Chapter 5

R134a Flow Boiling Heat Transfer in Small Diameter Tubes

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Boiling heat transfer in small diameter tubes has been experimentally investigated using R134a as the working fluid. The heat transfer experiments were conducted with two stainless steel tubes of internal diameter 4.26 mm and 2.01 mm respectively. Other parameters were varied in the range: mass flux 100 – 500 kg/m^2*s; pressure 8 – 14 bar; quality up to 0.9; heat flux 13 - 150 kW/m*^{2}*. The heat transfer coefficient was found to be independent of vapour quality when the quality was less than about 40% to 50% for the 4.26 mm tube and 20% to 30 % for the 2.01 mm tube. Above these quality values, the heat transfer coefficient decreases with vapour quality. Furthermore, at high heat flux values this decrease occurs for the entire quality range. The heat transfer rates were compared with existing correlations.*

INTRODUCTION

Flow boiling heat transfer has been studied extensively in the past. More recently and in view of the benefits of process intensification, researches turned their attention to the study of small to micro passages. This was encouraged by an increasing use of compact heat exchangers in a great number of applications including refrigeration and heating/cooling systems.

The effects of geometry and size on two-phase flow and heat transfer were examined by Kew and Cornwell (1997). Small tubes with diameters of 1.39 – 3.69 mm were tested using R141b. Their results showed that in 3.69 and 2.87 mm tubes, the boiling heat transfer coefficient decreased slightly or remained constant with vapour quality when $x < 0.2$, but increased monotonically with increasing vapour quality when $x > 0.2$. However, in the 1.39 mm tube, the heat transfer coefficient increased monotonically with increasing vapour quality at low mass flux ($G = 478$ kg/m^{2}s), but decreased rapidly with increasing vapour quality at a higher mass flux ($G = 1480$ kg/m^{2}s). The sudden decrease of heat transfer coefficient was attributed to overheating of the tube and subsequent local dry-out. They reported that when the confinement number, Co, defined by Equation (1) below, was in excess of 0.5, two-phase flow in such small hydraulic diameters exhibited different flow characteristics and heat transfer results compared with corresponding flow in traditional size passages. The confinement number can be directly influenced by pressure (temperature) mainly through the vapour density and to a lesser extent through surface tension. For typical working fluids like water and R134a, this equates to hydraulic diameters less than 5.4 and 2.1 mm, respectively (calculated at 30 °C for water and –20 °C for R134a).

$$Co = \frac{\left[\sigma/(g(\rho_l - \rho_g))\right]^{1/2}}{d_h} \tag{1}$$

The flow visualization experiments reported recently by Chen et al. (2004) indicated that reducing the tube diameter from 4.26 mm to 2.01 mm had a significant effect on the flow pattern transition boundaries obtained with R134a agreeing with the above and verifying the confinement effect and the increasing importance of surface tension at this size.

Two-phase flow boiling heat transfer in tubes/channels can be characterized by either the nucleate or convective component or both. All three possibilities have been reported based on experiments under different system parameters. In large diameter tubes/channels, the flow patterns are usually annular for the largest range of quality and the convective heat transfer mechanism dominates (Reid et al. 1987, Jung and Radermacher 1991 and Carey et al. 1992). In contrast, conclusions may differ among the various researchers as to the boiling heat transfer mechanisms in small diameter tubes/channels over the entire quality range. A study on boiling heat transfer of refrigerant R113 in a 2.92 mm diameter horizontal tube was carried by Wambsganss et al. (1993). Bao et al. (2000) studied flow boiling heat transfer coefficients for R11 and R123 in a copper tube with an inner diameter of 1.95 mm. They observed a strong dependence of the saturated boiling heat transfer coefficient upon heat flux and negligible influence of quality and thus concluded that the mechanism of nucleate boiling controlled the wall heat transfer process during saturated boiling. Boiling heat transfer experiments were performed by Tran et al. (1996) in a small circular channel ($d_{in} = 2.46$ mm) and a small rectangular channel ($d_h = 2.40$ mm) with R12. They concluded that over a broad range of heat flux, nucleation was the dominant heat transfer mechanism but at sufficiently low values of heat flux (very low wall superheat), forced convection dominates. Yan and Lin (1998) carried out experiments to investigate the characteristics of boiling heat transfer and pressure drop for refrigerant R134a flowing in a horizontal small circular pipe of 2.0 mm inside diameter. They noted that the boiling heat transfer coefficient was higher at a higher imposed wall heat flux except in the high vapour quality region, and also, the boiling heat transfer coefficient was higher at a higher mass flux and saturation temperature when the imposed heat flux was low. Vertical flow boiling of R134a in small multi-channels was investigated by Agostini and Bontemps (2004). Their experimental results indicated that heat transfer rates were greater than that reported in the previous literature for conventional tubes, while dry-out occurred at low qualities. However, from their results, it was very difficult to conclude which regime was dominant, nucleation or forced convection.

The local heat transfer coefficient, pressure drop, and critical heat flux were measured by Lazarek and Black (1982) for flow boiling of R113 in a round vertical tube with an internal diameter of 3.1 mm. A correlation based on their experiments was given as:

$$\alpha_{tp} = 30\,\mathrm{Re}_{lo}^{0.857}\,Bo^{0.714} k_l / d_h \tag{2}$$

where, Re_{lo} is the Reynolds number with only liquid flowing in the tube; Bo is the boiling number.

