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SOME CONSIDERATIONS REGARDING THE ENHANCEMENT OF NUCLEATE POOL BOILING

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Abstract. *The purpose of this paper is to present a review of some recent results for enhanced pool boiling applications using passive techniques of enhanced surfaces. The work begins with a review of the main results for confined boiling, particularly the principal dimensionless parameters and the different mechanisms involved in the nucleate regime and their effect on the heat transfer coefficient and the limit of heat flux or dryout phenomenon. We then consider some results for finned and microfinned surfaces in confined and non-confined conditions. Finally, we review some special kinds of structured surfaces for which the enhancement effect is produced by the combination of microfins and foils with pores placed over them or directly over smooth tubes.*

Keywords. *enhanced boiling, nucleate boiling, grooved surface, confined boiling, structured surface*

1. Introduction

The nucleate pool boiling regime continues to be the object of a great number of studies because of its recent application in cooling electric systems and electronic components as well as in the refrigeration industry. In particular, the latter has improved the development of compact evaporators for automotive air-conditioning. Kenning (1997) reviewed the papers presented at the Eurotherm Seminar 48 on Pool Boiling which was characterized by experimental studies into different types of geometries, surface treatments and refrigerant mixtures for industrial applications.

The boiling heat transfer coefficient can be increased by means of the area augmentation of the heat transfer surfaces using, for example, passive techniques such as the addition of microfins to the inner or outer tube surfaces, microfin plates and coatings, and finned surfaces (Passos and Reinaldo, 2000; Bergles, 1998; Stephan, 1992; Thome, 1990; Webb, 1994). The heat transfer coefficient, for low and moderate heat flux, can be increased also by simple confinement (Ishibashi and Nishikawa, 1969; Yao and Chang, 1983; Bonjour and Lallemand, 1997). Several proposals, including an augmentation of the area combined with the confinement of the cooling fluid, offer very interesting kinds of improvement of heat transfer mechanisms (Nakayama et al., 1980a-b; Chien and Webb, 1998a-b; Kim and Choi, 2001).

The idea that the heat transfer coefficient in nucleate boiling regime, can be increased to an extent greater than that utilizing special fin and micro-fin surfaces by means of, for example, a thin copper welded plate with pores over an enhanced tube has been formulated in patents registered in the sixties and seventies, as is described in Nakayama et al. (1980a). These surfaces are known as structured surfaces and can be obtained by techniques which are not so expensive (Webb, 1994).

In this work a current perspective on the nucleate boiling regime for different geometries and fluids is presented. The authors discuss the latest results from the literature, including those obtained in their laboratories with the main focus on the effects of confinement and enhancement of the surfaces allowing increased nucleate boiling.

2. Kinds of configurations, surfaces and dimensionless parameters

In order to better understand the main enhancement boiling mechanisms using passive techniques we will analyze the effect of the confinement on smooth or plain surfaces and on finned and micro-finned surfaces, with and without structured surfaces. Because this review is directed to the pool boiling mode of heat transfer, the finned and structured surfaces considered are those working, in general, in flooded evaporators, Kakaç and Liu (1998). A combination of factors such as the augmentation of the surface area, and the confinement can change the mechanisms of heat transfer and thereby promote a delay or an acceleration in the initiation of the reduction of the heat transfer or dryout phenomenon. A knowledge of the relation between these different mechanisms and the attempt to identify the main dimensionless groups can compensate the absence of general correlations due to the complexity of the physical problem and the still relatively small amount of experimental data, so contributing to the orientation of the design of new applications.

2.1. The confinement effect

It is well known that the heat transfer mechanism in the pool boiling regime on a simple plain surface can be modified by means of the confinement of the system using, for example, an unheated surface (Ishibashi and Nishikawa, 1969; Katto et al., 1977; Yao and Chang, 1983; Fujita et al., 1988; Nishikawa and Fujita, 1990; Palm, 1991; Bonjour, 1996, Bonjour and Lallemand, 1998). The general trend that characterizes these previous results is that the heat transfer coefficient, for moderate heat flux, can increase when the distance between the heated and unheated surfaces decreases. However, for high heat flux this enhancement effect disappears and the maximum heat transfer, corresponding to the critical heat flux or dry out, decreases when the distance between the heated and unheated surfaces decreases (which is equivalent to an increase in the confinement).

Ishibashi and Nishikawa (1969) have studied the pool boiling inside the annulus between a vertical copper cylindrical heater and various glass tubes with different internal diameters and shown that there are two main regimes in the nucleate pool boiling regime, one characterized by isolated vapor bubbles and the other by coalesced bubbles. For the isolated bubble regime, at a constant heat flux q, the heat transfer coefficient h depends on the characteristic length of confinement (s), represented by the gap between the heated cylinder and the glass tube, as shown by the following equation:

$$h \propto q^{\frac{2}{3}} s^{-0.13} \tag{1}$$

In narrow channels, for low and moderate heat flux, the bubbles become squeezed and the liquid film between the coalesced bubble and the heater surface causes a more efficient evaporative process in which the area of the liquid film surface increases with the confinement, and h depends on s as shown by the following equation:

$$h \propto q^{\frac{2}{3}} s^{-\frac{2}{3}} \tag{2}$$

An other important result obtained by Ishibashi and Nishikawa (1969) is the non dependency of h on the surface tension, in the coalesced bubble regime. We can observe from Eqs. (1) and (2) that the dependency of h on s is stronger for the coalesced bubbles regime compared with the isolated bubble regime. For the water nucleate pool boiling regime, at atmospheric pressure, inside a vertical annulus in which the internal surface was heated and with a gap of 0.97 mm, and in the coalesced bubbles regime, the rate of latent heat transport corresponding to the coalesced bubbles is approximately 9 % of the total heat flux.

