Two-phase Heat Transfer in Small Passages and Microfinned Surfaces – Fundamentals and Applications.

By
Professor Tassos Krayiannis, PhD, FEI, MIMechE, MInstR, MCIBSE (Brunel University), Professor John Rose, PhD, DSc, FIMechE, FASME (Queen Mary, London University) and Dr. Raya AL-Dadah, PhD, MInstR (University of Birmingham).

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Summary

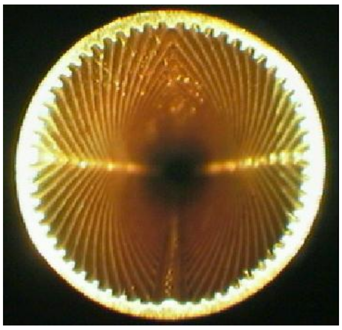
Micro channels and internally finned tubes are increasingly being utilized in the evaporators and condensers of refrigeration systems. The adoption of such geometries in the development of micro-cooling systems is first discussed in this paper. Recent work on flow boiling heat transfer and condensation in small to micro passages as well as on microfinned surfaces is then presented. The complex effect of diameter size on flow boiling patterns and heat transfer and correlations currently available in literature are summarized. Condensation in microfinned tubes and microchannels is then discussed.

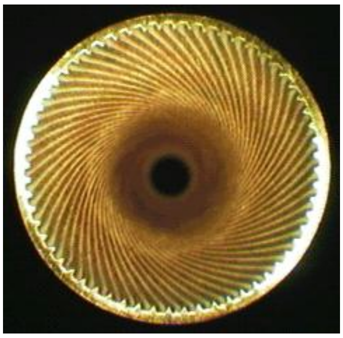
1. Introduction

Boiling and condensation heat transfer provide the most probable means of transferring the high heat fluxes encountered in many engineering applications including compact heat exchangers used in the refrigeration, power and process industries and, at an even smaller scale, in cooling systems for electronic devices.

In the condensers and evaporators of many refrigeration and air-conditioning applications, air or water flows over tubes while condensation or evaporation of refrigerant takes place inside the, often internally microfinned, tubes, see Fig. 1. The outside thermal resistance is generally higher than that of the phase change side, particularly in the air-cooled case. However, through improvements in external enhancement (louvered fins), a significant additional advantage can be obtained by increasing the phase change side heat-transfer coefficient.

Attention has also been given in recent years to condensation and evaporation in small (hydraulic diameter around 1 mm) channels, often termed micro- or mini-channels. The advantage of using





Herringbone microfins Spiral microfins

Fig.1. Microfinned tubes (Reproduced by permission of Professor H. Honda, Kysuhu University)

passages of the types illustrated in Fig. 2 has become empirically well established in motor vehicle air conditioner condensers and evaporators over the past 15 to 20 years. Condensers and evaporators employing small hydraulic diameter multi-channel tubes are presently under consideration for larger air-conditioning and refrigeration applications. Such tubes have additional advantages of reduced airside pressure drop and ability to support high internal pressure.

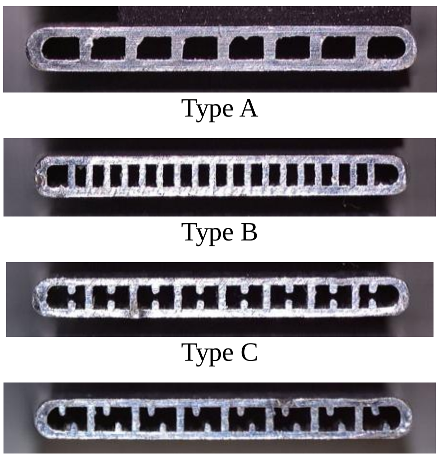


Fig. 2 Photographs of multi-port extruded tubes used by Koyama et al. [1, 2].

The use of such intricate passages with high heat transfer performance and high surface area per volume, is essential not only to the development of compact, lightweight and efficient large cooling systems but also to developing small capacity cooling systems. Cooling systems with overall dimensions of few centimetres and cooling load capabilities up to 500 W are termed mesoscopic-cooling systems while those with cooling capacities of few watts, and with dimensions of a few centimetres but with thickness of few millimetres are known as micro-cooling systems. Drost [3] and Drost and Frederich [4] reported on the development of a mesoscopic water / lithium bromide vapour absorption cooling system to be used for personal cooling. They used arrays of micro channels with channel widths up to 300 microns and channel depths up to 1 mm to construct the evaporator and condenser. In the evaporator, boiling heat transfer coefficients of 10 to 20 kW/m^2K and heat fluxes up to 1 MW/m^2 were obtained. These values exceed those of conventional evaporators by a factor of 4. In the condenser, heat fluxes in excess of 300 kW/m^2 were attained. For the absorber and desorber, they used a micro machined contactor to constrain the incoming solution to form an ultra thin film of approximately 100 μm thus enhancing the heat and mass transfer performance of both the absorber and desorber. The absorber and desorber developed were shown to have capabilities 10 times higher than the conventional ones. The cooling system has a total mass of 4.7 kg, dimensions of 20 cm x 22 cm x 8 cm, COP of 0.68-0.71 and cooling capacity of 350 W. The specific cooling density (100 kW/m^3) is 40 times higher than that of a conventional absorption system. US patent No. 0040129018 [5] describes a mesoscopic vapour compression cooling system with cooling density of up to 210 kW/m^3.