Tran et al. (1996) proposed a correlation based on their experiments with a wide range of parameters, i.e qualities up to 0.94, a mass flux range of 44-832 kg/m^{2}s, and a heat flux range of 3.6-129 kW/ m^2, given below as Equation (3).

$$\alpha_{tp} = 840000\left(Bo^2 We_l\right)^{0.3}\left(\frac{\rho_l}{\rho_g}\right)^{0.4} \tag{3}$$

where We_l is the Weber number, in which the surface tension is taken into account,

$$We_l = \frac{G^2 d_h}{\rho_l \sigma}$$

Gungor and Winterton (1986) developed a general correlation for forced convection boiling in vertical and horizontal tubes with the aid of a data base, which consists of over 4300 data points for water, refrigerants and ethylene glycol. The basic form of the correlation is:

$$\alpha_{tp} = E\alpha_l + S\alpha_{pool} \tag{6}$$ 4 and 5?

where α_l was given by the Dittus-Boelter equation for liquid only flowing in the tube.

$$\alpha_l = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4}\,k_l / d$$

and, α_{pool} was proposed by Cooper (1984) as

$$\alpha_{pool} = 55P_r^{0.12}(-\log_{10} P_r)^{-0.55} M^{-0.5} q^{0.67}$$

E is the enhancement factor given as

$$E = 1 + 24000 B_O^{1.16} + 1.37\left(\frac{1}{X_{tt}}\right)^{0.86}$$

and X_{tt} the Martinelli parameter $\left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_f}{\mu_g}\right)^{0.1}$

S is the boiling suppression factor given by

$$S = \left[1 + 115\times10^{-8} E^2 \,\mathrm{Re}^{1.17}\right]^{-1}$$

The proposed range of applicability of this correlation is: tube diameter d = 2.95 – 32 mm, system pressure P = 0.08 – 202.6 bar, mass flux G = 12.4 – 61518 kg/m^{2}s, heat flux q = 350 – 9.1534 × 10^7 W/m^2.

Kandlikar (1983) developed a correlation for predicting saturated flow boiling heat transfer coefficients inside horizontal and vertical tubes. It was based on a model utilizing the contributions due to nucleate boiling and convective mechanisms. It incorporated a fluid-dependent parameter F_{fl}. The earlier correlation was further refined in Kandlikar (1990) by expending the data based to 5246 data points from 24 experimental investigations with ten fluids. The form of the proposed correlation is:

$$\alpha_{tp} = \max\left(E', S'\right)\alpha_l \tag{7}$$

where

$$E' = 0.6683C^{-0.2} f\left(Fr_l\right) + 1058Bo^{0.7}F_{fl}$$

$$S' = 1.136C^{-0.9} f\left(Fr_l\right) + 667.2Bo^{0.7}F_{fl}$$

$$\alpha_l = \frac{0.023\,\mathrm{Re}_l^{0.8}\,\mathrm{Pr}_l^{0.4}\,k_l}{d_h}$$

$$C = \left(\frac{1-x}{x}\right)^{0.8}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}$$

$$Bo = \frac{q}{Gh_{fg}}$$

$$Fr_l = \frac{G^2}{\rho_l^2 g d_h}$$

$$f\left(Fr_l\right) = 1 \quad if \quad Fr_l \geq 0.04$$

An experimental facility was designed and constructed during this study to allow a detailed and accurate investigation of the effect of diameter on flow patterns, heat transfer mechanism and rates and pressure drop. The test rig can use a range of working fluids including refrigerants and water. In this study R134a was used. Some of our heat transfer results and the comparisons with the correlations described above are presented in this paper.

5.2 EXPERIMENTAL FACILITY AND PROCEDURE

The experimental facility is shown schematically in Figure 1. It consists of (a) a refrigerant circulating pump; (b) two Coriolis mass flow meters for measuring high and low flow rates thus ensuring high measurement accuracy; (c) a preheater, (d) a chiller for subcooling the refrigerant; (e) two test sections, namely the heat transfer rate measurement test section and the flow pattern observation test section; (f) condensers and a tank which is used to receive liquid refrigerant. A detailed description is available in Huo et al. 2004 and will not be repeated here – only a summary is given. A heater in the R134a tank enabled system pressure control. An energy balance based on the heat supplied and the enthalpy change enabled the exit quality to be calculated. The total enthalpy change across the test section was calculated based on the flow rate of the refrigerant and the pressure and temperature change measured by the differential pressure transducer and thermocouples, respectively, at two ends of the test section. Cooling at the condenser and chiller, see Figure 1, is provided by an R22 plant, the details of which are not shown here.

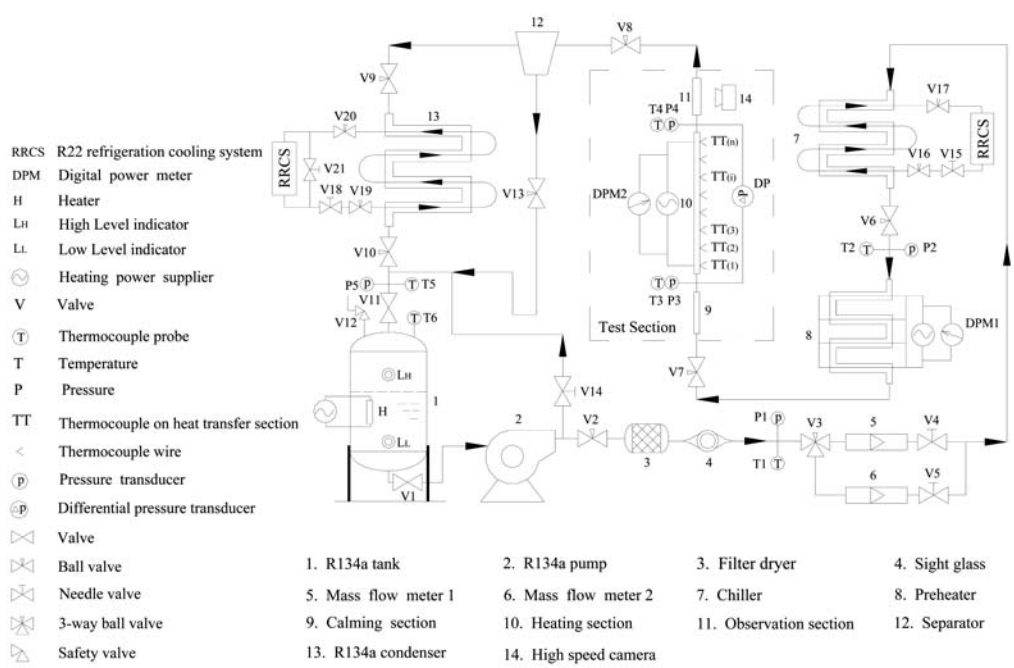

Figure 1. Schematic diagram of the flow boiling experimental facility

Local flow boiling heat transfer coefficients and flow patterns for R134a were obtained for the range: pressure 8, 10, 12, 14 bar, heat flux 13-150 kW/m^2, mass flux 100-500 kg/m^{2}s, vapour quality 0-0.9 and tube diameter 2.01 and 4.26 mm.

