CRITICAL HEAT FLUX, POST-DRYOUT AND THEIR AUGMENTATION

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Riassunto

Viene presentato uno stato dell'arte sulla crisi termica e sullo scambio termico post-crisi, che compendia, a livello di manualistica, una tradizione di ricerca presso i laboratori del CR. Casaccia.

I quattro capitoli in cui è suddiviso il presente lavoro riguardano più specificamente: a) la crisi termica in ebollizione sottoraffreddata; b) la crisi termica in ebollizione satura; c) lo scambio termico dopo la crisi termica; d) l’incremento del flusso termico critico e dello scambio termico post-crisi.

I primi due capitoli, per l’importanza dell’argomento, sono i più estesi e sono strutturati in maniera simile, fornendo, dopo una breve introduzione, informazioni sugli andamenti parametrici, ossia sull’influenza delle condizioni termoidrauliche e geometriche sulla crisi termica. Di seguito, vengono fornite in dettaglio le correlazioni più accreditate per il calcolo del flusso termico critico, sia in termini di affidabilità che di semplicità d’uso. Infine, vengono presentati i vari approcci teorici per la descrizione modellistica della crisi termica, riportando in dettaglio alcuni esempi rilevanti.

Il terzo capitolo riporta correlazioni e modelli teorici per la predizione dello scambio termico post-crisi.

Il quarto capitolo infine, descrive le varie tecniche disponibili per l’incremento del flusso termico critico e dello scambio termico in ultracrisi. Vengono presentate in dettaglio alcune tecniche passive ed alcune correlazioni applicabili per il calcolo delle condizioni di scambio termico.

Il lavoro di review compendia ricerche svolte presso l’Istituto di Termofluidodinamica del CR. Casaccia ENEA, con un’attenta lettura della letteratura più recente.

Abstract

The present work reports on the state-of-the-art review on the critical heat flux and the post-dryout heat transfer.

The four chapters of the report refer specifically to: a) critical heat flux in subcooled flow boiling; b) critical heat flux in saturated flow boiling; c) post-dryout heat transfer; and d) enhancement of critical heat flux and post-dryout heat transfer.

The first two chapters are somewhat tutorial, and are featured in a similar way. They provide, after a brief introduction, with information on parametric trends, i.e. on the influence of the thermal-hydraulic and geometric parameters on the thermal crisis. After that, the most widely used correlations are described in detail, either in terms of reliability and simplicity of use. Eventually, the various approaches for a modelling of the critical heat flux are reported.

The third chapter describes correlations and models available for the prediction of the post-dryout heat transfer, trying also to highlight the main drawbacks.

Finally, the fourth chapter describes the passive techniques for the enhancement of the critical heat flux and the post-dryout heat transfer, together with available correlations.

The present work is a merge of original researches carried out at the Institute of Thermal Fluid Dynamic of ENEA and a thorough review of the recent literature.

KEYWORDS: CRITICAL HEAT FLUX, SUBCOOLED FLOW BOILING, SATURATED FLOW BOILING, POST CRITICAL HEAT FLUX, AUGMENTATION OF CRITICAL HEAT FLUX, AUGMENTATION OF POST CRITICAL HEAT FLUX
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Introduction

The term critical heat flux (CHF) indicates an abrupt worsening of the heat transfer between a heating wall and a coolant fluid, generally with undesired consequences. This is typically due to the presence on the heated wall of a vapour layer which strongly reduces the heat transfer rate from the heater to the coolant.

In systems where the heat transfer is temperature-controlled (i.e., when a variation in the coolant thermal-hydraulic conditions implies only a variation in the heat flux, and not in the wall temperature) the sudden decrease of the heat transfer coefficient leads to a reduction in the performance of the heat exchanger and may cause chemical consequences for the wall (fouling, etc.) or safety consequences for the plant. This is immediately clear once we consider eq. (1):

$$q = \alpha (T_w - T_l)$$

As the wall-to-fluid temperature difference is imposed, a reduction in the heat transfer coefficient $\alpha$ will cause a decrease in the heat flux $\dot{q}$. A typical temperature controlled system is that where the wall is heated by a condensing fluid on one side and cooled on the other side.