Micro-cooling systems are being developed using silicon technology where the whole cooling system is etched on a silicon wafer. An example is the work currently undertaken at the University of Illinois and aims to establish a vapour compression system on a thin flexible wafer [6]. The development of such micro cooling systems may in the future be utilised for heating and / or cooling of buildings, cars, microprocessors or being integrated in products as diverse as prefabricated wall elements, or in special suits for use in extreme conditions.

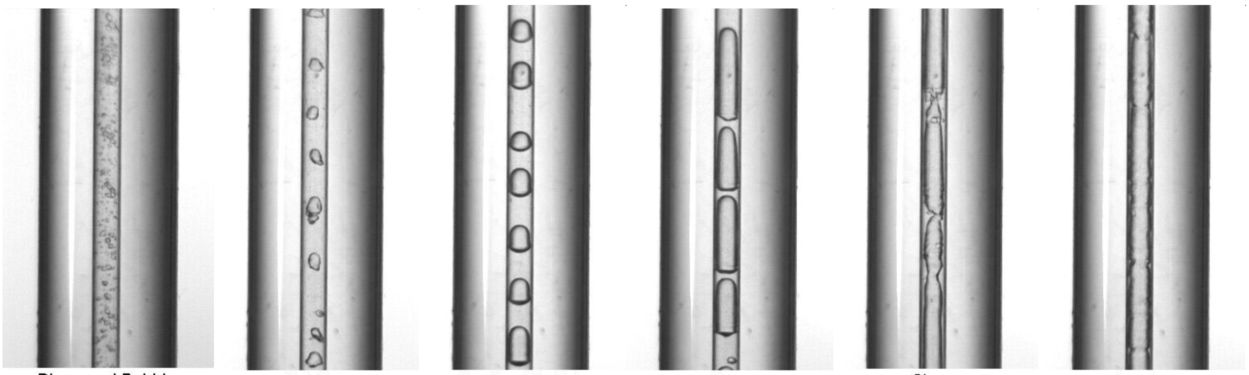
In this paper we report recent research undertaken to understand the two-phase heat transfer processes in small / micro channels and on microfinned surfaces, which is fundamental for progress in the above and other systems.

2. Flow Boiling Heat Transfer in single small / micro channels

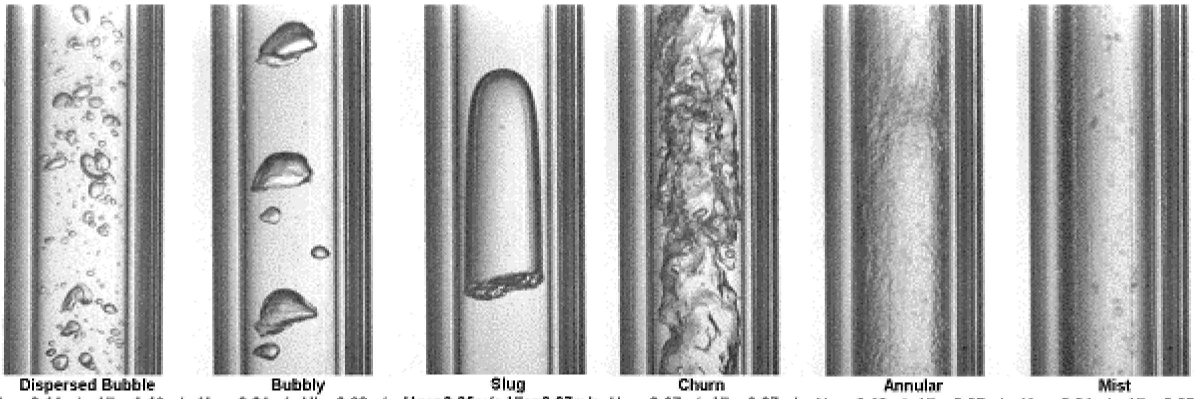
Boiling inside small / micro channels was investigated by a number of researchers since the early 80s. Huo et al. [7] reviewed the various aspects of boiling in small channels with particular attention to flow patterns, heat transfer mechanism (e.g. nucleate boiling or convective boiling) and critical heat flux. They concluded that the flow patterns in small passages are significantly different from those in traditional size tubes due to the increased influence of surface tension. A commonly used threshold criterion to distinguish between macro and micro scale effects was proposed by Kew and Cornwell [8] as a function of the confinement number, $Co = [\sigma / g(\rho_l - \rho_g)]^{1/2} / d$, which depends on the surface tension and the densities of the liquid and vapour and thus on the system pressure. As the diameter decreases or the Co number increases above the threshold given by this criterion, i.e. 0.5, bubble growth is confined by the channel to the point where individual bubbles grow in length rather than in diameter, i.e. confined bubble flow. The above is seen in Fig. 3 from Chen et al. [9], with the confined flow seen in the smaller of the two tubes. They reported that for R134a and P=6-14 bar confinement effects start at ~2.0 mm. For the same pressure range the Kew and Cornwell criterion gives threshold diameters of 1.7 and 1.4 mm.

