-dominant boiling.

When interpreting and correlating data, it is often important to be knowledgeable about two-phase flow patterns. Thus, flow pattern identification is included in Fig. 6. The transition boundaries were obtained from the flow pattern map developed by Damianides and Westwater (1988) for a 3-mm-diameter tube based on adiabatic tests with air-water mixtures. We recognize that the fluid
Fig. 6. Measured heat transfer coefficients as a function of mass quality for heat fluxes and mass fluxes of (a) $G = 50 \text{ kg/m}^2\text{s}$, (b) $G = 100 \text{ kg/m}^2\text{s}$, (c) $G = 150 \text{ kg/m}^2\text{s}$, (d) $G = 200 \text{ kg/m}^2\text{s}$, (e) $G = 250 \text{ kg/m}^2\text{s}$, and (f) $G = 300 \text{ kg/m}^2\text{s}$; $q_T = W/m^2$ (------ extrapolated behavior)
properties of air/water mixtures are different from Freon vapor/liquid mixtures, and that surface tension, as well as vapor/liquid density ratio, can be expected to influence flow patterns. However, the channel geometry studied by Damianides and Westwater (1988) was quite similar to the present case, and the comparison of the present data to the flow patterns in Fig. 6 is instructive.

Wambsganss et al. (1990a, 1990b, 1991) showed that for small rectangular channels, the transition from plug/bubble to slug flow is evident in measured two-phase pressure drop data as a well-defined local peak or inflection point in the curve of the normalized two-phase pressure gradient that is plotted as a function of the mass quality or Martinelli parameter. However, no similar indication is observed in the heat transfer coefficient data given in Fig. 6. Most of the data fall in the plug and slug flow pattern regions, with only a few measurements falling in the annular or wave flow pattern regions. Nucleation in the thick liquid layers of the plug and slug flow patterns is indicated by the heat flux dependence of the data. The question of nucleation suppression in the thin liquid films of annular flow is not a consideration for these data. However, there is an indication that the data tend toward heat flux independence as quality increases at each mass flux of Fig. 6 (see dashed-line asymptotes sketched on the figure). This trend shows a move toward convection-dominant heat transfer as the annular flow pattern is approached, but it does not appear to have been reached in the data obtained. The large slug flow pattern region that exists in the smaller channels of this study, compared with larger channels, is one important parameter leading to the dominance of the nucleation mechanism. The high boiling number is the second parameter, and it will be discussed further in the following sections.

In studying the effect of heat flux, it can be observed that, at the lowest mass flux of 50 kg/m²s (see Fig. 6a), the heat transfer coefficient is weakly dependent on heat flux for the range of heat fluxes tested (8,800 to 16,522 W/m²). However, at the higher mass fluxes, the heat transfer coefficient, in the mass quality range up to =0.8, is strongly dependent on, and monotonically increases with, increasing heat flux. Due to experimental limitations, high qualities of =0.8 were not obtainable at all flows and heat fluxes. In addition, the data were limited to approaching critical heat flux (CHF) at x = 0.8. In general, the flow pattern goes into annular flow at this upper limit of the data (x = 0.8), and there is the possibility of nucleation suppression in the thin liquid films. However, it is entirely possible to reach the CHF without experiencing nucleation suppression.

The present data show nucleation dominance at all heat fluxes. There is an indication that nucleation will become suppressed and the convection heat transfer mechanism will dominate at higher qualities than the data within the plug and slug flow pattern regions. There is also an indication that convection will dominate in the annular pattern above =0.8 quality. However, the data that
were obtained were essentially all in the nucleation-dominant region and only an indication was obtained of movement towards convection domination at higher qualities.

Only limited data sets with approximately the same input heat flux and varying mass flux were taken. One such data set, taken at a heat flux of $\approx 26,000$ W/m$^2$, and three different values of mass flux (100, 200, and 242 kg/m$^2$s) is shown in Fig. 7. The small-diameter tube of this study raises a question as to how efficient the nucleation mechanism will remain. The results of Fig. 7 show that the small-tube data behave as larger tube data in that they are essentially independent of mass flux in the nucleation-dominant heat transfer region.

