CHAPTER 4

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

In this chapter the measured heat transfer data and observed bubble characteristics for the saturated flow boiling of R-134a in the narrow annular duct affected by the refrigerant mass flux, system pressure and duct size are presented. In the tests we have saturated liquid R-134a at the inlet of the duct (x_{in} = 0). The experiments are performed for the refrigerant mass flux varying from 400 to 700 kg/m^{2}s (G = 400-500 kg/m^{2}s for δ=2.0 mm, G = 500-600 kg/m^{2}s for δ=0.5 & 1.0 mm and G = 600-700 kg/m^{2}s for δ=0.2 mm), imposed heat flux q from 1 to 55 kW/m^2 and the system pressure set at 414 kPa and 488 kPa (corresponding to the R-134a saturation temperature T_{sat}=10℃ and 15℃) for the annular gap of the duct δ=0.2 mm to 2.0 mm. The measured boiling heat transfer data are expressed in terms of the boiling curves and boiling heat transfer coefficient. Moreover, the overview of the boiling flow in the duct and the close view flow photos taken at a small region around the middle axial station z =80 mm are presented to illustrate the bubble characteristics in the boiling flow. Finally, empirical correlations will be proposed to correlate the present data for the saturated flow boiling heat transfer coefficient, mean bubble departure diameter, mean bubble departure frequency and average active nucleation site density.

4.1 Single-phase Heat Transfer

Before beginning the two-phase flow boiling experiments, the single-phase convective heat transfer experiments are conducted for liquid R-134a. The measured single-phase convection heat transfer coefficients are compared with the correlations proposed by Dittus-Boelter [62] and Gnielinski [70] for Re_l > 2,300 and by Choi et al. [71] for Re_l < 2,300. In the single-phase heat transfer tests the refrigerant mass flux is varied from 200 to 1,200 kg/m^{2}s, gap size δ from 0.2 to 2.0 mm(corresponding to the Reynolds number of the refrigerant flow from 714 to 12,506) for the refrigerant inlet temperature fixed at 15℃. Selected results from these tests are plotted in Figure 4.1.

The Dittus-Boelter correlation is

$$Nu_{1\phi} = 0.023 \cdot \mathrm{Re}^{0.8} \cdot \mathrm{Pr}^{0.4} \qquad \text{for Re} > 10^4 \tag{4.1}$$

and the Gnielinski correlation is

$$Nu_{1\phi} = \frac{\left(\frac{f_f}{2}\right)(\mathrm{Re}-1000)\,\mathrm{Pr}}{1.07 + 12.7\sqrt{\frac{f_f}{2}}\,(\mathrm{Pr}^{2/3}-1)} \qquad \text{for } 2{,}300 < \mathrm{Re} < 10^5 \tag{4.2}$$

where $f_f = (1.58 \ln \mathrm{Re} - 3.28)^{-2}$ (4.3)

and the Choi correlation is

$$Nu_{1\phi} = 0.000972 \cdot \mathrm{Re}^{1.17} \cdot \mathrm{Pr}^{1/3} \qquad \text{for Re} < 2{,}000 \tag{4.4}$$

The results from the single-phase heat transfer tests indicate that the energy balance between the heater and the refrigerant flow defined in Eq.(3.2) is within 5% for all run. This insures the heat loss from the test section is rather small and the design of test section is suitable for our measurement. The results in Figure 4.1 manifest that for δ =0.5-2.0 mm the present data for h_l can be well correlated with the Gnielinski correlations. But for the smaller gap of 0.2 mm the Choi correlation is closer to the present data.

4.2 Saturated Flow Boiling Curves

The effects of the refrigerant mass flux, gap size of the duct and refrigerant saturated temperature on the saturated flow boiling characteristics at the middle axial location (z =80 mm) of the narrow annular duct are shown in Figures 4.2-4.6 by presenting the boiling curves for various G, δ and T_{sat}.