In systems with imposed heat flux (i.e., when a variation in the coolant thermal-hydraulic conditions implies only a variation in the wall temperature and not in the heat flux) the sudden decrease in the heat transfer coefficient leads to a sharp increase in the wall temperature, as given by eq. (1). This latter may lead to the wall melting or its deterioration. A nuclear reactor core, an electrically heated rod or channel are typical heat flux controlled systems.

The term CHF, which is the limiting phenomenon in the design and operating conditions of water-cooled nuclear reactors as well as of much other thermal industrial equipment, will be used to represent the heat transfer deterioration above described, although different mechanisms of the thermal crisis might also suggest different names. Under subcooled or low-quality saturated flow boiling conditions, being nucleate boiling the main boiling mechanism, the onset of thermal crisis is following the departure from nucleate boiling (DNB), and this is often the name used in this case. Under high-quality saturated flow boiling conditions, typically characterized by the annular flow regime, the dryout of the liquid film adjacent to the heated wall is the leading mechanism to the thermal crisis, which is therefore named dryout.

In a heat flux-controlled situation (which will be the only one treated here), the rapid wall temperature rise may cause rupture or melting of the heating surface, which is termed as physical burnout. The burnout heat flux is generally different from the DNB or the dryout heat flux. Only in the case of extremely high heat fluxes under subcooled flow boiling conditions (expected to be faced in some components of the thermonuclear fusion reactor), the CHF is characterized by extremely high temperature differences. Failure of the heating wall is very often experienced and therefore the heat flux causing the DNB is practically identical with the physical burnout heat flux (Celata (1996)). This is absolutely not the case of situations where higher heat transfer coefficients and lower critical heat fluxes give rise to only reduced temperature excursions at the DNB or dryout (Hewitt (1978), Bergles et al. (1981), Hsu & Graham (1986), Weisman (1992), Collier & Thome (1994), Katto (1994)). If the temperature rise does not cause failure of the heating surface, a post-CHF heat transfer is thus possible, although the heat transfer rate will be much lower than that before the CHF occurring.

In the following sections, the CHF in subcooled and saturated flow boiling will be discussed, together with the post-CHF heat transfer. Finally, a section will deal with existing methods for CHF and post-CHF heat transfer augmentation.
1 CHF in Subcooled Flow Boiling

Simply speaking, forced convective subcooled boiling involves a locally boiling liquid, whose bulk temperature is below the saturation value, flowing over a surface exposed to a heat flux. Under such conditions the critical heat flux is always of the DNB type, resulting in a significant increase in the wall temperature, the larger the higher the heat flux.

Although relevant to the thermal-hydraulic design of Pressurized Water Reactor cores and therefore studied since the far past (Hewitt (1978), Bergles et al. (1981), Hsu & Graham (1986), Weisman (1992), Collier & Thome (1994), Katto (1994), Gambill (1968), Bergles (1977)), the CHF in subcooled flow boiling received a renewed attention in the recent past due to the possible use of water in subcooled flow boiling for the cooling of some components of the thermonuclear fusion reactor believed to be subjected to operating conditions characterized by extremely high thermal loads (Celata (1996), Boyd (1985a)). Hereafter the parametric trends experimentally observed, together with available correlations and theoretical models will be discussed.

1.1 Parametric Trends

The magnitude and the occurrence of the CHF are affected by many parameters such as thermal-hydraulic, geometric and external parameters. Among thermal-hydraulic parameters we have subcooling, mass flux, pressure, binary component fluids, while important geometry parameters are channel diameter, heated length, channel orientation, tube wall thickness and material. External parameters of interest are heat flux distribution and content of dissolved gas.

1.1.1 Influence of subcooling

As reported by Boyd (1985a), most of the early experimental studies reveal that the relationship between subcooling and CHF is almost linear, even though Bergles (1963) indicated that for very large subcooling at moderate to large liquid velocity (1 to 10 m/s) the relationship between CHF and subcooling is nearly linear, but becomes highly nonlinear as the subcooling decreases, showing a minimum at small positive subcooling. Recent experiments under conditions of high liquid subcooling confirmed the almost linear relationship between CHF and subcooling (Celata et al. (1993a), Nariai et al. (1987), Vandervort et al. (1992)). Figure 1-1 shows the CHF versus inlet subcooling for data carried out by Celata et al. (1994b) in 2.5 mm I.D. stainless steel tubes, 0.25 mm wall thickness, 10 cm long, uniformly heated by Joule effect, with vertical upflow of water. The functional dependence of the CHF on the subcooling is practically linear, up to very high subcooling and very high liquid velocity. The CHF versus \( \Delta T_{sub,in} \) curves, plotted at different liquid velocities, result parallel among each other, and no inter-relation between \( u \) and \( \Delta T_{sub,in} \) would seem to exist.