Effect of diameter on flow boiling regimes and heat transfer

The effect of diameter on flow patterns was presented by Chen et al. [9] for R134a, P=6-14 bar and d=1.1 mm to 4.26 mm. Twelve flow pattern maps were drawn and compared with models



(a) 1.10 mm internal diameter tube.



(b) 4.26 mm internal diameter tube

Fig. 3. Flow patterns observed in the 1.1 mm and 4.26 mm internal diameter tubes at 10 bar.

for normal size tubes indicating significant differences in the 4.26 mm tube and more so for the smaller tubes examined. The boundaries of slug to churn and churn to annular moved to higher vapour velocity whilst the dispersed bubble to bubbly boundary moved to higher liquid velocity when the diameter changed from 4.26 to 1.1 mm. The diameter did not affect the dispersed bubble to churn and bubbly to slug.

It has been reported that heat transfer increases as the tube diameter decreases, Yan and Lin [10]. They reported that their measured heat transfer rates for R134a were higher (by 30-80%) in a 2 mm diameter tube when compared to calculated values for an 8 mm tube. In large diameter tubes/channels, the flow patterns are usually annular for the largest range of quality and the convective heat transfer mechanism dominates. 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 and this is still a subject of investigation. Huo et al. [11] conducted heat transfer experiments with two stainless steel tubes of internal diameter 4.26 and 2.01 mm (same facility as Chen et al. [9]). They concluded that for the same heat flux, the heat transfer coefficient in the 2.01 mm tube was higher that that in the 4.26 mm tube, see Fig. 4, [12]. However, they obtained a complex dependence of the heat transfer coefficient on quality and heat flux. At