The dimensionless parameter resulting from the ratio between the characteristic length of a particular confinement condition and the capillary length (Carey, 1992), is known as the Bond number (Yao and Chang, 1983) and is given by the following equation:

$$Bo = \frac{s}{\sqrt{\dfrac{\sigma}{g(\rho_l - \rho_v)}}} \tag{3}$$

where the term in the denominator is the capillary length which is proportional to the departure diameter; a function of the square root of the ratio between the surface tension, σ, and $g(\rho_l - \rho_v)$, where g, ρ_l and ρ_v represent the acceleration of gravity, the density of the liquid and the density of the vapor, respectively. In general, when $Bo < 1$ the confinement effect is important and the bubbles tend to be deformed or squeezed whereas for $Bo \geq 1$ the boiling is not confined. Some references consider the Bond number as the square of the value for Bo derived from Eq. (3), also called Eötvos number (Rohsenow et al., 1998).

Katto et al. (1977) have analyzed and visualized the atmospheric water pool boiling in a horizontal gap between two horizontal disks, in which the bottom disk is heated and face is upward, and the reported results show the increase of the heat transfer coefficient when s decreases, for low and moderate heat flux, and a distance s between the disks higher or equal to 0.2 mm. The results of Katto et al. (1977) show that this enhancement effect on the boiling mechanisms is achieved when $Bo \leq 1$.

In general, the analysis of the Bond number is not enough to characterize the deformation of the vapor bubbles due to the confinement. Yao and Chang (1983), have presented an analysis, at atmospheric pressure, for R-113, water and acetone pool boiling, inside a vertical narrow annulus with heights of 25.4 and 76.2 mm and gaps of 0.32, 0.80 and 2.58 mm. In addition to the Bond number, these authors considered the effect of the aspect ratio of the channel in combination with the heat flux and define a modified boiling number that is given by the following equation:

$$Bn = \frac{qH}{h_{fg}\rho_v su} \tag{4}$$

where q, H, h_{fg} and s represent the heat flux delivered to the working fluid, the height of the channel, the heat of vaporization and the annulus gap, respectively, and u represents the velocity magnitude of the vapor bubbles calculated by means of the following equation:

$$u = \left[\frac{gD(\rho_l - \rho_v)}{\rho_l}\right]^{\frac{1}{2}} \tag{5}$$

where D represents the departure bubble diameter. The dimensionless parameter presented in Eq. 4 can be interpreted (Yao and Chang, 1983) as the ratio of the characteristic time of bubble rising through the confined space (H/u) to the characteristic time of vapor formation ($h_{fg}\rho_v s/q$). Plotting the values of Bn as a function of the Bond number, these authors have characterized their experimental results for three regimes: isolated deformed bubbles, coalesced deformed bubbles and slightly deformed bubbles.

Misale and Bergles (1997) studied the effect of the gap size and the degree of subcooling on FC-72 or Galden HT-55 nucleate pool boiling on three in-line vertical heaters, whose dimensions were 29x29 mm with a thickness of 2 mm, cooled from both sides, at atmospheric pressure, with a gap range of 0.5, 2 and 33 mm. The results showed a decrease of the confinement causes an increase in the heat transfer coefficient whereas the critical heat flux is decreased. At saturated boiling, moderate heat flux ($q < 20$ kW/m^2) and with gaps of 2 and 33 mm the heater placed in the upper part of the channel was better cooled than that placed in the bottom, whereas for the gap of 0.5 mm there was no clear difference in the heat transfer coefficient related to the placement of the heaters.

Bonjour (1996) and Bonjour and Lallemand (1997) considered the enhancement effect on R-113 boiling inside a rectangular vertical channel caused by the reduction of the gap size. The channel height was 120 mm and the gap-size ranged from 0.3 to 2 mm and included an unconfined configuration. The results showed an increase in the heat transfer coefficient as the gap size decreased, when the heat flux was lower than 3-5 kW/m^2. Using the hot-wire anemometry technique as a detector of liquid and vapor phases, Bonjour and Lallemand (1998) classified the confined boiling regimes as isolated deformed bubbles, coalesced bubbles and partial dryout. The transition from isolated deformed bubbles to coalesced bubbles corresponded to a heat flux of 20 % of the critical heat flux (or dryout heat flux) and that from coalesced bubbles to partial dryout to 70 % of the critical heat flux.

