COMPACT EXCHANGERS FOR PHASE CHANGE

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ABSTRACT

Compact heat exchangers are being increasingly considered for duties involving phase change processes of boiling and condensation. In many cases such applications are completely new and no previous experience or information on exchanger performance is available. Understanding the thermal-hydraulic characteristics of flow passages of compact heat exchangers for phase change duties is therefore vitally important. Even for those compact heat exchangers, which have long been used for boiling and condensation duties, design procedures based on local two-phase characteristics are clearly preferred. This paper reviews the currently available information about phase change in small diameter channels, which are typical of the compact heat exchanger passages. These channels often exhibit different two-phase heat transfer characteristics when compared to 'normal', large diameter tubes. These differences caused by narrowness, and in some cases, non-circular nature of the flow channels are discussed. Some general considerations regarding the effectiveness of single phase heat transfer enhancement features for two-phase applications, the use of fin efficiency term for two-phase duties and the introduction of two phase flow in compact heat exchanger passages are discussed. Finally, some areas of further work are delineated.

INTRODUCTION

Compact heat exchangers are gaining increasing attention in the industrial practice, and they are being considered as cost-effective alternatives in applications where shell and tube heat exchangers would normally be used. In the heat exchanger context, the term compact really implies compactness in exchanger size arising from two factors, namely, higher heat transfer area density (heat transfer area per unit exchanger volume) and better heat transfer characteristics. These two factors could, thus, be used in deciding whether an exchanger is compact or not. Although no definition is available of what better heat transfer characteristics mean, an area density figure of 200 m^2/m^3 is often used [1] as a rather arbitrary cut-off point to classify an exchanger as a compact. In the present paper this cut-off value is not rigorously followed but it is used as a general guideline only. Increased area density is related to smallness of the hydraulic diameters of flow passages. In the current paper the discussion is limited to passages of hydraulic diameters ranging from about 1 mm to 7mm.

A large proportion of the current compact heat exchanger applications is for single phase heat duties, but the proportion for phase change duties of boiling and condensation is increasing. In terms of the compact heat exchanger usage for two-phase applications, the following three categories can be noted.

1. Compact heat exchangers which have been used for phase change application for long time. Plate-fin heat exchangers used in cryogenic industry belong to this category, where the exchangers have long been used for boiling and condensing duties, along with the single phase vapour and liquid applications. Compact finned-tube exchangers of various designs have been used in air to two-phase service for many years in refrigeration and air-conditioning applications [2].
2. Some compact heat exchangers could previously handle only liquids and not gases or vapours but now they are also available in alternative forms capable of handling gases or vapours. The conventional gasketted plate frame heat exchanger belongs to this category. Their traditional application is limited to liquid-liquid duties. This is mainly due to the limitation of gaskets used for providing the seals for the flow passages, and to some extent, the limited mechanical integrity arising from the frame type construction. With the introduction of the brazed and welded variants, these heat exchangers are now capable of handing gaseous phases, and hence are being increasingly considered for phase change duties too. The plate heat exchanger manufacturers realise that the future growth in their market share is going to come from two-phase applications.
3. Entirely new types of compact heat exchanger, which are capable of handling single phase as well as phase change duties. These new exchangers are being increasingly used in single-phase applications, and at the same time, they are also being considered for two-phase duties. An example pertaining to this category is the printed circuit heat exchanger.

Except for the first two categories, the compact heat exchangers face a number of barriers to a more widespread and general acceptance. Two of the often stated barriers are the availability of reliable and independent methods and correlations for thermal-hydraulic characteristics of the compact heat exchanger passages, and the availability of independent software tools for the design of compact heat exchangers. The latter barrier concerning independent software is adequately addressed by commercial organisations such as HTFS, and therefore need not be discussed here. The former barrier related to thermal-hydraulic characteristics is important for single phase duties but even more so for two-phase applications. The reasons for this are as follows.

For single phase applications, the heat transfer coefficient changes along the flow length of the exchanger only as a result of changes in the physical properties of the fluid. For phase change duties, however, the coefficient can also change very significantly because of the changes in the local vapour phase mass fraction i.e. the vapour quality. Furthermore the local vapour quality changes can also result in different local two-phase flow patterns within the exchanger flow passages causing further impact on the local thermal-hydraulics. In case of boiling, the relative contributions of two-phase convective heat transfer and nucleate boiling heat transfer can also change as a result of changes in local quality and flow patterns.

In spite of the importance of thermal-hydraulic characteristics for two-phase processes, the amount of published data is very limited. The current paper reviews available data, methods and observations published by various researchers for phase change in compact heat exchanger passages; this is done in the context of heat transfer characteristics but some passing references are also made for pressure drop data. Single channels of small diameter tubes and non-circular flow passages are examined first, followed by more complex passages. An important issue of effectiveness of single-phase enhancement devices or geometries for two-phase duties is then examined. The issue of flow distribution in introducing a two-phase vapour-liquid mixture in compact exchanger flow passages is also reviewed. Finally, some examples of new applications of compact heat exchangers are covered and areas of further work are suggested.

There are other review papers related to the present subject area. Haseler and Butterworth [3] reviewed boiling in compact heat exchangers, focussing on industrial practice and problems. Huo et al [4] and Kandlikar [5] reviewed flow boiling heat transfer and two-phase flow patterns in channels with small hydraulic diameters. Srinivasan and Shah [6] reported a review on condensation heat transfer in compact heat exchangers.

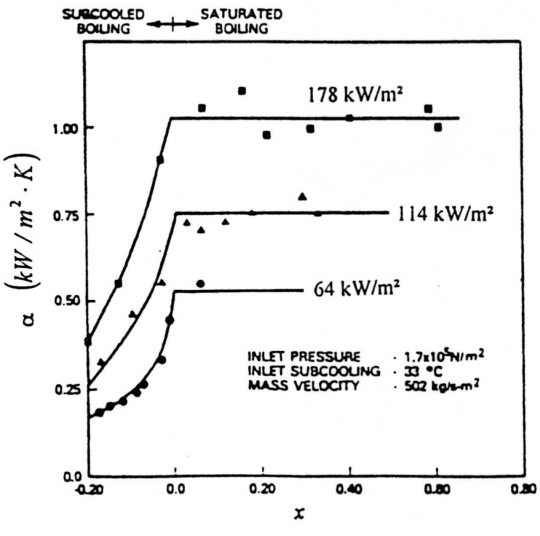

Figure 1 Local heat transfer coefficient as a function of local vapour quality for 3.1 mm diameter tube at different heat flux values, Lazarek and Black [7].

SINGLE CHANNELS WITH SMALL HYDRAULIC DIAMETER

Single channels with small hydraulic diameter, with either circular or rectangular cross-sections, are important in developing our understanding about the effect of narrowness on phase change in compact heat exchanger flow passages. In this section both vertical and horizontal flow passages are covered, although vertical orientation is more often used for phase change duties.

The local heat transfer coefficient, pressure drop, and critical heat flux were measured by Lazarek and Black [7] for flow boiling of R113 in a round vertical tube with an internal diameter of 3.1 mm, and heated lengths of 12.3 and 24.6 cm. The heat flux was varied from 1.4×10^4 to 3.8×10^5 W/m^2 ; the mass flux from 125 to 725 kg/m^{2}s and the pressure from 130 to 410 kPa. It can be seen in Figure 1, obtained from their experiments, that in the saturated boiling region the boiling heat transfer coefficient is independent of quality. From the strong dependence of the saturated boiling heat transfer coefficient upon heat flux and the negligible influence of quality, Lazarek and Black concluded that the mechanism of nucleate boiling controlled the wall heat transfer process. They correlated the saturated local boiling heat transfer coefficient data with the following simple correlation.

$$Nu = 30\,\mathrm{Re}^{0.875}\,Bo^{0.714} \qquad (1)$$

The results of a study on boiling heat transfer of refrigerant R113 in a 2.92 mm diameter horizontal tube were reported by Wambsganss et al. [8]. They observed that the heat transfer coefficient was a function of heat flux and was independent of quality over most of the quality range. Therefore, they suggested that nucleate boiling is the dominant heat transfer mechanism. They estimated that the predominant flow pattern regime was slug flow up to qualities of 0.6 to 0.8. Therefore, they concluded that the thick-liquid regions of slug flow are more likely to support nucleation than the thin liquid films of annular flow.

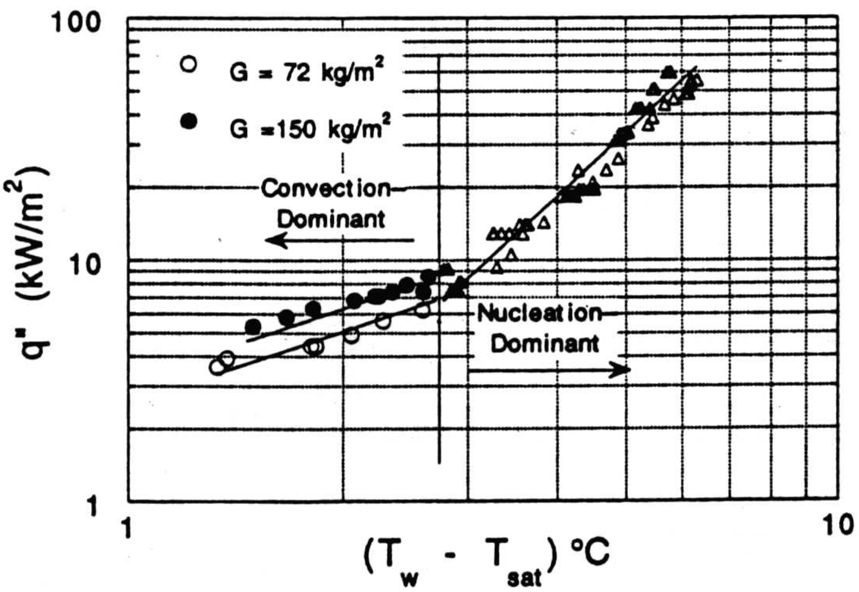

Figure 2 A graph of heat flux against wall superheat showing nucleate boiling and convective heat transfer dominated regions, Tran et al. [9].