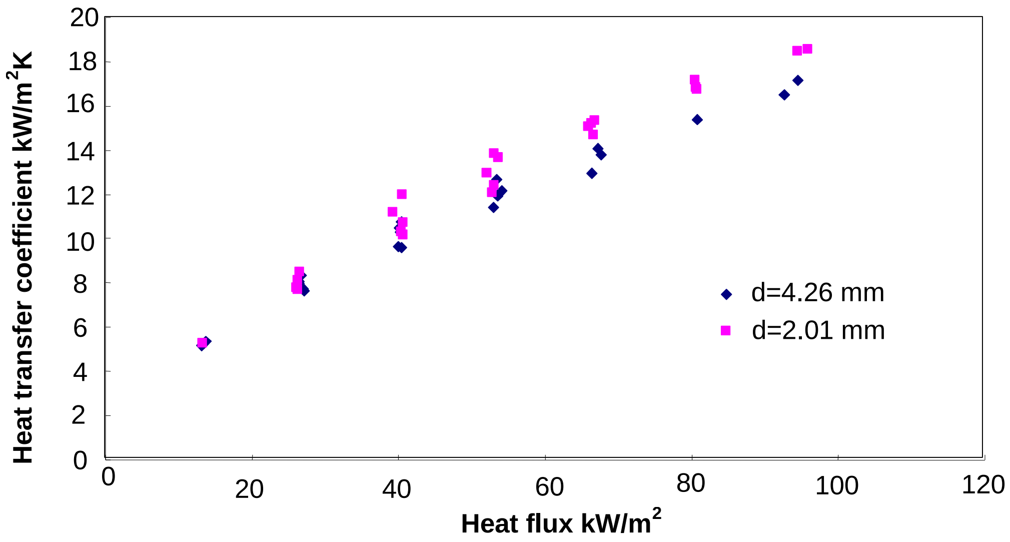


Fig. 4. Effect of diameter on heat transfer coefficient in small diameter tubes

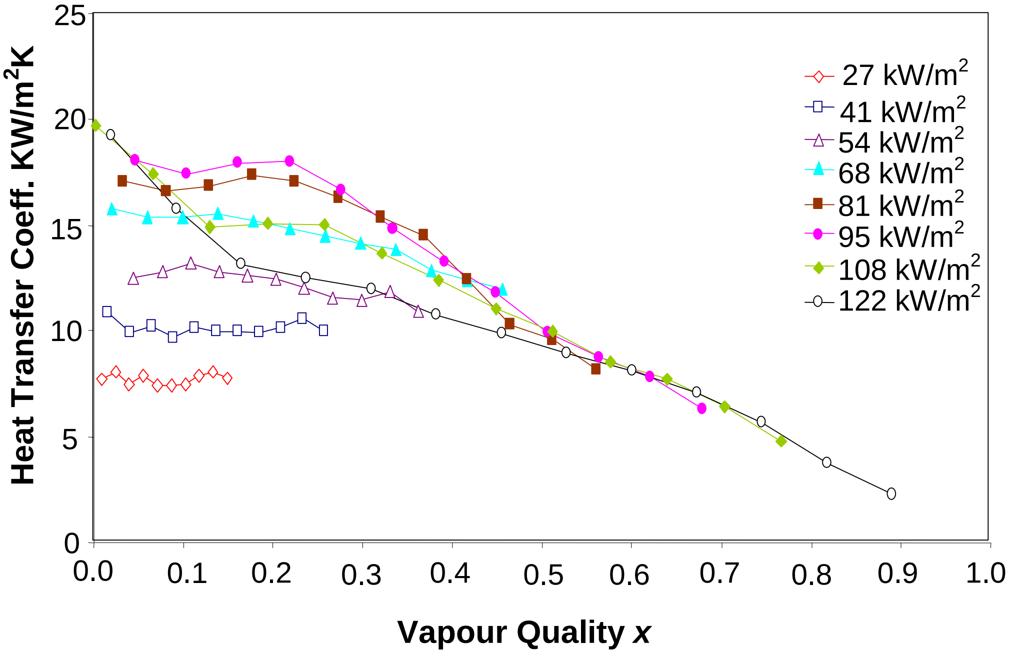


Fig.5. Local heat transfer coefficient as a function of vapour quality with different heat flux; G = 300 kg/m^{2}s, P = 8 bar, d = 2.01 mm.

low values of heat flux, when quality x <0.5 for the 4.26 mm, the heat transfer coefficient depends on the heat flux and is independent of quality. The detailed results for the 2.01 mm tube are depicted in Fig. 5 and demonstrate that this is so for a smaller range of quality, i.e. x < 0.3. At quality 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 monotonic decrease in the heat transfer coefficient with vapour quality in the 2.01 mm tube could be due to the fact that partial dryout occurs at these conditions. During this study, flow patterns were observed at the Pyrex glass tubes installed immediately after the stainless steel tube test sections. Partial dryout was seen at vapour quality above 40%-50% for the 4.26 mm tube and 20%-30% for the 2.01 mm tube.

The fact that the heat transfer coefficient was independent on quality and dependent on heat flux was generally interpreted as indicative of the dominance of nucleate boiling (see Lazarek and Black [13] and Tran et al. [14]). However, Thome et al. [15] noted that for small passages the same behaviour can be explained based on the transient evaporation of the thin liquid film surrounding the elongated bubble, see Fig. 3, being the dominant heat transfer mechanism and not nucleate boiling.

Dupont and Thome [16] discussed the effect of diameter in the microscale range in some detail. They noted the work of Owhaib and Palm [17] for R134a who found that the average heat transfer coefficient increased when the diameter decreased from 1.7 to 0.8 mm. Baird et al [18] on the other hand reported that there was no significant effect for a diameter ranging from 0.92 to 1.95 mm for R123 (slightly outside the range of micro according to Kew and Cornwell). Dupont and Thome [16] used the three-zone model, see next section, to study the effect of diameter. They reported that the effect of diameter on the heat transfer coefficient α depends on the vapour quality, i.e. for $x < 0.04$ α decreases with d: for $0.04 < x < 0.18$ α increases with d to a maximum and then decreases and for $x > 0.18$ it increases with d. As mentioned above, such complex behaviour requires further research and clarification.

Correlations of flow boiling in small channels

Correlations cited frequently in literature for predicting the flow boiling heat transfer coefficients in small/micro channels are given in Table 1. Some of the correlations involve a number of equations and steps until the final form given in the table. These are not shown here due to space limitation and the interested reader is referred to the original references.

For conventional channels, the correlations of Chen [19], Shah [20], Gunger and Winterton [21], Kandlikar [22], Liu and Winterton [23] and Steiner and Taborek [24] are widely used. These correlations were evaluated for predicting heat transfer coefficients obtained in small channels by Zhang et al. [25] who used 1203 data points from 13 experimental investigations using water and refrigerants (such as R11, R113, R12) inside horizontal and vertical circular and rectangular channels of hydraulic diameters 0.78-6 mm. Zhang et al. [25] concluded that the Chen correlation gave the best predictive accuracy and produced a generalized form that takes into account the nature of flow in the small channels. The generalized correlation predicted the 1203 data bank with mean absolute deviation of 18.3%.

Owhaib et al [17] used the Lazarek and Black [13], Kew and Cornwell [8], the Tran et al. [14] and Yu et al. [26]

Table 1. Correlations of flow boiling in small channels.