First, the effects of the refrigerant mass flux on the saturated flow boiling curves are shown in Figures 4.2 and 4.3. The results indicate that at a low imposed heat flux the wall superheat is lower than that for the onset of nucleate boiling (ONB) and no bubble nucleates from the heating surface. Hence heat transfer in the flow results completely from the single-phase forced convection. As the imposed wall heat flux is raised gradually, the wall superheat increases correspondingly. At a certain wall superheat bubbles start to nucleate from the heating surface and we have onset of nucleate boiling in the flow. Beyond the ONB there is a significant increase in the slope of the boiling curves, implying that a small rise in the wall superheat causes a large increase in the heat transfer rate from the wall to refrigerant. Note that at increasing refrigerant mass flux the boiling curves shift slightly to the left when the gap size reduces (Figure 4.3), which indicates that at a higher refrigerant mass flux the heat transfer in the saturated boiling is slightly better. This small increase in the heat transfer rate with the mass flux is mainly due to the bubbles in the liquid refrigerant move more vigorously and turbulently due to the higher refrigerant mass flux. The results also indicate that the required imposed heat flux to achieve ONB is influenced noticeably by the change in the mass flux. Specifically, the required imposed heat flux to achieve ONB is slightly higher for a higher mass flux except for the smallest δ of 0.2 mm tested here (Figure 4.3(b)). At this δ the ONB heat flux is nearly the same for G = 600 & 700 kg/m^{2}s. When compared with the results for the subcooled flow boiling to be presented in chapter 5, no apparent temperature undershoot occurs during the onset of boiling when the flow is saturated.

Then, the effects of the duct size on the saturated flow boiling curves are shown in Figure 4.4. It is noted that the boiling curve shifts significantly to the left as the gap in the duct is reduced from 2 mm to 0.2 mm, indicating that the boiling heat transfer in the smaller duct is substantially better. This also agrees with the findings of Qiu et al. [39] and Aritomi et al. [40]. It is also evident from the data that a lower imposed heat flux is needed to initiate boiling on the heated surface for the smaller duct. Then, the data shown in Figures 4.5 and 4.6 suggest that the effects of the refrigerant saturation temperature on the boiling curves including the single-phase forced convection and nucleate boiling regimes are insignificant. Finally, the relation between the imposed heat flux and wall superheat temperature at the incipient boiling is shown in Figure 4.7. The results indicate that at the same q_{ONB} the wall superheat at ONB is smaller for the narrower duct.

4.3 Saturated Flow Boiling Heat Transfer Coefficient

The effects of the refrigerant mass flux, duct size and refrigerant saturated temperature on the saturated flow boiling heat transfer of R-134a at the middle axial location (z =80 mm) in the narrow annular duct are shown in Figures 4.8-4.12 by presenting the saturated flow boiling heat transfer coefficient against the imposed heat flux for various G, δ and T_{sat}. The results indicate that at given G, δ and T_{sat} the saturated boiling heat transfer coefficient increases substantially with the imposed heat flux. For example, at T_{sat} =15℃, δ=0.5 mm and G =500 kg/m^{2}s, the saturated boiling heat transfer coefficient for q =46 kW/m^2 is about 91% higher than that for q = 6 kW/m^2 (Figure 4.9(a)). This large increase in the saturated boiling heat transfer coefficient is ascribed to the higher active nucleation site density on the heating surface and higher bubble departure frequency for a higher imposed heat flux. The effects of the refrigerant mass flux on the saturated flow boiling heat transfer coefficient shown in Figures 4.8 and 4.9 manifest that the saturated boiling heat transfer coefficient rises slightly with the refrigerant mass flux only at a high q for δ=0.5 and 0.2 mm. For instance, the saturated boiling heat transfer coefficient at q =46 kW/m^2, T_{sat} =15℃ and δ=0.5 mm for G =600 kg/m^{2}s is about 5% higher than that for G =500 kg/m^{2}s (Figure 4.9(a)). For the larger δ of 1.0 and 2.0 mm the effects of G and h_r are negligible, as evident from the data in Figure 4.8.

