CHAPTER 5

SUBCOOLED FLOW BOILING OF R-134a IN A HORIZONTAL NARROW ANNULAR DUCT

The present data for the subcooled flow boiling heat transfer and associated bubble characteristics of refrigerant R-134a flowing in the horizontal narrow annular duct are inspected in this chapter. The experiments were conducted for the refrigerant mass flux G varying from 500 to 600 kg/m^{2}s, imposed heat flux q from 0 to 50 kW/m^2, inlet liquid subcooling ΔT_{sub} from 3 to 6℃, duct gap δ from 0.2 to 2.0 mm (corresponding to the hydraulic diameter D_h from 0.4 to 4.0 mm) and the refrigerant saturated temperature T_{sat} from 10 to 15 ℃ (corresponding to the R-134a saturated pressure from 414 to 488 kPa). Attention will be focused on how the flow boiling heat transfer and associated bubble behavior are affected by the size of the duct.

In what follows, the heat transfer characteristics in the R-134a subcooled flow boiling are expressed in terms of the boiling curves, which are the plots for the imposed heat flux q versus the wall superheat (T_w-T_{sat}) for various flow and thermal conditions. Moreover, selected experimental data and flow photos from the present study are presented to illustrate the subcooled flow boiling heat transfer coefficient and associated bubble characteristics in the boiling flow including the mean bubble departure diameter, departure frequency and active nucleation site density. Finally, empirical correlation equations are proposed to correlate the present data for the subcooled flow boiling heat transfer coefficient, average bubble departure diameter, bubble departure frequency and active nucleation site density.

5.1 Subcooled Flow Boiling Curves

The effects of the refrigerant mass flux, inlet subcooling, saturated temperature, and gap size of the duct on the subcooled flow boiling characteristics at the middle axial location (z =80mm) of the annular duct are shown in Figures 5.1-5.7 by presenting the boiling curves for various G, ΔT_{sub}, T_{sat} and δ.

First, the effects of the refrigerant mass flux on the subcooled flow boiling curves are illustrated in Figures 5.1 and 5.2. The results indicate that for a given boiling curve, at low imposed heat flux the temperature of the heated wall is also below the saturated temperature of R-134a and heat transfer in the duct is completely due to the single-phase forced convection. As the imposed heat flux is raised gradually, the heated wall temperature increases slowly to become above T_{sat} at a certain q and we have a positive wall superheat ΔT_{sat} (= $T_w - T_{sat}$). When the positive wall superheat reaches ceratin critical level, a smaller increase in q causes boiling to suddenly appear on the heated wall and the heated wall temperature drops immediately to a noticeable degree. Thus there is a significant temperature undershoot during the onset of nucleate boiling (ONB) in the subcooled flow boiling. Note that the temperature undershoot can be as high as 8.5℃ for G =400 kg/m^{2}s, δ=2.0 mm, T_{sat} =15 ℃ and ΔT_{sub} =3℃ (Figure 5.1(a)). Note that the refrigerant mass flux substantially affects the magnitude of the temperature undershoot during ONB for δ=1.0 & 2.0 mm. But for δ=0.5 & 0.2 mm the influence is slight. Specifically, at a lower mass flux the temperature undershoot is somewhat larger. Besides, a slightly higher wall superheat is needed to initiate the nucleate boiling for a lower G for δ=1.0 & 2.0 mm but for δ=0.2 mm we have opposite trend (Figures 5.1 & 5.2). Beyond the ONB a small rise in the wall superheat causes a large increase in the wall heat transfer rate and the slopes of the boiling curves are much steeper than those for the single-phase convection. Checking with the data in Figures 5.1 and 5.2 further reveals that beyond ONB the refrigerant mass flux exhibits rather slight effects on the boiling curves. But in the single-phase region the heated wall temperature is somewhat affected by the refrigerant mass flux. Note that at a higher mass flux the imposed heat flux needed to initiate ONB is larger. Next, the effects of the inlet liquid subcooling on the subcooled boiling curves are shown in Figures 5.3 and 5.4. The results indicate that during ONB a substantial increase in the temperature undershoot occurs when the inlet liquid subcooling is raised from 3℃ to 6℃. Thus a higher wall superheat is needed to initiate the boiling on the heated surface for a higher ΔT_{sub}. It is also noted that the boiling curves are not affected to a noticeable degree by the subcooling in the nucleate boiling region. It is evident that a higher imposed heat flux is needed to initiate boiling on the heated surface for a higher inlet liquid subcooling for a given G. However, in the single-phase region a higher liquid subcooling results in a higher heat transfer from the wall so that at a given wall superheat the imposed heat flux is significantly higher for a higher liquid inlet subcooling. This reflects the fact that at a given wall superheat the temperature difference between the wall and the bulk liquid increases with the inlet subcooling.