Boiling heat transfer experiments were performed by Tran et al. [9] in a small circular tube with 2.46 mm diameter and a small rectangular channel with hydraulic diameter of 2.4 mm using R12 as a test-fluid. They observed that the local heat transfer coefficient was a function of heat flux and effectively independent of quality and mass flux for quality greater than 0.2 and wall superheats above 2.75 K. At lower wall superheats, the mass flux effect became important. Therefore, they concluded that over a broad range of heat flux, nucleation was the dominant heat transfer mechanism for flow boiling in the small passages of their study, and at sufficiently low values of heat flux (very low wall superheat), forced convection dominated. These two regions of heat transfer are shown in Figure 2. Tran et al also noted that there was very little difference between the heat transfer coefficients in rectangular and circular channels. Their mean heat transfer data for R12 and R113 (studied previously, Wambsganss et al. [8]) were correlated in the nucleation dominant region as:

$$\alpha = 8.4 \times 10^{-5} Bo^2 We_l^{0.3} \left(\frac{\rho_l}{\rho_v} \right)^{0.4} \qquad (2)$$

Recently, Bao et al. [10] studied flow boiling heat transfer for R11 and R123 in a copper tube with an inner diameter of 1.95 mm. The range of parameters examined was: heat flux from 5 to 200 kW/m^2; mass flux from 50 to 1800 kg/m^{2}s; vapour quality from 0 to 0.9; system pressure from 200 to 500 kPa; and experimental heat transfer coefficients from 1 to 18 kW/m^{2}K. They too reported that the heat transfer coefficient was a strong function of the heat flux and the system pressure, while the effects of mass flux and vapour quality were negligible in the range examined.

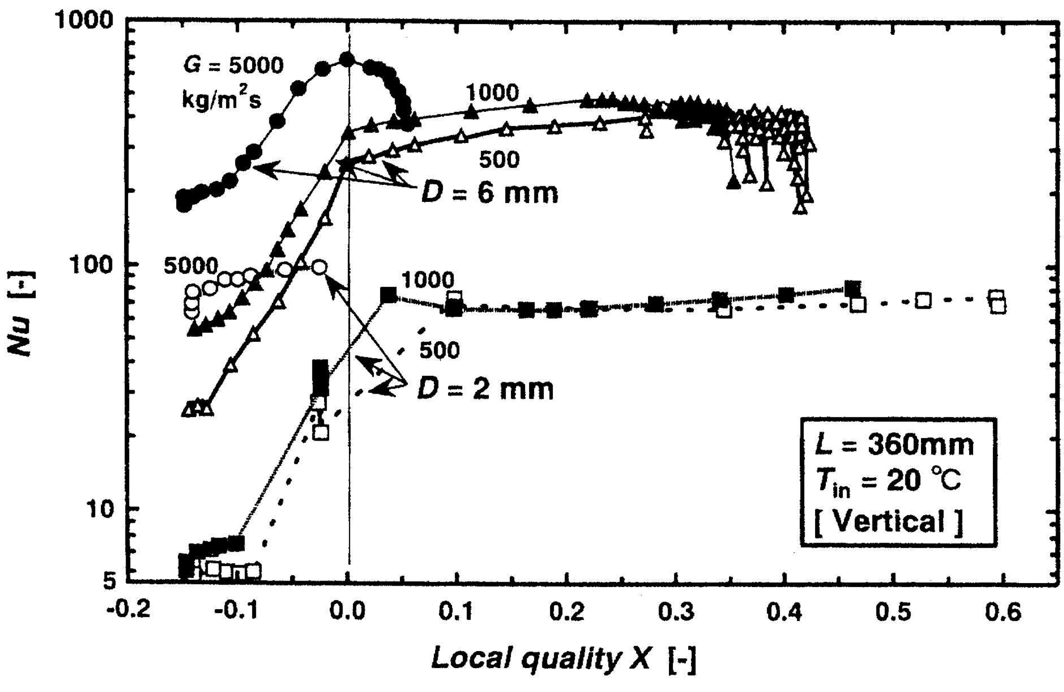

Figure 3 Local Nusselt number against local vapour quality, Kureta et al [11]

The general conclusion that can be drawn from the studies mentioned above is that the flow boiling heat transfer in small diameter tubes is dominated by nucleate boiling. Kureta et al [11] reported data for flow boiling of water in 2 and 6 mm diameter tubes at relatively high heat fluxes. They correlated their data in the saturated boiling region using the following equation.

$$Nu = \left(9.7 \times 10^6 D^{1.93}\right) x + 1342 D \,\mathrm{Re}_{in}^{0.44} \qquad (3)$$

Note that Re_{in} is liquid Reynolds number at the inlet conditions. This correlation suggests that the flow boiling heat transfer coefficient was only a function of vapour quality and it was independent of heat flux, indicating complete absence of nucleate boiling.

Figure 3 shows a graph of the Nusselt number against the local vapour quality obtained from their work. The graph contains the data at three mass fluxes, namely, 500, 1000 and 5000 kg/m^{2}s under subcooled and saturated boiling conditions. Most of the data for the highest mass flux of 5000 kg/m^{2}s are in subcooled boiling; the remaining data, which are in saturated boiling show a decreasing Nusselt number with increasing vapour quality. For a tube diameter of 6mm, the data at two other mass fluxes also show decreasing Nusselt number at higher vapour qualities. Kureta et al noted that this behaviour arises from the occurrence of DNB i.e. departure from nucleate boiling leading to local dryout. For a more detailed examination, their saturated boiling data are plotted as heat transfer coefficient against vapour quality in Figure 4, leaving out the data in both the subcooled boiling and DNB region.

It can be seen from Figure 4 that for a tube diameter of 6mm the data appears to indicate the trends of two-phase convective heat transfer because the coefficient increases with mass flux and vapour quality, and is independent of heat flux. For a tube diameter of 2 mm the heat transfer coefficient was only weakly dependent on

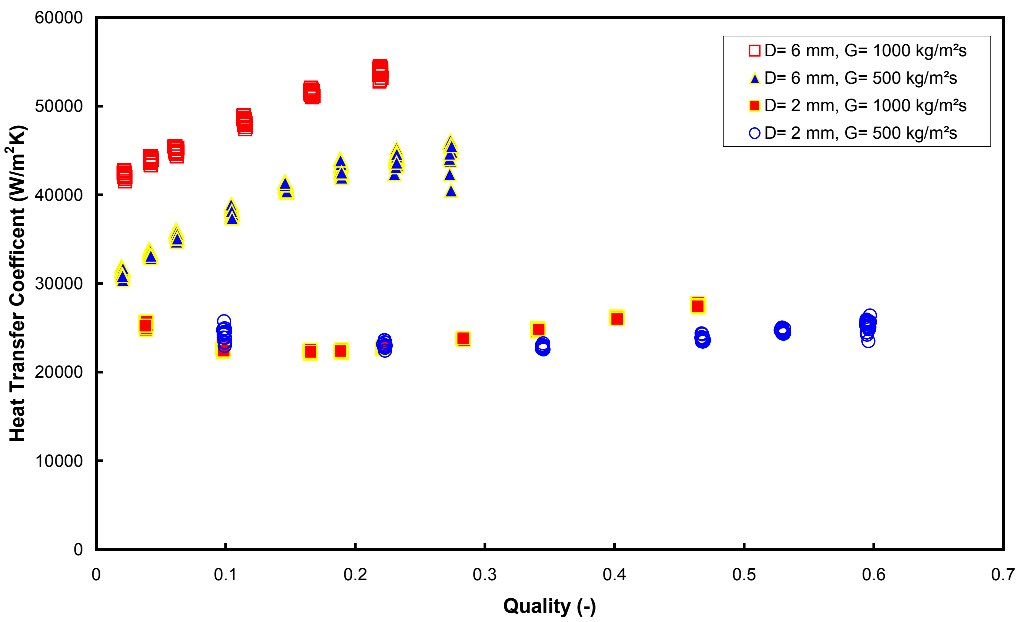

Figure 4 Local heat transfer coefficient as a function of vapour quality using the data in saturated flow boiling, adapted from Kureta et al [11]

vapour quality. This could be taken as an indication that these data are dominated by nucleate boiling heat transfer. However, with nearly a four-fold variation in heat flux, there is very little or no effect on the measured coefficient.

Oh et al [12] reported the results of flow boiling of R134a in horizontal tubes of 0.75, 1 and 2 mm diameter. It is interesting to note that, under the experimental conditions investigated, they did not observe nucleate boiling region. Their typical results are shown in Figure 5. It can be seen that their data show two distinct regions of convective heat transfer depending on the vapour quality. In the low quality region the measured coefficient increases rather slowly with quality whereas in the higher quality region there is relatively rapid increase in the coefficient with quality. The decreasing heat transfer coefficients at very high quality are associated with local dryout. They correlated the data using a single convective heat transfer correlation, ignoring the difference between the two regions mentioned above.

Experiments were carried out by Yan and Lin [13] to investigate the characteristics of boiling heat transfer and pressure drop for refrigerant R134a flowing in a horizontal small circular pipe with 2 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. However, from their results, it was very difficult to conclude which regime was dominant, nucleation or forced convection.

Cornwell, Kew and co-workers [14,15] suggested a simple criterion for the onset of narrow channel effect based on a Confinement number, Co defined as:

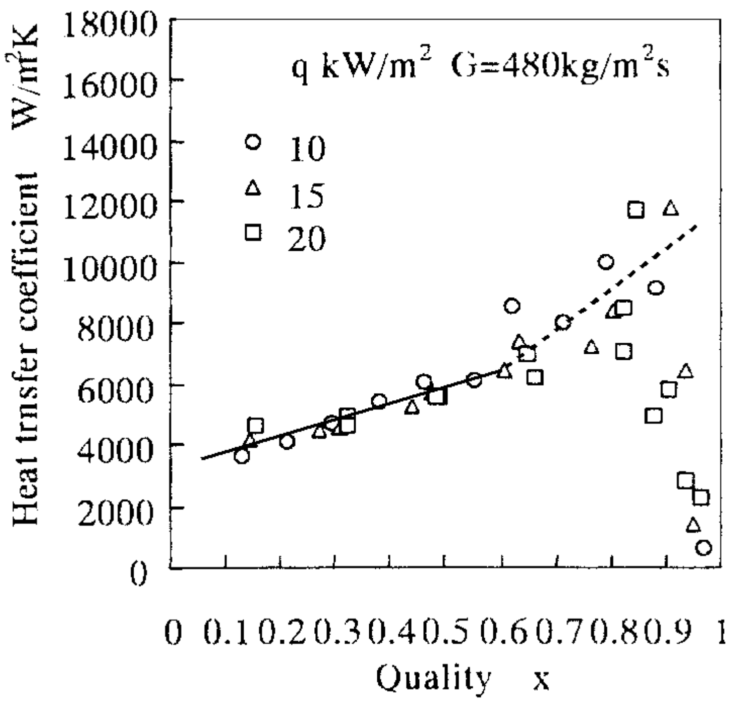

Figure 5 Measured data of Oh et al [12] for flow boiling of R134a in a 2 mm diameter tube. Best fit lines show two regions of convective heat transfer.

$$Co = \frac{\left[\sigma/(g(\rho_l - \rho_g))\right]^{1/2}}{d_h} \tag{4}$$

They divided flow boiling in narrow passages into three regions, namely, isolated bubble, confined bubble and annular-slug flow regions, see Figure 6. They proposed that, when the channel diameter is small enough to give rise to Confinement number of greater than 0.5, the confined bubble flow occurs. Further discussion on this approach is presented later.

