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Abstract

Experimental measurements of void fraction, bubble frequency, and velocity were obtained in subcooled R-134a flowing over a heated flat plate near an unheated wall and compared to analytical predictions. The measurements were obtained for a fixed system pressure and mass flow rate (P=2.4 MPa and w=106 kg/hr) at various inlet liquid temperatures. During the experiments, electrical power was applied at a constant rate to one side of the test section. The local void fraction data, acquired with a hot-film anemometer probe, showed the existence of a significant peak near the heated wall and a smaller secondary peak near the unheated wall for the larger inlet subcoolings. Local vapor velocity data, taken with the hot-film probe and a laser Doppler velocimeter, showed broad maxima near the centerline between the heated and unheated plates. Significant temperature gradients near the heated wall were observed for large inlet subcooling. Bubble size data, inferred from measurements of void fraction, bubble frequency and vapor velocity, when combined with the measured bubble chord length distributions illustrate the transition from pure three dimensional spherical to two-dimensional planar bubbly flow, the latter being initiated when the bubbles fill the gap between the plates. These various two-phase flow measurements were used for development of a multidimensional, four-field calculational method; comparisons of the data to the calculations show reasonable agreement.

Nomenclature

d_b	local bubble diameter
d_s	spacing between HFA probe sensors
D_h	hydraulic diameter
$E(\tau)$	cross-correlation factor
f_b	local bubble frequency
I	gamma beam intensity through two-phase mixture
I_o	gamma beam intensity with empty test section
P	pressure
t	test section spacing (0.25 cm) or time
$t_{E(\tau)max}$	time associated with maximum cross-correlation factor
T	time interval
T_{in}	average liquid inlet temperature
V	voltage
V_g	local vapor velocity
V_i	interfacial velocity
w	mass flow rate
X	streamwise position
Y	transverse position
Z	spacing position
α	local void fraction
μt	product of fluid attenuation coefficient and fluid thickness in gamma densitometer system (GDS) void fraction measurement
ρ_g	vapor density
ρ_l	liquid density
$\rho_{2\phi}$	two-phase mixture density

1. Introduction

Gas-liquid two-phase flows are encountered in a variety of industrial applications, including chemical processing, petroleum transport and power generation. Computer codes are often used for process optimization and equipment design. However, successful application of the next generation of computer codes based on a multi-field formulation relies on the availability of an experimental database that can be used to develop an understanding of fundamental physical phenomena. Such a database is also required to rigorously assess a code's predictive capability. For subcooled boiling flows, the quantities and phenomena that need to be modeled include lift and wall forces, turbulent dispersion, bubble induced turbulent viscosity, bubble size, wall heat partitioning, interfacial heat transfer, bubble drag, interfacial area density and bubble nucleation inception.

A number of experimental investigations have been conducted to obtain local data in subcooled boiling flows. Jiji and Clark (1964) measured mean and fluctuating temperature distributions in boiling flows of water over a flat plate contained within a vertical flow channel. Walmet and Staub (1969) measured temperature distributions in low pressure boiling water flows through a duct heated on one side. These authors also reported liquid velocity profiles, measured with miniature pressure probes, and void fraction profiles normal to the heated surface measured with an x-ray densitometer. Local measurements of void fraction in subcooled boiling flows of R-114 through an annular duct were made by Shiralkar (1970) and Dix (1971) using constant temperature hot-film anemometry. Later, Delhaye *et al.* (1973) used a microthermocouple to measure the local void fraction distribution in subcooled flow boiling of water, based on the probability density function of the fluid temperature.

A significant number of the more recent fundamental investigations of subcooled boiling flows has been conducted by two groups. Roy and co-workers (Jain and Roy, 1983; Hasan *et al.*, 1991; Roy *et al.*, 1993; Roy *et al.*, 1994; Velidandla *et al.*, 1995) investigated turbulent subcooled boiling flows of R-113 through a vertical annular duct whose inner wall was heated. The quantities measured included: line-average void fraction using an x-ray densitometer, local vapor and liquid temperatures using microthermocouples, local void fraction using a constant temperature hot-film anemometer and fiber-optic probe, bubble velocity using a dual-sensor fiber-optic probe, and mean and fluctuating vapor and liquid velocities using a laser Doppler velocimeter. From the output signals of the dual-sensor fiber-optic probe, local bubble size distributions and estimates of interfacial area concentration were also obtained. Shoukri and co-workers (Shoukri *et al.*, 1991; Zeitoun *et al.*, 1994; Zeitoun *et al.*, 1995; Zeitoun and Shoukri, 1996; Zeitoun and Shoukri, 1997) studied subcooled boiling and condensing flows of water in an annular test section with the inner wall heated, mostly under low pressure and low flow conditions. Measurements reported by these researchers included axial void fraction profiles measured using a single-beam gamma densitometer, and bubble size and interfacial area concentration using high speed video and image processing techniques.