Author(s)	Correlation	Fluid/ Test Parameters	Channel	Remarks
Lazarek and Black, 1982 [13]	$\alpha_{tp} = 30\,\mathrm{Re}_{lo}^{0.857}\,Bo^{0.714} k_l / d_h$	R113	vertical tube of d= 3.1mm	Small channel correlation
Kew & Cornwell (1994) [8]	$\alpha_{tp} = 30\,\mathrm{Re}_{l}^{0.857}\,Bo^{0.714}(1-\chi)^{-0.143} k_l / d_h$	R141b		
Tran et al., 1996 [14]	$\alpha_{tp} = 840000\left(Bo^2 We_l\right)^{0.3}\left(\frac{\rho_l}{\rho_g}\right)^{0.4}$	x up to 0.94, G = 44-832 kg/m²s, q = 3.6-129 kW/ m²		Small Channel correlation
Yu et al. (2002) [26]	$\alpha_{tp} = 640000\left(Bo^2 We_l\right)^{0.27}\left(\rho_l / \rho_g\right)^{0.2}$	Water G = 50 – 200 kg/m^{2}s	d = 2.98 mm	
Warrier et al. (2002) [27]	$\alpha_{tp} = \mathrm{E}\alpha_l$ $\mathrm{E} = 1 + 6\mathrm{Bo}^{1/16} + \mathrm{f}(\mathrm{Bo})\mathrm{x}^{0.65}$ $\mathrm{f}(\mathrm{Bo}) = -5.3(1-855\mathrm{Bo})$	FC84, x = 0.03 – 0.55 G = 557 – 1600 kg/m^{2}s q = 0 – 59.9 kW/m^2	d = 0.75 mm, L/d = 409.8	
Zhang et al, 2004 [25]	$\alpha_{tp} = E\alpha_l + S\alpha_{pool}$ $S = 1/(1+2.53*10^{-6})\,\mathrm{Re}_l^{1.17}),\ F = Max(F',1)$ $\mathrm{F}' = 0.64\phi_l,\ \phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}$	Water and refrigerants P=0.101-1.21MPa, q=2.95-2511kW/m^2 G=23.4-2939kg/m^{2}s	Vertical and horizontal round and rectangular channels	Macrochannel correlation generalised for microchannels
Thome et al. (2004) [15]	$\alpha(z) = \frac{t_l}{\tau}\alpha_l(z) + \frac{t_{film}}{\tau}\alpha_{film}(z) + \frac{t_{dry}}{\tau}\alpha_g(z)$ $t_l = \tau/(1+\frac{\rho_l}{\rho_g}\frac{x}{1-x}),\ t_v = \tau/(1+\frac{\rho_g}{\rho_l}\frac{1-x}{x})$, $t_{dry\ film}(z) = \frac{\rho_l \Delta h_{lg}}{q}\left[\delta_0(z) - \delta_{min}\right]$, $\alpha_{film}(z) = \frac{\lambda_1}{\delta_0 - \delta_{end}}\ln\left(\frac{\delta_0}{\delta_{end}}\right)$	R11, R12, R113, R123, R134a, R141b and CO_2 x=0.01-0.99 G=50-564kg/m^{2}s P=124-5766kPa q=5-178kW/m^2	Round tubes with d=0.77-3.1mm	α_l and α_g were determined using the London and Shah and the Gnielinski correlations respectively

correlations to predict their results of boiling R134a inside vertical round tubes of d = 1.7-0.8 mm. The Tran et al. [14] correlation overpredicted their results by more than 30%. The Yu et al. [26] correlation predicted the 0.8 mm tube results within 30% and overpredicted the larger tubes results. On the other hand, both the Lazarek and Black [13] and Kew and Cornwell [8] correlations predicted the large tubes (d=1.7 and 1.2 mm) within 30% and underpredicted the small diameter tube results by more than 30%.

Huo et al. [11] compared their experimental results (R134a, P=8-12 bar and d=2.01 and 4.26 mm) with the correlations of Lazarek and Black [13], Gungor and Winterton [21], Tran et al. [14] and Kandlikar [22]. The Lazarek and Black correlation underpredicted the data by more than 30%. The Tran et al. correlation predicted the experimental data for the 4.26 mm tube within 30% but underestimated significantly the 2.01 mm data. The Kandlikar correlation underestimated the data for both the tubes by about 30 to 50%. The disagreement was similar for both diameters.