In the heat transfer experiments, the fluid entered the test section in a subcooled state and was evaporated to a quality of about 90% or less in most cases, depending on the mass flux and the heat flux. Direct electric heating was applied to the test section. Thirteen K-type thermocouples were soldered to the outside of the tube to provide the wall temperatures. T-type thermocouples and pressure transducers were used to measure inlet and outlet temperatures and pressures. A Pyrex glass tube for flow pattern observation was located immediately downstream of the heat transfer test section. A digital high-speed camera (Phantom V4 B/W, 512 x 512 pixels resolution, 1000 pictures/sec with full resolution and maximum 32000 pictures/sec with reduced resolution, 10ms exposure time) was used to observe the flow patterns. The results of the flow visualization part of this study is presented in detail in Chen et al. (2004).

All the instruments used were carefully calibrated. The uncertainty in temperature measurement was ± 0.2 K, flow rate measurements ± 0.4%, and pressure measurements ± 0.15 %. The average error in the heat transfer coefficient was ± 6 %. A series of flow boiling tests were performed at different mass flux and heat flux. During these tests, the inlet temperature was controlled by adjusting the capacity of the chiller and heating power to the preheater. The flow rate was set to the required value and the heat flux was increased gradually until superheated flow. The data were recorded after the system was steady - normally it took about 15 minutes but sometimes longer. Each recorded parameter was the average of 20 data. The test was then repeated at a different flow rate.

The local heat transfer coefficient at each thermocouple point was determined from the following equation:

$$\alpha = \frac{q}{T_w - T_f} \tag{8}$$

where T_w is the local inner wall temperature, T_f is the local fluid temperature and q is the inner wall heat flux to the fluid. T_f was deduced from the fluid pressure, which was determined based on the assumption of a linear pressure drop through the test section. T_w was calculated based on the outside surface temperature, recorded by the thermocouples, the heat generated by the direct electric heating and the tube wall thermal resistance. The heat lost to the ambient, ΔQ, was included in the calculation. It was obtained from single-phase experiments, Huo et al. 2004. The vapour quality (x) was determined based on the heat transferred to the fluid. It is given by:

$$x_i = \frac{h_i - h_l}{h_g - h_l} \tag{9}$$

where h_l and h_g are the specific enthalpy of saturated liquid and vapour, respectively. hi is the local specific enthalpy of the fluid. This was determined from the enthalpy of the previous section and the heat transferred to the fluid, i.e.

$$h_i = h_{i-1} + \frac{L_i}{mL}(Q - \Delta Q)$$

where the heat transfer (Q) is equal to the product of the voltage and the current applied directly to the test section.

5.3 EXPERIMENTAL RESULTS

The local heat transfer coefficient is plotted as a function of quality in Figures 2 and 3 for the 4.26 and 2.01 mm tubes respectively. At low values of heat flux, when x <0.5 for the 4.26 mm and x <0.3 for the 2.01 mm tube, the heat transfer coefficient depends on the heat flux and is independent of quality. As mentioned earlier, flow patterns were observed with Pyrex glass tubes installed immediately after the stainless steel tube test section. Partial dryout was

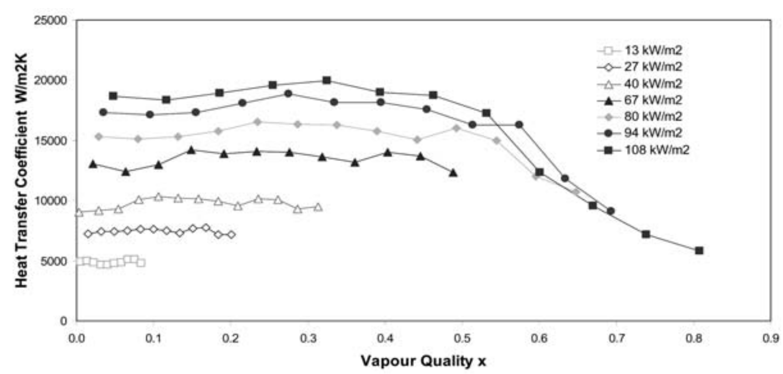

Figure 2. Local heat transfer coefficient as a function of vapour quality with different heat flux; G = 300 kg/m^{2}s, P = 8 bar, d = 4.26 mm.