Similar to the experimental programs of Roy *et al.* and Shoukri *et al.*, the primary objective of the present work was to provide local data needed to develop mechanistic models for boiling two-phase flows in narrow spaces. The experiments were performed in a vertical test section at a single mass flow rate and pressure condition. A fixed wall heat flux was applied to one side of the test section, and the inlet flow temperature was varied to create a range of dispersed vapor flow fields. A variety of advanced measurement techniques was applied to ascertain the detailed, local characteristics of these flows. Local and line-average void fraction data were obtained using a hot-film anemometer (HFA) and gamma densitometer system (GDS), respectively. Measurements of local interfacial velocity were acquired in the direction of flow along the parallel plates using laser Doppler velocimetry (LDV) and dual-sensor HFA techniques, as well as bubble frequency measurements with the latter. Important modeling information such as bubble size was inferred from measurements of the local vapor volume fraction, vapor velocity and bubble frequency. The HFA instrumentation was additionally operated in the constant current mode to obtain measurements of the local liquid temperature profiles across the narrow test section dimension. These various measurements were compared to calculated results from a two-fluid, four-field computer code.

2. Test Facility and Measurement Techniques

The test section, illustrated schematically in Figure 1, was a vertical duct with a hydraulic diameter of 0.485 cm and aspect ratio (width-to-spacing) of 22.5. These test section dimensions facilitate the use of thin, transparent heater films that enable visual observations and are consistent with the flow and control capabilities of the test loop. The working fluid was R-134a (1,1,1,2-tetrafluoroethane) which is one of the relatively new class of nonchlorinated refrigerant fluids that do not deplete the ozone layer. Key components of the flow loop were a chiller and pressurizer to maintain the liquid phase at the inlet of a circulating canned rotor pump, a large CO_2 heat exchanger, loop heaters, high/low range throttle valves, flow meters, and a vertical test section. Loop conditions were set by programmed logic controllers. The measured operating conditions included mass flow rate, temperature, pressure, heater power, and test section pressure drop. The loop design pressure ranged from 0.4 to 2.5 MPa, and the temperature ranged from 0° to 80°C. Optical access to the flow in the test section was provided by eight quartz windows, each 3.8 cm thick by 7.6 cm wide by 27.9 cm long. The center 5.7 cm of the window width comprised the transverse (Y) dimension of the internal flow passage. The measurement ports between each pair of windows contained 2.5 cm diameter inserts in the side of the test section. These inserts permitted access to the flow for thermocouple rakes or other instrumentation.

An instrument scanning mechanism was used to position the gamma densitometer system (GDS) and laser Doppler velocimeter (LDV) instrumentation along three axes: the Z axis (horizontal scans along the test section spacing dimension), the Y axis (horizontal scans across the transverse width of the test section) and the X axis (vertical, or streamwise position). To measure void distributions in either the thickness (Z) or the width (Y) directions, the gamma densitometer was rotated 90 degrees about the test section. Both gamma beam and laser tests have shown that the GDS and LDV positioning accuracy is approximately ±0.03 mm. A small offset in the measurement position (usually less than ±0.05 mm), introduced by thermal expansion of the test section, was corrected for, as necessary.

All GDS data were acquired at X = 51.7 cm (X/D_h = 106.5; Measurement Level 4). In addition to measurements of the cross-sectional averaged void fraction, line-averaged edge scan data were obtained at seven positions across the narrow test section dimension. Also, a line-averaged measurement was taken at the center of the test section transverse dimension, to compare to the integrated average of the local Z-scan void fraction data acquired with the hot-film anemometer (HFA) probe. The HFA instrumentation consisted of a dual-sensor probe mounted through a hole centered near the top of Window 6 at X = 57.9 cm (X/D_h = 119.4). In addition to local void fraction data, measurements were made of local bubble frequency, vapor velocity and liquid temperature at up to 32 locations across the narrow (Z) test section dimension. Z-dimension LDV vapor velocity scans were also obtained at the center of the transverse (Y) dimension, using a backscatter fiber optic probe at X = 55.4 cm (X/D_h = 114.2). It was not possible to take LDV data at exactly the same streamwise position as the dual-sensor HFA probe due to spurious reflections originating from the aluminum block used to mount the HFA probe in Window 6.