The last row in Table 1 summarizes a three-zone model developed by Thome et al. [15]. The model differs from the above, which are based on nucleate and convective heat transfer mechanism in the tubes. It evaluates the local time-averaged heat transfer coefficient at fixed locations along the channel based on the evaporation of elongated bubbles without any contribution from nucleate boiling. The three zones refer to a dry zone (caused by local dry out), liquid slug zone and elongated bubble surrounded by a thin liquid film zone. The model was evaluated by Dupont [28] using 1591 data points covering seven refrigerants, tube diameters 0.77 mm -3.1 mm, mass flux G = 50-564 kg/m^{3}s, heat flux q = 5-178 kW/m^2, x = 0.01-0.99 and operating pressure of 124-5766 kPa. They found that 67% of the data bank was predicted within ±30%. Very recently, Shefiraw et al. [29] compared the experimental data of Huo et al. [11] with this model. The model consistently overpredicted the data for the 4.26 mm tube by 20% to 40%, which is not surprising as this tube is outside the range of the expected applicability of the model. For the 2.01 mm tube the data are predicted within 20% at 8 bar and underpredicted by up to 30% at 12 bar. This recent work demonstrated that a model without a nucleate boiling contribution may provide a reasonably successful approximate prediction of “apparently nucleate boiling” heat transfer regime. Some suggestions were made by Shiferaw et al. [29] for improving the model which is indicative of the fact that further research - both experimental and theoretical - is necessary in order to develop a model capable of predicting heat transfer rates in micro passages.

3. Enhanced In-Tube Condensation

Condensation heat transfer experiments are notoriously difficult. They often involve determination of small temperature differences and are susceptible to large errors due to the presence of non-condensing gases in the vapour and insufficient accuracy in measurement of surface temperature. In many cases surface temperatures are not measured directly and data are analyzed using “Wilson Plot” techniques based on overall measurements, sometimes when the accuracy, number and range of the measured quantities may not justify this and when the vapour-side thermal resistance is small compared with the measured overall resistance (see Rose [30]). When correlations have no theoretical basis and are derived from data for fluids having similar properties and sometimes of questionable accuracy, it is not surprising that different "models" can give widely different predictions. Analytical approaches to enhanced condensation are difficult and require assumptions and approximations. However, some condensation problems lend themselves to analysis since in many instances condensate films are thin and laminar flow can be assumed.

Condensation in internally enhanced tubes

For relatively large (> about 5 mm) diameter tubes experimental data have become available in recent years and empirical correlations have been proposed. Recent reviews by Cavallini et al. [31, 32] cover measurements and calculation methods for both heat transfer and pressure drop. The experimental data exhibit relatively large scatter and differ significantly between different investigations. Vapour-side, heat transfer coefficients and pressure drops are found to be roughly 1.5 to 2 and 1.1 to 1.5 respectively, times those for smooth tubes under similar conditions. As well as wide uncertainty limits in the measurements in many cases, a major problem in correlating data for enhanced tubes is the large number of geometric variables involved (tube diameter, fin height, pitch, thickness, fin flank slope angle, helix angle). This, together with the fact that there are several relevant driving mechanisms for condensate motion (gravity, vapour shear stress, surface tension), indicates that a comprehensive heat transfer correlation would have upwards of 10 independent dimensionless parameters! Additionally, much of the available data is for fluids with similar properties (refrigerants) so that the general validity of correlations cannot be fully tested and their application to new fluids cannot be relied upon.

Correlations of heat-transfer and pressure drop data are generally compared with data on an experimental value *versus* calculated value basis. Predicted dependence on individual geometric variables (while others are held constant) is seldom explored. The performance of different correlations/models depends on the data chosen for comparison. Correlations generally agree with the data sets used in their development to within around 20%, but agree less well with data from other sources.

It seems clear that a successful general correlation for either heat transfer and pressure drop should have a theoretical basis. The mechanism of surface tension enhancement for condensation in a microfin tube is the same as that for external condensation on finned tubes. In the latter case theoretical results (Honda and Nozu [33], Rose [34]) are in good agreement with data for wide ranges of fluid properties (steam and ethylene glycol as well as refrigerants) and geometries and predict correct trends for dependence on fin geometry (pitch, thickness, height). For in-tube condensation the problem is complicated by the involvement of vapour shear stress, swirling flow near the tube wall due to the helical microgrooves and changes of flow regime along the tube. Nozu and Honda [35], Honda et al. [36] and Wang et al. [37] have attempted to develop a theoretical approach to the problem of condensation in spiral microfinned tubes. This, together with various correlations, is compared by Wang and Honda [38] with data from six investigations in which six microfin tubes and five refrigerants were used. In all cases heat-transfer coefficients were derived from directly-measured wall temperatures. The theoretical approach was found to be superior, with heat-transfer coefficients generally being predicted to within 15%.

Although various correlations for pressure drop are based on different data sets, the predications are somewhat less widely spread than for heat transfer. Wang et al. [39] have compared some of these against a refrigerant database. It is surprising to note that the best performing correlation in this comparison was that of Goto et al. [40] which is the simplest and does not include some of the variables used in other correlations. For herringbone internally microfinned tubes vapour shear stress causes the condensate film to be directed along the grooves leading to non-uniform distribution of condensate around the tube. Encouraging results have been obtained by Ebisu and Torikoshi [41] and Miyara et al. [42] showing improved heat transfer performance over spiral fins with moderate increase in pressure drop. Goto et al. [43] have recently reported heat-transfer coefficients for herringbone tubes twice those measured by the same authors for spiral microfin tubes but with rather large bounds of uncertainty.