Kenning and co-workers reported a number of studies (for example, Kenning and Yan [16]) on flow boiling of water in a single channel with a rectangular cross-section 2mm x 1 mm and heated length of 260mm. The channel was heated directly by passing electric current through the thin stainless steel sheet that formed three sides of the channel, bonded to a glass window by silicone rubber. Local measurements of the fluctuating pressure and wall temperature were made at five positions along the channel and synchronised (to within one frame) with video recordings at 500 Hz of the local flow. In all cases, the flow was unsteady and was dominated by the growth of single confined bubbles near the inlet to the channel, which produced a disturbance in pressure and wall temperature that propagated over the full heated length of the channel. At a given location, heat transfer occurred at different times by convection to the liquid slugs between confined bubbles, convective evaporation of the thin liquid film around the bubbles or nucleate boiling in this liquid film. The "large channel" concept of different flow and heat transfer regimes developing in the axial direction did not apply. The onset of boiling also differed from "large-channel" behaviour. There were only a few active sites and, instead of developing unstable rapid growth at sub-visible sizes, bubble nuclei could be seen growing slowly or not at all and sometimes moving with the flow until growth or coalescence caused them to be squashed against the heated wall. This triggered the rapid growth of a confined bubble. The head of this bubble accelerated rapidly, reaching velocities of 5 – 8 m/s towards the downstream end of the channel, where individual bubbles appeared to lose their identity in an unstable annular flow.

The above observations made by Kenning and co-workers are useful in

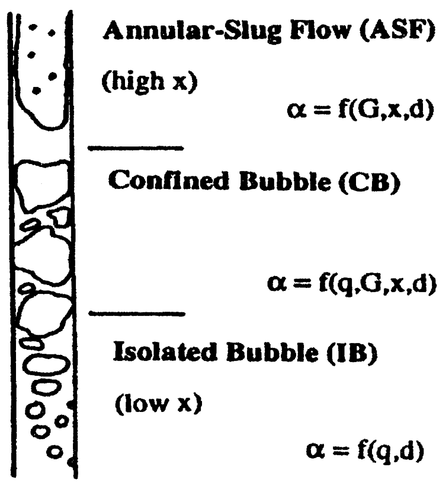

Figure 6 Flow regimes in a narrow diameter tube, Cornwell, Kew and co-workers [14,15]

obtaining a general understanding of two-phase flow behaviour in small diameter channels.

In contrast to flow boiling, there are only limited studies of condensation in small diameter channels. One example of recent work is that of Fiedler and Auracher [17] where they reported reflux condensation of R134a in a tube of 7mm diameter. They found that Wang and Ma's semi-empirical correlation for condensation in large diameter tubes showed good agreement with the data. However, the prediction of optimum angle of inclination was different to that predicted by Wang and Ma correlation. Therefore they proposed the following modified correlation.

$$\frac{\overline{Nu}}{\overline{Nu_K}} = \left(\frac{L}{R}\right)^{\cos(\beta/4)} \left(0.125 + 1.46\times10^{-2}\beta - 7.27\times10^{-5}\beta^2\right) \qquad (5)$$

$$\text{where, } \overline{Nu_K} = \mathrm{Re}/(1.47\,\mathrm{Re}^{1.22} - 1.3)$$

Baudoin and co-workers (Russeil et al, [18]) reported condensation from humid air between flat vertical plates of 1.5, 3.2 and 8 mm spacing. Their results are relevant in providing a base case for condensation in finned tube exchangers as well as plate-fin and plate frame exchangers. In a separate paper (Tribes et al, [19]), the group reported flow visualisation results concerning condensate retention and draining using a similar experimental set up.

MORE COMPLEX PASSAGES

In this section investigations on more complex channels of compact heat exchangers are reviewed. Such channels range from multiple parallel subchannels of plate-fin exchanger passages to cross-corrugated channels of plate heat exchangers to full heat exchanger assemblies. Here we face the complete complexity of the compact heat exchanger passages in terms of smallness of cross-sectional area, non-circular nature, and in some cases, tortuousness of flow paths. These investigations can be divided into two categories, one relating to local measurements and other relating to overall measurements.

Local Measurements

Table 1 shows various boiling and condensation studies that were undertaken with passages of plate-fin exchanger. As shown in the table only single passages were investigated in these studies, but the passages themselves contained parallel flow subchannels arising from fins sandwiched between the primary surfaces.

In the studies conducted at HTFS by Robertson, Wadekar and co-workers, the test-sections were representative of single channel used in aluminium brazed plate heat exchangers. The mass fluxes and heat fluxes used were typical of those used in industrial practice. Robertson [20,21] and Robertson and Clarke [22] performed tests with perforated and serrated fins using boiling liquid nitrogen. Robertson and Wadekar [23] and Wadekar [24] reported their tests with a perforated fin test-section using cyclohexane and heptane as test-fluids. Figure 7 shows typical results obtained by Robertson and Wadekar at two different mass fluxes. All these test performed at HTFS indicated that the dominant mechanism of heat transfer was convective heat transfer.

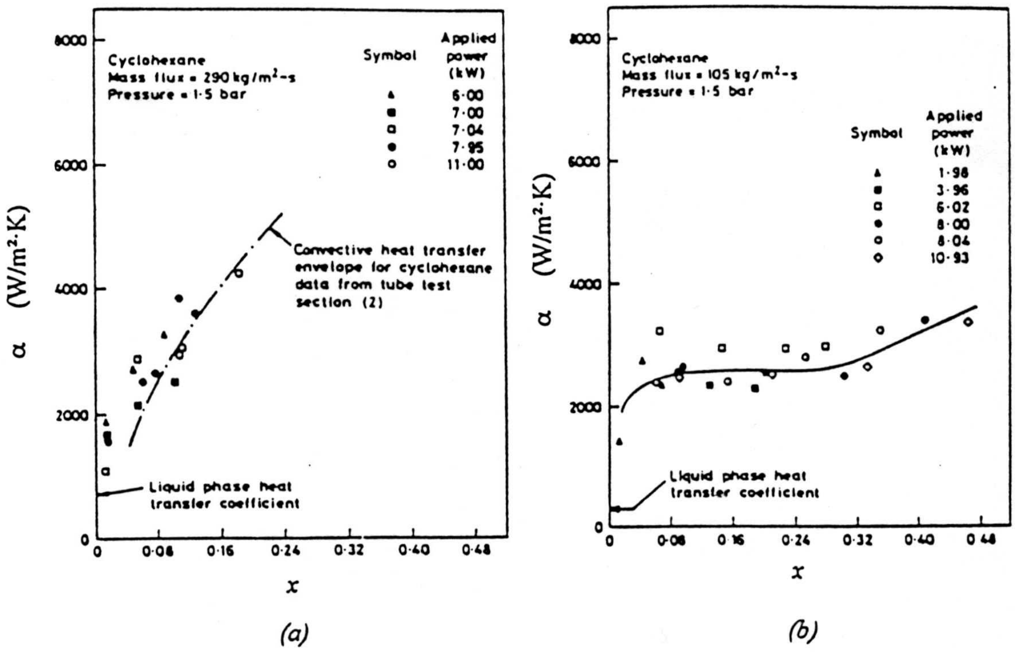

Figure 7 Plot of boiling heat transfer coefficient against quality for two different mass flux values, Robertson and Wadekar [23].

Carey and co-workers [25,26] investigated flow boiling heat transfer with methanol, butanol and water in serrated fin test-sections made up from copper block. The test-section was electrically heated on one side with glass cover plate on the opposite side to allow for flow visualisation. The test-section geometry and heating arrangement was not typical of that employed in practice for plate-fin heat exchangers.

Feldman et al [27] reported data for flow boiling of R114 in perforated and serrated fin test-sections of different fin frequencies and fin heights. They observed

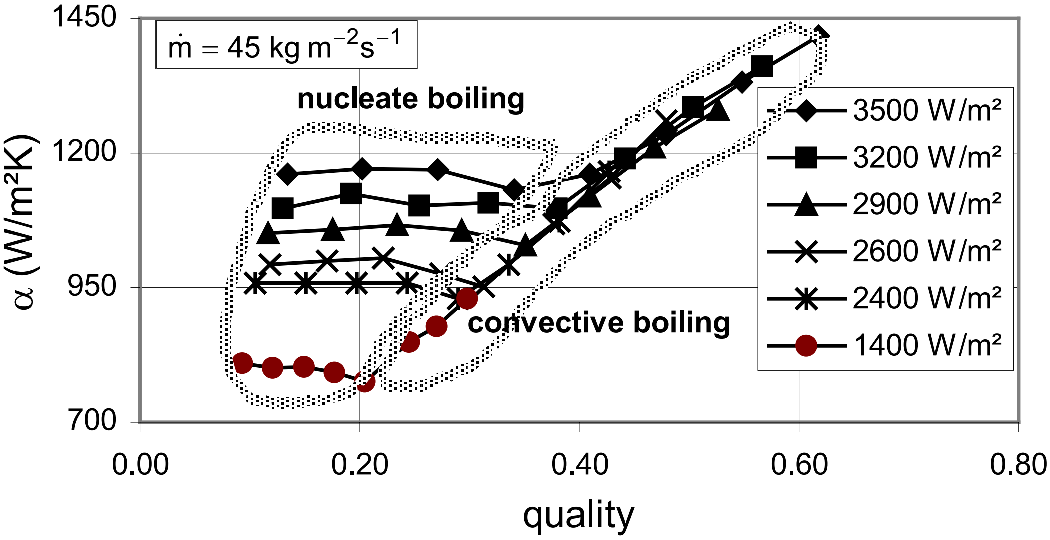

Figure 8 A graph of heat transfer coefficient against heat flux showing nucleate boiling and convective heat transfer regions, Feldman et al [27]

only convective heat transfer region for serrated fins whereas for perforated fins both nucleate boiling and convective heat transfer regions were reported, see Figure 8. Watel and Thonon [28] reported experiments with boiling propane in a fluid heated test-section. Details of the test-section and test conditions are given in Table 1.

Geometry effects on flow boiling heat transfer in narrow channels were reported by Mertz and Groll [29], e.g. their multi-channel arrangement of aspect ratio 2 gave the best results. They also observed a different behaviour between single and multi-channel arrangements, i.e. the heat transfer coefficient increased with increasing heat flux in the single channel while it decreased or remained about constant with increasing heat flux in the multi-channel heat exchanger.

Roberston et al [30,31] reported heat transfer and pressure gradient data for condensation on plain fins. Clarke [32] studied downflow condensation of nitrogen in a serrated plate-fin test-section under conditions similar to those used in industrial practice. He found that the condensation correlations developed for large diameter tubes under high vapour shear tended to overpredict the condensation coefficient. A proprietary correlation developed by HTFS for condensation in plate-fin passages was found to predict the data well.