The triple track window heater design consisted of three transparent indium-tin-oxide conductive films vapor deposited onto the inside window surface and an anti-reflective coating on the outside surface. The bridges between the quartz windows contained the brushes to carry current to the window heater films. Three silver epoxy buses carried the current around both ends of the windows and connected with the silver graphite brushes. Since the power to each of the three window heater strips was independently controlled, experiments were performed with various non-uniform heating profiles.

The two-phase flow field was created by phase change in the test section due to heat addition through the window heater strips. The inlet temperature was controlled by means of a heater located upstream of the test section inlet. Subcooled R-134a was introduced to the test section and heat was applied to only the two lower windows on one side of the test section at a time. The movement of the dual sensor HFA probe was limited to about half the test section Z dimension, in order to avoid contacting the curved part of the sensor support needles with the bottom of the hole in the quartz window. However, it was desired to take measurements over as much of the spacing dimension as possible. This was accomplished by first heating Windows 1 and 2 to obtain data from one wall to the duct center. Then Windows 5 and 6 were heated to obtain data from the opposite wall to the duct center to cover the entire range.

Prior to initiating these measurements, the HFA probe was calibrated for liquid temperature measurement by running the probe as a constant current anemometer (CCA). Based on previous testing experience, the LDV technique is difficult to apply when the incident laser beams are directed through a window heater strip with vapor generation. Therefore, LDV data were acquired only when Windows 5 and 6 were heated.

The primary features of the specialized instruments used in the subcooled boiling flows experiments are briefly discussed below.

Gamma Densitometer System (GDS)

The gamma densitometer provides a direct measurement of the density of the two-phase mixture in the path of the gamma beam through the following relationship:

$$\rho_{2\phi} = \frac{\ln\left(\frac{I_o}{I}\right)}{\mu t} \qquad (1)$$

where I_o and μt are calibration constants obtained from gamma count measurements at each desired measurement position with an empty test section and a subcooled liquid-filled test section, and I is the count rate measured for the two-phase test condition. The local void fraction is related to the two-phase mixture density, and vapor and liquid component densities, through the following relationship:

$$\alpha = \frac{\rho_l - \rho_{2\phi}}{\rho_l - \rho_g} \qquad (2)$$

where α is the local void fraction, ρ_l is the density of the liquid phase, and ρ_g is the density of the vapor phase. In the subcooled region, Eq. 2 can only be applied if the liquid density at the measurement location is known; otherwise, the two-phase mixture density must be used directly for comparisons with calculations. Although liquid temperature data were not obtained at the GDS measurement locations, a good estimate of the subcooled liquid density was obtained from the liquid temperature distribution measured at the HFA location, which was 6.2 cm downstream of the GDS location.

Hot-Film Anemometer (HFA)

The constant temperature hot-film anemometer technique has been used previously for various thermal-hydraulics measurements and is described in detail by Trabold *et al.* (1997). For the present test sequence, dual-sensor probes were installed for void fraction, bubble frequency and vapor velocity measurements along a single spacing (Z) scan. These HFA probes were comprised of two active sensing elements separated in the streamwise (X) direction by a known distance, d_s. The HFA probes used in the present study had platinum film sensors with 25 μm diameter and 254 μm active length. The sensor separation distance of the probe was 0.144 ± 0.0013 cm. The use of two sensors permits acquisition of interfacial velocity measurements based on the cross-correlation between two output voltage signals V_1 and V_2:

$$E(\tau) = \lim_{T \to \infty} \frac{1}{T} \int_0^T V_1(t) V_2(t+\tau) dt \qquad (3)$$

The peak in the $E(\tau)$ versus time plot corresponds to the most probable time ($t_{E(\tau)max}$) required for a gas-liquid interface to travel between the HFA sensors, from which the mean interfacial velocity may be calculated by

$$V_i = V_g = \frac{d_s}{t_{E(\tau)max}} \qquad (4)$$

In addition to the various HFA measurements described in this paper, it is also possible to obtain local liquid temperature measurements by operating the probe in the constant current mode. The calibration procedure for this measurement is discussed in Section 3.

