CHAPTER 3

DATA REDUCTION

In the present study of heat transfer and bubble characteristics in the flow boiling of R-134a in a narrow annular duct, the gap bwtween the inner and outer circular pipes is varied from 0.2 to 2.0 mm with the refrigerant saturated temperature set at 10℃ and 15 ℃. The refrigerant mass flux ranges from 400 kg/m^{2}s to 700 kg/m^{2}s, the imposed heat flux from 0 to 55KW/ m^2, and inlet liquid subcooling from 0 to 6℃. A data reduction analysis is needed to calculate the flow boiling heat transfer coefficient from the raw data measured in the horizontal annular duct.

3.1 Flow Boiling Heat Transfer Coefficient

The imposed heat flux to the refrigerant flow in the annular duct is calculated on the basis of the total power input and the total outside heat transfer area of the inner pipe of the annular duct. The total power input is computed from the product of the measured voltage drop across the cartridge heater and the electric current passing through it. The imposed heat flux at the outside surface of the inner pipe is then evaluated from the relation

$$q = VI / A_s \tag{3.1}$$

where V and I respectively represent the measured voltage drop and current. Before the two-phase experiments, the total heat loss from the test section is evaluated by comparing the total power input from the power supply Q_s with the total heat transfer rate to the single-phase refrigerant flow in the duct expressed as $G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T$, where A_{cs} is the cross-sectional area of the annular duct and ΔT is the difference in the refrigerant temperature at the exit and inlet of the test section. The relative heat loss from the test section is defined as

$$\varepsilon = (Q_s - G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T) / Q_s \tag{3.2}$$

The results from this heat loss test indicate that the heat loss from the test section is generally less than 5% of the total power input and neglect this loss. Then, the average single-phase convection heat transfer coefficient over the entire heated surface is defined as

$$h_{r,l} = \frac{Q_n}{A_s \cdot (\overline{T}_w - T_{r,ave})} \tag{3.3}$$

and

$$T_{r,ave} = \frac{T_{r,i} + T_{r,o}}{2} \tag{3.4}$$

where Q_n is the net power input to the liquid refrigerant in the annular duct, which is estimated from the product $G \cdot A_{cs} \cdot c_{pr} \cdot \Delta T$, and $T_{r,ave}$ is the average of the measured inlet and outlet temperatures of the refrigerant flow through the test section, which is taken as the average bulk liquid refrigerant temperature. Note that $\overline{T_w}$ denotes the average outside surface temperature of the inner pipe measured at the selected thermocouple locations. The outside surface temperature at each thermocouple location is deduced from the measured inside surface temperature of the inner pipe by subtracting the radial temperature drop due to the radial heat conduction in the pipe wall. Thus we have

$$T_w = T_{w,i} - Q_n \frac{\ln(D_o / D_i)}{2\pi k_w l} \tag{3.5}$$

In the two-phase test, the single-phase heat loss may be not suitable for estimating the two-phase heat loss. We estimate the heat loss from the adiabatic cotton by measuring the temperatures of thermal insulation material surface and ambient temperature. The measured temperature drop is all less than 2^oC and the two- phase heat loss can be neglected. The local subcooled and saturated flow boiling heat transfer coefficients at a given axial location are respectively defined as

$$h_{r,sub} = \frac{Q_n / A_s}{(T_w - T_r)} \tag{3.6}$$

and

$$h_{r,sat} = \frac{Q_n / A_s}{(T_w - T_{sat})} \tag{3.7}$$

Here T_r is the local mean liquid refrigerant temperature and is estimated by assuming that it varies linearly in the axial direction.

3.2 Flow Boiling Bubble characteristics

To explore the bubble characteristics, we move further to estimate the average bubble departure diameter and frequency and the average active bubble nucleation site density on the heating surface for the cases dominated by the bubbly flow and the mean axial speed of big bubbles for the cases prevailed by the slug flow in the duct from the images of the boiling flow stored in the video tapes. Specially, for a given case the average bubble departure diameter is determined by measuring the diameters of departing bubbles on a small heated surface area and it is defined as

$$d_p = \frac{\sum d_p}{N} \tag{3.8}$$

where N is number of measured bubbles. In this determination of d_p, N is not less than 3. Similarly, the mean bubble departure frequency is estimated by averaging the number of bubbles departing from the heated surface n_b over a certain period of time t_b at a few bubbles departing sites N. Thus f_b is defined as

$$f_b = \frac{\sum f_b}{N} = \frac{\sum n_b / t_b}{N} \tag{3.9}$$

here N is also greater than 3. The average active nucleation site density is estimated by counting the number of bubble nucleation sites n_s over a small heated surface area A_s, and defined as

$$N_{ac} = n_s / A_s \tag{3.10}$$

The mean axial speed of the big bubbles is estimated by measuring the axial speed of the big bubbles and by averaging the above results. Hence

$$v_s = \frac{\sum v_s}{N} \tag{3.11}$$

Here N is the number of big bubbles whose speeds are measured and it is also greater than 3.

3.3 Uncertainty Analysis

Uncertainties of the heat transfer coefficients are estimated according to the procedures proposed by Kline and McClintock for the propagation of errors in physical measurement [69]. The results from this uncertainty analysis are summarized in Table 3.1.

Table 3.1 Summary of the uncertainty analysis

Parameter	Uncertainty
Annular pipe geometry	
Length, width and thickness (%)	±1.0%
Gap size (%)	±5.0%
Area (%)	±2.0%
Parameter measurement	
Temperature, T (℃)	±0.2
Temperature difference, △T (℃)	±0.28
System pressure, P (MPa)	±0.002
Mass flux of refrigerant, G (%)	±2
Single-phase heat transfer on inner pipe	
Imposed heat flux, q (%)	±4.5
Heat transfer coefficient, $h_{r,l}$ (%)	±12.5
Subcooled flow boiling heat transfer on inner pipe	
Imposed heat flux, q (%)	±4.5
Heat transfer coefficient, $h_{r,sub}$ (%)	±14
Saturated flow boiling heat transfer on inner pipe	
Imposed heat flux, q (%)	±4.5
Heat transfer coefficient, $h_{r,sat}$ (%)	±14.5