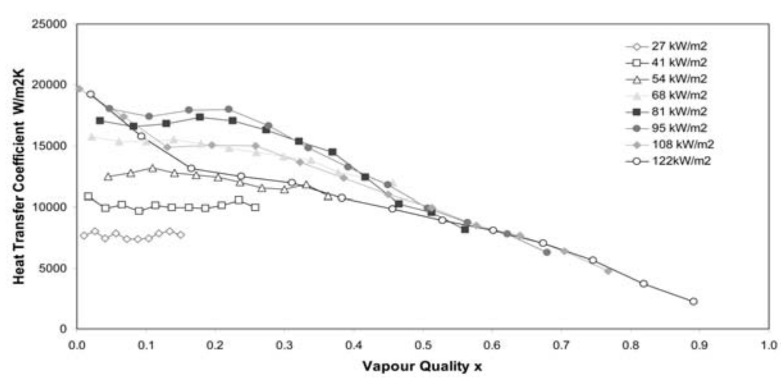

Figure 3. Local heat transfer coefficient as a function of vapour quality with different heat flux; G = 300 kg/m^{2}s, P = 8 bar, d = 2.01 mm.

seen at vapour quality above 40%-50% for the 4.26 mm tube and 20%-30% for the 2.01 mm tube. This finding is very similar to what is reported in Agnostini and Bontemps (2004). At values greater than those mentioned above, the heat transfer coefficient does not depend on heat flux and is strongly dependant on quality. In the small tube, this dependence increases with heat flux and for $q > 100$ kW/m^2 the heat transfer coefficient decreases monotonically with $x > 0$. The decrease in the heat transfer coefficient with vapour quality in the 2.01 mm tube could be due to the fact that the bubble creation frequency is too high and numerous bubbles occupy the tube wall, which may result in a suppression of the nucleate boiling.

The heat transfer coefficient is depicted in Figures 4 and 5 as a function of system pressure for the 4.26 and 2.01 mm tubes respectively. As seen in the figures, it increases with

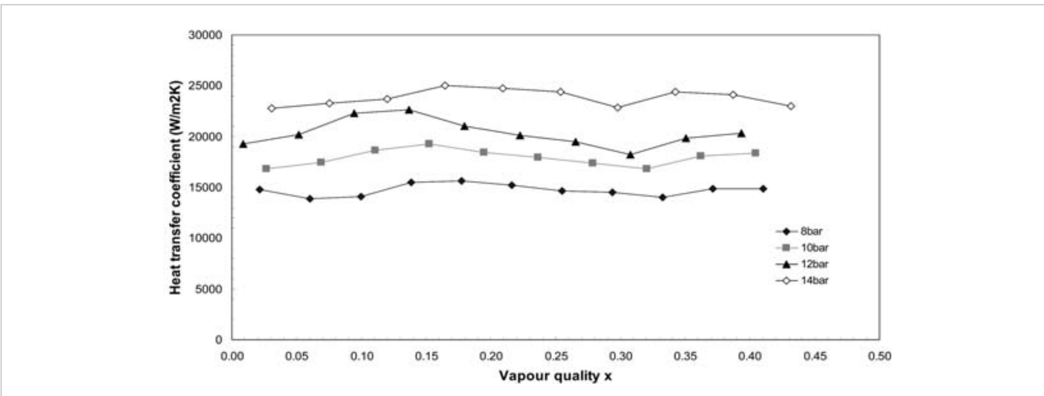

Figure 4. Local heat transfer coefficient as a function of vapour quality with different system pressure, G = 400 kg/m^{2}s, q = 52 kW/m^2, d = 4.26 mm.

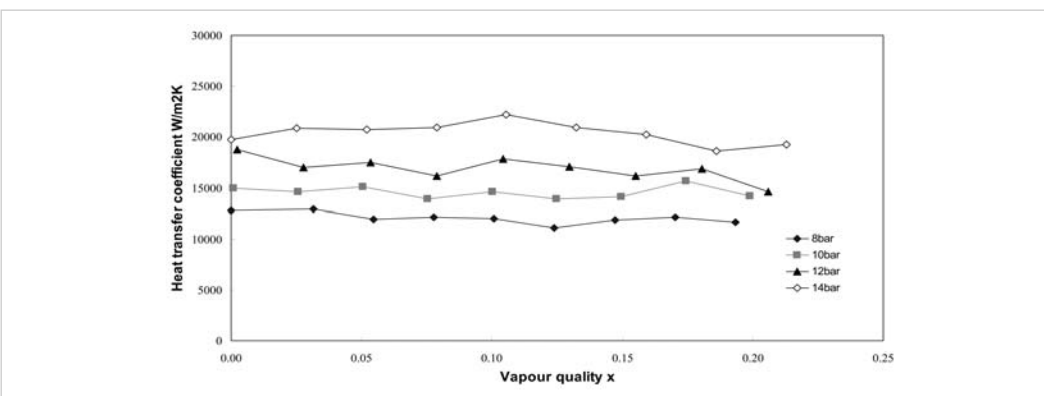

Figure 5. Local heat transfer coefficient as a function of vapour quality with different system pressure, G = 400 kg/m^{2}s, q = 54 kW/m^2, d = 2.01 mm.

system pressure for both tubes. The bubble departure diameter decreases as the system pressure increases. The bubble departure frequency also increases with increase in pressure (Sharma et al. 1996). Therefore, bubble growth and departure from the tube wall is faster at high pressure values for the same heat flux. In nucleate boiling, the disturbance caused by the bubbles growing and escaping from the wall contributes significantly to the total heat transfer rate.

The dependence of the heat transfer coefficient on mass flux is depicted in Figures 6 and 7. As clearly seen in the figure, the heat transfer coefficient is almost independent of the mass flux when the vapour quality is less than about 50% in the 4.26 mm tube and about 40% in the 2.01 mm tube.

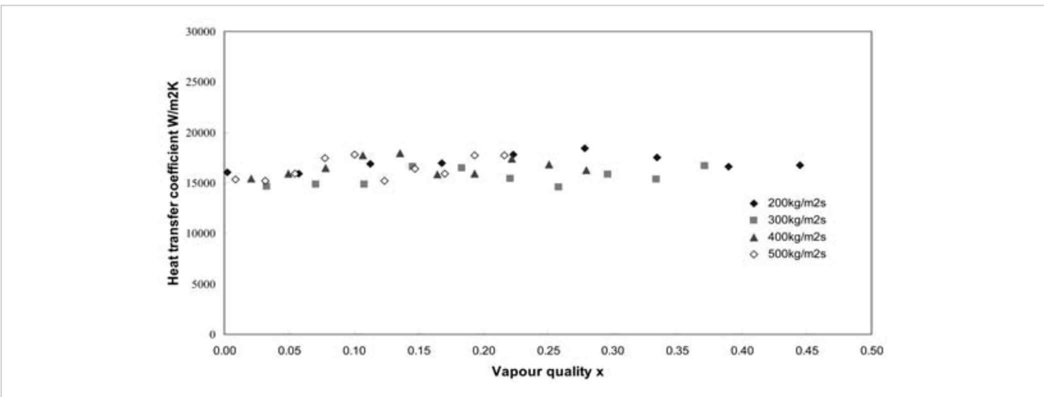

Figure 6. Local heat transfer coefficient as a function of vapour quality with different mass flux, P = 12 bar, q = 54 kW/m^{2}s, d = 4.26 mm.