<u>Laser Doppler Velocimeter (LDV)</u>

A backscatter fiber optic LDV probe was also used to obtain local vapor velocity measurements. The probe was equipped with a short focal length (122 mm) lens to produce a measurement volume about 0.025 cm long. The probe was mounted on a traversing slide to enable motion across the narrow (Z) test section dimension, with a positioning uncertainty of approximately ±0.127 mm. The beam power at the probe exit was between 40 and 75 mW.

The Doppler signals were analyzed using a counter-timer signal processor. To reduce noise, this processor requires that the input signals exceed a threshold voltage. If a signal is greater than this minimum amplitude, it is processed for velocity; however, if the processor gain is set too high, noise may be misinterpreted as a valid signal. To ensure that noisy signals are not processed, the signal processor performs a timing comparison for portions of each Doppler burst to ensure that the frequency is constant for each burst. Also, an additional data quality check was performed by periodically blocking one of the LDV beams to confirm that the data rate fell to zero. If noise appeared in the velocity histograms (usually in the form of stray velocity samples separated from the main peak) it was removed by a manual editing procedure using the data analysis software. These edited samples were usually less than 10% of the total number of samples. Typical data rates for the vapor velocity measurements were between 2 and 20 Hz. Between 500 and 1000 velocity samples were obtained at each measurement location over a period of 2 to 5 minutes.

Measurement Uncertainty

The measurement uncertainty, U, for all test section instrumentation was calculated based on the root-sum-square uncertainty interval for 95% confidence (Dieck, 1992):

$$U = \pm[B^2 + (t_{95}S_{\bar{x}})^2]^{1/2} \tag{5}$$

where B is the bias limit (systematic error) and $t_{95}S_{\bar{x}}$ is the precision limit (random error). The ranges of experimental parameters investigated, and the uncertainty associated with each, are summarized in Table 1.

3. Results and Discussion

The experimental results discussed below were acquired at a mass flow rate and pressure of 106 kg/hr and 2.4 MPa. All data are presented in terms of the dimensionless distance from the wall (*Z/t*).

Void Fraction

Measurements of local void fraction are of primary importance in two-fluid model development and code qualification. As outlined above, the R-134a test facility has the capability to make a variety of void fraction measurements using both the gamma densitometer system (GDS) and hot-film anemometer (HFA). In the present investigation, attention was focused on the narrow (Z) test section dimension profiles obtained near the top of the second heated window elevation (X/D_h = 106.5 and 119.4, respectively).

The void fraction data are presented for different inlet liquid temperatures with mass flow rate, pressure and wall heat rate maintained relatively constant. In Figure 2, the line-averaged void fraction data scans acquired using the GDS at X/D_h = 106.5 are presented. For average inlet temperatures (T_{in}) of 43.3, 48.9 and 54.4 ^{o}C, the void fraction profiles are peaked in the near-wall region with comparatively little variation over the range $0.40 \leq Z/t \leq 0.85$. Conversely, for the test condition with the highest inlet temperature of 65.6 ^{o}C, the void fraction data display a transition to a center-peaked profile that is somewhat skewed toward the heated wall. This profile is similar to that previously observed in low mass flux adiabatic flows (Trabold *et al.*, 1997), wherein it was conjectured that the shape of the profile is related to the fact that dispersed vapor volumes occupy nearly the entire duct spacing. This would most likely be the case for T_{in} = 65.6 ^{o}C where the local void fraction at the GDS measurement position was relatively high and the flow was saturated, allowing bubble coalescence to more readily occur.

The local void fraction data acquired with the HFA probe are given in Figure 3. As outlined in Section 2, since the HFA probe could only be moved from Z/t = 0.06 to 0.50, these data were obtained by first heating Windows 1 and 2 and then heating Windows 5 and 6. Hence, in the latter configuration, the probe scan extended from Z/t = 0.50 to 0.94 from the heated wall. Due to this test procedure, the local void fraction profiles display a slight mismatch at the Z/t = 0.50 measurement location; however, the average difference of 0.016 in void fraction was considered quite acceptable in light of the difficulty involved in reestablishing exactly the same thermal-hydraulic condition at low mass flux. In particular, it was necessary to wait as long as eight hours for the flow to reach a thermal equilibrium after the initiation of window heating.