1.1.2 Influence of mass flux

The CHF is an increasing function of the mass flux (or fluid velocity) with a less than a linear fashion. This was observed up to very high values of mass flux (90 Mg/m\(^2\)s). Figure 1-2 shows the results of experiments carried out by Boyd (1988, 1989, 1990) using water as a fluid in horizontal test sections of amzirc (copper-zirconium alloy) with an inner diameter of 3.0 mm, wall thickness around 0.5 mm, and a heated length of 0.29 m (Boyd (1988, 1989)), or 10.2 mm I.D., 0.125 mm wall thickness, 0.5 m long, and copper as a material (Boyd (1990)). Tests were performed at a constant inlet temperature of 20 °C. Similar results were obtained by Celata et al. (1993a).
Fig. 1-1  CHF versus inlet subcooling, Celata et al. (1993a)

Fig. 1-2  CHF versus mass flux, Boyd (1988, 1989, 1990)
1.1.3 Influence of pressure

Recent experiments (Celata et al. (1993a, 1994b), Nariai et al. (1992), Vandervort et al. (1992)) showed that in the range 0.1-5.0 MPa, direct influence of the pressure on the CHF is weak, other conditions being equal (i.e., for same subcooling and liquid velocity). This is demonstrated in Fig. 1-3a, where the CHF is plotted versus exit pressure $p$, for Vandervort et al. data (1992), obtained with stainless steel tubes of 1.07 mm I.D., 26.75 mm long. Virtually, no pressure effect was noted; in fact, there seemed to be a very slight decrease of the CHF with increasing pressure. Figure 1-3b shows the results of Celata et al. (1994b) obtained with stainless steel tubes of 8.0 mm I.D., 10 cm long, with uniform heating. The CHF versus subcooling data lie on a unique curve independent of the pressure, evidencing the negligible effect of this parameter.

Boyd (1985a) reported how other researchers found a maximum in the CHF versus pressure trend in the vicinity of a reduced pressure of 0.75, being this value somewhat variable with the mass velocity.

1.1.4 Binary component fluids

Tolubinsky & Matorin (1973) used ethanol-water, aceton-water, ethanol-benzene, ethylene-glycol-water with a 4 mm I.D., 60 mm long tube; Andrews et al. (1968) tested acetone-tuolene and benzene-tuolene with an annulus 6.35 mm I.D. 20.9 mm O.D. 76 mm long; Sterman et al. (1968) used mono-iso-propyldiphenyl-benzene with an annulus 10 mm I.D. 16 mm O.D. 110 mm long; Naboichenko et al. (1965) tested the same fluids as Sterman et al. (1968) using an annulus 6 mm I.D. 16 mm O.D. 80 mm long; Carne (1963) used acetone-tuolene and benzene-tuolene with an annulus 6.35 mm I.D. 19.05 mm O.D. 76.2 mm long; and finally Bergles & Scarola (1966) tested water-1-pentanol using a 6.26 mm I.D. 170 mm long tube. Typical trends of CHF are shown in Fig. 1-4; the CHF tends to reach a maximum value increasing the mole fraction and increases with subcooling and velocity. The maximum corresponds to the maximum difference between the vapour and liquid composition of the more volatile component ($y-x$). As the difference between the more volatile component concentration in the vapour and the liquid phase increases (in absolute value), a reduction occurs in the vapour bubble departure diameter, in the bubble rate of growth, and in the number of active nucleation sites. This results in a reduction of the vapour content of the wall layer of the boiling fluid and, therefore gives rise to an increase in the CHF (Tolubinsky & Matorin (1973)).