Then, the effects of the gap size of the duct on the saturated flow boiling heat transfer coefficient are shown in Figure 4.10. The data show that the saturated boiling heat transfer increases noticeably with a decrease in the channel gap. For example, at q =46 kW/m^2, T_{sat} =15℃ and G =500 kg/m^{2}s the saturated boiling heat transfer coefficient for δ=0.5 mm is about 31% higher than that for δ=2.0 mm (Figure 4.10(a)). Besides, at G =600 kg/m^{2}s and the same q and T_{sat} the saturated boiling heat transfer coefficient for δ=0.2 mm is about 32% higher than that for δ=1.0 mm (Figure 4.10(b)). Since the shear stress of the flow acting on the heated surface in a smaller channel becomes higher, the cavities on the heating surface can be more easily wetted. Moreover, the flow pattern changes from the bubbly flow for δ= 2.0 & 1.0 mm to become a slug flow for δ= 0.5 & 0.2 mm. These effects are thought to be the main reasons for the enhancement of nucleate and convection boiling heat transfer when the channel size is reduced. Finally, the data shown in Figures 4.11 and 4.12 suggest that the saturation temperature of the refrigerant exhibits negligible effects on the boiling heat transfer coefficient.

4.4 Bubble Characteristics in Saturated Flow Boiling

To illustrate the bubble behavior in the duct, selected photos of the boiling flow of R-134a from the side and top view covering the whole narrow duct to illustrate the effects of the gap size of the duct are shown in Figures 4.13 to 4.16. The results in Figure 4.13 for the large duct with δ = 2.0 mm clearly indicate that in the relatively upstream region near the duct inlet a great number of bubbles already exist in the flow. Bubbles of varying size can be seen. Specifically, in the upper part of the duct larger bubbles dominate obviously due to the buoyancy effects. These photos suggest that for δ = 2.0 mm the onset of nucleation boiling occurs immediately after the refrigerant enters the heated section of the duct. Because of the presence of the flanges, this bubble cannot be seen from the photos (Figure 2.2). Note that for a high G of 600 kg/m^{2}s and a smaller δ of 1.0 mm some delay in the bubble nucleation is seen in Figure 4.14. Besides, the bubble nucleation is earlier in the lower part of the duct. This is due to the difference in the buoyancy effect in different parts of the duct. More specifically, in the lower portion of the duct the flow is heated from above and hence is thermally stable. This in turn results in a lower convection heat transfer coefficient and obviously the heated surface temperature is higher for a fixed wall heat flux. This higher T_w causes the earlier inception of the bubbles from the surface in the lower portion of the duct. Because the ONB is somewhat unstable, the ONB location is not symmetrical with respect to the vertical central plane through the duct axis. In fact, the ONB locations move back and forth with time in an irregular manner. But for a further reduction of the duct gap to 0.5mm an opposite trend appears with bubbles nucleated on the heated surface earlier in the upper part of the duct (Figure 4.15). This results from the fact that at the smaller Re for the smaller δ of 0.5 and 0.2 mm laminar and transitional forced convective liquid flows dominate in the duct since the buoyancy-to-inertia ration is also small ($Gr/Re^2 = 4.14\times10^{-2}$ for δ = 0.5 mm and $Gr/Re^2 = 6.63\times10^{-3}$ for δ = 0.5 mm). In this laminar and transitional flow (Re = 2,670 for δ = 0.5 mm and Re = 1,070 for δ = 0.2 mm) subject to the inner duct heating Ciampi et al. [72] found that a helicoidal flow motion appears in the duct which maintains a continuous supply of warm liquid to the top of the cylinder and cold liquid to the bottom[72], which causes higher T_w in the upper portion of the duct. Thus the bubble nucleation takes place earlier in the upper part of the heating surface. Comparing the results in Figures 4.13-4.15 clearly shows that in the smaller duct the coalescence of bubbles is more pronounced. For δ = 1.0 mm in the exit half of the duct many big bubbles exist. When the duct gap is reduced to 0.5 mm Figure 4.15 shows that big bubbles even dominate in the upper portion of the exit half of the duct. Thus in this region we have a slug flow. For a further reduction of δ to 0.2 mm the bubble coalescence is so intense that the slug flow dominates the upper portion of the entire duct (Figure 4.16). The above results manifest that in the smaller ducts of 0.5& 0.2 mm the bubbly flow and slug flow coexist.