2.2. Finned surfaces and confinement

A passive enhancement surface with fins, micro-fins and grooves allows the reduction of tube length and therefore more compact heat exchangers. For industrial applications it is very important to analyze the behavior of new enhanced tube surfaces in the heat flux range of 10 to 50 kW/m^2.

Hahne and Müller (1983) have determined the R-11 boiling heat transfer coefficient, at atmospheric pressure, on the outside of a horizontal finned (thickness=0.4 mm, heigth=1.5 mm and fin-pitch=1.35 mm) single tube and an in-line 18 horizontal tube bundle arrangement with six rows, in which the fins are straight and the mean surface roughness of the tube is R_p=0.3 µm. For moderate heat flux, between 3 and 20 kW/m^2, the results show a dependency on the position of the tubes in the bundle, whereas for the fully developed nucleate boiling regime, when the heat flux is between 20 and 40 kW/m^2, the heat transfer coefficient as a function of the heat flux is well represented by the same curve. Equations (6) and (7), presented in Tab. (1), represent the experimental correlations for these two heat flux regions. The results for a single heated tube tested in the middle of the bundle, as indicated in Fig. (1), are correlated by Eq. (8) of Tab. (1). For the entire bundle arrangement, the data can be correlated by

Eq. (9), which is very close to the data for a tube tested in the middle of roll 4, for example, as is shown in Fig. (2). Therefore, it is possible to obtain the heat transfer coefficient for a bundle through the experimental analysis of a single heated tube placed in the middle of the bundle. These results may be very useful for the planning of new experimental analysis in bundles due to the practical interest in this kind of arrangement. In Eqs. (6)-(9), in Tab. (1), h_{single} represents the heat transfer coefficient for a single heated tube in W/m^{2}K, and q the heat flux in W/m^2 considering the total heat transfer area.

Hübner and Künstler (1997) presented experimental results for the heat transfer coefficient as a function of the heat flux for pool boiling on the outside of plain and GEWA-straight fin tubes with a trapezoidal shape and GEWA tubes with T or Y-shaped fins, considering the total surface area of the tubes. These authors showed that the tube surfaces as manufactured contain regions with distinct roughness level, since the top of the fins are rougher than the flanks and the base of the fins. In particular, in the case of the straight trapezoid-shaped fin, it was shown that with uniform surface roughness, similar to that of plain tubes, and achieved by sandblasting, the enhancement effect on the heat transfer coefficient, obtained for commercial tubes when compared with plain tubes, vanished. It is necessary to consider that the effect of roughness was studied only in the case of straight fins, without the confinement effect produced by the tunnels of the structured surfaces, as will be seen in the next section.

Table 1. Correlations obtained by Hahne and Müller (1983) for a horizontal finned tube and a finned tube bundle

Geometry	Heat flux range, q (W/m^2)	Empirical correlation, h (W/m^{2}K)	Equation
Single tube	3000 – 20000	$h_{single} = 0.697\,q^{0.79}$	(6)
	20000 – 40000	$h_{single} = 8.53q^{0.54}$	(7)
Single tube in the bundle	1000 – 40000	$h_{tube-bundle} = 1.331q^{0.735}$	(8)
Bundle	1000 – 40000	$h_{bundle} = 10.0q^{0.53}$	(9)

Test conditions: Atmospheric pressure (T_{sat} = 23.4 ^{o}C), Working fluid: R-11. Fin geometry: thickness=0.4 mm, heigth=1.5 mm and fin-pitch=1.35 mm

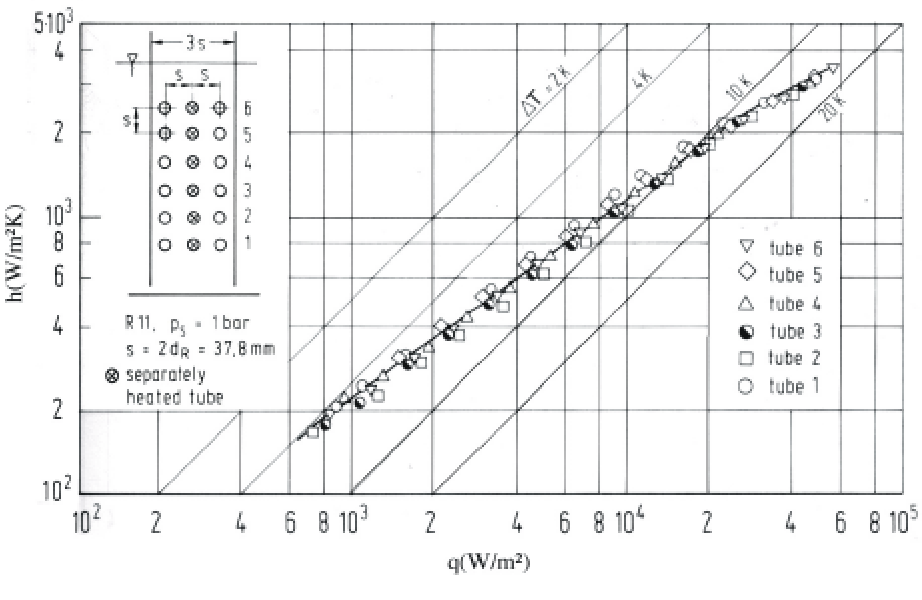


Figure 1. Heat transfer coefficient h_n for middle tubes heated separately in a bundle arrangement (from Hahne and Müller, 1983)