Overall Performance Measurements

This category covers studies performed with a full heat exchanger rather than a test-section containing a single or multiple flow passages. Because of difficulties involved in making local measurements within full heat exchangers, researchers would generally resort to overall performance measurements. This is also standard practice adopted in validating commercial codes for design and simulation of heat exchangers. Two recent examples of research pertaining to this category are discussed here; both of them deal with condensation in plate heat exchangers.

Wang et al [33] carried out an experimental investigation of condensation heat transfer in plate heat exchangers. The details of the plate heat exchangers and the plate geometries covered in their experimental programme are given in Table 2. Wang et al conducted experiments in the seven different exchanger configurations listed in Table 2 as PHE1, PHE2 etc. They measured the overall performance of these exchangers for steam condensation using water as a cooling medium.

Table 2: Geometric details of test sections used by Wang et al. [33]

	Units	PHE1	PHE2	PHE3	PHE4	PHE5	PHE6	PHE7
Plate Width	m	0.21	0.178	0.178	0.615	0.615	0.615	0.1
Plate Height	m	0.484	0.448	0.448	0.678	0.678	0.678	0.267
Port Diameter	m	0.06	0.054	0.054	0.21	0.21	0.21	0.032
Hydraulic Diameter	m	0.006	0.005	0.005	0.0074	0.0074	0.0074	0.0048
Chevron Angle	°	30	30	60	30	45	60	30
Surface Area	m^2	0.122	0.095	0.095	0.5	0.5	0.5	0.032
Number of Channels	-	21 or 7	19	19	9	9	9	79

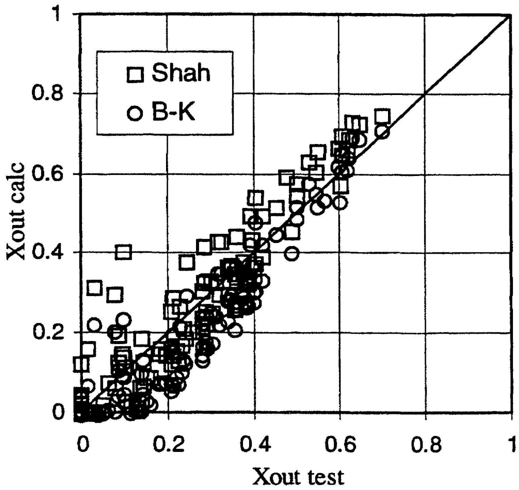

Figure 9: Comparison of measured and calculated exit qualities using Shah correlation and Boyko-Kruzhlin correlation by Wang et al. [33]

Figure 9 shows typical results reported by Wang et al as a comparison of measured and calculated exit qualities using two correlations for condensation heat transfer, one of which was the Boyko-Kruzhilin correlation [34]. Wang et al obtained the calculated exit qualities by performing stepwise heat transfer and pressure drop calculations along the exchanger. They concluded, from the comparison of measured and calculated exit qualities, that the overprediction of the exchanger performance was the result of the overprediction of the condensation coefficient by the Boyko-Kruzhilin correlation.

Gitteau et al [35] studied condensation of single component and binary mixtures of hydrocarbons in welded cross-corrugated heat exchangers. The corrugation angle was 45^0 and the hydraulic diameter was 10 mm. From their

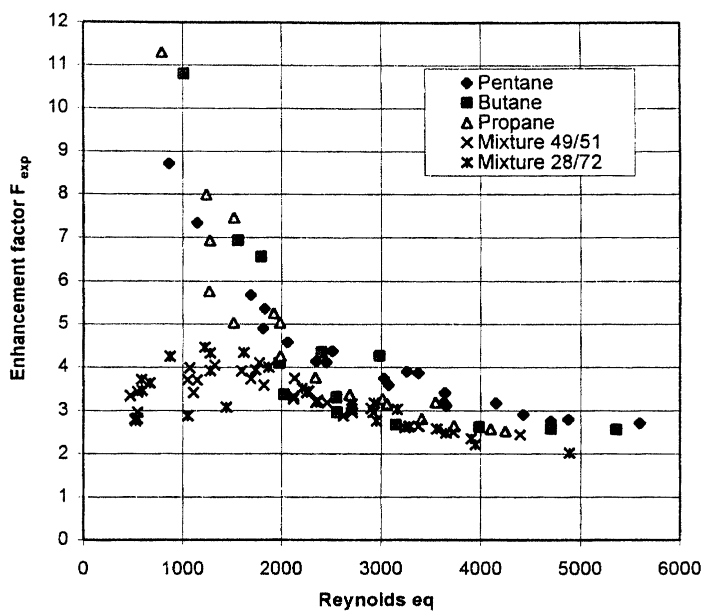

Figure 10: Condensation in a cross-corrugated compact heat exchanger by Gitteau et al [35]

experimental data, shown in Figure 10, they concluded that for single component fluids (propane, butane and pentane) two regimes could be identified. At low Reynolds numbers the heat transfer coefficient decreased with the Reynolds number, indicating a laminar flow regime. At higher Reynolds number the heat transfer coefficient decreased less rapidly, indicating a transition regime. For binary mixtures of propane and butane at low Reynolds number there was a sharp reduction in heat transfer coefficient when compared with single component fluids. At higher Reynolds number the heat transfer coefficients were comparable with single component fluids. From these results they concluded that at higher Reynolds numbers the mass transfer effect was negligible in cross-corrugated flow passages.

INTERPRETATION OF DATA BASED ON FLOW PATTERNS IN CHANNELS WITH SMALL HYDRAULIC DIAMETERS

In this section an attempt is made to explain the differing, and at times apparently contradictory observations made by some researchers regarding phase change heat transfer, especially boiling heat transfer, in single or multiple channels with small hydraulic diameter. Flow boiling studies, reviewed in the previous section, can be grouped into two categories.

(i) Studies showing conventional trends: In these studies classical trends of nucleate boiling or two-phase convective heat transfer or both were reported. Typical examples of this category are the work reported by Lazarek and Black [7], Bao et al [10] and Feldman et al [27].

(ii) Studies showing non-conventional trends: The apparently contradictory trends observed in these studies refer to the two phase convective heat transfer region. In this region the heat transfer coefficient is normally expected to be a strong function of the vapour quality and mass flux. These studies, however, report a relatively weak dependence on the vapour quality, and occasionally, also on the mass flux. Typical examples of this category are the data of Kureta et al [11] for 2 mm diameter tube, low

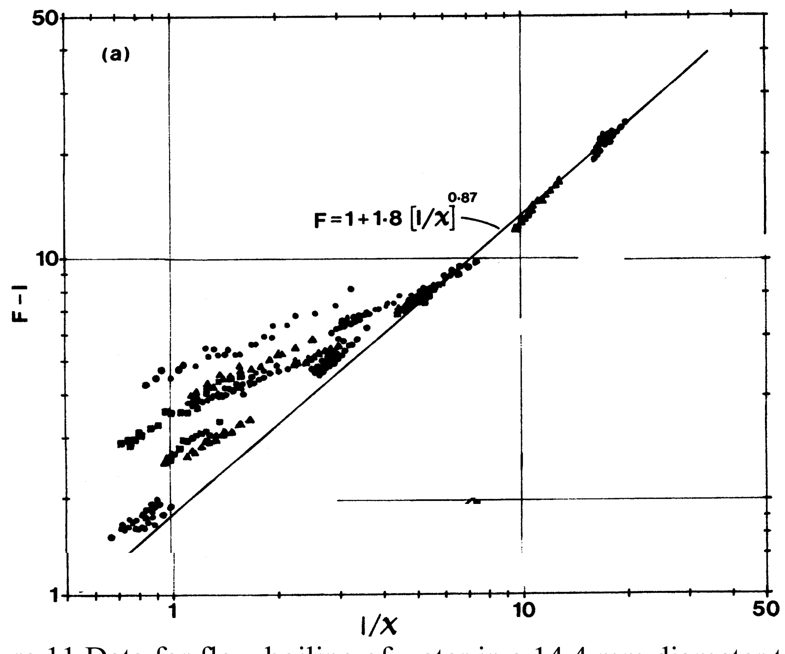

Figure 11 Data for flow boiling of water in a 14.4 mm diameter tube, Kenning and Cooper [36]

to medium vapour quality data of Oh et al [12] and that of Robertson and Wadekar [23] in Figure 7b.

It should be noted that, under certain conditions large diameter tubes (i.e. tubes with diameter >10 mm) also show similar non-conventional trends in boiling heat transfer data. Figure 11 shows an example of such data from the work of Kenning and Cooper [36]. These data are plotted as F-1, where F is the two-phase heat transfer multiplier, against the reciprocal of the Lockhart-Martinelli parameter. Kenning and Copper reported that the data at high values of the reciprocal of the Lockhart-Martinelli parameter and high mass flux are well represented by the following equation.

$$F = 1 + 1.8(1/\chi)^{0.87} \qquad (6)$$

when both phases are turbulent $\chi = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_v}{\rho_l}\right)^{0.5} \left(\frac{\eta_l}{\eta_v}\right)^{0.1}$

At low values of the reciprocal of the Lockhart-Martinelli parameter, however, the data showed increasing deviation from Equation (6) with decreasing mass flux. Selected data from Figure 11 are plotted in Figure 12 as heat transfer coefficient against vapour quality to highlight the two different trends exhibited by these data. Figure 12 is adapted from the work of Wadekar and Kenning [37]. The dashed line corresponds to Equation (6) and clearly shows that it represents a very different trend compared to the data at the low mass fluxes of 65 and 90 kg/m^{2}s. In comparison to the dashed line, the data at these two mass fluxes show relatively weak dependence on mass flux and quality. Wadekar and Kenning proposed that the differences between the two trends arise from the different thermal-hydraulic characteristics of intermittent and non-intermittent two-phase flow. They modelled the intermittent flow characteristics with a slug flow model. The solid lines shown in Figure 12 are predictions of the slug flow model. This model was later extended to cover intermittent two-phase flow with superimposed nucleate boiling [38]. In order to