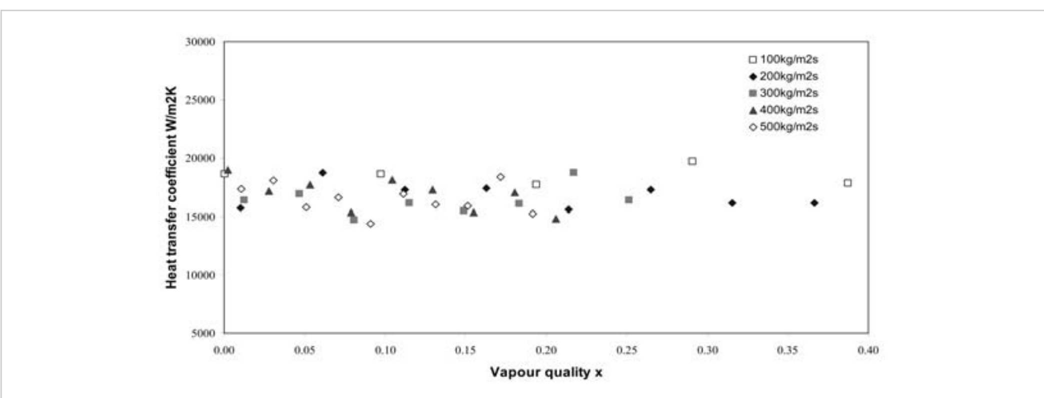

Figure 7. Local heat transfer coefficient as a function of vapour quality with different mass flux, P = 12 bar, q = 54 kW/m^{2}s, d = 2.01 mm.

5.4 COMPARISON WITH EXISTING CORRELATIONS

The heat transfer results were compared with some of the existing correlations. The results of this comparison are presented below.

5.4.1 The Lazarek and Black Correlation

Figures 8 – 11 depict the comparison of the present experimental results for the 4.26 mm and 2.01 mm tubes for system pressures of 8 bar and 12 bar with the Lazarek and Black (1982) correlation presented in the introduction. As seen in this set of figures, the correlation under

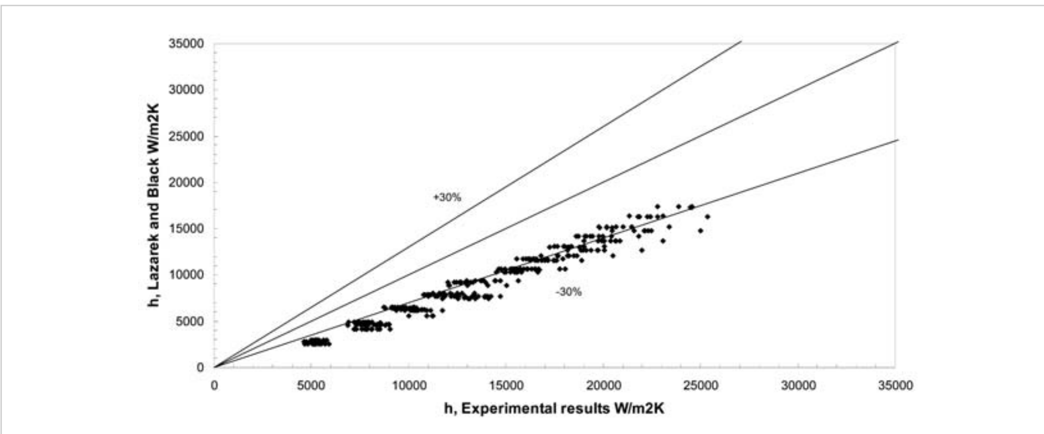

Figure 8. Heat transfer results compared with the Lazarek and Black correlation, P = 8 bar, d = 4.26 mm.

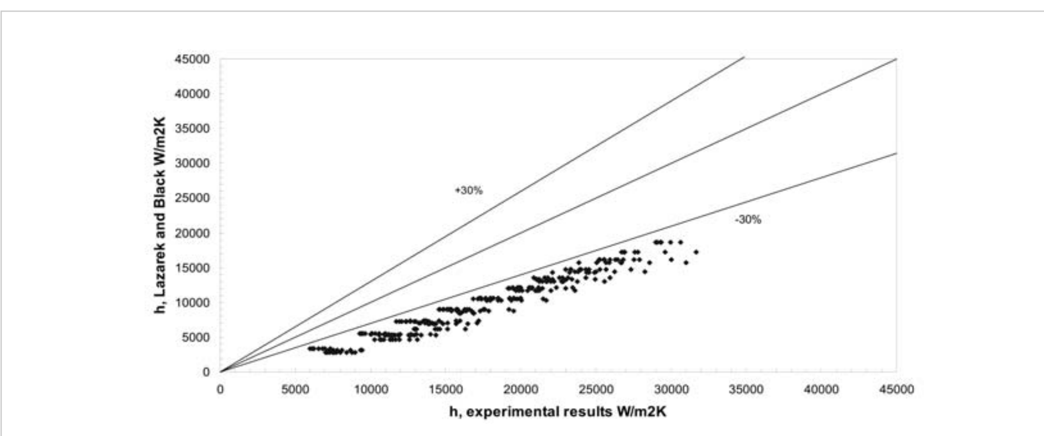

Figure 9. Heat transfer results compared with the Lazarek and Black correlation, P = 12 bar, d = 4.26 mm.