Chang and You (1997) have analyzed FC-87 and R-123 atmospheric pool boiling up to critical heat flux on the outside surface of six enhanced tubes horizontally placed and immersed in saturated liquids (FC-87 at 302.3 K and R-123 at 301 K). In the case of a triangular finned surface tube with a fin pitch (=open top fins) and height of 1.39 and 0.89 mm, respectively, the heat transfer coefficient based on a nominal diameter was 170 to 220% higher than the plain tube, which is greater than the 100 % increase in the finned surface area. This result is related to the roughness created during the machining process of the tube. The critical heat flux related to the dryout phenomenon is 86% greater than the experimental value for the plain tube and 50 % greater than the value predicted by the Zuber's correlation (Carey, 1992).

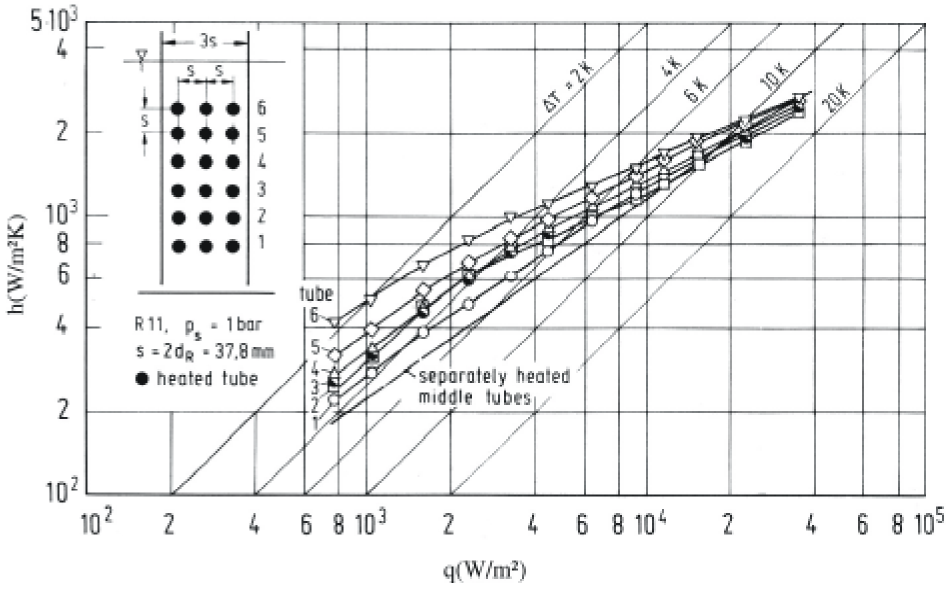


Figure 2. Heat transfer coefficient for a bundle arrangement (from Hahne and Müller, 1983)

2.3. Structured surfaces

The structured surface is characterized by the existence of a continuous tunnel over the outside surfaces of tubes connected to the outside liquid by narrow passages such as gaps and pores, as shown in Fig. (3).

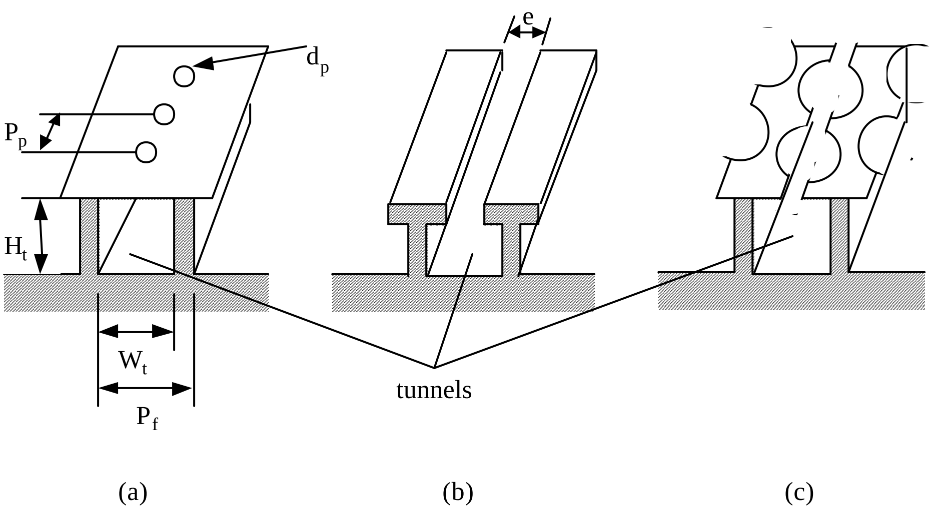


Figure 3. Diagram showing structured surfaces (from Kim and Choi, 2001).