5 Evaporation Heat Transfer Correlations

Reid et al. (1987) identified six in-tube evaporation heat transfer correlations that are commonly associated with refrigerants: the correlations were found in Chen (1966); Chaddock-Brunemann (see Reid et al., 1987); Pujol-Stenning (1969);

![Diagram](image)

**Fig. 7.** Heat transfer coefficient as a function of mass quality for approximately constant heat flux and variable mass flux; $q^*_T = W/m^2$, $G = kg/m^2s$
Shah (1976); Kandlikar (1983); and Gungor-Winterton (1986). Reid et al. evaluated the six correlations with data they obtained from evaporation of R-113 (the same refrigerant used in this study) in tubes with diameters of 8.71 and 10.92 mm. They concluded that several of the correlations showed reasonable agreement with the data at qualities above 0.2. For the 10.92-mm tube, five correlations (the Chen (1966) correlation being the exception) were comparable, with mean deviations ranging from 22 to 33%. For the 8.71-mm tube, the Gungor-Winterton (1986) correlation had the lowest mean deviation, 9.6%. However, the Chaddock-Brunemann (see Reid et al., 1987) and Kandlikar (1983) correlations were close, with mean deviations of 12.6 and 13.5%, respectively. These correlations were also compared with the small-tube data of the present study. However, the original Gungor-Winterton (1986) correlation used by Reid et al. (1987) was replaced by a newer version due to Liu and Winterton (1990). The original correlation did not predict the present data well, in contrast to the good performance with the data of Reid et al. (1987). There are physical reasons for the poor performance of the Gungor and Winterton (1986) correlation, and these will be discussed subsequently. The correlations given in Table 2 were used as developed by the original authors, and turbulent single-phase heat transfer correlations were used as specified, even if the Reynolds number indicated laminar flow. It should be noted, however, that $Re$ is based on the superficial vapor velocity in the test channel, and flow patterns such as slug and annular can be in the turbulent regime at low values of such Reynolds numbers.

The six in-tube evaporation correlations evaluated by Reid et al. (1987), with the Gungor-Winterton (1986) correlation replaced by a more recent version (Liu and Winterton, 1990), are included in Table 2. These correlations were evaluated with data from this investigation. The results are given in Figs. 8-11. A statistical comparison of the correlations, based on the mean deviation and percentage of data points predicted within ±20%, is given in Table 3. In Table 3, the various correlations are ranked, with the mean deviation as the measure of accuracy. This parameter allows direct comparison of results with previous studies.

The Chen (1966) correlation was the first to use the superposition principle of nucleate- and convection-dominated heat transfer. It is widely used with boiling water and was included here because it is widely used and serves here as a benchmark. The results of all of the data of the present study are compared with this correlation in Fig. 8. Considerable scatter is present in the plot, but the predictions are well centered in the data.

The predictions of both the correlations of Chaddock and Brunemann (see Reid et al., 1987) and Pujol and Stenning (1969) underpredicted the data significantly, with the Chaddock and Brunemann correlation the better performing of the two, as shown in Fig. 9 and Table 3.
<table>
<thead>
<tr>
<th>Source</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chen (1966)</td>
<td>$h_{TP} = h_{conv} + h_{NB}$</td>
</tr>
<tr>
<td></td>
<td>$h_{TP} = F h_\ell + S h_{pool}$,</td>
</tr>
<tr>
<td></td>
<td>where</td>
</tr>
<tr>
<td></td>
<td>$F = 2.35 \left( \frac{1}{X_{tt}} + 0.213 \right)^{0.736}$, for $\frac{1}{X_{tt}} &gt; 0.1$</td>
</tr>
<tr>
<td></td>
<td>$h_{pool} = 0.00122 \left( \frac{k_{\ell}}{\mu_{\ell}} \frac{c_{p,\ell}}{\rho_{\ell}} \frac{g_c}{\Delta T_x} \Delta \rho_{sat} \right)^{0.24} \Delta x^{0.75}$</td>
</tr>
<tr>
<td></td>
<td>$S = \left( 1 + 0.12 \text{Re}<em>{TP}^{1.14} \right)^{-1}$, for $\text{Re}</em>{TP} &lt; 32.5$</td>
</tr>
<tr>
<td></td>
<td>$\text{Re}<em>{TP} = \frac{G(1-x)D}{\mu</em>{\ell}} (F^{1.25})(10^{-4})$</td>
</tr>
<tr>
<td>Chaddock and Brunemann (1967)</td>
<td>$h_{TP} = 1.91 \left[ (B_0 \times 10^4) + 1.5 \left( \frac{1}{X_{tt}} \right)^{0.67} \right]^{0.6} h_\ell$</td>
</tr>
<tr>
<td>Pujol and Stenning (1969)</td>
<td>$h_{TP} = 4.0 \left( \frac{1}{X_{tt}} \right)^{0.37} h_\ell$</td>
</tr>
<tr>
<td>Shah (1976)</td>
<td>$h_{TP} = \psi h_\ell$</td>
</tr>
<tr>
<td></td>
<td>$Fr_\ell &gt; 0.04, \quad Bo &gt; 0.25 \times 10^{-4}$</td>
</tr>
<tr>
<td></td>
<td>$\psi$: From chart (Shah, 1982)</td>
</tr>
<tr>
<td>Stephan and Abdelsalam (1980)</td>
<td>$h_{TP} = C_4 q^{-0.745}$</td>
</tr>
<tr>
<td></td>
<td>$C_4 = 1.2$, for R-113, and P in the range of 140-180 kPa</td>
</tr>
</tbody>
</table>
Table 2. (Cont’d)