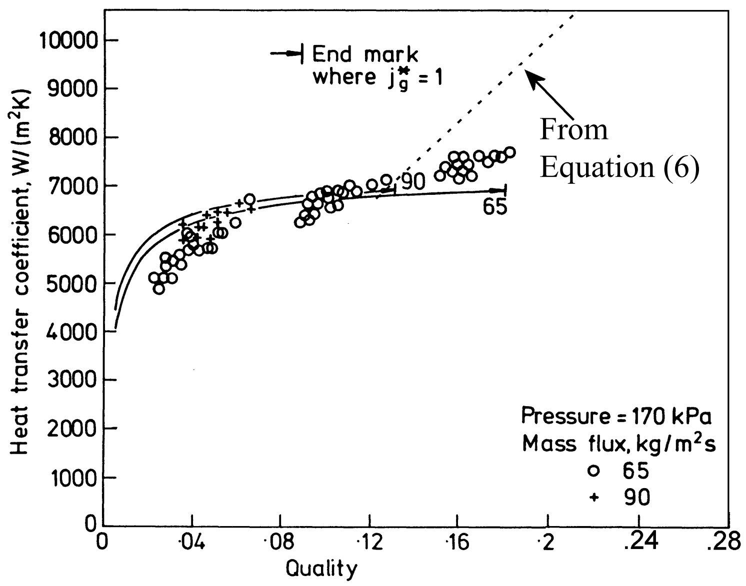

Figure 12 Two distinct regions of different trends of two-phase convective flow boiling heat transfer, adapted from Wadekar and Kenning [37]

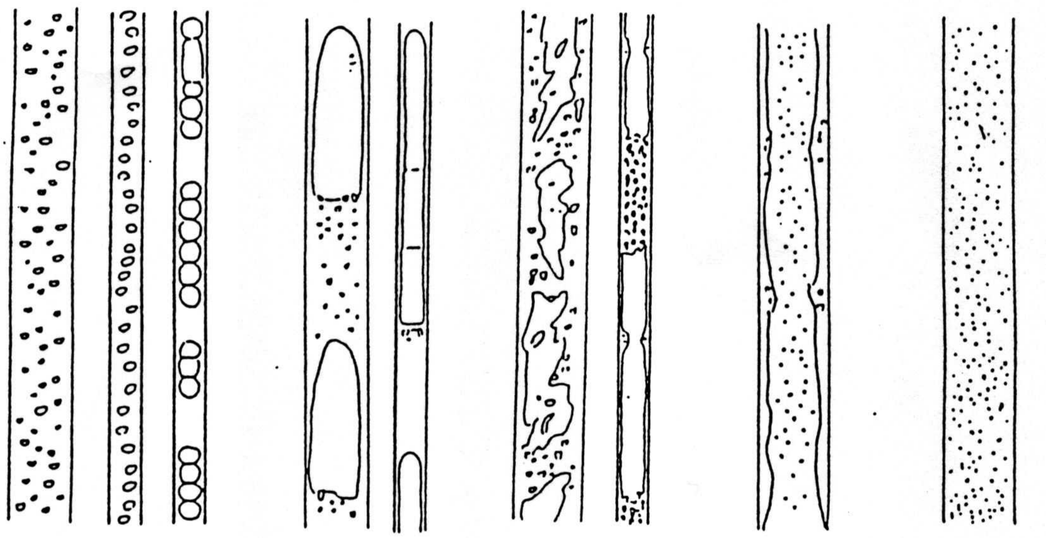

Figure 11 Effect of tube diameter on flow patterns, from Mishima and Hibiki [39]
(* Asterisks mark flow patterns which are different for small diameter tubes)

explore a possibility that a similar explanation may work for small diameter channels, two phase flow pattern studies for such channels need to be discussed.

The flow regime classification, reported by Cornwell, Kew and co-workers in Figure 6, is an appropriate starting point for this discussion. Similar to the bubble flow in large diameter tubes, the isolated bubble flow in small diameter channels would normally occur at very low vapour qualities and therefore it is of little or no relevance to the industrial practice. In connection with the confined bubble flow, it is worth referring to Figure 11 from Mishima and Hibiki [39]. It compares the flow patterns obtained for small diameter channels with those for larger diameter channels. The flow patterns marked with asterisks are those where Mishima and Hibiki found the flow structure to be different for small diameter tubes. From this figure it appears that the sketches for slug and churn flow patterns, marked with asterisks, are more likely to be true representations of confinement of bubbles.

Cornwell, Kew and co-workers proposed the use of the Confinement number as a basis for bubble confinement, see Equation (4). In this proposal there is a conceptual problem in interpreting the Confinement number as a ratio of bubble departure diameter to the channel diameter. The problem lies in the fact that the bubble departure diameter is for a vapour bubble growing on a horizontal surface, free from superimposed flow field. It is difficult to imagine how this could be relevant to a flow situation where the bubble departure would take place under the influence of the inertial rather than gravitational force. However there could be alternative interpretation of Equation (4) based on a similar dimensionless number, called Eötvos number, Equation (6). Gibson [40] showed from theoretical considerations that when a physical property group, very similar to the Eötvos number, is sufficiently small then the free rise velocity of a bubble in stagnant liquid is zero. Later Hatori [41] and Bretherton [42] independently derived a quantitative criterion for the free bubble rise velocity to be zero. In terms of the Eötvos number the criterion is as follows

$$N_{E\ddot{O}} \leq 3.37 \qquad (6)$$

where, $N_{E\ddot{O}} = \dfrac{gd_h^2\ (\rho_l - \rho_g)}{\sigma}$

It should be noted that Gibson [40], Hatori [41] and Bretherton [42] provided experimental verification for their theoretical treatments.

The narrowness of a channel leading to the bubble stagnation, i.e. zero free rise velocity of a bubble, appears to be a more meaningful interpretation of the bubble confinement than that involving a bubble departure diameter. Thus the Eötvos number based criterion in Equation (6) is suggested here as a rational basis for bubble confinement. A rearrangement of Equation (6) also shows that the bubble confinement would occur when the Confinement number, as defined by Equation (4), is greater than 0.54, a value close to 0.5 proposed by Cornwell, Kew and co-workers.

Slug flow comprising of confined vapour bubbles is, in one significant respect, different from the slug flow occurring in a large diameter tube. In large diameter tubes the Taylor bubble travels at a velocity, u_b, greater than the centre line flow velocity:

$$u_b = 1.2(u_g + u_l) + u_s \qquad (8)$$

Wallis [43] reported a general equation for the Taylor bubble rise velocity, u_s, in the inviscid region, irrespective of the tube diameter.

$$u_s = 0.345\left(\frac{gD(\rho_l - \rho_g)}{\rho_l}\right)^{0.5}\left[1 - EXP\{(3.37 - N_{E\ddot{o}})/10\}\right] \qquad (9)$$

According to Equation (9) when the Eötvos number is much greater than 3.37, i.e. the likely case for large diameter tubes, the square bracketed quantity becomes unity, and Eqution (9) reduces to the equation proposed by Nicklin et al [44] for large diameter tubes. When the Eötvos number is equal to 3.37, i.e. for small diameter tubes, the rise velocity, u_s, becomes zero because the square bracketed quantity becomes zero. Thus for small diameter tubes the Taylor bubble would travel at the same velocity of the centre line flow whereas for larger diameter tubes the Taylor

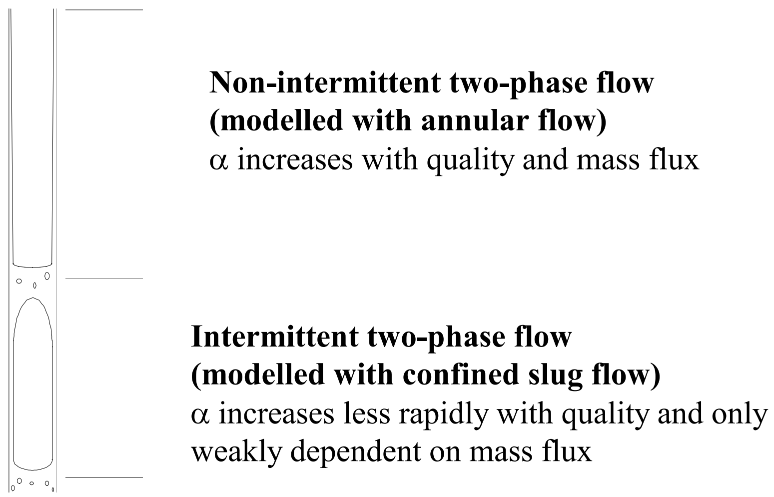

Figure 12 Schematic diagram showing two main flow patterns for small diameter tube and parametric trends in absence of nucleate boiling

bubble travels faster. This is a fundamental difference in terms of hydrodynamics, and in order to recognise this, the slug flow in small diameter channels needs to be referred to by a different name. Confined slug flow name is suggested for this purpose and used in the subsequent discussion. At higher vapour qualities transition from confined slug flow into annular flow could occur. These two main flow patterns shown in Figure 12 may be sufficient to characterise and model the heat transfer in small diameter tubes and channels. The slug flow heat transfer model [37, 38] for larger diameter tubes showed that in the slug flow region the heat transfer coefficient is not a strong function of mass flux and vapour quality. It is most likely that the confined slug flow too would exhibit similar trends. Depending upon whether the flow is confined slug flow or annular flow, and whether nucleate boiling heat transfer dominates, there would be four heat transfer regions. These regions, along with the appropriate examples chosen from the previous sections can be described as follows.

1. Confined slug flow, convective heat transfer region: This is likely to occur at low vapour quality and low wall superheat. Here the heat transfer coefficient will be independent of heat flux, and may increase only gradually with vapour quality and mass flux. Examples: Data for 2 mm diameter tube shown in Figure 4; low to medium vapour quality data marked by the solid line in Figure 5; low to medium vapour quality data in Figure 7b.
2. Confined slug flow, nucleate boiling dominated region: This is likely to occur at low vapour quality and high wall superheat. Here the coefficient will depend on heat flux and will be independent of mass flux and vapour quality. Example: Data marked as nucleate boiling in Figure 8.
3. Annular flow, convective heat transfer region: This is likely to occur at high vapour quality and low wall superheat. Here the coefficient will increase rapidly with mass flux and vapour quality. Examples: High vapour quality data marked by dashed line in Figure 5; data marked as convective boiling in Figure 8; data for 6 mm diameter tube shown in Figure 4; data in Figure 7a and high vapour quality data in Figure 7b.
4. Annular flow, nucleate boiling dominated region: This is likely to occur at high vapour quality and high wall superheat. Heat transfer coefficient will exhibit trends similar to region 2. Example: High vapour quality, high heat flux data shown in Figure 1.

It should be noted that the above discussion is presented in qualitative terms. Other factors such as operating pressure, critical pressure of the test-fluid concerned will also need to be taken into account in deciding the exact region. In spite of this limitation, the above classification appears to be successful in reconciling the different trends observed by various researchers. Note that the discussion in this section was presented largely in the context of boiling heat transfer. However, due to similar nature of two-phase convective heat transfer processes, it could also apply for condensation heat transfer.

SOME GENERAL CONSIDERATIONS

In this section some general considerations regarding the effectiveness of single phase heat transfer enhancement features for two-phase applications, the use of fin efficiency term for two-phase duties and the introduction of two phase flow in compact heat exchanger passages are discussed.