Then, the effects of the duct gap on the boiling curves are shown in Figure 5.5. It is found that a substantial reduction in the temperature undershoot during ONB occurs when the duct gap is reduced from 2.0 to 0.2 mm. Besides, a significantly lower wall superheat is needed to initiate the boiling on the heated surface for a smaller δ. This mainly results from the fact that for given G, q, Tsat and ΔT_{sub} the mass flow rate through the duct is low for a smaller δ. Thus the axial temperature rise of the refrigerant flow is larger for a smaller δ, which in turn results in a required lower wall superheat at ONB. It is also noted that the boiling curves are shifted to the left in the nucleate boiling region as the gap size is decreased, which indicates that the boiling heat transfer in the duct with a small gap is better. Moreover, a substantially lower imposed heat flux is needed to initiate boiling on the heated surface for a duct with a smaller gap size for a given G. However, in the single-phase region the effect of δ on the boiling curves is relatively slight except for δ=0.2 mm. Finally, the effects of the saturated temperature on the subcooled boiling curves are shown in Figures 5.6 and 5.7. The results clearly manifest that the temperature undershoot during ONB is larger for a lower T_{sat}. This is attributed to the fact that at a lower T_{sat} the surface tension of R-134a is higher (Table 2.2). At the higher surface-tension the liquid refrigerant is more difficult to completely flood the cavities, which in turn retards the bubbles to nucleate from the cavities on the heated surface. Thus a higher wall superheat is needed to activate the cavities. This trend is more prominent for the smaller δ (Figure 5.7). Otherwise, the effect of T_{sat} on the boiling curves is rather slight. Finally, the relation between the imposed heat flux and wall superheat temperature at the incipient boiling is shown in Figure 5.8. The results indicate that the wall superheat at ONB increases with q_{ONB} for most cases.

5.2 Subcooled Flow Boiling Heat Transfer Coefficient

The effects of the refrigerant mass flux, inlet subcooling, saturated temperature, and duct size on the subcooled flow boiling heat transfer coefficient measured at the middle axial location (z =80 mm) of the annular duct are shown in Figures 5.9-5.15 by presenting the subcooled flow boiling heat transfer coefficients against the imposed heat flux for various G, ΔT_{sub}, T_{sat} and δ.

The results in Figures 5.9-5.15 indicate that the increase of the subcooled flow boiling heat transfer coefficient with the imposed heat flux is relatively significant for all cases. We also note from Figures 5.9 and 5.10 that the refrigerant mass flux exhibits a slight effect on the boiling heat transfer coefficient for δ =1.0 & 2.0 mm. But for the smaller ducts with δ = 0.5 & 0.2 mm and at a high imposed heat flux, a higher h_r results for a higher *G*. This is attributed to the change from the bubbly flow to slug flow under these conditions, as that observed in saturated flow boiling examined in Chapter 4. Next, the boiling heat transfer is much better for a smaller inlet liquid subcooling especially at low imposed heat flux (Figures 5.11 and 5.12). For instance, at q =40 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and δ = 2.0 mm, h_r for ΔT_{sub}=3℃ is about 22% higher than that for ΔT_{sub}=6℃ (Figure 5.11(a)). It is of interest to note from the data in Figure 5.13 that reducing the duct size can effectively enhance the subcooled boiling heat transfer in the duct. For the specific case with q =40 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and ΔT_{sub}=3℃, h_r for δ = 0.2 mm is about 43% higher than that for δ = 2.0 mm (Figure 5.13(a)). This is considered to mainly result from the fact that in the narrower duct the radial gradient of the liquid axial velocity is larger, which in turn exerts higher shear force on the bubbles nucleated from the wall and causes them to depart from the heating surface at a higher rate. Besides, in the smaller ducts with δ = 0.5 & 0.2 mm the slug flow prevails and the boiling heat transfer in the ducts is much better than that for the bubbly flow. Finally, the refrigerant saturated temperature shows slight effects on the boiling heat transfer coefficient (Figures 5.14 and 5.15).