Figure 4 shows the main dimensions for a fin cross section of a GEWA-T tube, a diagram of which is given in Fig. 3b. The tunnel effect caused by the T-shaped fin promotes greater action of the vapor bubbles inside the helical channel or tunnel in comparison to straight-shaped fins (Stephan, 1992), because of the greater restriction to vapor bubbles escaping from the tunnels.

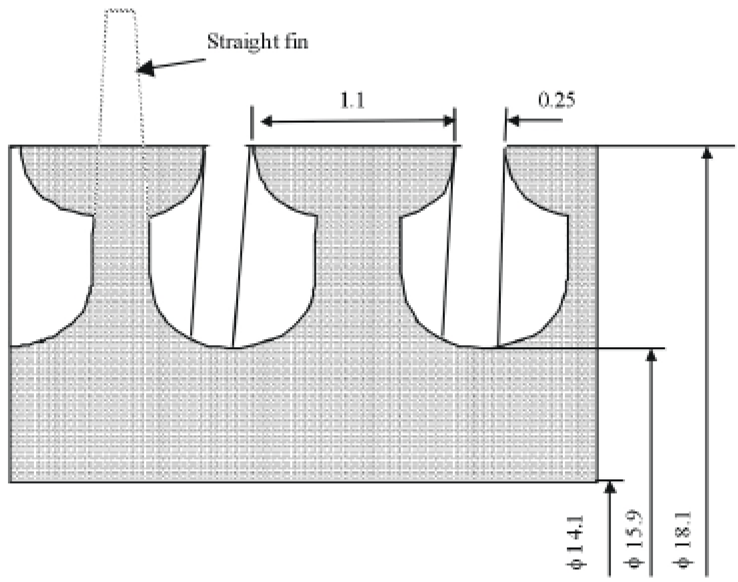


Figure 4. Structured surface (GEWA-T-tube), area factor equal to 3.18 (from Stephan, 1992)

Marto and Lepere (1982) analyzed the atmospheric pool boiling of R-113 and FC-72 for three commercial enhanced surfaces, two of them being structured surfaces: the GEWA-T tube (similar to Figs. 3b and 4) and the Thermoexcel-E tube (similar to Fig. 3-c) and compared the results with those for a horizontal tube with 15.8 mm outside diameter (OD) and a plain surface. The results show that the wall superheating necessary for the onset of nucleate boiling for R-113 is greater than for FC-72, and the superheat ratios varied between 1.64 and 2.4. The ratio between the heat transfer coefficients for each structured surface and for the plain tube shows that the enhancement effect for the Thermoexcel-E tube is higher (nearly 10 times) than for the GEWA-T tube (nearly 2.5 times), when the heat flux is low (4 kW/m^2) and the fluid is R-113. However, the enhancement effect deteriorates more rapidly for the Thermoexcel-E tube than for the GEWA-T tube when the heat flux is increased. These different behaviors show that it is very important to know fully the nucleate regime, from the onset boiling until the dryout heat flux, for a particular combination of working fluid and surface, in order to optimize the operation of a thermal system. For the FC-72, these authors present experimental results showing the deterioration of the heat transfer for heat fluxes close to the critical heat flux of 161 kW/m^2 calculated by Zuber's correlation (Carey, 1992) except for the GEWA-T tube where a heat flux of 188 kW/m^2 has been achieved with no rapid increase in the wall temperature.

Palm (1991, 1992) presents results in pool and forced flow boiling for five refrigerants and pressures between 1 and 8 bar on a single smooth copper tube and on a copper single smooth tube with perforated metal foil kept in place over the smooth tube by thin copper wires in order to create a very narrow gap between the smooth tube and the foil. The latter method of enhancing boiling is protected by US and European patents. Different pore diameters between 0.10 to 0.21 mm, rim heights of the pores between 0.04 to 0.09 mm and pore densities between 50 to 250 cm^{-2} were tested. The results with R-22 and R-134a are very close and the heat transfer is increased by 30 to 70 %, whereas with R-11 the heat transfer coefficient is increased by up to 300 %, compared with the single smooth tube. For tests under 5 bar the dryout heat flux is around 20 kW/m^2 for R-134a and around 35 kW/m^2 for R-22, Palm (1991). This author concludes that the optimal pore diameter is between 0.10 to 0.22 mm and the pore density is not important for heat fluxes less than 20 kW/m^2. The results under pressures of 2, 5 and 8 bar show an augmentation of the heat transfer coefficient with an increase of the pressure, however, for R-22 it seems that the dryout heat flux starts around 25 kW/m^2 at 8 bar and around to 32 kW/m^2 at 5 bar.

Hübner and Künstler (1997) showed for GEWA tubes with T or Y-shapes, in which the gaps at fin tops varied between 0.23 to 0.35, a clear enhancement of the heat transfer coefficient, considering the total surface area of the tubes, for propane pool boiling at 4.24 bar (p_r=0.1), when compared with a plain tube. The heat transfer enhancement trend reduces with the increasing of the heat flux and is negligible when the heat flux is near 10 kW/m^2.