Effectiveness of Single phase Heat Transfer Enhancement Features for Two-phase Applications

Many of the features of compact heat exchanger surfaces and channels have been developed or evolved for enhancing single phase heat transfer. For example, serrated fins are specifically aimed at increasing the low gas phase heat transfer coefficient. A key issue is about the effectiveness of these features in phase change applications. A similar issue arises in the context of heat transfer enhancement techniques, which are specifically developed for single phase duties. An understanding of the effectiveness of these techniques for phase change duties could perhaps be useful for the similar problem associated with compact heat exchanger surfaces and passages.

In order to obtain such understanding, single phase and flow boiling studies reported by Wadekar [45] with a test-section containing a single phase heat transfer enhancement device can be examined. The test-section was a 3 m long electrically heated copper tube of 19.2 mm inside diameter. It contained heat transfer enhancing wire inserts in the top and bottom one metre lengths, leaving one metre length in the middle of the test-section empty, i.e. a plain tube. Local thermal measurements allowed a direct comparison of heat transfer with or without inserts under similar thermal hydraulic conditions.

Typical single phase heat transfer results from this test-section, corresponding to a Reynolds number of about 9200, are shown in Figure 13. It can be seen that the value of heat transfer coefficient in the top and bottom 1 m length, where wire inserts were present, is about 1200 W/m^2 K; in the middle plain part, without inserts, the value is about 300 W/m^2 K. This direct comparison, thus, shows a four-fold enhancement in single phase heat transfer.

Typical flow boiling results from the same composite test-section are shown in Figure 14. It can be seen that the local heat transfer coefficient remains relatively constant over the entire length of the tube and the boiling heat transfer is not affected by the presence of wire inserts. As noted above, under similar conditions of pressure and flowrate, the same test-section showed a four-fold increase in the single phase heat transfer from the use of wire inserts. It should also be noted that although the

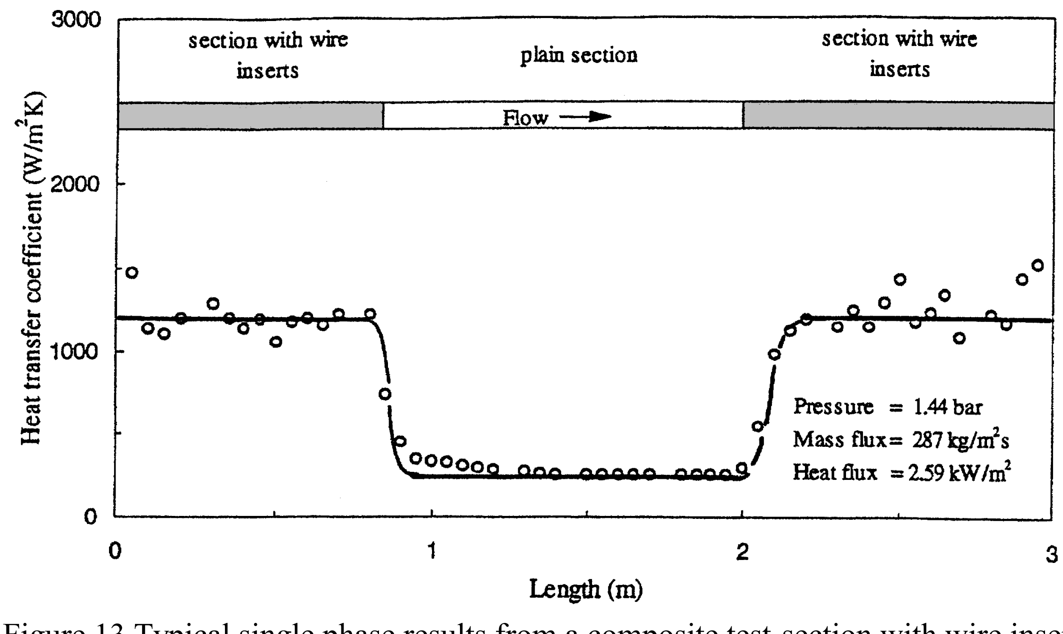

Figure 13 Typical single phase results from a composite test-section with wire inserts [45]

wire inserts did not affect the flow boiling heat transfer, the pressure drop penalty was nonetheless there. Some parallels may now be drawn between these observations and boiling in compact passages. These are illustrated for flow boiling in plate heat exchangers.

Plate heat exchangers, more specifically brazed plate heat exchangers, are used for boiling or condensation of single component fluids in the refrigeration industry. These exchangers are made with softer or harder plates (in thermal-hydraulic sense) by varying the chevron angle of the corrugations. For single phase duties harder plates provide both increased heat transfer as well as increased pressure drop when compared to softer plates. However, the results with the tube inserts indicate that for flow boiling of single component fluids there may be little or no difference between the heat transfer characteristics of hard plates and soft plates. Therefore, undue pressure drop penalty can be avoided by using soft plates for boiling duties involving a single component fluid. HTFS has measured data on boiling of refrigerants in plate heat exchangers to support this.

It is worth noting that the extra pressure drop associated with the wire inserts or a similar single phase heat transfer enhancement device may be dissipated in creating extra vapour-liquid interfacial area and also in increasing interfacial turbulence. Although this may not have any effect on phase change heat transfer to single component fluids, it may be beneficial for multicomponent mixtures. Therefore harder geometries such as hard plates in plate heat exchangers and serrated fins in plate-fin exchangers are likely to be better choices, when dealing with mixtures with a wide boiling range. The extra pressure drop associated with hard plates or serrated fins is likely to be utilised in increasing the interfacial area and interfacial turbulence, thereby minimising the reduction in heat transfer coefficient arising from mixture effects.

The proprietary correlations that HTFS is developing for boiling and condensation in compact heat exchanger passages incorporate these concepts of hard and soft geometries. As a result of this, the correlations become more general in terms of applicability to a wider range of soft and hard geometries. For example, the

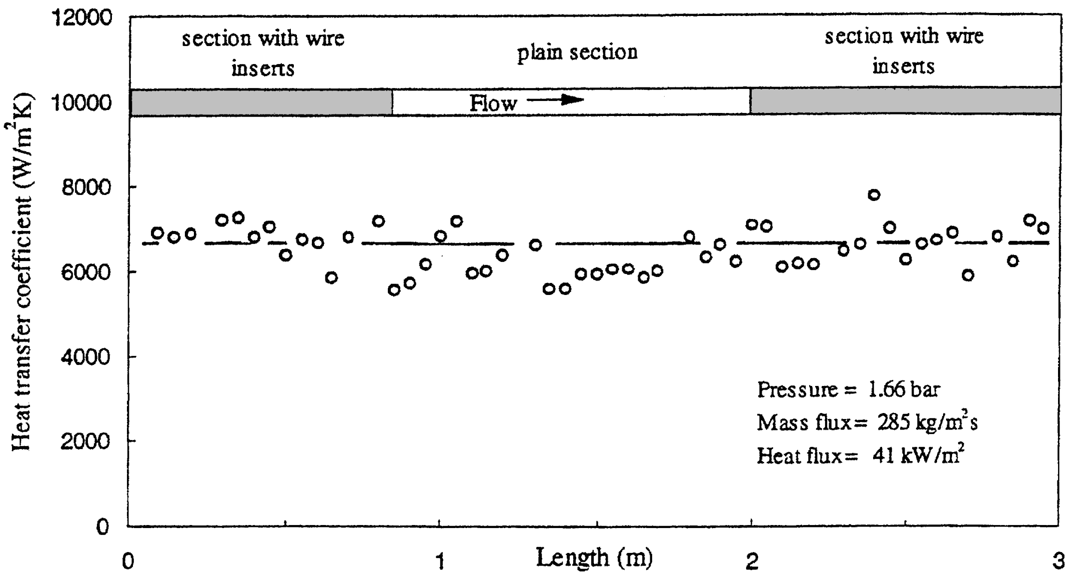

Figure 14 Typical flow boiling results from a composite test-section with wire inserts [45]

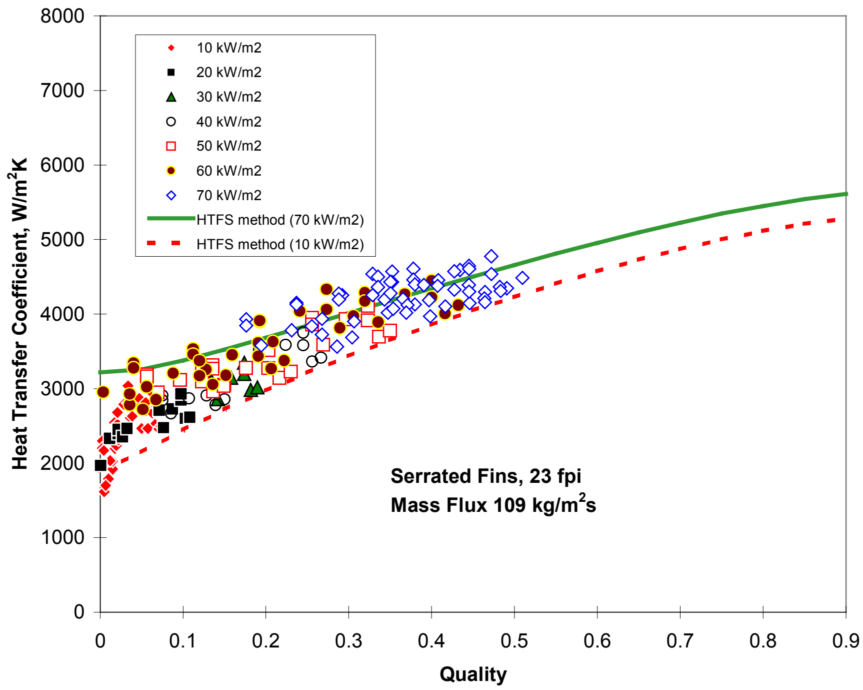

Figure 15 Predictions of the HTFS method and recent HTFS data for flow boiling of pentane

correlation for boiling in plate-fin passages is applicable to a variety of fin types encompassing a wide range of fin geometries. Figure 15 shows a comparison of the predictions against recently obtained data with pentane as a test fluid.

Fin Efficiency for Two-phase Duties

Many compact heat exchangers employ extended heat transfer surfaces for heat transfer enhancement. The standard derivation of fin efficiency for extended surfaces is based on the assumption of constant heat transfer coefficient over the extended surface. This assumption is generally valid for single phase heat transfer but may not be necessarily so for phase change duties. An extreme example of a large variation in heat transfer coefficient on the fin surface is that of boiling exhibiting nucleate, transition and film boiling on various parts of the fin surface. Such case is truly an extreme case and can only happen when very large heat flux is applied on the primary surface. In case of condensation the variation in the heat transfer coefficient over the fin surface can arise from the variation in condensate film thickness.