5.3 Bubble Behavior in Subcooled Flow Boiling

When the wall superheat exceeds the incipient boiling temperature, it is noted from the experiment that tiny bubbles form on the active cavities and grow continuously until they depart from the heating surface. The bubble growth and departure are somewhat regular and the bubbles are nearly spherical in shape at a low imposed heat flux. The bubble formation, growth and detachment processes in the duct obviously depend on the flow and thermal conditions and on the geometry of the cavities.

To illustrate the effects of the gap size on the bubble behavior in the entire duct, photos of the R-134a boiling flow from the side and top views covering the whole duct at G=500 kg/m^2s, T_{sat}=15℃, and q=40 kW/m^2 are shown in Figures 5.16-5.19. Due to the difference in the buoyancy effect in different parts of the duct the flow in the lower portion of the duct is heated from above and hence is thermally stable. This in turn results in a lower convection heat transfer coefficient and high heating surface temperature. This higher T_w causes the earlier inception of the bubbles from the surface in the lower portion of the duct. The results in Figures 5.16 and 5.17 clearly indicate that the onset of nucleate boiling first appears in the lower part of the heating surface for the duct with δ = 1.0 & 2.0 mm. But for the smaller ducts with δ = 0.5 & 0.2 mm the locations of ONB depend only weakly on the circumferential position of the heating surface (Figures 5.18 and 5.19). This is attributed to the much weaker buoyancy effects in the smaller duct in view of the fact that the buoyancy is directly proportional to δ^3 (Gr/Re^2 = 6.96×10^{-2} for δ = 0.5 mm and Gr/Re^2 = 1.11×10^{-2} for δ = 0.2 mm). Besides, the bubble motion in the upper and lower parts of the duct are significantly different. In the lager ducts with δ = 1.0 & 2.0 on the upper part of the heated surface, bubbles are noted to either lift off directly from the active nucleation sites or slide for a short distance, and then accelerate to a greater speed over that of the surrounding bulk liquid flow. Collision and coalescence of bubbles are insignificant. But in the low part of the duct, the bubbles departing from the nucleation sites slide circumferentially along the heating surface. And the collision and coalescence of bubbles are rather intense except at the low imposed heat flux. The bubble sliding appears to be responsible for heat transfer augmentation from the heating surface. In the smaller ducts with δ = 0.5 & 0.2 mm, in the lower part of the ducts the bubbles are discrete and small and the bubbly flow prevails. However, in the upper part of the duct coalescence of bubbles are very intense and big bubbles dominate, resulting in the slug flow (Figures 5.18 and 5.19).