<table>
<thead>
<tr>
<th>Source</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Shah (1982) (modified)</td>
<td>$h_{TP} = \psi h_L,$ where $\psi = 230 \text{Bo}^{0.5}$, for $\text{Bo} &gt; 3.0 \times 10^{-5}$</td>
</tr>
<tr>
<td>Lazarek and Black (1982)</td>
<td>$h_{TP} = 30 \text{Re}_L^{0.857} \text{Bo}^{0.714} \left( \frac{k_f}{D} \right)$</td>
</tr>
<tr>
<td>Kandlikar (1983)</td>
<td>$h_{TP} = \left[ D_1(\text{Co})^{D_2} (25F_{\text{Fr}<em>f})^{D_5} + D_3(\text{Bo})^{D_4} 25(F</em>{\text{Fr}<em>f})^{D_6} (F</em>{\text{Fr}}) \right] h_f,$ where $D_5 = D_6 = 0$, for $F_{\text{Fr}} &gt; 0.04$</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Constant</th>
<th>For Co &lt; 0.65</th>
<th>For Co &gt; 0.65</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_1$</td>
<td>1.091</td>
<td>0.809</td>
</tr>
<tr>
<td>$D_2$</td>
<td>-0.948</td>
<td>-0.891</td>
</tr>
<tr>
<td>$D_3$</td>
<td>887.46</td>
<td>387.53</td>
</tr>
<tr>
<td>$D_4$</td>
<td>0.726</td>
<td>0.587</td>
</tr>
</tbody>
</table>

$\text{Fr}_f = 1.24$, for R-113

\[ Fr_f = \frac{G^2}{\rho_f^2 gD} \]
Table 2. (Cont'd)

<table>
<thead>
<tr>
<th>Source</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Liu and Winterton (1990)</td>
<td>( h_{TP} = [((Eh_t)^2 + (Sh_{pool})^2)^{1/2}, )</td>
</tr>
<tr>
<td></td>
<td>where</td>
</tr>
<tr>
<td></td>
<td>( h_{pool} = 55 \left( \frac{P}{P_{cr}} \right)^{0.12} \left[ -\log_{10} \left( \frac{P}{P_{cr}} \right) \right]^{-0.55} m^{-0.5} q^{0.67} )</td>
</tr>
<tr>
<td></td>
<td>( E = \left[ 1 + (x) \left( \frac{P}{P_{cr}} \right) \left( \frac{\rho_t}{\rho_g} - 1 \right) \right]^{0.35} )</td>
</tr>
<tr>
<td></td>
<td>( S = \left[ 1 + 0.055 E^{0.1} (Re_t)^{0.16} \right]^{-1} )</td>
</tr>
<tr>
<td>Jung and Radermacher (1991)</td>
<td>( h_{TP} = h_{NB} + h_{conv} )</td>
</tr>
<tr>
<td></td>
<td>( h_{TP} = Nh_{sa} + F_{ph_t} )</td>
</tr>
<tr>
<td></td>
<td>where</td>
</tr>
<tr>
<td></td>
<td>( h_{sa} = 207 \frac{k_T}{(bd)} \left[ \frac{q''(bd)}{k_T T_{sat}} \right]^{0.745} \left( \frac{\rho_g}{\rho_t} \right)^{0.581} Pr^{0.533} )</td>
</tr>
<tr>
<td></td>
<td>((bd) = 0.146 \beta \left[ \frac{2\sigma}{g(\rho_t - \rho_g)} \right]^{-0.5}, ; \beta = 35^\circ )</td>
</tr>
<tr>
<td></td>
<td>( N = \begin{cases} 4048 X_{tt}^{1.22} B_o^{1.13}, &amp; \text{for } X_{tt} &lt; 1 \ 2.0 - 0.1 X_{tt}^{0.28} B_o^{0.33}, &amp; \text{for } 1 &lt; X_{tt} \leq 5 \end{cases} )</td>
</tr>
<tr>
<td></td>
<td>( F_p = 2.37 \left( 0.29 + \frac{1}{X_{tt}} \right)^{0.85} )</td>
</tr>
<tr>
<td></td>
<td>( T_{sat} ) is expressed in K</td>
</tr>
</tbody>
</table>