1.1.5 Influence of channel diameter

Works to identify the dependence of the CHF on the channel diameter have been conducted up to the recent past (Vandervort et al. (1992), Bergles (1963), Kramer (1976), Celata e al. (1993c), Nariai & Inasaka (1992)). It is well established that the CHF is inversely related to the channel diameter. Figure 1-5 shows the CHF versus the channel diameter $D$, for Vandervort et al. data (1992). As observed by previous researchers, for given values of exit thermal hydraulic conditions, heated length and liquid velocity, the CHF increases with the decrease of the tube inside diameter, but the effect was less significant for decreased mass flux. A threshold is observed beyond which the effect of the tube inside diameter may be considered negligible, that is a function of the channel geometry and thermal hydraulic conditions. To explain the observed dependence of the CHF on the tube inside diameter it is worth reporting here three different reasons proposed by Bergles (1963). For a tube with a smaller inside diameter we have: (1) a small bubble diameter, (2) an increased velocity of the bubbles with respect to the liquid, and (3) the fluid subcooled bulk closer to the growing bubbles (collapsing in the bulk). From the analysis of experimental data of void fraction in narrow tubes, Nariai & Inasaka (1992) concluded that, as tube inside diameter decreases and mass velocity increases, the diameter of generated bubbles or, better, the thickness of the two-phase boundary layer becomes smaller due to the intense condensation effect by subcooled water at core region, and the void fraction becomes smaller,
Fig. 1-3  CHF versus pressure, Vandervort et al. (1992) (left graph) and versus inlet subcooling, Celata et al. (1994b) (right graph).
making the CHF higher. The decrease in the diameter gives rise to an increase in the slope of the velocity profile in the two-phase boundary layer, making the detachment of growing bubbles and the consequent condensation in the core region easier. The higher the mass flux the most consistent the effect.

\[
\frac{(Y - X)}{\%}
\]

\[
\Delta T_{\text{sub}} = 70 \text{ K}
\]

\[
\text{azeotrop composition}
\]

\[
\begin{align*}
u &= 5 \text{ m/s} \\
L/D &= 15 \\
p &= 0.33 - 1.32 \text{ MPa}
\end{align*}
\]

**Fig. 1-4**  CHF versus mixture composition for forced convection boiling of benzene/ethanol mixtures, Tolubinsky & Matorin (1973). In the top figure, \(Y-X\) represents the difference between the composition of the vapour phase, \(Y\), and the liquid phase, \(X\), for the more volatile component.

1.1.6 Influence of channel heated length

The heated length of the channel seems to be inversely related to the CHF. Generally, investigators use the ratio of the heated length to the inside (or equivalent) diameter of the channel \(L/D\), as the characteristic non-dimensional length, but this still needs to be established. Recent experiments were carried by Nariai et al. (1987) and by Vandervort et al. (1992). Figure 1-6 reports the results of Nariai et al. (1987) showing the CHF versus \(L/D\). The CHF increases as \(L/D\) decreases, and the
effect is more significant for smaller channel diameter. As the effect seems to be greatest for \( \frac{L}{D} < 20 \) (depending on the diameter), this would indicate that the CHF is related to the state of development of the bubble-boundary layer. Vandervort et al. (1992) verified that the functional dependence between CHF and \( \frac{L}{D} \) is independent of mass flux. As for the case of the channel diameter, experiments showed the presence of a threshold beyond which the CHF is practically independent of \( \frac{L}{D} \) and this limit (between 20 and 40) is related to flow parameter since \( \frac{L}{D} \) is related directly to the flow development.

1.1.7 Influence of channel orientation

The effect of flow orientation (e.g., horizontal versus vertical upflow) may be significant if the buoyant force is a nonnegligible percentage of the axial inertial force in flow boiling. Quantitatively, this can be evaluated by considering the modified Froude number \( Fr \), defined as:

\[
Fr = \frac{\frac{m \cos \phi}{\rho_l \left[ gD \left( \frac{\rho_l - \rho_g}{\rho_l} \right) \right]^{1/2}}}{Fr_{\phi = 0}}
\] (1-1)

where \( \phi = 0 \) represents the horizontal case. For modified Froude number greater than 5-7, effects of stratification and orientation may disappear. Wherever flow orientation plays a relevant role, the CHF for horizontal flow is always less than the value for vertical flow (Merilo (1977), Cumo et al. (1978)). Recent experiments carried out by Celata et al. (1994b) using water under conditions relevant to the NET/ITER divertor \((p \text{ around } 3.5 \text{ MPa})\) showed that for a liquid velocity greater than 5.0 m/s, horizontal and vertical data do not show any remarkable difference (at 5.0 m/s the modified Froude number is greater than 20).