The photos of the boiling flow taken for the cases at different duct sizes and imposed heat fluxes in the small region around the middle axial location marked on Figures 4.13(a)-4.16(a) are shown in Figure 4.17. First of all, it is noted from the photo taken from the duct for δ=1.0 mm shown in Figure 4.17(a) for the case at T_{sat}=15℃ and *G*=600 kg/m^{2}s at the imposed heat flux q=15 kW/m^2 that a number of discrete bubbles nucleate from the cavities and slide along the heating surface. As the imposed heat flux is increased to q=25 kW/m^2, the active bubble nucleation density increases and a lot more bubbles appear and they move faster (Figure 4.17(b)). Many coalescence bubbles are seen as the heat flux is raised to q=35 kW/m^2 (Figure 4.17(c)). Then, the photos taken from the smaller duct with δ=0.5 mm shown in Figure 4.17(d) for the same *G*, q, and T_{sat} indicate that a large number of bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. As the bubbles become larger, they become distorted and elongated as they slide on the heating surface. As the imposed heat flux is increased slightly to q=25 kW/m^2 (Figure 4.17(e)), the active bubble nucleation density increases and bubbles collide and coalesce more frequently. 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=35 kW/m^2 (Figure 4.17(f)), coalescence of the bubbles occurs irregularly at a very high rate. The coalescence bubbles can be very large. In fact, the liquid slugs and discrete bubbles coexist in the duct. At even higher imposed heat flux for q>30 kW/m^2, the bubble departure frequency is very high so that it is difficult to clearly distinguish the individual bubbles. In general, the bubble departure frequency increases substantially with the imposed heat flux due to the fact that an increase in 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 diameter increases slightly at increasing imposed heat flux due to the rise in the wall superheat. Then, the corresponding photo taken from the even smaller duct with δ=0.2 mm are shown in Figures 4.17(g)~(i) for the same G, q and T_{sat}. Note that for the smaller δ more coalescence bubbles are seen and they are even bigger especially at a higher imposed heat flux. This causes less bubbles nucleated at the heated surface.

The bubble characteristics in the narrow duct around the middle axial location for δ = 0.5 mm affected by the refrigerant mass flux and saturated temperature are illustrated by the photos in Figure 4.18. The results in Figures 4.18(a)-(f) indicate that at a higher mass flux the liquid refrigerant flow moves at a higher speed, which in turn tends to sweep the bubbles more quickly away from the heating surface. Collision and coalescence of bubbles are still significant. Besides, the bubble departure frequency is higher and the bubbles are smaller and in violent agitating motion. However, the active nucleation site density is lower. Note that at the low mass flux the bubble coalescence is more important and a number of bigger bubbles form in the duct. Then, the effects of the refrigerant saturation temperature on the bubble characteristics are illustrated by comparing the photos in Figures 4.18(a)~(c) with Figures 4.18(g)~(i). The results indicate that at a lower saturation temperature the bubbles grow bigger and move slower due to the higher surface tension. Small bubbles are easier to merge into big bubbles. The flow patterns observed in the present test for $0.22 \leq N_{conf} \leq 2.22$ and $5.5\times10^{-5} \leq Bo \leq 6.8\times10^{-4}$ are summarized in a flow regime map shown in Figure 4.19.

To be more quantitative on 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 with the bubbly flow dominated in the duct from the images of the boiling flow stored in the video tapes. The results from this estimation are examined in the following. The effects of the three parameters, namely, the refrigerant mass flux, duct size and refrigerant saturated temperature, on the mean bubble departure diameter for the saturated flow boiling of R-134a at the middle axial location (z =80 mm) in the annular duct are shown in Figures 4.20-4.24 by presenting the average bubble departure diameter against the imposed heat flux for various G, δ and T_{sat}. First, the effects of the refrigerant mass flux on the average bubble departure diameter shown in Figures 4.20 and 4.21 indicate that the average departing bubble is only slightly larger for a lower refrigerant mass flux. For example, at q =30 kw/m^2, T_{sat} =15℃ and δ = 0.5 mm, the average bubble departure diameter for G =500 kg/m^{2}s is only about 11% larger than that for G =600 kg/m^{2}s (Figure 4.21(a)). Then, the data given in Figure 4.22 also suggest that the average bubble departing from the heated surface is slightly larger in the smaller duct only at the lower G of 500 kg/m^{2}s for the imposed heat flux q > 20 kw/m^2. For example, at q =30 kw/m^2, T_{sat} =15℃ and G =500 kg/m^{2}s, the average bubble departure diameter for δ = 0.5 mm is only about 18% higher than that for δ = 2.0 mm (Figure 4.22(a)). Otherwise the effects of the duct gap on the bubble departure diameter are relatively small. Finally, the results in Figures 4.23 and 4.24 indicate that the average bubble departure diameter is smaller for a higher refrigerant saturated temperature. For instance, at q =30 kw/m^2, G =500 kg/m^{2}s and δ = 2.0 mm, and the average departing bubble for T_{sat} =10℃ is about 18% larger than that for T_{sat} =15℃ (Figure 4.23(a)).