Source
Table 2. (Cont’d)

<table>
<thead>
<tr>
<th>Source</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steiner and Taborek (1992)</td>
<td>( h_{TP} = \left( (h_{nb})^n + (h_{cb})^n \right)^{1/n} ) or ( h_{TP} = \left( (h_{nb,o}F_{nbf})^n + (h_{LFTP})^n \right)^{1/n} ), where ( n = 3 )</td>
</tr>
<tr>
<td></td>
<td>( F_{TP} = \left[ (1 - x)^{1.5} + 1.9(x)^{0.6} \left( \frac{\rho_l}{\rho_g} \right)^{0.35} \right]^{1.1} ), for ( x \leq 0.6 ) and ( F_{TP} = \left[ (1 - x)^{1.5} + 1.9(x)^{0.6} \left( 1 - x \right)^{0.01} \left( \frac{\rho_l}{\rho_g} \right)^{0.35} \right]^{2.2} )</td>
</tr>
<tr>
<td></td>
<td>[ F_{TP} = \left[ \left( \frac{h_G}{h_L} \right)(x)^{0.01} \left( 1 + 8(1 - x)^{0.7} \left( \frac{\rho_l}{\rho_g} \right)^{0.67} \right) \right]^{-2} \right)^{-0.5} ]</td>
</tr>
<tr>
<td></td>
<td>for ( x &gt; 0.6 )</td>
</tr>
<tr>
<td></td>
<td>( h_{nb,o} = 2180 \text{ W/m}^2\text{K} ), for R-113</td>
</tr>
<tr>
<td></td>
<td>( F_{nbf} = f(m, x)F_{pf} \left( \frac{q''}{q_{of}} \right)^{nf(pr)} ) ( F(d)F(R_a)F(M) )</td>
</tr>
<tr>
<td></td>
<td>( f(m, x) = 0.37 )  (assumed constant value in this study)</td>
</tr>
<tr>
<td></td>
<td>( q_{of} = 20,000 \text{ W/m}^2 )</td>
</tr>
</tbody>
</table>
Table 2. (Cont'd)

<table>
<thead>
<tr>
<th>Source</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{pf} = \left[ 2.816(P_r)^{0.45} + \left( 3.4 + \frac{1.7}{1-(P_r)^7} \right)(P_r)^{0.37} \right]$, for $P_r \leq 0.95$</td>
<td></td>
</tr>
<tr>
<td>$n_f(pr) = 0.8 - (0.1)\exp(1.75P_r)$, for all fluids except cryogenics</td>
<td></td>
</tr>
<tr>
<td>$F(d) = \left( \frac{d}{d_0} \right)^{-0.4}$, where $d_0 = 0.01m$</td>
<td></td>
</tr>
<tr>
<td>$F(R_a) = \left( \frac{R_a}{R_{a,0}} \right)^{0.133} = 1$</td>
<td></td>
</tr>
<tr>
<td>$F(m) = 0.377 + 0.119 \ln(M) + 2.8427 \times 10^{-5}(M)^2$</td>
<td></td>
</tr>
</tbody>
</table>

*In the above correlations, the following definitions are used:

**Dittus-Boelter correlation for single-phase heat transfer:**

$$h = 0.023 \left( \frac{k_f}{D} \right)(Re_f)^{0.8}(Pr_f)^{0.4}, \quad h_L = 0.023 \left( \frac{k_f}{D} \right)(Re_L)^{0.8}(Pr_f)^{0.4},$$

$$h_G = 0.023 \left( \frac{k_G}{D} \right)(Re_G)^{0.8}(Pr_G)^{0.4};$$

**Boiling number:** $Bo = \frac{q^*}{G_{fg}}$;

**Martinelli parameter:** $X_{tt} = \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{\rho_G}{\rho_f} \right)^{0.5} \left( \frac{\mu_f}{\mu_G} \right)^{0.1}$;

**Reynolds number:** $Re_f = \frac{G(1-x)D}{\mu_f}, \quad Re_L = \frac{GD}{\mu_f}, \quad Re_G = \frac{GD}{\mu_G}$;

**Prandtl number:** $Pr_f = \frac{c_p f \mu_f}{k_f}, \quad Pr_G = \frac{c_p g \mu_g}{k_G}$.

All fluid properties are based on saturation temperature.
Fig. 8. Comparison of measured heat transfer coefficients with prediction of Chen (1966)

Fig. 9. Comparison of measured heat transfer coefficients with predictions of Chaddock and Brunemann (see Reid et al., 1987) and Pujol and Stenning (1969)
Fig. 10. Comparison of measured heat transfer coefficients with prediction of Shah (1976)

Fig. 11. Comparison of measured heat transfer coefficients with predictions of Kandlikar (1983) and Liu and Winterton (1990)
Table 3. Statistical comparison of correlations; total number of data points
$N = 105$

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Mean Deviation$^a$ (%)</th>
<th>Percentage of Data in ±20% range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lazarek and Black (1982)</td>
<td>12.7</td>
<td>84.8</td>
</tr>
<tr>
<td>Stephan and Abdelsalam (1980)</td>
<td>12.9</td>
<td>83.8</td>
</tr>
<tr>
<td>Shah (1976), as modified</td>
<td>13.1</td>
<td>81.0</td>
</tr>
<tr>
<td>Liu and Winterton (1990)</td>
<td>15.5</td>
<td>74.3</td>
</tr>
<tr>
<td>Jung and Radermacher (1991)</td>
<td>16.0</td>
<td>75.2</td>
</tr>
<tr>
<td>Steiner and Taborek (1992), as modified</td>
<td>17.5</td>
<td>66.7</td>
</tr>
<tr>
<td>Shah (1976)</td>
<td>17.6</td>
<td>68.6</td>
</tr>
<tr>
<td>Kandlikar (1983)</td>
<td>22.0</td>
<td>57.1</td>
</tr>
<tr>
<td>Chaddock and Brunemann (1967)$^b$</td>
<td>24.0</td>
<td>50.5</td>
</tr>
<tr>
<td>Chen (1966)</td>
<td>36.0</td>
<td>41.9</td>
</tr>
<tr>
<td>Pujol and Stenning (1969)</td>
<td>53.6</td>
<td>6.7</td>
</tr>
</tbody>
</table>

$^a$Mean deviation $= \frac{1}{N} \sum \frac{|h_{meas.} - h_{pred.|}}{h_{meas.}} \times 100$

$^b$See Reid et al., 1987

The Shah (1976) correlation involves the use of graphs given in that reference. It was used in that form, and is presented as such in Table 2. [In a later publication, Shah (1982) fit curves to the graphs.] The predictions of the Shah (1976) correlation are compared with the present data in Fig. 10. The results are similar to those from the Chen (1966) correlation shown in Fig. 8, where the predictions are centered in the data; however, there is considerable scatter in the Shah (1976) predictions, albeit less than in the Chen (1966) predictions (see Table 3).

Winterton has presented three correlations in the literature. The initial correlation (Gungor and Winterton, 1986) was used by Reid et al. (1987) in evaluating their large-tube R-113 data. The correlation was revised in 1987 (Gungor and Winterton, 1987) and again in 1990 (Liu and Winterton, 1990). The most recent correlation (Liu and Winterton, 1990) was used in this investigation. It is given in Table 2, and predictions from it are compared to the present data in Fig. 11. The data are seen to be consistently underpredicted, but the scatter band is small. Also shown in Fig. 11 are the predictions of the Kandlikar (1983) correlation. Here, the data are mostly overpredicted, but with a small scatter band.
Three other correlations were evaluated with the present data: Lazarek and Black (1982), Jung et al. (1989, 1991), and Steiner and Taborek (1992). Each has a feature that suggested its inclusion in the present evaluation.