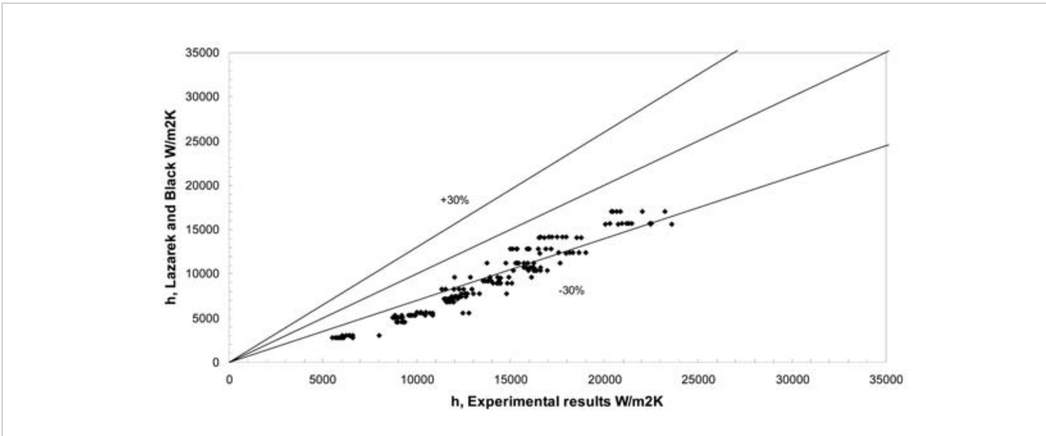

Figure 10. Heat transfer results compared with the Lazarek and Black correlation, P = 8 bar, d = 2.01 mm.

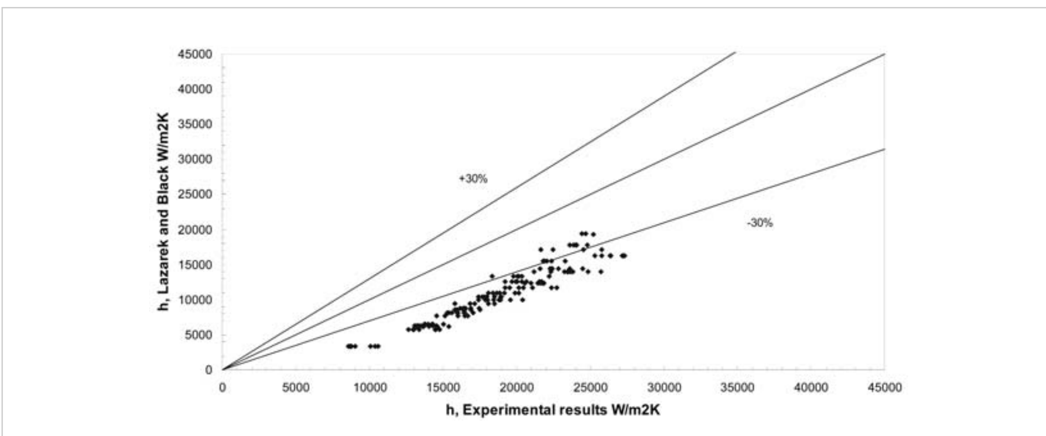

Figure 11. Heat transfer results compared with the Lazarek and Black correlation, P = 12 bar, d = 2.01 mm.

predicts by more than 30% the present data. However, there is some improvement in the agreement at lower pressure (8 bar) and in the smaller diameter tube.

This could be due to the fact that this correlation was proposed based on very limited experimental results, in which the pressure varied from 1.3 to 4.1 bar. The reduced pressure defined as P/P_c was from 3.8% to 12%. In our experiments, the corresponding reduced pressure for the 8 bar and 12 bar values were 19.7% and 29.6%. These are out of the possible application range of this correlation; but 8 bar is closer than 12 bar.

5.4. 2 The Gungor and Winterton Correlation

The comparison of the present data with the data predicted by the correlation presented by Gungor and Winterton (1986) is depicted in Figures 12-13 for the pressure of 8 bar. The comparison is somewhat better than that obtained with the Lazarek and Black (1982) correlation. The results are mostly lower that 30% of the predicted values for the high pressure of 12 bar in the 4.26 mm tube. However, they are mostly within 30% for the lower pressure (8 bar) in the 4.26 mm tube and for both lower and higher pressure in the 2.01 mm tube. The reason for this is because this correlation was developed based on a very large data bank, which almost covers our experimental parameter range. However, R134a was not included in this data bank, and very importantly most data were from tubes with diameter larger than 5 mm.

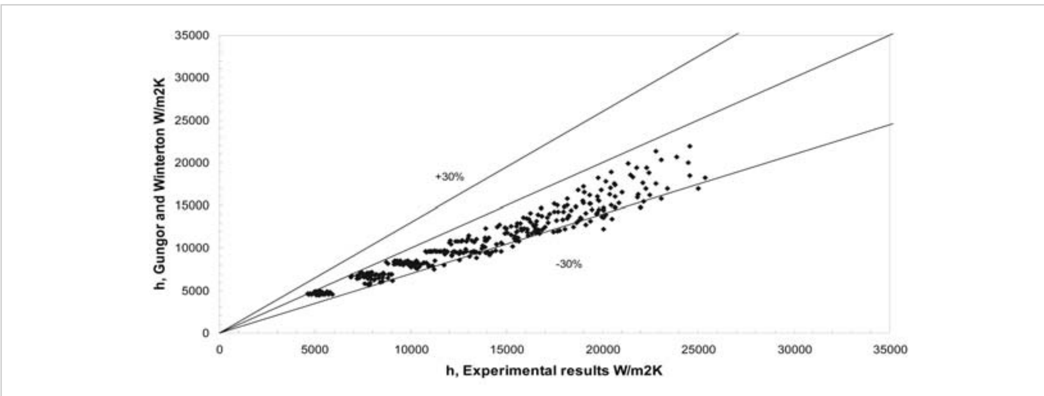

Figure 12. Heat transfer results compared with the Gungor and Winterton correlation, P = 8 bar, d = 4.26 mm.