How the bubble departure frequency is affected by the three parameters for the saturated flow boiling of R-134a at the middle axial location (z =80 mm) in the annular duct are shown in Figures 4.25-4.29 by presenting the average bubble departure frequency against the heat flux for various G, δ and T_{sat}. Note that the increase of the bubble departure frequency with the imposed heat flux is rather significant for all cases presented here. First, the effects of the refrigerant mass flux on the saturated flow boiling average bubble departure frequency are shown in Figures 4.25 and 4.26. The results indicate that the average bubble departure frequency is somewhat higher for a higher refrigerant mass flux especially at high imposed heat flux. For example, at q =26 kw/m^2, T_{sat} =15℃ and δ = 0.2 mm, the average bubble departure frequency for G =700 kg/m^{2}s is about 14% higher than that for G =600 kg/m^{2}s (Figure 4.26(b)). Then, the effects of the duct size on the saturated flow boiling average bubble departure frequency shown in Figure 4.27 manifests that the average bubble departure frequency is noticeably higher in the smaller duct. For instance, at q =26 kw/m^2, T_{sat} =15℃ and G =500 kg/m^{2}s the average bubble departure frequency for δ = 0.5 mm is about 23% higher than that for δ = 2.0 mm (Figure 4.27(a)). Besides, at G =600 kg/m^{2}s and at the same q, and T_{sat} the average bubble departure frequency for δ = 0.2 mm is about 29% higher than that for δ = 1.0 mm (Figure 4.27 (b)). Finally, the data given Figures 4.28 and 4.29 indicate that the average bubble departure frequency is significantly higher for a higher saturated temperature. As an example, at q =26 kw/m^2, G =600 kg/m^{2}s and δ = 0.2 mm the average bubble departure frequency for T_{sat} =15℃ is about 21% higher than that for T_{sat} =10℃ (Figure 4.29(b)).

The number density of the active nucleation sites for ONB affected by the three parameters for the saturated flow boiling of R-134a at the middle axial location (z =80 mm) in the annular duct are shown in Figures 4.30-4.34 by presenting the average active nucleation site density against the imposed heat flux for various G, δ and T_{sat}. The data clearly show the substantial increase of the active nucleation site density with the imposed heat flux for all cases examined here. First, the effects of the refrigerant mass flux on the saturated flow boiling average active nucleation site density are shown in Figures 4.30 and 4.31. The results indicate that the average active nucleation site density is significantly higher for a smaller refrigerant mass flux especially at high imposed heat flux for the smaller ducts with δ = 0.5 & 0.2mm. For example, at q =30 kw/m^2, T_{sat} =15℃ and δ = 0.5 mm, the average active nucleation site density for G =500 kg/m^2s is about 22 % higher than that for G =600 kg/m^2s (Figure 4.31(a)). Then, the effects of the duct size on the saturated flow boiling average active nucleation site density shown in Figure 4.32 manifest that the average active nucleation site density is significantly higher for a larger duct when the imposed heat flux exceeds 20 kw/m^2. For instance, at q =30 kw/m^2, T_{sat} =15℃ and G =500 kg/m^2s the average active nucleation site density for δ = 2.0 mm is about 45% higher than that for δ = 0.5 mm (Figure 4.32(a)). In addition, at G =600 kg/m^2s and at the same q and T_{sat} the average active nucleation site density for δ = 1.0 mm is about 50% higher than that for δ = 0.2 mm (Figure 4.32 (b)). These results seem to introduce adverse effects for heat transfer for a reduction in the duct size. But a large portion of bubbles are merged to become large confined bubbles in the smaller duct for δ = 0.5 & 0.2 mm at higher imposed heat flux. Hence in the small duct the boiling heat transfer is better. Finally, the data shown in Figures 4.33 and 4.34 suggest that the average active nucleation site density increases significantly with T_{sat} especially at high heat flux. As an example, at q =30 kw/m^2, G =500 kg/m^2s and δ = 0.5 mm the average active nucleation site density for T_{sat} =15℃ is about 22% higher than that for T_{sat} =10℃ (Figure 4.34(a)).