The correlation of Lazarek and Black (1982) was used in this study because the data base for it, viz., the boiling of R-113 inside of 3.1-mm-diameter tubes, was the closest found to the present experiments. The experiments of Lazarek and Black (1982) were different from those in the present study in that they used a vertically oriented (U-tube) test section with both upflow and downflow occurring simultaneously by means of a curved connecting section. The heat transfer correlation that was developed for these data is presented in Table 2 and predictions are compared with the present data in Figs. 12 and 13. With a mean deviation of 12.7% (Table 3), the results are the best of all the correlations presented thus far. As shown in Fig. 13, there is some larger error at low qualities, but 85% of the data are correlated to ±20% (Table 3), and this result is considered very good.

Jung et al. (1989, 1991) recently developed a correlation, given in Table 2, specifically for refrigerants boiling in a relatively large (9-mm-diameter) tube; the authors report a mean deviation of 7.2%, from their experimental data obtained with R-22, R-12, R-152a, and R-114. In Fig. 14, the correlation is evaluated against the present data. The correlation shows good agreement with the data except for a few measurements, which are all at low quality. Seventy-five percent of the data fall within 20% of predictions.

The final correlation that was considered for application to the present data is a new treatment due to Steiner and Taborek (1992). This correlation was developed from a data base of more than 12,000 measurements. It was developed along the superposition principle of Chen (1966), but it added the Churchill (1974) technique for introducing the proper asymptotic predictions well into the nucleation- or convection-dominated heat transfer regimes. The correlation was developed strictly for vertical flow boiling. In applying it to the horizontal data of this study, the parameter that accounts for differences between nucleate-pool boiling and nucleation-dominated flow boiling in the correlations was much too large, and the correlation overpredicted the present data by a factor of ≈3. No reason was discovered for this anomaly. Therefore, a constant value of 0.37 was assumed for the horizontal correction factor and the correlation as presented in Table 2 was applied to predict the present data. The correlation is defined as the Steiner and Taborek (1992) correlation as modified. The results are shown in Fig. 15, where it can be observed that the correlation is well centered, with a mean deviation of 17.5% and 67% of the data points predicted within 20%.
Fig. 12. Comparison of measured heat transfer coefficients with prediction of Lazarek and Black (1982).

Fig. 13. Ratio of Nusselt number predicted by Lazarek and Black (1982) to measured Nusselt number as a function of mass quality.
Fig. 14. Comparison of measured heat transfer coefficients with prediction of Jung and Radermacher (1991)

Fig. 15. Comparison of measured heat transfer coefficients with prediction of Steiner and Taborek (1992) correlation, as modified for the present study
§ Discussion

Evaluation of the applicability of existing correlations and/or the development of new correlations for heat transfer require that we know whether a nucleate boiling mechanism or a convective mechanism dominates a particular quality range or length of heat transfer channel. By definition, in compact heat exchangers the channel cross-sectional area is small and the channel length is short. To avoid unacceptably high pressure drops, the small channel cross section leads to low flow velocities (low mass fluxes), with Reynolds numbers typically in the laminar regime for a subcooled liquid. The short channel length leads to a high heat flux if superheated vapor is to be produced. This combination of high heat flux and low mass flux results in high boiling numbers for small channels. All of the boiling numbers in the tests of this study were relatively high, ranging from $5 \times 10^{-4}$ to $25 \times 10^{-4}$. This parameter played a key role in interpreting the differences among the predictions of the various correlations that were tested.

Lazarek and Black (1982) obtained heat transfer data over the entire range of qualities up to CHF. They concluded that in their experiments, the nucleate boiling mechanism was the dominant heat transfer mechanism that was controlling the wall heat transfer process in all their tests. This conclusion was based on an observed strong dependence of heat transfer coefficient on heat flux, with negligible influence of quality. Pujol and Stenning (1969) showed that as the boiling number increases, the transition quality, from nucleation-dominated to convection-dominated boiling, also increases. Following the work of Pujol and Stenning (1969), Lazarek and Black (1982) suggested that the occurrence of nucleation-dominated heat transfer all the way to CHF could be attributed to high boiling numbers, which were above $5 \times 10^{-4}$. As a result of the high boiling numbers, the critical quality was reached before the transition to nucleation-suppressed boiling occurred.