Chang and You (1997) have also analyzed FC-87 and R-123 atmospheric pool boiling up to critical heat flux on the Turbo-B enhanced tube surface (Thome, 1992; Webb, 1990). The heat transfer coefficient in nucleate boiling was higher than that for the plain surface tube. In particular, for a heat flux of 100 kW/m^2 the wall superheating for FC-87 was 10 K and for R-123 was 6 K whereas for the plain tube the corresponding values were around 27 K. For this commercial enhanced surface, with an outer diameter of 17.2 mm, the experimental critical heat fluxes were 91%, for FC-87, and 46%, for R-123, higher than the respective Zuber's correlation predicted values (Carey, 1992).

Chien and Webb (1998a,b) analyzed the effect of the tunnel dimensions, pore diameter and pore pitch using R-11 and R-123 at 26.7 ^{o}C on horizontal structured tubes, similar to that shown in Figure 3a, for heat fluxes between

2 to 70 kW/m^2. These authors show that the fin pitch has a secondary effect on boiling performance whereas the pore size and the pore pitch are the primary factors influencing the boiling heat transfer performance. A comparison of two rectangular tunnels, one having a circular and the other a rectangular fin base with pore diameter, pore pitch and fin height equal to 0.23, 1.5 and 0.6 mm, respectively, shows that the heat transfer coefficient is 20 % higher in the case of a rectangular fin base than a circular one. In fact this enhancement effect is because in the case of a rectangular fin base the meniscus radius is less than that for the circular fin base, causing conditions in which the thin liquid film can vaporize more easily than in the case of a circular fin base. Figure (5) shows the main results of the analysis of Chien and Webb (1998b) concerning the effects of pore diameter and pitch pore. For a fixed pore diameter, d_p=0.23 mm, the heat transfer coefficient increases, whereas the dryout heat flux decreases with an increase in pore pitch. For a fixed pore pitch, P_p=1.5 mm, as shown by the solid curves, the dryout heat flux increases as pore diameter increases. For operation points at moderate heat fluxes, which is the case for a great number of applications, it is interesting to have higher heat transfer performance and high values of dryout, which based on the results presented in Fig. (5), leads to the choice of d_p=0.23 mm and P_p=1.5 mm as the optimum combination.

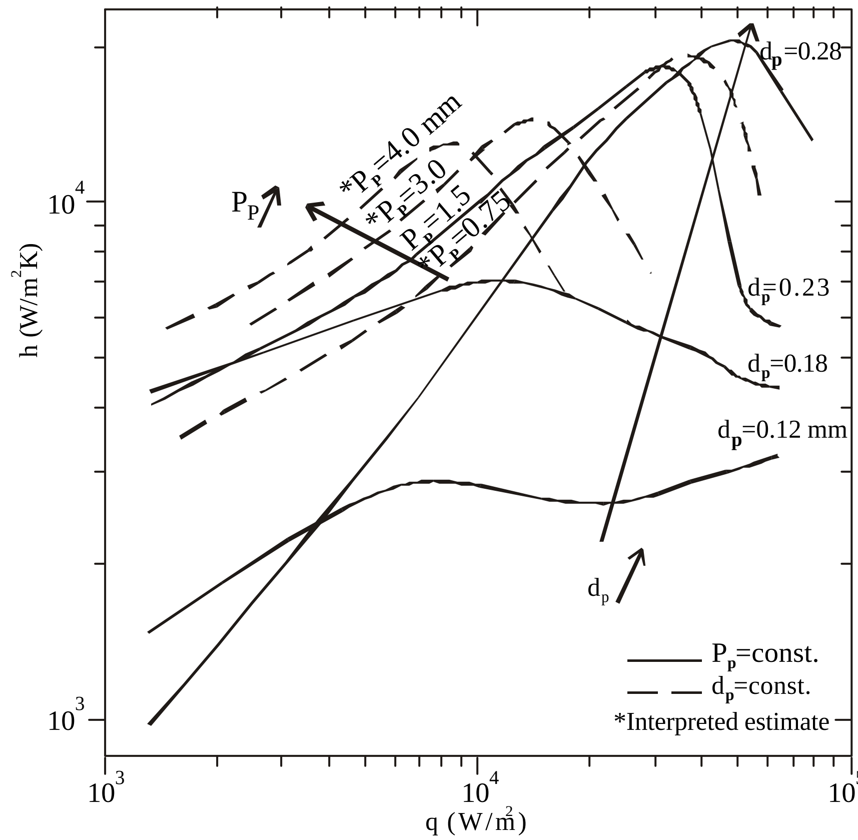


Figure 5. Combined effect of pore diameter (d_p) and pore pitch (P_p) (from Chien and Webb, 1998b)