Condensation in microchannels

There are as yet relatively few reliable experimental heat-transfer data for condensation in microchannels. In most experimental investigations vapour-side, heat-transfer coefficients are found from overall measurements using "Wilson plots" and consequently have large uncertainty. No generally accepted prediction methods are available. Quoting Garimella [44], "The art of the utilization of microchannels for achieving high heat transfer rates is in fact much further ahead than the science of obtaining a comprehensive understanding of phase change in these channels". Recent reviews have been given by Cavallini et al. [45] and Garimella [44]. Various models have been found to underestimate the heat-transfer coefficient at higher mass fluxes (mass velocities).

In a recent and detailed investigation into condensation of R134a, Koyama et al. [1, 2] used several types of tube (see Fig. 2) and a test section divided into four separately-cooled sections, each of length 150 mm and fitted with heat flux meters and embedded wall thermocouples. Specimen results are shown in Fig. 6 for tube B (Fig. 2) with channels of hydraulic diameter 0.81 mm. In this study the tube wall temperature was measured directly. It is clear that calculation of the small vapour-side temperature difference from the much larger overall temperature difference would be prone to very large uncertainty in this case.

The problem has been approached from a purely theoretical point of view by Wang et al. [46, 47], and Wang and Rose [48,49,50,51]. The treatment uses the Nusselt [52] approximations for the condensate film and includes the surface tension generated pressure gradient at right angles to the flow direction as well as the effects of gravity and vapour shear stress on the surface of the film. Fig. 7 shows specimen calculated condensate film profiles at different distances along a horizontal square channel of side 1 mm for condensation of R134a. Thinning of the film near the corners is evident

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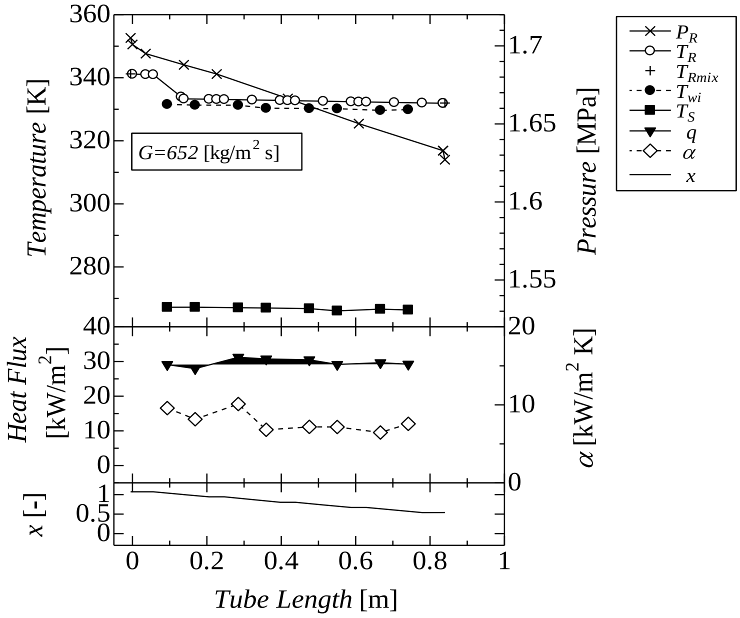

Fig. 6. Condensation of R134a in multi minichannel tube (type B in Fig. 2). Koyama et al [2].

towards the channel inlet and the effect of gravity (thicker condensate film at the bottom) is seen at greater distance along the channel. Fig. 8 shows the corresponding variation of the average (around the channel perimeter) heat-transfer coefficient in the streamwise direction for uniform wall temperature and saturation conditions at the inlet. Results have also been obtained for rectangular and triangular channels. More experimental data, for several fluids covering a wide range of properties are needed to evaluate theories and to develop reliable, theoretically-based correlations.

4. Conclusions

Flow boiling in minichannels and microchannels produces higher heat transfer coefficients than conventional channels. The channel diameter was found to affect the transition boundaries between flow regimes and confined flow was reported. The dependence of the heat transfer coefficient on diameter at different heat flux values is rather complex and further research is required to clarify and establish this relation. Correlations developed based on experimental results of boiling inside small/micro channels were found to be limited and can only predict results under similar operating conditions. Most of the models developed were based on the dominance of nucleate or convective heat transfer mechanisms. However, recent modelling work based on convective heat transfer in the confined bubble regime without a nucleate boiling contribution provided a reasonable approximate prediction of experimental data. Further work is underway to provide an improved correlation for this size of passages.