The HFA data display roughly the same trends as described above for the GDS results. As expected, the HFA void fractions are somewhat higher because the probe is located at the center of the transverse (Y) test section dimension at a streamwise distance of X/D_h = 119.4. The net heat addition at this location is greater than at the GDS measurement plane which is located at a streamwise distance of X/D_h = 106.5. The four lowest inlet temperature runs all have maximum void fractions in the range Z/t = 0.06 to 0.08, while a center-peaked profile (also slightly skewed toward the heated wall) was observed for T_{in} = 65.6 ^{o}C. Several additional features of these flows are discernible from the HFA results. For instance, with an inlet temperature of 32.2 ^{o}C, the two-phase region extends only to Z/t = 0.15 from the heated wall. When the inlet temperature is increased to 43.3 and 48.9 ^{o}C, the two-phase region is found to extend over the entire cross-section of the duct. In addition, the data for T_{in} = 48.9 ^{o}C indicate the presence of a secondary void fraction peak around Z/t = 0.85. This is not unexpected since as the inlet subcooling is decreased (inlet temperature increased) the bubbles entering the subcooled liquid away from the heated wall condense at a slower rate. Consequently, on average, a greater population of smaller bubbles exist throughout the duct away from the heated wall. These smaller bubbles tend to be spherical, and it has been shown by many investigators (e.g., Nakoryakov *et al.*, 1986; Serizawa *et al.*, 1987; Liu and Bankoff, 1990) that spherical bubbles in a shear flow will experience a transverse (lift) force that drives them towards the duct walls. Consequently, as long as the bubbles do not condense, void peaking will occur in the vicinity of the unheated wall.

Bubble Frequency

Bubble frequency (f_b) measurements provide important information for understanding the local two-phase flow structure. For a given average void fraction, bubble frequency data offer some insight into the magnitude of the bubble size (discussed in detail below), the bubble size distribution and interfacial area concentration. In Figure 4, the local bubble frequency data are illustrated for subcooled boiling flow runs with varying inlet liquid temperature. For the four lowest values of T_{in}, the profiles are similar to those measured for the local void fraction, with peaks occurring near the heated wall and secondary peaks in the vicinity of the unheated wall for some test conditions. For T_{in} = 65.6 ^oC, the f_b data follow a trend which is different than that for the associated α data. A broad bubble frequency maximum is measured near the heated wall and a secondary f_b maximum is observed near the unheated wall, with a local minimum occurring at the duct center line. The higher frequency near the heated surface is likely due to the interaction with the HFA probe of newly "born" vapor bubbles. Since the local void fraction for this run reaches a maximum near Z/t = 0.50, bubble coalescence must take place in the direction normal to the heated surface to account for the reduction in measured frequency. Bubble dispersion and the lift force possibly are responsible for the secondary peak near the unheated wall.

Vapor Velocity

Vapor velocity (V_g) data were acquired using both laser Doppler velocimeter (LDV) and dual-sensor hot-film anemometer (HFA) instrumentation at streamwise positions of $X/D_h = 114.2$ and 119.4, respectively. The vapor velocity results for four subcooled boiling test conditions are presented in Figure 5. A polynomial regression line was fit to the HFA data to illustrate the overall data trends. With the exception of the data for $T_{in} = 65.6\ ^{o}C$ which shows some scatter in the LDV data at large Z/t, the results from the two measurement techniques agree reasonably well, especially near the duct center plane. Unlike the very steep gradients observed in the void fraction profiles, all of the velocity data scans are quite flat with broad maxima near the center of the test section. Additionally, it is observed that the location of maximum local vapor velocity does not coincide with the location of maximum void fraction. This behavior seems to be consistent with the measurements of subcooled boiling flow reported by Velidandla *et al.* (1995). These authors also reported difficulty in applying a dual-sensor optical probe for measurement of local velocity via the cross-correlation method, due to low bubble number densities away from the heated wall. This same limitation was only observed in the present experiments with the HFA probe for conditions of highly subcooled liquid at the test section inlet.

Figure 5 illustrates the rather weak dependence of local vapor velocity on the liquid inlet temperature over the range $43.3 \leq T_{in} \leq 54.4\ ^{o}C$. For these conditions, the maximum measured velocity was between 0.34 and 0.40 m/s. However, a nearly twofold increase in V_g was measured upon increasing T_{in} to 65.6 ^{o}C. For this condition, simultaneous high speed video footage results show that the slug flow regime is the predominant flow structure. By contrast, for the cases between $T_{in} = 43.3$ and $54.4\ ^{o}C$, the high speed video record showed that the flow structure was primarily bubbly. For the low mass flow rate investigated in the present test sequence, vapor phase buoyancy dominates the structure of the flow, and vapor velocity becomes more dependent on the individual bubble rise velocity which significantly increases in going from the bubbly to the slug flow regime (Vassallo *et al.*, 1995).