In the small ducts with δ = 0.5 & 0.2 mm subject to a high imposed heat flux, the slug flow prevails and the flow is dominated by the big bubbles. The mean axial speeds of these bubbles are measured for the saturated flow boiling of R-134a at the middle axial location (z =80 mm) in the annular duct. The data are shown in Figures 4.35-4.37. First, the effects of the refrigerant mass flux on the average speed of the bubbles shown in Figure 4.35 indicate that the average bubble speed is substantially larger for higher refrigerant mass flux and higher heat flux. For example, at q =40 kw/m^2, T_{sat} =15℃ and δ = 0.5 mm, the average bubble speed for G =600 kg/m^{2}s is about 30% larger than that for G =500 kg/m^{2}s (Figure 4.35(a)). For instance, at T_{sat} =15℃, δ =0.2 mm and G =600 kg/m^{2}s, the average bubble speed for q = 28 kW/m^2 is about 54% higher than that for q = 46 kW/m^2 (Figure 4.35(b)).Then, the data given in Figure 4.36 suggests that the average bubble speed is substantially larger when δ is reduced from 0.5 mm to 0.2 mm. This is more pronounced at high heat flux. For example, at q =40 kw/m^2, T_{sat} =15℃ and G =600 kg/m^{2}s, the average bubble speed for δ = 0.2 mm is about 29% higher than that for δ = 0.2 mm (Figure 4.36). Finally, the results in Figure 4.37 indicate that the average bubble speed is significantly higher for a higher saturated temperature especially at a high heat flux. For instance, at q =40 kw/m^2, G =500 kg/m^{2}s and δ = 0.5 mm the average bubble speed for T_{sat} =10℃ is about 25% lower than that for T_{sat} =15℃ (Figure 4.37(a)).

4.5 Correlation Equations

According to flow boiling mechanisms, the heat transfer in the flow boiling in the bubbly flow regime can be roughly considered as a combination of single-phase convection heat transfer q_c and pool boiling heat transfer q_b. Thus the total heat flux input to the boiling flow q_t can be expressed as

$$q_t = q_b + q_c \tag{4.5}$$

Here q_b and q_c can be calculated from the relations

$$q_b = \rho_g V_g f N_{ac} \, i_{fg} \tag{4.6}$$

and

$$q_c = E \, h_{1\varphi} \, \Delta T_{sat} \tag{4.7}$$

Note that q_b expressed in Equation(4.6) in fact represents the latent heat carried away from the heating surface during the departure of bubbles from the surface. The single-phase forced convection heat transfer coefficient is estimated from the correlation for the enhanced factor E and the Nusselt number $Nu_{1\varphi}$ as

$$h_{1\varphi} = Nu_{1\varphi} \, k_l / D_h \tag{4.8}$$

with

$$E = 0.97 N_{conf}^{0.28} Fr_l^{0.01} (1-330Bo)^{5.51} \tag{4.9}$$

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

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

and

$$Nu_{1\varphi} = 0.000972\, Re_l^{1.17} Pr_l^{1/3} \quad \text{, for } Re_l < 2{,}300 \tag{4.10}$$