For condensation in microfinned and minichannels, the problem of data correlation is complicated by the large number of variables involved and also by uncertainty in much of the data. More data, using fluids covering a wider range

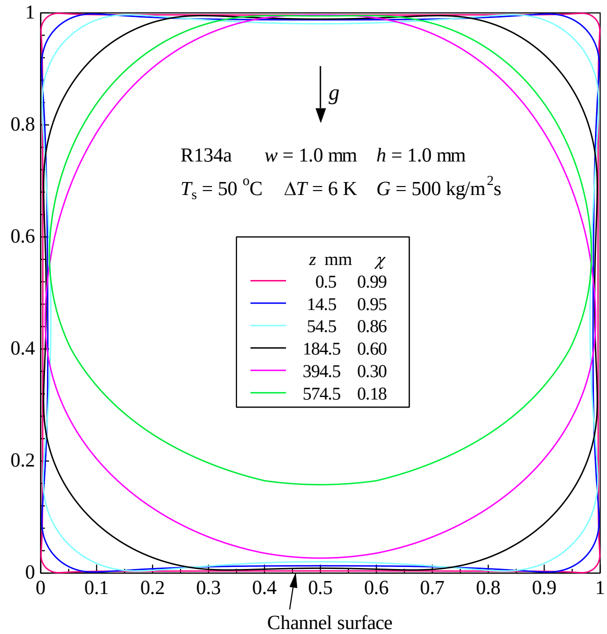

Fig. 7 Condensate film surface profiles at different distances along a square microchannel (Wang and Rose, [47, 48, 51]).

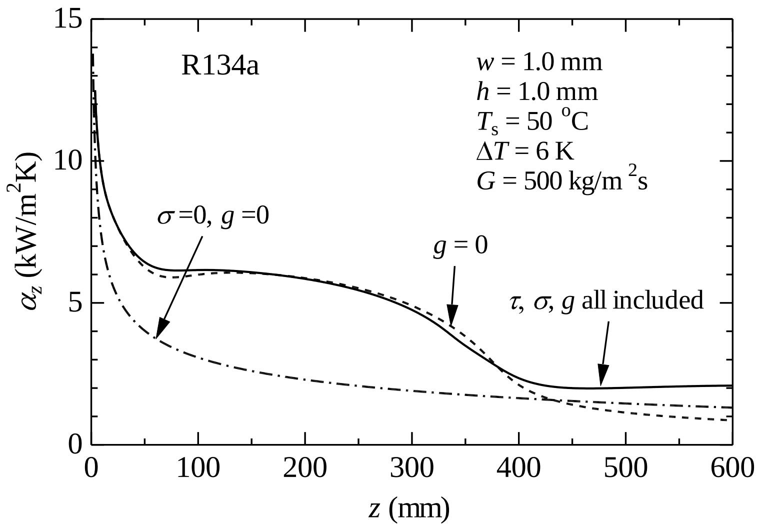

Fig. 8 Condensate film surface profiles at different distances along a square microchannel (Wang and Rose, [47, 48, 51]).

of properties and of sufficient accuracy, are needed. Careful assessment of accuracy of the data is important and results of different investigators with the same fluid and closely similar conditions should agree within the uncertainty estimates; such detailed comparisons are not often found in the literature. In view of the many variables involved, correlations should be theoretically based. It should also be demonstrated that correlations and theories show the correct trends with respect to each of the relevant geometrical parameters when the other variables are held constant. Systematic experimental measurements of this kind are, of course, time consuming and expensive. Even in the absence of such detailed data it is useful to verify that correlations give plausible results and to examine sensitivity of the heat-transfer coefficient to the separate variables.

5. Acknowledgements

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6. Nomenclature

b	side dimension of square channel
Bo	Boiling number, q/Gh_{lg}
Co	Confinement number
Co	Convection number, $[(1-\chi)/\chi]^{0.8}[\rho_g/\rho_l]^{0.5}$
d	Diameter
Fr_l	liquid Froude number, $G^2/\rho_l^2 gd$
G	mass flux
g	gravitational acceleration
H	height of rectangular channel
H	Enthalpy
k	thermal conductivity
L	Length
M	molecular weight
M	mass flow rate
Nu	Nusselt number, $Nu = \alpha d/k$
P	Pressure
P_R	pressure of (Fig. 6)
Pr	Prandtl number, $Pr = C_p \mu / k$
Q	Heat
q	heat flux
Re	Reynolds number, $Re = G \cdot d / \mu$
t	time
T	Temperature
T_s	coolant temperature (Fig. 6) or vapour temperature (Figs. 6, 7)
T_R	temperature of vapour (Fig. 6)
T_{wi}	inside wall temperature (Fig. 6)
w	width of rectangular channel
We_l	Weber number, $G^2 d/\rho_l \sigma$
x	quality
X	Lockhart Martinelli Parameter
z	distance along surface

GREEK SYMBOLS

α	heat transfer coefficient
α_z	local heat-transfer coefficient at distance z along channel
δ	liquid film thickness
Δ	finite increment
ρ	density
σ	Surface tension
ϕ^2	two-phase friction multiplier
ΔT	vapor-to-surface temperature difference
τ	shear stress on surface of condensate
τ	pair period
χ	quality

SUBSCRIPTS

f	fluid
g	gas
h	hydraulic
i	index
in	inside
l	liquid
lo	liquid only
pool	Pool boiling
r	reduced
tp	two phase
w	wall

7. References

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