Bubble Diameter and Chord Length Distributions

As shown by previous investigators (e.g., Spindler *et al.*, 1988; Trabold *et al.*, 1994), the characteristic size of the dispersed vapor or liquid field can be estimated from local measurements of void fraction, velocity and frequency through the relation:

$$d_b = 1.5\frac{V_g \alpha}{f_b} \tag{6}$$

where α, f_b and V_g are obtained by the HFA probe. A similar expression was obtained experimentally by Lim and Agarwal (1990) and can also be obtained analytically by a method based on Galaup's (1976) work.

Equation (6) is based upon the following assumptions:

- All bubbles are spherical. For ellipsoidal bubbles, the factor $E^{2/3}$ would be included in the denominator of Eq. 6, where E is the ratio of the minor to major bubble axes (Trabold *et al.*, 1994).

- The bubbles are randomly distributed about the HFA probe. This assumption strictly applies only when the measured bubbles are quite small. Once the bubbles reach the size of approximately half the test section thickness this assumption is no longer valid. Additionally, the assumption does not apply near the wall, since the chord length along which the bubble intersects the HFA sensor is not a random variable (Kalkach-Navarro *et al.*, 1992).

- The non-streamwise (i.e., Y and Z dimensions) vapor velocity components are negligible. This is a reasonable assumption for most test conditions along a Z-dimension scan since the HFA probe should not be exposed to any recirculating flow regions.

Mean bubble diameters, based on Eq. 6, were calculated for experiments conducted with inlet liquid temperatures of 43.3, 48.9, 54.4 and 65.6 °C. These data are first presented in Figure 6 without an assessment of the applicability of the assumptions stated above. Data for T_{in} = 32.2 °C are not available due to the inability to obtain vapor velocity data at this condition. For the three lowest inlet liquid temperatures, the data profiles are nearly flat across the narrow test section dimension, and the magnitude of the diameter is only weakly dependent on the level of inlet subcooling. Since the mean diameters calculated for these conditions are all considerably less than the duct spacing, it is likely that the vapor field is primarily comprised of dispersed spherical bubbles, with little or no coalescence occurring in the direction normal to the heated wall. When the inlet temperature is increased to 65.6 °C, the structure of the flow changes dramatically. The maximum calculated diameter exceeds 3 mm and there is a large gradient from the near-wall to centerline region. Clearly, for this condition, "planar" bubbles exist which are confined by the test section walls, and the transition from pure bubbly to bubbly-slug flow has been reached.

Although the results appear reasonable and consistent with the flow structure, the fact that a significant population of planar bubbles exists at the highest inlet temperature limits the quantitative utility of the bubble size data. Planar bubbles are confined by the test section walls and are, therefore, "flat" in the width (Y) direction and curved in the spacing (Z) direction with a radius of curvature roughly equal to half the spacing dimension. The presence of planar bubbles also infers that a significant number of spherical bubbles exist that are constrained by or interact with the test section walls; consequently, they are not randomly distributed about the HFA probe as assumed. In view of this, the bubble size data are presented for selected runs in a more basic form in terms of the measured chord length distributions. In this way, a better understanding of the characteristic dispersed vapor size is obtained.

The bubble chord length distributions for T_{in} = 43.3 and 65.6 °C are presented in Figure 7. In each figure, the chord length distributions measured at Z/t = 0.05, 0.25, 0.50, 0.75 and 0.94 from the heated wall are presented. For T_{in} = 43.3, the chord lengths measured at all locations are generally less than 1 mm, with the exception of a small population of larger chord lengths at Z/t = 0.05. As Z/t increases from the heated wall toward the duct centerline, the concentration of bubbles clearly decreases, but the mean chord length does not change significantly. At Z/t = 0.75, a very small population of bubbles with chord lengths less than 0.6 mm is present. This finding is not surprising, since the associated local void fraction and bubble frequency are only 0.013 and 13.8 Hz, respectively.