The high boiling numbers obtained during this study are expected to be typical of those that can be expected in compact heat exchangers operating in the parameter ranges of this study, as previously discussed. It is, therefore, also probable that the nucleation mechanism will dominate the boiling heat transfer. Under such conditions, it would be expected that the best correlation of the data would come from models that accentuate the nucleation heat transfer component or use it exclusively. In an attempt to verify this condition, all of the correlations considered in this study were carefully examined to determine the relative contributions of the nucleation and convective terms relative to the present data, all of which exhibit nucleation domination. It was found that, in general, the correlations that best predicted the data were dominated by the nucleation term.
The converse was also found to be true; correlations that predicted the data poorly had large convective boiling terms. Each correlation is discussed below.

The Chen (1966) correlation showed considerable spread when compared with the present data; see Fig. 8. It was found that the nucleation and convection components of the heat transfer were predicted to be of roughly equal magnitudes for most of the data. This condition is contrary to the data, and the spread in the predictions is a consequence.

The Shah (1976) correlation has a basic nondimensionalization problem typical of many correlations. As pointed out by Steiner and Taborek (1992), the use of a convective heat transfer coefficient to nondimensionalize the nucleation contribution to heat transfer does not agree with the physics of the situation. The heat transfer coefficient $h_1$ includes both mass flux and quality effects that are inappropriate in this regime. However, the Shah (1976) correlation has a mass flux effect in the boiling number of the nucleation heat transfer model. The mass flux from the boiling number and the convection coefficient used for nondimensionalization purposes produce a net small mass flux effect of $G^{0.3}$. The fact that there is no separate convection term in the Shah (1976) correlation makes it appropriate for the data of this study. The result is that the correlation predicts the data reasonably well; however, there is some scatter, as seen in Fig. 10. There is some place for improvement with respect to removing convection mechanisms from the model.

The correlations of both Chaddock and Brunemann (see Reid et al., 1987) and Pujol and Stenning (1969) are nondimensionalized with a convective heat transfer coefficient. The mass flux effects in both correlations are larger than the Shah (1976) correlation, and the data predictions are not good, as shown in Fig. 9.

The correlation of Kandlikar (1983) is also nondimensionalized by $h_1$. Two terms represent nucleation and convection contributions to the heat transfer in this correlation. These terms are of comparable magnitude at higher qualities, with a mass flux effect in both terms giving rise to the deviation from the data shown in Fig. 11 and quantified in Table 3.

The correlations due to Winterton started with the work of Gungor and Winterton (1986), which was similar in form to the Chen (1966) correlation. There were two terms, one each for the contributions of nucleation and convection heat transfer, and the mass flux effect appeared in both. This correlation was replaced (Gungor and Winterton, 1987) with a form similar to that of Chaddock and Brunemann (see Reid et al., 1987) and most recently replaced by the work of Liu and Winterton (1990). The latest correlation is similar to the Steiner and Taborek (1992) correlation in the sense that it is based on the correct asymptotes for the
Chen (1966) form of the correlation. Because there are two terms in this correlation, one for nucleation and one for convection, the relative magnitudes must be improved to satisfactorily predict the present data which, despite a relatively small mean deviation of 15.5%, are consistently underpredicted by this correlation in its present form, as shown in Fig. 11.

The correlation of Lazarek and Black (1982) contains only a nucleation term because it was derived from the nucleation data base. The mass flux appears both in a Reynolds number and in the boiling number so that the net effect is $G^{0.14}$. This minimal mass flux effect, coupled with no explicit convective term in the correlation, was consistent with the data of the present investigation. Figures 12 and 13 show that, of all the correlations tested, the correlation of Lazarek and Black (1982) best predicts the present data, with a mean deviation of 12.7%, as given in Table 3. It also suggests that the 3-mm tube diameter is small enough for surface tension forces to dominate gravitational forces inasmuch as the Lazarek and Black (1982) correlation, developed for vertical tubes, predicts the present horizontal tube data so well.