The characteristics of bubbles in the subcooled flow boiling in a small section around the middle axial location (z =80 mm) of the annular duct are illustrated in Figure 5.20 by showing the side view photos taken from the cases at δ =0.5 mm for different imposed heat fluxes, refrigerant mass fluxes, inlet subcoolings and saturation temperature. First of all, the bubbles at the low q of 20 kw/m^2 for the case at T_{sat}=15℃, G=500 kg/m^2s, δ =0.5 mm and $\triangle T_{sub}$=3℃ can be seen from Figure 5.20(a). Checking with the video tapes recording the bubble motion reveals that the bubbles form and grow at the active nucleation sites while they experience a short period of stationary growth to a certain size. And then the bubbles detach from the heating surface and accelerate into the subcooled liquid. As the imposed heat flux is increased slightly to q=30 kW/m^2 (Figure 5.20(b)), more bubbles are nucleated and bubbles are observed to collide and coalesce occasionally. The coalescence bubbles rise faster than the tiny bubbles due to the larger buoyancy force associated with them. As the heat flux is raised to q=40 kW/m^2 (Figure 5.20(c)), coalescence of the bubbles occurs more frequently and irregularly. At even higher heat fluxes the bubble nucleation density becomes too large to visually distinguish the individual nucleation sites. In general, increasing the imposed heat flux directly provides more energy to the cavities and more cavities on the heating surface can be activated. Besides, the bubble departure frequency also increases substantially with the imposed heat flux. Moreover, the bubble departure diameter increases slightly with the imposed heat flux due to the higher wall superheat. Note that the buoyancy and shear force cause the bubbles to lift off the heating surface, but the surface tension trends to keep bubble on the heating surface. When the wall superheat increases, the buoyancy force increases in the upper part of the heating surface but the surface tension decreases. Next, Figures 5.20(d)~(f) show the bubble characteristics around the middle axial location affected by the refrigerant mass flux by presenting the photos for the higher mass of 600 kg/m^2s but at the same q, T_{sat}, δ and $\triangle T_{sub}$ as that for Figures 5.20(a)-(c). A close inspection of the corresponding photos and video tapes at different mass fluxes reveals that at a higher G the higher speed of the subcooled liquid tends to condense the bubbles intensively in an earlier stage of the bubble growth. This mechanism results in a lower bubble departure frequency and smaller bubble departure diameter. Thus the partial nucleate boiling dominates in the flow at a high G and low q. But the higher liquid speed for a higher G can sweep the bubbles away from the cavities in an easier way resulting in a higher bubble generation frequency. Besides, at a higher G the liquid temperature is lower for a given imposed heat flux at a given ΔT_{sub}. Hence less bubble nucleation is activated on the heated wall and the bubble nucleation density is lower. Then, the effects of the inlet liquid subcooling on the bubble characteristics around the middle axial location are illustrated by comparing the photos shown in Figures 5.20(g)~(i) with Figures 5.20(a)~(c) respectively for $\triangle T_{sub}$=6℃ and 3℃ at q=20~40 kW/m^2, G=500 kg/m^{2}s, δ=0.5 mm and T_{sat}=15℃. In general, the mean bubble diameter is larger at a lower liquid subcooling. The larger bubbles are due to the weaker vapor condensation and more bubble coalescence at a lower inlet liquid subcooling. In addition, an increase in the inlet subcooling results in a reduction in the bubble departure frequency. This is due to the fact that at a higher inlet liquid subcooling the liquid R-134a temperature at the subcooled liquid-vapor interface is relatively low compared to the hot heated surface. Hence at the same imposed heat flux, the wall superheat is not high enough to sustain the continuing growth of the bubbles when the inlet liquid subcooling is high. Besides, the active nucleation sites decrease with increasing inlet subcooling. The above results clearly reveal that the significant influences of the liquid inlet subcooling on the bubble characteristics are associated with the more prominent vapor condensation at the subcooled liquid refrigerant-bubble interface for a higher inlet subcooling. This significant vapor condensation can impede the growth of the bubbles to a noticeable degree. Finally, the effects of the refrigerant saturation temperature on the bubble characteristics around the middle axial location are illustrated by comparing the photos shown in Figures 5.20(j)~(l) with Figures 5.20(a)~(c) respectively for T_{sat} =10℃ and 15℃ at q=20~40 kW/m^2, G=500 kg/m^{2}s, δ=0.5 mm and $\triangle T_{sub}$=3℃. In general, bubbles are larger at a lower saturation temperature. The larger bubbles are due to the higher surface tension and more bubble coalescence at a lower saturation temperature. In addition, an increase in the R-134a saturation temperature results in an increase in the bubble departure frequency. Besides, the active nucleation sites increase with increasing saturation temperature.

The bubble characteristics around the middle axial location affected by the duct size are shown in Figures 5.21(a)~(l). Due to the space limitation bubbles are squeezed and deformed in the smaller ducts. Besides, more large bubbles are formed from the coalescence of the small bubbles in the smaller ducts particularly at a high imposed heat flux. Note that in this middle portion of the small ducts the slug flow prevails, which is characterized by the liquid slugs interdispersed with big bubbles along with some smaller bubbles in the liquid slugs, as evident from the photos in Figures 5.21(i), (k) and (l). The flow patterns observed in the present test for $0.22 \le N_{conf} \le 2.22$ and $1\times10^{-4} \le Bo \le 7.3\times10^{-4}$ are summarized in a flow regime map shown in Figure 5.22.