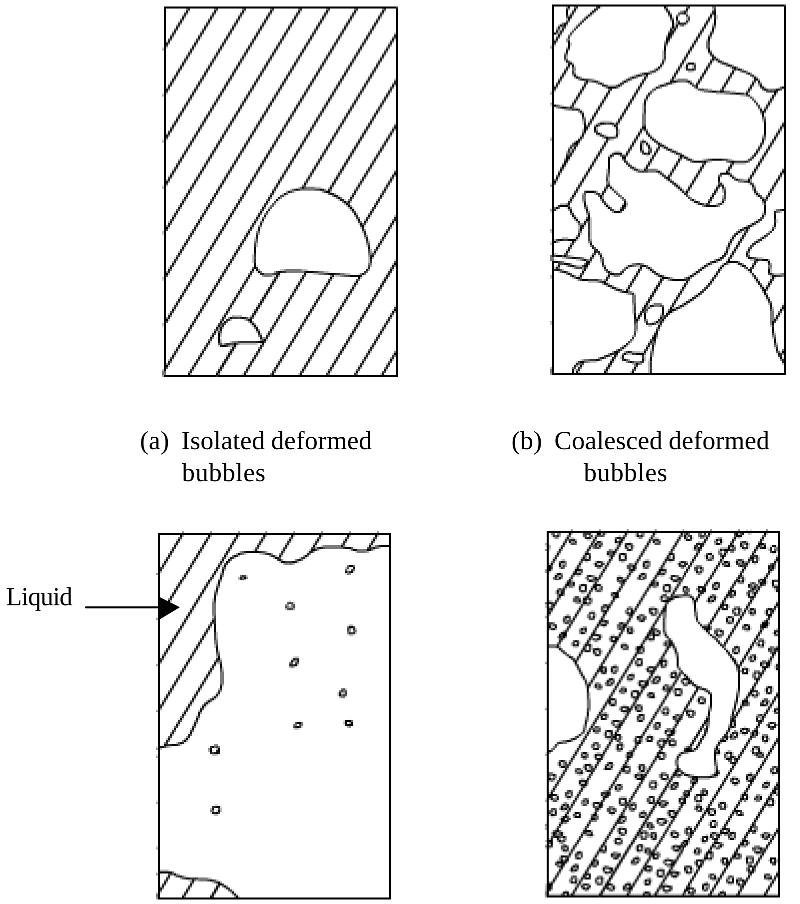
Kim and Choi (2001) analyzed the pool boiling heat transfer coefficient using R-11, R-123 and R-134a, in structured enhanced tubes having pores with connecting gaps, similar to that in Fig. 3c. Three geometries are considered in which the pore diameters are 0.20, 0.23 and 0.27 mm, the gaps are between 0.04 and 0.10 mm and the tunnel dimensions are W_t=0.25 mm and H_t=0.54 mm, see Fig. 3. The heat transfer coefficient for the enhanced surfaces, when the heat flux is 40 kW/m^2 is five to six times that of the smooth tube.

3. Results of visualization studies

The main mechanisms and configurations reviewed later in this section represent the principal sources of inspiration for the development of models in order to predict the boiling heat transfer mechanisms in confined and reentrant channels.

3.1. Confined boiling

In Fig. (6), the main configurations of pool boiling visualized by Yao and Chang (1983) inside vertical narrow annuli (heights of 25.4 and 76.2 mm and gap sizes of 0.32, 0.80 and 2.58 mm) with closed bottoms, are summarized. Figure (6a) shows isolated deformed bubbles with thin liquid film between the bubble and the wall for a low Bond number (Bond < 1, small gap) and low heat flux. Figure (6b) shows coalesced and deformed bubbles caused by the increase in heat flux. For a supplementary increase in heat flux, arriving close to the dryout value, since the deformed bubbles cover the majority of the heated wall, rapid evaporation of the liquid film can lead to the dryout phenomenon and the heat transfer coefficient decreases progressively. Finally, in Fig. (6d), a large number of isolated bubbles and some deformed bubbles are presented. This diagram is representative of a large gap (e=2.58 mm), therefore a large Bond number, and at high heat flux.



(a) Isolated deformed bubbles

(b) Coalesced deformed bubbles

(c) Post-dryout liquid

(d) Bubble nucleated on solid surface

Figure 6. Diagrams of the two-phase configurations in the confined space between two vertical plates, one heated and the other unheated (from Yao and Chang, 1983)

A similar visualization study was carried out by Ishibashi and Nishikawa (1969) who have dassified two regimes in nucleate boiling, isolated bubbles and coalesced bubbles, depending on the fluid, the gap annulus and the heat flux.

3.2. Reentrant channel of structured surfaces

A visualization study of the phenomena in a tunnel, similar to the Fig. 3a, was carried out by Nakayama et al. (1980a) in a transparent test section consisting of a horizontal channel (height varying between 0.5 to 1.0 mm and width of 1 mm) on which the base was heated, with a cover plate with pores with diameters between 0.05 and 0.5 mm. The tests were performed in R-11 pool liquid, at atmospheric pressure. A new study has been conducted by Nakayama et al. (1982) with a micro-finned tube covered by a thin copper plate having rows of pores with dimensions of 0.05, 0.10 and 0.15 mm and pressures of 0.04 to 0.23 MPa. By means of an optical probe these authors analyzed the bubble behaviors and determined the distribution of bubble departure diameter and the frequency of bubble formation. Three two-phase configurations or mechanisms were considered by these authors and are summarized as follows:

a) dried-up; when the tunnel is filled with vapor and the vapor bubbles are forming and growing attached to the pores they cause an increase of the tunnel pressure in the order of the $4\sigma/d_p$, where σ and d_p represent the surface tension and the pore diameter, respectively;
b) suction-evaporative; it represents the highest heat transfer performance and corresponds to the configuration with a liquid film covering the tunnel wall whereas the tunnel is quasi-filled by vapor; and the bubbles growing in active pores create a suction effect of the liquid from the outside into the tunnel space through the inactive pores, allowing the formation of menisci on the corners of the channel;
c) flooded; when most of the tunnel space is occupied by liquid and an active pore operates as an isolated nucleation site.