Here the friction factor f_f is evaluated from the Gnielinski correlation

$$f_f = (1.58\ln Re_l\text{-}3.28)^{-2} \tag{4.11}$$

Moreover, the Reynolds number of the liquid flow is defined as

$$Re_l = GD_h(1\text{-}x)/\mu_l \tag{4.12}$$

In the above equations ρ_g is the vapor density, V_g is the mean vapor volume of a departing bubble which is equal to $\frac{4\pi}{3}\left(\frac{d_p}{2}\right)^3$, f is bubble departure frequency, N_{ac} is the active nucleation site density, i_{fg} is the enthalpy of vaporization. Because the range of experimental Re_l is between 800 to 6000, we use Gnielinski correlation for $Re_l > 2{,}300$ but Choi correlation for $Re_l < 2{,}300$ to represent single phase convection heat transfer term($h_{1\varphi}$). 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>40 kW/m^2 at δ = 2.0 mm, q>35 kW/m^2 at δ = 1.0 mm, q>30 kW/m^2 at δ = 0.5 mm, and q>25 kW/m^2 at δ = 0.2 mm.

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

$$\frac{d_p}{\sqrt{\sigma/(g\Delta\rho)}} = 0.605\left(\frac{\rho_l}{\rho_g}\right)^{0.5} Re_l^{-0.2} \cdot Bo^{0.195} \cdot N_{conf}^{-0.19} \tag{4.13}$$

Figure 4.38 shows that almost all the present experimental data for d_p fall within ±25% of the above correlation and the mean absolute error is 9.5%. Besides, an empirical equation is proposed for the product of the mean bubble departure diameter and departure frequency as

$$\frac{f \cdot d_p}{\mu_l/(\rho_l D_h)} = 1.61 Re_l^{1.33} \cdot Pr_l^{2} \cdot Bo^{0.725} \cdot N_{conf}^{0.59} \tag{4.14}$$

Note that more than 85 % of the experimental data for $f \cdot d_p$ collected in this study can be correlated within ±30% by Equation (4.14) and the mean absolute error is 13.1%(Figure 4.39). Finally, we propose an empirical correlation for the average active nucleation site density in the saturated flow boiling of R-134a as

$$N_{ac} d_p^{2} = -0.009 + 1095.25 Bo^{1.23} Re_l^{0.049} N_{conf}^{0.058} \tag{4.15}$$

Figure 4.40 shows that the present experimental data fall within ±30% of the above correlation and the mean absolute error is 29.6%.

When the correlations for d_p, f, and N_{ac} given in Equations (4.13)-(4.15) are combined with Equations (4.5)-(4.12) for q_t, the heat transfer data measured in the present study fall within ±30% of the correlation proposed here with the mean absolute error of 29.17%(Figure 4.41).

For the smaller ducts with δ = 0.5 & 0.2 mm the slug flow (confined bubbles) prevails at a high imposed heat flux, the associated flow boiling heat transfer data from the present measurement are correlated by modifying the correlation of Cornwell and Kew [41] for the confined bubble regime as

$$Nu/Nu_{lo} = 2.5 Bo^{0.02} N_{conf}^{0.15} \tag{4.16}$$

with

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

and

$$Nu_{lo} = 0.000972\, Re_l^{1.17} Pr_l^{1/3} \qquad , \text{for } Re_l < 2{,}300 \qquad (4.17)$$

Figure 4.42 shows that the present experimental data fall within ±30% of the above correlation and the mean absolute error is 9.4%.

4.6 Concluding Remarks

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

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

(2). The saturated boiling heat transfer coefficients increase with a decrease in the gap size. Besides, raising the imposed heat flux can cause a significant increase in the boiling heat transfer coefficient. 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, mean bubble departure diameter, mean bubble departure frequency and mean active nucleation site density in the saturated flow boiling.

(4). The results from the flow visualization show that the mean diameter of the bubbles departing from the heating surface decreases slightly with increasing refrigerant mass flux. Besides, at a high imposed heat flux many bubbles generated from the cavities in the heating surface tend to merge together to form big bubbles. The bubble departure frequency increases with the increasing refrigerant mass flux and saturated temperature and with the decreasing duct size. The active nucleation site density is much lower at a higher refrigerant mass flux and lower saturation temperature and smaller gap. Besides, in the smaller ducts with δ = 0.5 & 0.2 mm the slug flow prevails at a high imposed heat flux. Many big bubbles appear in the duct.