To be quantitative on the bubble characteristics, we move further to estimate the mean bubble departure diameter and frequency and the active nucleation site density in the bubbly flow and mean speed of the big bubbles in the slug flow by carefully tracing the motion of the bubbles from the images of the boiling flow stored in the video taps. These quantitative data illustrating the bubble behavior are examined in the following. The effects of the refrigerant mass flux, inlet liquid subcooling, duct size and saturated temperature on the mean bubble departure diameter for the subcooled flow boiling of R-134a at the middle axial location (z =80 mm) of the annular duct are shown in Figures 5.23-5.29 by presenting the average bubble departure diameter against the imposed heat flux for various G, ΔT_{sub}, δ and T_{sat}. Note that the increase of the bubble departure size with the heat flux is very significant for all cases presented here. First, the effects of the refrigerant mass flux on the mean bubble departure diameter are shown in Figures 5.23 and 5.24. The results indicate that the average bubble departure diameter is slightly larger for a smaller refrigerant mass flux. For example, at q =30 kw/m^2, T_{sat} =15℃, ΔT_{sub} = 3℃ and δ = 0.5 mm, the average bubble departure diameter for G =500 kg/m^{2}s is only about 7 % larger than that for G =600 kg/m^{2}s (Figure 5.24(a)). Next, the effects of the inlet subcooling on the R-134a subcooled flow boiling average bubble departure diameter are shown in Figures 5.25 and 5.26. Note that the average bubble departure diameter is somewhat larger for a smaller liquid subcooling. For example, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and δ = 0.5 mm, the average bubble departure diameter for ΔT_{sub} = 3℃ is about 14 % larger than that for ΔT_{sub} = 6℃ (Figure 5.26(a)). Then, the effects of the duct size on the subcooled flow boiling average bubble departure diameter are shown in Figure 5.27. The results indicate that the average departing bubbles are slightly larger in the smaller duct. For instance, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and ΔT_{sub} = 3℃, and the average bubble departure diameter for δ = 0.2 mm is about 12% higher than that for δ = 2.0 mm (Figure 5.27(a)). Finally, the effects of the refrigerant saturated temperature on the subcooled flow boiling average bubble departure diameter are shown in Figures 5.28 and 5.29. The results show that at a higher T_{sat} the average departing bubble is slightly smaller.

How the refrigerant mass flux, inlet subcooling, duct size and refrigerant saturated temperature affect the measured mean bubble departure frequency for the subcooled flow boiling of R-134a at the middle axial location (z =80 mm) of the annular duct are illustrated in Figures 5.30-5.36 by presenting the average bubble departure frequency against the imposed heat flux for various G, ΔT_{sub}, δ and T_{sat}. The increase of the bubble departure frequency with the imposed heat flux is clearly seen from the data in Figures 5.30-5.36. First, the effects of the refrigerant mass flux on the subcooled flow boiling mean bubble departure frequency are shown in Figures 5.30 and 5.31. The results indicate that the average bubble deaprture frequency is somewhat higher for a larger refrigerant mass flux in the larger ducts with $\delta \geq 0.5$ mm especially at high imposed heat flux. For example, at q =30 kw/m^2, T_{sat} =15℃, ΔT_{sub} = 3℃ and δ = 0.5 mm, the average bubble departure frequency for G =600 kg/m^{2}s is about 13% higher than that for G =500 kg/m^{2}s (Figure 5.31(a)). But for the small duct with δ = 0.2 mm the bubble departure frequency is not influenced by the refrigerant mass flux to a noticeable degree except at a high heat flux (Figure 5.31(b)). Next, the effects of the inlet subcooling on the subcooled flow boiling average bubble deaprture frequency are shown in Figures 5.32 and 5.33. Note that the average bubble departure frequency is also somewhat higher for a smaller liquid subcooling. For instance, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and δ = 0.5 mm, the average bubble departure frequency for ΔT_{sub} = 3℃ is about 15% higher than that for ΔT_{sub} = 6℃ (Figure 5.33(a)). Then, the effects of the duct size on the subcooled flow boiling average bubble departure frequency are shown in Figure 5.34. The results indicate that the average bubble departure frequency is significantly higher for a smaller duct. For instance, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and ΔT_{sub} = 3℃, and the average bubble departure frequency for δ = 0.2 mm is about 55% higher than that for δ = 2.0 mm (Figure 5.34(a)). Finally, the effects of the refrigerant saturated temperature on the subcooled flow boiling average bubble departure frequency are shown in Figures 5.35 and 5.36. The results indicate that the average bubble departure frequency is slightly highr for a higher refrigerant saturation temperature. As a example, at q =30 kw/m^2, G =500 kg/m^{2}s, δ = 0.5 mm and ΔT_{sub} = 3℃, and the average bubble departure frequency for T_{sat} =15℃ is about 8% higher than that for T_{sat} =10℃ (Figure 5.36(a)).