A useful paper on the topic of fin efficiency of extended surfaces in two-phase flow is by Srinivasan and Shah [46]. They note that in a worst case scenario a full numerical solution over the fin surface may be required to derive the true fin efficiency. Haseler and Butterworth [3] note that for aluminium plate-fin heat exchangers the assumption of constant heat transfer coefficient over the fin surface may not result in significant errors because of the standard design practice of ensuring that the relatively high fin efficiencies of about 0.7 or higher.

Introduction of Two-phase Flow into Compact Heat Exchanger

With the increasing use of compact heat exchanger for two phase duties, new challenges and problems arise, especially for those compact heat exchangers which were, until recently, only used for single phase applications. One example of such new challenges is the introduction of two-phase flow into compact heat exchangers. Because of multiple passages, and in some cases multiple subchannels, the introduction of even single phase liquid has attracted a great deal of attention from the manufacturers. This is reflected, for example, in many complex designs of distributors developed for plate-fin and plate heat exchangers. The problem with the introduction of two-phase flow is likely to be more complex.

It is instructive to examine how a similar problem is addressed for plate-fin heat exchangers which have long been used for two-phase applications. As a rule two-phase vapour-liquid mixture is never directly introduced in a plate-fin heat exchanger, but the phases are first separated and then introduced using special devices such as a spray remixer or bubbler remixer, as shown in Figure 16. There are other more complex ways of introducing two phases into a plate-fin heat exchanger [47].

One approach, which is often adopted for plate heat exchangers in refrigeration industry, is to pass the two-phase flow mixture through a proprietary mixing device, before introducing it into the exchanger. Such approach is likely to be effective for low to medium vapour qualities. Bernoux et al [48] conducted studies in a specially designed test-section. The purpose of this study was to understand the problem of phase distribution in the introduction of two-phase flow into a plate heat exchanger. They observed that, irrespective of the flow pattern in the inlet nozzle and the flow structure in the inlet manifold, the distribution of phases was never uniform. But at low vapour quality the liquid was more uniformly distributed and at high quality the vapour was more uniformly distributed. In another work related to reduction of phase maldistribution in a large rising film plate evaporator, Holm et al [49] showed that with a specially designed device using a deflector plate it was possible to reduce the maldistribution significantly.

CONCLUDING REMARKS

The use of compact heat exchangers for phase change duties will continue to grow and they will increasingly replace the conventional shell and tube exchangers. They will also establish themselves in entirely new and novel applications; many of the novel

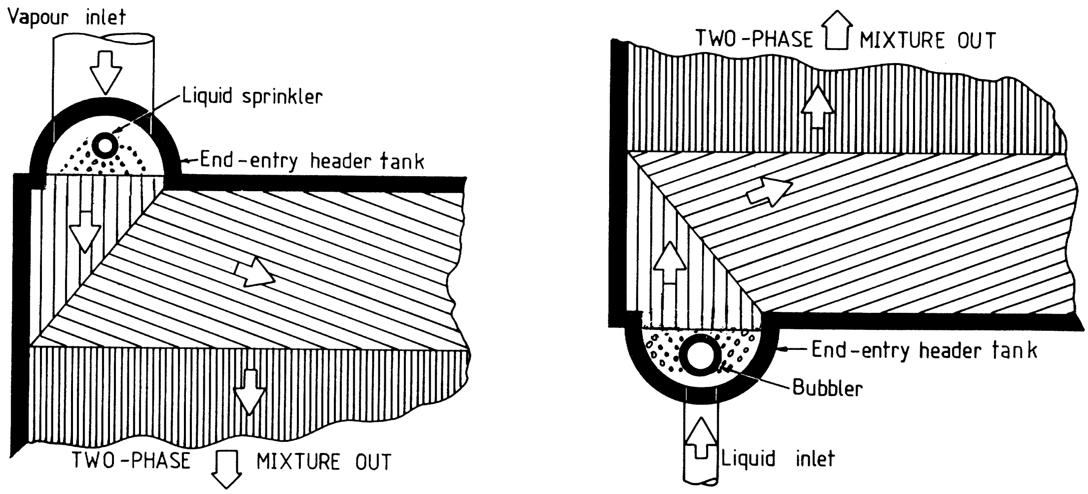

Figure 16 Spray remixer and bubbler remixer for distributing the phase in the exchanger header tank [47]

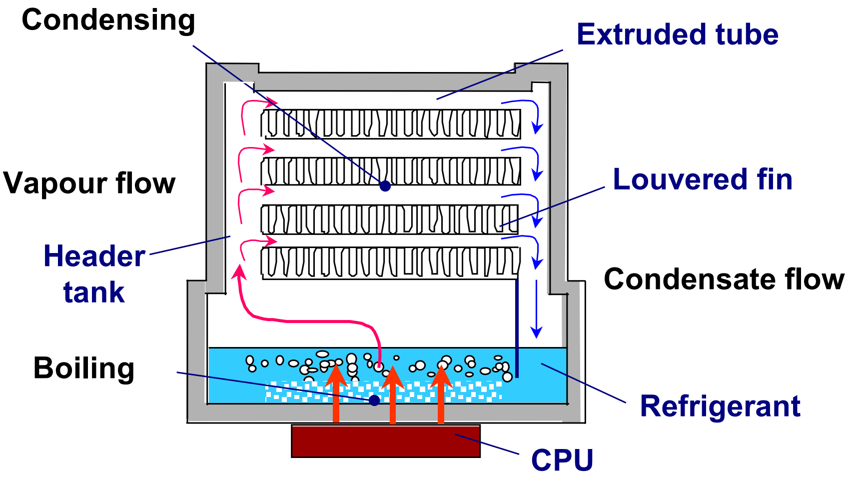

Figure 17 A novel plate-fin design for high heat flux of cooling of electronic devices [52]

applications could be in the area of process intensification. Cooling of electronic and power devices, fuel cell applications could be the areas of future growth for new and novel applications of compact heat exchangers for phase change duties. Before delineating the areas of further work two more examples of recent work are described; one relates to the type of collaborative work required for the replacement type application and the other relates to new applications.

Cailloux et al [50] demonstrated the feasibility of replacing a shell and tube overhead condenser for hydrocarbon duties with a compact welded heat exchanger. In their collaborative project with industry the investigation ranged from conceptual plant design stage, through study of condensation in the welded condenser to final verification of the compact condenser performance through the use of commercial software for process simulation. Ando et al [51] reported a novel plate-fin design for the removal of heat from power electronic devices at high heat flux. In this application the principle of heat pipe is used with a hermetically sealed refrigerant fluid inside the evaporator-condenser. A similar design is reported by Tanaka et al [52] and is shown in Figure 17, where air flows through the finned channels in a crossflow manner with the condensing refrigerant.

In order to further accelerate the use of compact heat exchangers for phase change duties some of the suggested areas of further research work are as follows.

1. More two-phase flow pattern studies are required for the compact heat exchanger passages, especially for cross corrugated channels of plate heat exchangers. This information would be useful for developing flow pattern specific models for compact heat exchangers.
2. Use of modern measurement techniques such as liquid crystal thermography etc to establish the heat transfer coefficient variation. These studies should be focussed on obtaining better predictive methods for time averaged local heat transfer coefficients
3. It is observed that the data collected for single small diameter channels tends to be at higher mass fluxes than those for normal diameter tubes. This makes it difficult to compare the performance of the tubes of different diameters. Therefore

systematic work covering the same mass fluxes with a wide range of tube diameters is required.

4. One of the effective ways of improving the performance of compact heat exchangers for two-phase duties is to provide uniform flow distribution in the exchanger passages. This is especially true for plate heat exchangers where various proprietary and non-proprietary designs exist for improving the phase distribution; providing a more logical basis for their selection will be useful.

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Table 1: Summary of test-section geometry, conditions, test-fluids covered for phase change in plate-fin channels.

Fin size h/ t/ (L_s) (mm)	Fin frequency (fins/metre)	D_h (mm)	Mass flux (kg/m^2s)	Heat flux (kW/m^2)	Test-fluids	Fin Types	Orientation/ Heating/Cooling	Pressure (kPa)	Ref
Boiling									
6.15/0.2/(3.18)	591	2.4	11-110	<4	LN_2	Serrated	Vertical upflow Both sides electrically heated	300	[20]
6.15/0.2	590	2.4	37-120	<4	LN_2	Perforated	Vertical upflow Both sides electrically heated	140-520	[21,22]
6.15/0.2	590	2.4	50-290	1-10	Cyclohexane	Perforated	Vertical upflow Both sides electrically heated	150	[23]
6.15/0.2	590	2.4	50-290	1-10	*n*-heptane	Perforated	Vertical upflow Both sides electrically heated	150	[24]
3.8/1.59/(12.7) 9.52/1.91/(12.7) 1.91/1.59/(12.7)	105 98 105	5.14 8.86 3.08	3-320 3-320 3-320		Methanol, Butanol and Water	Serrated	Vertical upflow One side electrically heated	100	[26]

3.8/1.59/(12.7)	105	5.14	4-60		Methanol, Butanol and Water	Serrated	Horizontal One side electrically heated	100	[25]
6.93/0.2/(3.18) 6.93/0.2/(9.52) 6.93/0.2 3.23/0.2 6.93/0.2	709 736 709 709 866	2.06 1.985 2.06 1.76 1.68	19-49	1.4-3.54	R114	Serrated Serrated Perforated Perforated Perforated	Vertical upflow Both sides electrically heated	300	[27]
6.93/0.2/(3.18)	919.2	1.574	12-71	0.2-7.4	Propane	Serrated	Vertical upflow fluid Heated by glycol solution	800	[28]
Condensation									
6.15/0.2	710	2.4	100	<4	LN_2	Plain	Vertical downflow Cooled by boiling stream on both sides	Up to 700	[30,31]
6.15/0.2/(3.18)	710	2.4	30-80	<4	LN_2	Serrated	Vertical downflow Cooled by boiling stream on both sides	600	[32]