The correlation of Jung and Radermacher (1991) is of the Chen (1966) type, with two terms. Based on their study of 13 different refrigerants, they concluded that, in the quality range >20%, nucleate boiling is predicted to be fully suppressed for all refrigerants, indicated by the fact that the heat transfer coefficient increases with quality. Thus, most of the data base for this correlation was in the convection-dominated region at qualities above 0.2. However, the very good comparison of the correlation with the present data, shown in Fig. 14, is a result of the predicted dominance of the nucleate boiling term. This term was adopted from the general work of Stephan and Abdelsalam (1980); however, that work also included simpler but fluid-specific correlations for nucleate boiling heat transfer. The correlation for R-113 is listed in Table 2 for the pressure range of the present data. The correlation is heat-flux-dependent and mass-flux-independent (for this natural convection boiling situation), and it predicts the data very well, as shown in Fig. 16. This comparison supports the nucleate boiling domination trend of the data, as do all other comparisons of this study where improved data prediction accompanies a prediction of nucleate boiling as the dominant heat transfer mechanism.

The Stöiner and Taborek (1992) work is based on a combination of the superposition model with the technique of Churchill (1974) for producing the proper asymptotes to the data. Although the correlation is based on a very large data base, the spread of predictions from the present data is due to a reasonably large predicted convective term.
Although the Lazarek and Black (1982) correlation, with the proper physical parameters, predicted the present data well, an attempt was made for further improvement by using the Shah (1976) correlation form. As mentioned previously, the mass flux effect in the Shah correlation can be as low as $G^{0.3}$, which is predicted for very large boiling numbers. To maintain this small effect of mass flux, the Shah (1976) correlation was modified to impose the high boiling number form at all times. A second modification was made in the form of the heat transfer coefficient that was used for nondimensionalizing. The coefficient $h_f$ was based on $G$ rather than $G(1 - x)$ to eliminate a convective-type quality dependence from the correlation. These modifications are presented in Table 2 as the modified Shah correlation. With only these two changes, and with all other constants as in Shah's original correlation (1976), the data were predicted very well, as shown in Fig. 17, with a mean deviation of 13.1% (Table 3).

7 Summary and Concluding Remarks

The evaporation heat transfer of refrigerant R-113 in a uniformly-heated, small-diameter (3 mm), horizontal tube was studied. Results of the study are reported and compared with the results of Reid et al. (1987) from a similar investigation of flow boiling of R-113, but in 8.71- and 10.92-mm diameter tubes.
Data were also compared with eight correlation equations from the literature for predictive two-phase heat transfer upstream of CHF, and with two modified correlations (the modifications were made as part of this study).

In agreement with the results of Reid et al. (1987), the results of this study showed that the local heat transfer coefficient for the evaporation of R-113 in a small-diameter, horizontal tube is a strong function of heat flux, and only weakly dependent on mass quality, although trends with mass quality could be observed. Qualitatively, the values measured for heat transfer coefficients were in good agreement with the measurements of Reid et al. (1987). The compared data were at relatively high boiling numbers, in the range of $3 \times 10^{-4}$ to $6 \times 10^{-4}$, giving rise to large nucleation-dominant regions of the flow.

The flow pattern map of Damianides and Westwater (1988) for a 3-mm-diameter tube was superimposed on the heat transfer data of this study. The map revealed that most of the data were in the plug and slug flow regions. Nucleation in the thick liquid layers of these regions was found not to be suppressed for all of the data obtained, although the trends indicated that suppression would occur at higher qualities for each heat flux tested.
The dominance of the nucleation mechanism in the present data was attributed to the high boiling number of the data. As a result, two-phase heat transfer correlations that predicted this dominance also best predicted the data if they also properly modeled the physical parameters. The correlation of Lazarek and Black (1982) and of Shah (1976) as modified in this study predicted the data well; mean deviations were approximately 13%, and more than 80% of the data were predicted within ±20%. The simplified correlation of Stephan and Abdelsalam (1980) also predicted the data well where heat flux and not mass flux was the controlling parameter. In a similar manner, the recent correlation of Jung and Radermacher (1991), based on data of many refrigerants in larger diameter tubes, predicted the present data well (with the exception of very low qualities) due to the dominance of the nucleation term.

In general, the small-tube boiling heat transfer coefficient exhibited the same characteristics and trends as found in larger tubes, but the dominating mechanisms showed a significant change, which was strongly reflected in the predictions of the correlations. However, the designs of compact heat exchangers can lead to high boiling numbers and extended slug flow regimes similar to those of this study. This situation leads in turn to nucleation-dominant heat transfer in round channels, a result that is the opposite of the predominantly convective-dominated heat transfer of conventional refrigeration evaporators (Jung and Radermacher, 1991).

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