The effects of the refrigerant mass flux, inlet subcooling, duct size and refrigerant saturated temperature on the number density of the active bubble nucleation sites in the subcooled flow boiling of R-134a at the middle axial location (z =80mm) of the annular duct are shown in Figures 5.37-5.43. For all cases the increase of the active nucleation site density with the imposed heat flux is rather pronounced. The results also indicate that the average active nucleation site density is significantly higher for a smaller refrigerant mass flux. For example, at q =30 kw/m^2, T_{sat} =15℃, ΔT_{sub} = 3℃ and δ = 0.5 mm, the average active nucleation site density for G =500 kg/m^{2}s is about 28% higher than that for G =600 kg/m^{2}s (Figure 5.38(a)). Next, the effects of the inlet subcooling on the subcooled flow boiling average active nucleation site density shown in Figures 5.39 and 5.40 indicate that the average active nucleation site density is somewhat higher for a smaller liquid subcooling. As an example, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and δ = 0.5 mm, the average active nucleation site density for ΔT_{sub} = 3℃ is about 14% higher than that for ΔT_{sub} = 6℃ (Figure 5.40(a)). Then, the results in Figure 5.41 show that the average active nucleation site density is substantially lower for the smaller gap especially at higher imposed heat flux. But at lower imposed heat flux the influence is insignificant. For instance, at q =30 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and ΔT_{sub} = 3℃, the average active nucleation site density for δ = 2.0 mm is about 39% higher than that for δ = 0.2 mm (Figure 5.41(a)). Finally, the data shown in Figures 5.42 and 5.43 indicate that the average active nucleation site density is significantly higher for a higher refrigerant saturation temperature. As a example, at q =30 kw/m^2, G =500 kg/m^{2}s, δ = 0.5 mm and ΔT_{sub} = 3℃, the average active nucleation site density for T_{sat} =15℃ is about 28% higher than that for T_{sat} =10℃ (Figure 5.43(a)).

The effects of the refrigerant mass flux, inlet subcooling, duct size and saturated temperature on the mean axial speed of the big bubbles in the slug flow regime for the subcooled flow boiling of R-134a at the middle axial location (z =80 mm) of the small annular ducts with δ = 0.5 & 0.2 mm are shown in Figures 5.44-5.47. First, the effects of the refrigerant mass flux on the bubble speed are shown in Figure 5.44. The results indicate that the average bubble speed is somewhat higher for a larger refrigerant mass flux especially at high heat flux. For example, at q =41 kw/m^2, T_{sat} =15℃, ΔT_{sub} = 3℃ and δ = 0.5 mm, the average bubble speed for G =600 kg/m^{2}s is about 16 % larger than that for G =500 kg/m^{2}s (Figure 5.44(a)). Next, the effects of the inlet subcooling on the average bubble speed are shown in Figure 5.45. Note that the average bubble speed is slightly higher for a smaller liquid subcooling. For example, at q =41 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and δ = 0.5 mm, the average bubble speed for ΔT_{sub} = 3℃ is about 11 % higher than that for ΔT_{sub} = 6℃ (Figure 5.45(a)). Then, the effects of the duct size on the average bubble speed are shown in Figure 5.46. The results manifest that the average bubble speed is significantly higher in the smaller duct. For instance, at q =41 kw/m^2, T_{sat} =15℃, G =500 kg/m^{2}s and ΔT_{sub} = 3℃, the average bubble speed for δ = 0.2 mm is about 23% higher than that for δ = 0.5 mm (Figure 5.46(a)). Finally, the effects of the refrigerant saturated temperature on the average bubble speed are shown in Figure 5.47. The results indicate that the average bubble speed is somewhat higher for a higher refrigerant saturation temperature. As a example, at q =41 kw/m^2, G =500 kg/m^{2}s, δ = 0.5 mm and ΔT_{sub} = 3℃, the average bubble speed for T_{sat} =15℃ is about 14% higher than that for T_{sat} =10℃ (Figure 5.47(a)).