A striking change in the structure of the two-phase flow field occurs when the liquid inlet temperature is increased to 65.6 °C. Although small (i.e., < 1 mm) chord length samples still dominate the distributions, very large bubbles are present. At Z/t = 0.25, 0.50 and 0.75 from the heated wall, many measured bubble chord lengths lie between 3 and 6 mm[1]. These bubbles fill the entire narrow test section dimension and are becoming elongated in the streamwise (X) dimension.

Liquid Temperature

1. The illustrated chord length distributions were limited to 6 mm to make the data easily presentable. For T_{in} = 65.6 °C at Z/t = 0.25, 0.50 and 0.75, the maximum measured chord lengths were 18.8, 22.1 and 13.0 mm, respectively.

When the HFA equipment is operated as a constant current anemometer (CCA), the small platinum sensor acts as a resistance thermometer; hence, it can be used to measure local fluid temperature. In the current test program, the HFA probe was used to measure temperature along a narrow (Z) test section dimension scan. The experimental procedure was to first operate the HFA probe in a constant temperature mode for acquisition of void fraction, frequency and velocity data, and then reverse the data scan with the anemometer operated in constant current mode.

Previous researchers have used both HFA sensors and microthermocouples to measure local fluid temperature in subcooled boiling flows of refrigerant fluids. Hasan *et al.* (1991) demonstrated that a microthermocouple and hot-film sensor (TSI model 10A; the same as used in the current experiments) gave similar temporal response, with an effective time constant of approximately 7 ms. Since in the present investigation, the residence times of bubbles in subcooled flows generally ranged from 1 to 5 ms, it may be concluded that the HFA sensor measured only the liquid phase temperature.

Prior to the acquisition of HFA local temperature measurements, it was necessary to calibrate the constant current anemometer by relating the output voltage to a known temperature. This was accomplished by introducing single phase subcooled liquid R-134a and measuring the centerline temperature at the inlet and exit using thermocouple rakes at X/D_h = 2.6 and 254 (Figure 1). The local liquid temperature at the HFA measurement location was determined by assuming a linear streamwise temperature gradient. As expected, there was a linear relationship between CCA output voltage and local liquid temperature. However, during the course of the subsequent subcooled boiling temperature measurements, it was realized that the output voltage was drifting. This may have resulted from small amounts of loop contaminants accumulating on the platinum film sensor. Similar drift problems due to fluid contamination are commonly encountered when hot-film devices are used for liquid velocity measurements. The other two-phase flow measurements of interest (void fraction, frequency, velocity) are based on the voltage differences arising from vapor and liquid phases in constant temperature mode; drift has no perceptible effect on these measurements.

To account for the effect of drift on temperature measurements, the calibration procedure was repeated at the end of the test sequence (14 days after the initial calibration). To facilitate normalization of the CCA output voltage data, the magnitude of the drift was calculated as a function of the number days from the initial calibration. The CCA output voltage was then corrected for drift for each data run. The uncertainty associated with the calibration procedure has been incorporated into an overall liquid temperature measurement uncertainty (Section 4).

The local liquid temperature data for the five subcooled boiling test conditions are given in Figure 8. As for the data discussed previously, there is a slight discontinuity in the profiles at Z/t = 0.50 due to the separate runs made first with Windows 1 and 2 heated and then with Windows 5 and 6 heated. The average temperature difference at Z/t = 0.50 for these five runs was 0.4 °C. For the lowest temperature case (T_{in} = 32.2 °C), the measurement nearest the heated wall at Z/t = 0.06 was about 11 °C below the saturation temperature (nominally 76 °C at P = 2.4 MPa), and there is a relatively large temperature gradient across the half of the flow field nearest the heated wall. Beyond about Z/t = 0.70, the temperature remains constant at about 21°C below the saturation temperature. Upon increasing the inlet liquid temperature to 43.3 °C, there is an approximately 8 °C increase in the near-wall temperature measurement, and a significant flattening of the profile. In fact, the temperature gradient is confined almost entirely within a region extending to only Z/t = 0.30 from the heated wall. As one would expect, with further increases in

T_{in}, the local liquid temperature data approach the saturation temperature.