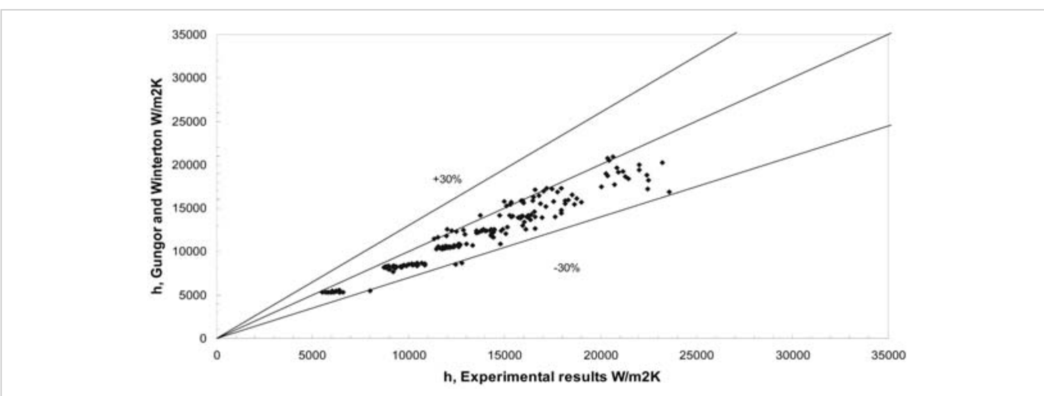

Figure 13. Heat transfer results compared with the Gungor and Winterton correlation, P = 8 bar, d = 2.01 mm.

5.4. 3 The Tran et al. Correlation

Figures 14-15 depict the comparison of the results obtained for the pressure of 8 bar with the correlation proposed by Tran et al. (1996). As seen in the figures, the correlation predicts the present experimental results for the 4.26 mm tube well, i.e. within ±30%.

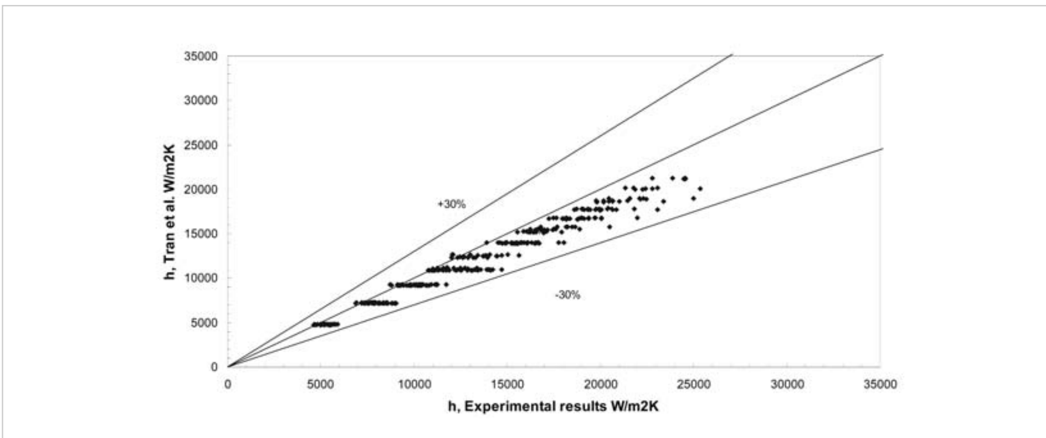

Figure 14. Heat transfer results compared with the Tran et al. correlation, P = 8 bar, d = 4.26 mm.

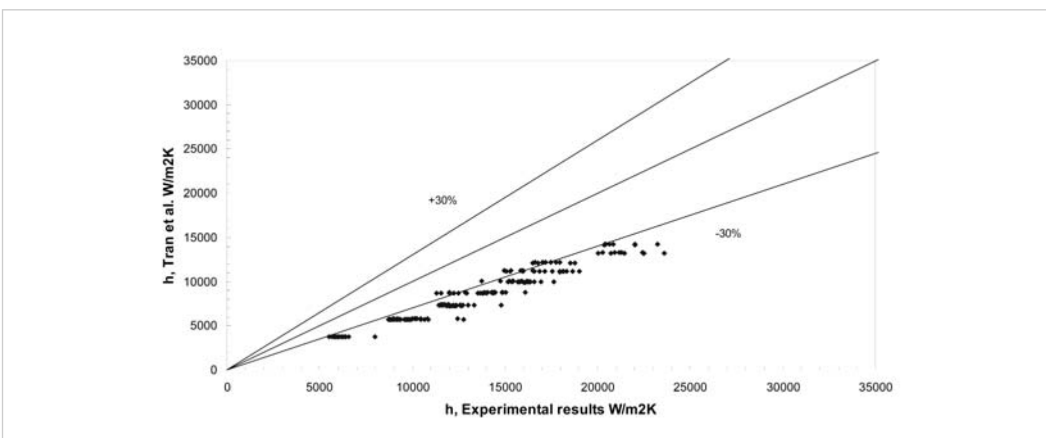

Figure 15. Heat transfer results compared with the Tran et al. correlation, P = 8 bar, d = 2.01 mm.

However, the data for the 2.01 mm tube are underestimated by this correlation. There was no pressure effect in these comparisons.

5.4.4 The Kandlikar Correlation

Figures 16-17 depict the comparison of the results predicted by the correlation proposed by Kandlikar (1990) and the present experimental results. This correlation underestimates the experimental results both for the 4.26 mm and 2.0 1mm tubes, i.e. by about -30% – 50%. The prediction is better for the lower pressure studied, seen in the figures. There is no obvious diameter effect in this comparison, i.e. the disagreement is similar for both diameters studied.