REFERENCES

1. Shah R.K., Classification of heat exchangers, in: Heat Exchangers: Thermal-Hydraulic Fundamentals and Design, Kakac, S., Bergles A.E. and Mayinger, F., pp 9-46, 1986.
2. Webb, R. L., Principles of Enhanced Heat Transfer, John Wiley and Sons Inc., New York, 1994
3. Haseler, L. E. and Butterworth, D., 1995, Boiling in compact heat exchangers/Industrial practice and problems, Keynote Paper IV, Int. Conference on Convective Flow Boiling, Banff, Canada.
4. Huo, X., Tian, Y. S., Wadekar, V. V. and Karayiannis, T. G. 2001 Review of Aspects of Two Phase Flow and Boiling Heat Transfer in Small Diameter Tubes, Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 335-346
5. Kandlikar, S. G. 2001 Two-phase flow patterns, pressure drop and heat transfer during boiling in minichannel and microchannel flow passages of compact evaporators, Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 335-346
6. Srinivasan, V and Shah R.K., 1997, Condensation in compact heat exchangers, Journal of Enhanced Heat Transfer, Vol 4, pp 237-256.
7. Lazarek, G. M. and Black, S. H., 1982, Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113, Int. J. Heat Mass Transfer, Vol. 25, No. 7, pp. 945-960.
8. Wambsganss, M. W., France, D. M., Jendraejczyk, J. A. and Tran, T. N., 1993, Boiling heat transfer in a horizontal small-diameter tube, Journal of Heat Transfer, Vol. 115, pp. 963-972.
9. Tran, T. N., Wambsganss M. W. and France D. M., 1996, Small circular- and rectangular-channel boiling with two refrigerants, Int. J. Multiphase Flow, Vol. 22, No. 3, pp. 485-498.
10. Bao, Z. Y., Fletcher, D. F. and Haynes, B. S., 2000, Flow boiling heat transfer of Freon R11 and HCFC123 in narrow passages, Int. Journal of Heat and Mass Transfer, Vol. 43, pp. 3347-3358.
11. Kureta, M., Kobayashi, T., Mishima, K. M. and Nishihara, H., 1998, Pressure drop and heat transfer for flow boiling of water in small-diameter tubes: JSME Int. J. B, Vol.41, 4, pp.871.
12. Oh H., Katsuta M., Shibata K., Heat transfer characteristics of R134a in a capillary tube heat exchanger, Proc. of 11th International Heat Transfer Conference Vol. 6, 1998, pp. 131-136.
13. Yan, Y. Y. and Lin, T. F., 1998, Evaporation heat transfer and pressure drop of refrigerant R-134a in a small pipe, Int. J. Heat Mass Transfer, Vol. 41, pp. 4183-4194.
14. Lin, S., Kew, P. A. and Cornwell, K., 1998, Two-phase flow regimes and heat transfer in small tubes and channels, Heat Transfer 1998, Proceedings of 11th IHTC, Vol. 2, pp. 45-50.
15. Kew, P. A. and Cornwell, K., 1994, Confined bubble flow and boiling in narrow spaces, Heat Transfer 1994, Proceedings of 10th IHTC, Vol. 7, pp. 473-478.
16. Kenning, D.B.R. and Yan, Y., 2001, Saturated flow boiling of water in a narrow channel: experimental investigation of local phenomena, IChemE Trans.A, Chem.

Eng. Research and Design 79, 425-436.

17. Fiedler, S. and Auracher, H. 2001 `Heat Transfer During Reflux Condensation of R134a in a Small Diameter Inclined Tube.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 369-376
18. Russeil, S., Tribes, C. and Baudoin B. 2001 `Experimental Study of Condensation on Two Flat Vertical Plates.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 377-383
19. Tribes, C., Russeil, S. and Baudoin B. 2000 `Retention and drainage of condensate on heat exchanger surface' 3rd European Thermal-Sciences Conference, Heidelberg, 2000, pp 893-898.
20. Robertson J.M., 1979, Boiling heat transfer with liquid nitrogen in brazed aluminium plate-fin heat exchangers, National Heat Transfer Conference, AlChE Symposium Series 189, Vol.75, San Diego, pp 151-164
21. Robertson J.M., 1983, The boiling characteristics of perforated plate-fin channels with liquid nitrogen in upflow, Heat Exchangers for Two-phase Applications, HTD 10, Proc. of the ASME Heat Transfer Division, pp 35-40.
22. Robertson J.M. and Clarke R.H., 1985, The general prediction of convective boiling coefficients in plate-fin heat exchanger passages, Heat Transfer, AlChE Symposium Series 81 Vol 245, pp 129-134.
23. Robertson J.M and Wadekar V.V., 1988, Boiling characteristics of cyclohexane in vertical upflow in perforated plate-fin passages, 25th National Heat Transfer Conference, AlChE Symposium Series 84 Vol 263, pp120-125.
24. Wadekar V.V., 1992, Flow boiling of heptane in a plate-fin heat exchanger passage, Compact Heat Exchangers for Power and Process Industries, HTD 201, Proc. of the ASME Heat Transfer Division, pp 1-6.
25. Mandrusiak G. D., Carey V. P. and Xu X., 1988, An experimental study of convective boiling in a partially heated horizontal channel with offset strip fins, ASME J. Heat Transfer 110 pp 229-236.
26. Mandrusiak G. D. and Carey V. P., 1989, Convective boiling in vertical channels with different offset strip fin geometries, ASME J. Heat Transfer 111, 156-165.
27. Feldman, A., Marvillet, C. and Lebouché, M., 2000, "Nucleate and convective boiling in plate fin heat exchangers", Int. J of Heat and Mass Transfer, Vol 43, pp 3433-3442.
28. Watel B. and Thonon B., Flow boiling in plate fin heat exchanger, 3'"European Thermal-Sciences Conference, Heidelberg, 2000, Vol 2, pp1171-1176.
29. Mertz, R. and Groll, M., 2000, Flow boiling heat transfer in narrow channels, Heat and Technology, Vol. 18, Supplement n. 1, pp. 75-79.
30. Robertson J. M., Blundell, N. and Clarke R. H., 1986, The condensing characteristics of nitrogen in plain brazed aluminium plate-fin heat exchanger passages, Heat Transfer 1986, Proc. 8th International Heat Transfer Conference, Vol. 4, pp 1719-1724.
31. Robertson J. M., Blundell, N. and Clarke R. H., 1987, Characteristics of pressure gradients in downflow condensing of nitrogen in plain brazed aluminium plate-fin heat exchanger passages, AlChE Symposium Series 83 Vol.257, pp 120-125.
32. Clarke, R. H. 1994, The condensation heat transfer characteristics of liquid nitrogen in serrated plate-fin passages, IChemE Symposium Ser. No. 129, Paper 96.

33. Wang, L, Christensen, R. and Sundén, B 2000, 'An Experimental Investigation of Steam Condensation in Plate Heat Exchangers', International Journal of Heat Exchangers, Vol 1, pp.125-150
34. Boyko, L. D. and Kruzhilin, G. N. 1967, 'Heat Transfer and Hydraulic Resistance during Condensation of Steam in a Horizontal Tube and in a Bundle of Tubes', Int. J. Heat Mass Transfer, Vol 10, pp.361-373
35. Gitteau, J., Thonon, B. and Bontemps, A. 2001 `Condensation of Pure Hydrocarbons and their Mixtures in a Compact Welded Heat Exchanger.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 487-494.
36. Kenning, D. B. R. and Cooper, M. G. 1989, Saturated flow boiling of water in vertical tubes, Int. J. Heat Mass Transfer, Vol.32, pp. 445-458.
37. Wadekar, V. V. and Kenning, D. B. R., 1990, "Flow boiling heat transfer in vertical slug and churn flow region", Proc. 9th International Heat Transfer Conference, Jerusalem, Vol. 3, pp. 449-454.
38. Wadekar, V. V., 1991, Vertical slug flow heat transfer with nucleate boiling, ASME, HTD-Vol. 159, pp. 157 - 161.
39. Mishima, K. M. and Hibiki, T., 1996, Some characteristics of air-water two-phase flow in small diameter vertical tubes, Int. J. Multiphase Flow, Vol. 22, No. 4, pp. 703-712.
40. Gibson, A. H., 1913, On the Motion of Long Air Bubbles in a Vertical Tube, Philosophical Magazine, 26-156, pp. 952--965.
41. Hattori, S., 1935, On the Motion of a Cylindrical Bubble in a Tube and its Application to the Measurement of the Surface Tension of a Liquid, Report 115, Aeronaut. Res. Inst. Tokyo Imp. Univ.
42. Bretherton, F. P., 1961, The Motion of Long Bubbles in Tubes, J. Fluid Mech., pp. 166-188.
43. Wallis, G. B. One-dimensional Two-phase flow, McGraw Hill Inc, 1969.
44. Nicklin, D. J. , Wilkes, J. O. and Davidson, D. F., 1962, Two phase flow in vertical tubes, Trans. Institution Chem. Engrs, Vol 40, pp. 61-68.
45. Wadekar, V. V., 1998, Improving industrial heat transfer - Compact and not-so-compact heat exchangers. *Enhanced Heat Transfer*, Vol. 5, pp. 53-69.
46. Srinivasan, V and Shah R. K., 1995, Fin efficiency of extended surfaces in two-phase flow, Fin efficiency of extended surfaces in two-phase flow, Proc. of the 1st Int. Symp. on Two Phase Flow Modelling and Experimentation, Rome, Italy, 9-11 Oct. 1995, G. P. Celata, R. K. Shah (Eds.), Edizioni ETS, Vol. 2, pp. 1107-1118.
47. Taylor M. A., Ed., 1987, Plate-fin heat exchangers: Guide to their specification and use, Published by HTFS
48. Bernoux, P., Bernier, P. and LeBouche, M 2001 `Two-Phase Flow Distribution in a Compact Heat Exchanger.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 347-352
49. Holm, M., Nilsson, I. and Wilhelmsson, B. 2001 `Reduction of Maldistribution in Large Rising Film Plate Evaporators.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 519-524
50. Cailloux, B. D., Lee, J. M. G. and Jachuck, R. J. 2001 `Process Intensification: Performance Studies and Integration of Multifunctional Compact Condenser in Distillation Processes.' Compact Heat Exchangers and Enhancement Technology

for the Process Industries, Davos, Switzerland, pp. 533-538

51. Ando, K. 2001 'Development and application of semiconductor device cooler using high performance plate heat exchanger design.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 509-514.
52. Tanaka, H., Ohara, T., Kawaguchi, K. and Terao, T. 2001 `Boiling Refrigerant Type Compact Cooling Unit for Computer Chip.' Compact Heat Exchangers and Enhancement Technology for the Process Industries, Davos, Switzerland, pp. 515-518

NOMENCLATURE

Bo	Boiling number
Co	Confinement number
d_h	Hydraulic diameter
D	Tube diameter, m
F	Two phase heat transfer multiplier, α_c/α_l
g	Gravitational acceleration, m/s^2
G, $\dot{m}$	Mass flux, kg/m^{2}s
h	Fin height, mm
L	Tube length, m
L_s	Serration length, mm
$N_{EÖ}$	E tvos number
Nu	Nusselt number
$\dot{q}$	Heat flux, W/m^2
Re	Reynolds Number
t	Fin thickness, mm
T	Temperature, K
u_b	Bubble velocity, m/s
u_s	Bubble rise velocity, m/s
u	Superficial velocity, m/s
We	Weber number
x	vapour quality

Greek

α	Heat transfer coefficient, W/m^{2}K
β	Angle of inclination
χ	Lockhart-Martinelli parameter
η	Viscosity, kg/m s
ρ	Density, kg/m^3
σ	Surface tension, N/m

Subscript

l	Liquid
c	Convective
g,v	Gas or vapour
sat	Saturation
w	Wall