The experimental results concerning the limits of the nucleate regime for the structured surfaces show that the mechanisms of dryout are the cause of the reduction in the heat transfer coefficient when the heat flux is high. This limit condition can be attained when the liquid film of case b, above, is vaporized.

The flooded configuration is characteristic of the high pore diameter pores and in the study of Nakayama et al. (1980a) for d_p=0.5 mm.

Thome (1990) presents pictures and diagrams representative of the visualization study carried out by Arshad and Thome (1983) in horizontal reentrant channels formed by grooves (triangular, rectangular and circular with 1 mm height or diameter) on the top of a brass block covered by a very thin copper plate with pores of 0.15 and 0.25 mm diameters. Their visualizations span the nucleation and growing of a bubble up to the dry-out of the wall. Before the dry-out a thin liquid film covers the channel walls taking their shape. The main results are: the nucleation process inside the channel is not controlled by the pore diameters of the covering plates; and the flooded phenomenon is very dependent on the pore diameters and the channel geometry.

The visualization of R-22 pool boiling carried out by Gorenflo's group (Gorenflo, 2001; Gorenflo et al. (1998), on the outside surface of horizontal GEWA-T tubes has been presented in Thome (1990). The pictures for a test at 5 bar and heat flux of 52 kW/m^2, with a gap opening of 0.45 mm, show isolated and coalesced bubbles inside the channel. Increasing the pressure to 32.5 bar, with the same heat flux, but with a gap opening of 0.35 mm, the bubbles are more numerous and the flow inside the channel appears to be a slug flow regime, without evidence of a thin liquid film, Thome (1990).

4. Discussion

In order to discuss some of the main results presented above the values of the capillary length, $\left\{\sigma\left[g\left(\rho_l - \rho_v\right)\right]^{-1}\right\}^{0.5}$, for the majority of fluids and test conditions considered above are presented in Tab. (2). These values show that the capillary lengths are, in general, of the same magnitude or less than the main dimensions of the tunnels (H_t or W_t) of the structured surfaces discussed earlier, in section 2.3, characterizing Bond numbers less than one and, consequently favorable conditions to develop coalesced deformed bubbles, as schematized in Figure 6, and, for relatively higher heat fluxes, the dryout phenomenon.

Table 2. Capillary length for different fluids

Fluid	Surface tension, σ (J/m^2)	T_{sat} (^{o}C) (p (bar))	Capillary length $\left\{\sigma\left[g\left(\rho_l - \rho_v\right)\right]^{-1}\right\}^{0.5}$ (mm)
FC-87	0.0089	29.1 (1)	0.8
FC-72	0.0100	56 (1)	0.8
R-11	0.0183	23.4 (1)	1.1
R-22	0.0142	0.14 (5)	1.1
R-113	0.0190	47 (1)	1.1
R-123	0.0148	27.8 (1)	1.0
R-134a	0.0095	15.6 (5)	0.9
Water	0.0588	100 (1)	2.5

For R-22 and R-134a, used as the working fluid in Palm's work, with pressures between 2 to 8 bar the capillary length was approximately 1 mm, and considering the height of the gap created between the smooth tube surface and the perforated foil equal to the rim height between 0.04 to 0.09 mm the corresponding Bond number is less than one and the boiling can be considered confined. The heat transfer coefficients for these fluids are very close and qualitatively coherent with the trends observed in the studies of Ishibashi and Nishikawa (1969), Katto et al. (1977) and Yao and Chang (1983), presented in section 2.1 of the present work. However, the differences in the dryout heat fluxes obtained by Palm (1991), for R-134a and R-22, cannot be interpreted as a simple effect of the confinement.

For the structured surfaces with tunnels, the confined boiling conditions should be considered and can explain in some cases the enhancement effect on the heat transfer coefficient. However, the evaporation of menisci in the corner of the tunnel base, can play an important role on the increase in the heat transfer and has been analyzed in the works of Chien and Webb (1998c) and Jiang et al. (2001), for example. Due to space restrictions, an analysis of the existing models for structured surfaces is not presented in this paper.

5. Conclusions

We have presented a review of recent results, particularly those for enhanced structured surfaces. This synthesis considers confined boiling and different regimes, including the coalesced boiling regime that enhances the boiling coefficient and reduces the dryout heat fluxes. For the micro finned surfaces we present results showing the enhancement of boiling resulting from the machining process and an enhancement even higher than the simple augmentation of the surface area. For the majority of structured surfaces the main dimensions are close to or less than a millimeter and are of the same order of magnitude or less than the capillary length for a corresponding fluid.

6. References

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