The current comparisons include the data for which the heat transfer coefficient does not depend on quality, i.e. up to values of vapour quality 50% and 30% for the 4.26 mm and 2.01 mm tube respectively (see also figures 2 and 3). Current work is underway to include all data and compare with more recent correlations such as the one proposed by Thome at al. (2004).

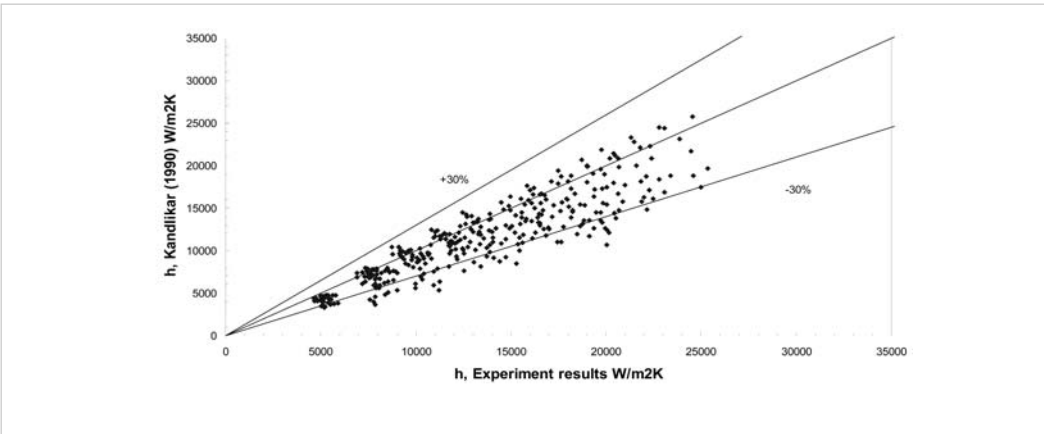

Figure 16. Heat transfer results compared with the Kandlikar correlation, P = 8 bar, d = 4.26 mm.

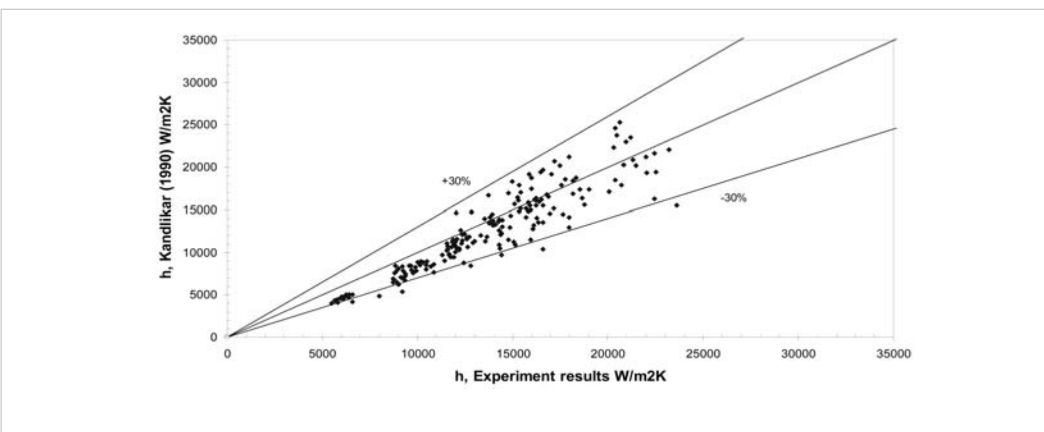

Figure 17. Heat transfer results compared with the Kandlikar correlation, P = 8 bar, d = 2.01 mm.

5.5. CONCLUSIONS

The experimental results, clearly demonstrate that in the 4.26 mm tube when the vapour quality was less than about 40% to 50%, the heat transfer coefficient increases with heat flux and system pressure, but does not change with vapour quality. For the 2.01mm tube, this boundary moves to 20% - 30% vapour quality. Partial dryout was seen after the vapour quality above, i.e. 40%-50% for the 4.26 mm tube and 20%-30% for the 2.01 mm tube. A comparison of the present results was made with existing correlations. The conclusion reached is that the existing correlations can not predict heat transfer data to any satisfactory degree. This could be due to a variety of reasons including the range of the current parameters in relation to the database used by other researchers, e.g. pressure, diameter and fluids used. Further work is needed and is underway to provide a correlation that represents better data at this range for refrigerants. Further work is also needed to clarify the heat transfer mechanism and the corresponding flow patterns and dependence of the heat transfer coefficient on the controlling parameters.

NOMENCLATURE

Bo	Boiling number, $q/G\ h_{fg}$
Co	Confinement number
d	Diameter, m
Fr	Froude number, $G^2/p_1^2 g_{dh}$
G	Mass flux, kg/m² s
g	Gravitational acceleration, m/s²
h	Enthalpy, J/kg
k	Thermal conductivity, W/m• K
L	Length, m
M	Molecular weight
m	Mass flow rate, kg/s
Nu	Nusselt number $Nu = {}^{d}/_{k} \cdot \alpha$
P	Pressure, Pa
Pr	Prandtl number, $Pr = cp \cdot \mu/k$
Q	Heat, W
q	Heat flux, W/m^2
Re	Reynolds number, $Re = G \cdot d/\mu$
T	Temperature, K
We	Weber number, $G^2 d_h/\rho_l \sigma$
x	Quality

Greek Symbols

α	Heat transfer coefficient, W/(m^2 K)
Δ	Finite increment
ρ	Density, kg/m^3
σ	Surface tension, N/m

Subscripts

c	critical
g	fluid
g	gas
h	hydraulic
i	index
in	inside
l	liquid
lo	liquid only
pool	Pool boiling
r	reduced
w	wall

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