5.4 Correlation Equations

An empirical equation to correlate the present data of the heat transfer coefficient in the subcooled flow boiling of R-134a in the horizontal annular duct with a narrow gap is proposed here when the bubbly flow dominates in the duct. Based on the present data, the total heat flux input to the boiling flow q_t is considered to consist of two parts： one resulting from the bubble nucleation q_b and another due to the single-phase forced convection q_c. Thus

$$q_t = q_b + q_c \tag{5.1}$$

Here q_b and q_c can be calculated from the quantitative data for the bubble characteristics examined in section 5.3 and single phase forced convection as

$$q_b = \rho_g V_g f N_{ac}\ i_{fg} \tag{5.2}$$

and

$$q_c = E\ h_{1\varphi}(T_w - T_r) \tag{5.3}$$

Note that in the above equation an enhancement factor E is added to q_c to account for the agitating motion of the bubbles which can enhance the single-phase convection heat transfer. Empirically, E and $h_{1\phi}$ can be correlated as

$$E = N_{conf}^{\ 0.48} Fr_l^{0.15} (1+100Bo)^{1.5} \tag{5.4}$$

and

$$h_{1\varphi} = Nu_{1\varphi} \cdot k_l / D_h \tag{5.5}$$

Note that $Nu_{1\phi}$ is estimated from the Gnielinski and Choi correlations [70, 71],

$$Nu_{1\varphi}=\frac{(f_f/2)(Re_l-1000)Pr_l}{1.07+12.7\sqrt{f_f/2}\left(Pr^{2/3}-1\right)} \qquad \text{for } Re_l \geq 2{,}300$$

and

$$Nu_{1\varphi} = 0.000972\ Re_l^{1.17} Pr_l^{1/3} \qquad \text{for } Re_l < 2{,}300 \tag{5.6}$$

Here f_f is the friction factor evaluated from the Gnielinski correlation [70] and is correlated as

$$f_f=(1.58\ln Re_l-3.28)^{-2} \tag{5.7}$$

Where ρ_g is the vapor density, V_g is the mean vapor volume of the departing bubble which is equal to $\frac{4\pi}{3}\left(\frac{d_p}{2}\right)^3$, f is the mean bubble departure frequency, N_{ac} is the mean active nucleation site density, i_{fg} is the enthalpy of vaporization. Because the experimental Re_l ranges from 700 to 5000, we use the Gnielinski correlation for $Re_l > 2{,}300$ but use the Choi correlation for $Re_l < 2{,}300$ to evaluate the single-phase forced convection heat transfer. It is difficult to distinguish the individual bubbles at a higher imposed heat flux. Hence the above correlations do not apply to the data for q>45 kW/m^2 at δ = 2.0 mm, q>40 kW/m^2 at δ = 1.0 mm, q>35 kW/m^2 at δ = 0.5 mm, and q>30 kW/m^2 at δ = 0.2 mm.

To enable the usage of the above correlation for computing the flow boiling heat transfer in the bubbly flow regime, the mean bubble size and departure frequency and the mean active nucleation site density on the heating surface need to be correlated in advance. The average bubble departure diameter in the subcooled flow boiling of R-134a in the narrow annular duct estimated from the present flow visualization can be correlated as

$$\frac{d_p}{\sqrt{\sigma/(g\Delta\rho)}} = \frac{115N_{conf}(\rho_l/\rho_g)^{0.333}}{Re^{0.5}\left[Ja'+\dfrac{165(\rho_l/\rho_g)^{0.333}}{Bo^{0.5}Re^{1.4}}\right]} \tag{5.8}$$

Here Ja_l is Jakob number defined as

$$Ja' = \frac{\rho_l \cdot C_p \cdot \Delta T_{sub}}{\rho_g \cdot i_{fg}} \tag{5.9}$$