4. Contribution of R-134a Data to Bubbly Flow Modeling

The R-134a test facility has provided a comprehensive database in support of the effort to develop a calculation method for multi-dimensional, heated two-phase flows (Siebert, *et al.* (1995)). The quantities and/or processes that need to be modeled to describe the bubbly flow regime include lift and wall forces, turbulent dispersion, bubble induced turbulent viscosity, bubble size, wall heat partitioning, interfacial heat transfer, bubble drag, interfacial area density and bubble nucleation inception. Data obtained to qualify the bubbly flow models based on both "separate effects" and "integrated effects" include local void fraction, local liquid temperature, local bubble frequency, local vapor velocity, local bubble size and duct pressure drop. In addition, high speed videos obtained for each of the experimental conditions proved invaluable in determining the extent of the bubbly flow regime for the conditions investigated. Calculations to qualify the bubbly flow models are described below. First, it is useful to characterize the bubbly flow regime in narrow duct geometries.

For duct geometries with high aspect ratios (> 10), the bubbly flow regime is comprised of two subregimes depending on the bubble size. These subregimes are referred to as the "spherical" and "planar" bubbly regimes. The spherical bubbly regime exists when the bubble equivalent diameter is less than or equal to the duct spacing. The planar bubbly regime is initiated once the bubble equivalent diameter exceeds the duct spacing, at which point the bubble becomes disk-like or planar. Of course, this transition as described is an idealization, since elongated (in the width-wise direction) bubbles will exist that are neither spherical nor planar, but whose equivalent diameter exceeds the duct spacing.

The planar bubbly regime ends and the planar slug regime begins when the characteristic dimension of the bubble reaches a critical size which can be defined either from stability considerations or rise velocity (drag) considerations. Based on the work of Maneri (1995), the drag characteristics of spherical and planar bubbles are mechanistically the same. What differentiates these subregimes are the non-drag forces such as lift and wall forces, the interfacial area density and, under heated conditions, the interfacial heat transfer characteristics.

Initial calculations of void fraction, bubble size, vapor velocity and liquid temperature were performed for those experimental cases where only the spherical bubbly flow regime was observed in the videos. The two-fluid calculational model was constructed as two-dimensional, with no variation in the width dimension. Since the local data were obtained in the spacing dimension at a single midwidth location, the experimental duct-average void fraction at the measurement location was not matched in the model. Furthermore, since these are initial calculations, only estimates of the heat losses were used to adjust the input power. The calculations are compared with the experimental measurements in Figures 9 through 11 for T_{in} = 43.3, 48.9 and 54.4 ^{o}C. While these comparisons are generally good, some trends do emerge. First, the local void fraction near the heated wall tends to be underpredicted and the liquid temperature overpredicted as the liquid subcooling is increased. This suggests a deficiency in the wall heat partition model incorporated in the calculational method. Second, the bubble size is consistently overpredicted, although the experimental bubble size tends to be biased on the low side because it is an average based on the number of bubbles, although a volume-weighted average may be more appropriate. Finally, it was necessary to vary the wall force coefficients with bubble size in order to obtain the degree of comparison shown.

5. Summary of Results

Subcooled boiling flow experiments were conducted in a R-134a test facility. The experiments were run with a fixed nominal system pressure, mass flow and wall heat rate, and various inlet liquid temperatures. The void fraction data, acquired with both gamma densitometer system (GDS) and hot-film anemometer (HFA) instrumentation, displayed the existence of a significant peak near the heated wall in the spacing dimension scans, especially for large inlet subcooling. Conversely, for only slightly subcooled liquid inlet conditions, the void fraction profiles were center-peaked, with some skewness toward the heated wall. The shape of the bubble frequency scans extracted from the HFA output voltage were generally consistent with the void fraction data trends. However, local bubble velocity data obtained with both a laser Doppler velocimeter (LDV) and dual-sensor HFA showed broad maxima near the duct centerline. There was no apparent correlation between the locations of maximum void fraction and bubble velocity. By running the HFA probe in constant current mode, the local liquid temperature was also measured. Significant gradients near the heated wall were observed for large subcooling. At higher inlet temperatures, the local temperature profiles became flat as they approached the saturation temperature.

Bubble sizes were inferred from local measurements of void fraction, bubble frequency and vapor velocity. These data, combined with the measured bubble chord length distributions, illustrate the transition from pure bubbly flow to bubbly-slug flow in which the dispersed vapor volumes span the entire duct spacing and become elongated in the streamwise dimension.

The experimental data for cases where only the spherical bubbly flow regime was observed in the high speed video record were compared with two-fluid computer code calculations of void fraction, bubble size, vapor velocity and liquid temperature obtained with an integrated set of bubbly flow models. The resulting comparisons, while generally good, indicate the need for further improvements in subcooled bubbly flow modeling.

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