Figure 5.48 shows that almost all the present experimental data for d_p fall within ±25% of the above correlation and the mean absolute error is 18.4%. Besides, an empirical equation is proposed for the product of the mean bubble departure diameter and frequency as

$$\frac{f \cdot d_p}{\mu_l/(\rho_l D_h)} = 1642 \cdot Re_l^{0.887} \cdot Ja'^{-0.05} \cdot Bo^{0.887} \cdot N_{conf}^{0.01} \tag{5.10}$$

Note that more than 90 % of the experimental data for $f \cdot d_p$ collected in this study can be correlated within ±30% by Equation (5.10) and the mean absolute error is 15% (Figure 5.49). Finally, we propose an empirical correlation for the average active nucleation site density in the subcooled flow boiling of R-134a as

$$N_{ac} d_p^{\ 2} = 0.0022 + 2231.73 Bo^{1.705} Re_l^{0.345} Ja^{-0.149} N_{conf}^{-0.01} \tag{5.11}$$

Figure 5.50 shows that the present experimental data fall within ±30% of the above correlation and the mean absolute error is 18.9%.

When the correlations for d_p, f, and N_{ac} given in Equations (5.8)-(5.11) are combined with Equations (5.1)-(5.7) for q_t, more than 90% of the heat transfer data for the bubbly flow regime measured in the present study fall within ±30% of the correlation proposed here with a mean deviation of 17.5%(Figure 5.51).

For the smaller ducts with δ = 0.2 & 0.5 mm the slug flow prevails at high imposed heat flux. The correlation for the flow boiling heat transfer in this slug flow regime(confined bubbles) is modified from that of Cornwell and Kew [41] for the confined bubble regime as

$$Nu/Nu_{lo} = 2.4 Bo^{0.01} N_{conf}^{0.1} \tag{5.12}$$

with

$$Nu_{lo} = \frac{(f_f/2)(Re_l - 1000) Pr_l}{1.07 + 12.7\sqrt{f_f/2}\left(Pr^{2/3} - 1\right)} \qquad \text{for } Re_l \geq 2{,}300$$

and

$$Nu_{lo} = 0.000972\, Re_l^{1.17} Pr_l^{1/3} \qquad \text{for } Re_l < 2{,}300 \tag{5.13}$$

Figure 5.52 shows that the present experimental data fall well within ±30% of the above correlation and the mean absolute error is 11.3%.

5.5 Concluding Remarks

The experimental heat transfer data and the associated bubble behavior for the subcooled flow boiling of R-134a in a horizontal narrow annular ducts have been presented here. The effects of the imposed heat flux, refrigerant mass flux, inlet subcooling, system pressure and duct size on the subcooled flow boiling heat transfer coefficient and associated bubble characteristics have been examined in detail. In addition, empirical equations to correlate the measured heat transfer data, bubble departure diameter, bubble departure frequency, and active nucleation site density were proposed. The major results obtained in the present study can be summarized in the following.

(1) The temperature undershoot at ONB are significant for the subcooled flow boiling of R-134a in the narrow annular duct.

(2) The subcooled boiling heat transfer coefficient increases with a decrease in the duct size, but decreases with an increase the inlet subcooling. Besides, raising the imposed heat flux can cause a significant increase in the boiling heat transfer coefficients. However, the effects of the refrigerant mass flux and saturated temperature on the boiling heat transfer coefficient are small.

(3) Correlation equations were provided for the boiling heat transfer coefficient, bubble departure diameter, bubble departure frequency and active nucleation site density in the subcooled flow boiling.

(4) Visualization of the bubble motion in the boiling flow reveals that the bubbles are suppressed by raising the refrigerant mass flux and inlet subcooling. Moreover, raising the imposed heat flux produces positive effects on the bubble population, coalescence and departure frequency. The bubble departure frequency in the bubbly flow and the mean speed of the confined bubbles in the slug flow decrease with the decreasing refrigerant mass flux, inlet subcooling and saturated temperature and with the increasing duct size. Besides, the bubbly flow dominates in the duct except when the smaller ducts with δ = 0.5 & 0.2 mm are subject to a high imposed heat flux.

Under that situation the slug flow prevails. Between the big confined bubbles and heating surface thin liquid films exist and the interfacial evaporation of the liquid films is rather effective for heat transfer enhancement by reducing the duct size.