1.1.8 Influence of tube wall thickness and material

Celata et al. (1997) tested a number of SS 304 tubes having almost the same inner diameter but different wall thicknesses (from 0.25 to 1.75 mm) and found a slight effect of the tube wall thickness on the CHF: a slight decrease in the CHF as the wall thickness increased was observed, but within 20% passing from the smallest to the largest thickness. Vandervort et al. (1992) used five different materials in their experiments, such as SS 304, SS 316, nickel 200, brass 70/30 and inconel 600, and, under very similar geometric and thermal-hydraulic conditions, did not observe any significant effect of the tube material on the CHF.

1.1.9 Influence of heat flux distribution

The optimum axial heat flux distribution for subcooled flow boiling is one where the peak heat flux occurs near the inlet (Boyd (1985a)). Groeneveld (1981) notes that a short pulse spike has a significant effect on subcooled flow boiling CHF, finding a CHF increase but a critical power decrease. Doroschuk et al. (1978) found that the CHF was lower for cosine distribution than for uniform ones. Ad hoc experiments were recently performed by Nariai et al. (1992) and by Gaspari (1993) to investigate the effect of the circumferential heat flux distribution on the CHF. In particular, Gaspari made a comparison between peripherally full and half-heated tubes, straight flow, analysing the CHF at both inlet and exit thermal hydraulic conditions. Using a 10 mm I.D. channel, 0.15 m long, Gaspari observed that, under constant inlet liquid subcooling, higher CHF values were observed for half-heated tubes. Plotting the CHF versus exit liquid subcooling such a difference tends to disappear, as reported in Fig. 1-7, where the CHF is plotted versus inlet/outlet subcooling.
Fig. 1-5  CHF versus channel diameter, Vandervort et al. (1992)

Fig. 1-6  CHF versus channel heated length, Nariai et al. (1987).
1.1.10 Influence of dissolved gas

On the basis of previous literature it is reasonable to conclude that dissolved gas has no effect on CHF. However, for experiments carried out with small diameter tubes, the bubble boundary layer may be smaller, and it is conceivable that even small amounts of dissolved air coming out of solution could affect the CHF. Specific tests were performed by Vandervort et al. (1992) using 1.07 mm I.D. channels, at a mass flux of 25 Mg/m$^2$s, an exit pressure of 0.6 MPa and an exit subcooling of 100 K. No significant change was observed in the CHF results over the range of dissolved gas concentration in water from near zero (2 ppm) up to the saturation level (~ 9.5 ppm).

1.2 Available Correlations for the Prediction of Subcooled Flow Boiling CHF

Many different types of correlational approaches have been proposed. These include empirical, dimensional analysis or similitude-based, analytical, tabular, and graphical, being the first two categories the most widely used. A thorough review of them has been given by Boyd (1985b), listing as many as 38 correlations. We just report here the most widely used, also on the basis of their possible extrapolation to conditions different from the originating ones, (Celata et al. (1994a), Inasaka & Nariai (1996)) although this must be done with great care.

- Gunther (1951)

\[
q_{CHF} = 71987u^{0.5} \Delta T_{sub, ex}\]

(1-2)

(recommended ranges: $p = 0.1 - 1.1$ MPa; $u = 1.5 - 12.1$ m/s; $CHF = 0.4 - 11.4$ MW/m$^2$; $\Delta T_{sub} = 11-139$ K)
\[ q_{CHF} = (0.23 \times 10^6 + 0.094 m) (3 + 0.018 \Delta T_{sub}) [0.435 + 1.23 \exp (- 0.0093 L/D)] \times \left( 1.7 - 1.4 \exp \left[ - 0.532 \left( \frac{h_{l} - h_{lg}}{h_{lg}} \right)^{3/4} \left( \frac{\rho_g}{\rho_l} \right)^{-1/3} \right] \right) \]  
\[ (1-3) \]

(recommended ranges: \( p = 5.5 - 19.0 \) MPa; \( u = 0.3 - 12.1 \) m/s; \( CHF = 0.4 - 4 \) MW/m\(^2\); \( L/D = 21 - 365 \); \( \Delta T_{sub} = 0 - 126.7 \) K)

Tong-75, Tong (1975)

\[ q_{CHF} = 0.23 \bar{m} h_{lg} (1 + 0.0216 (\frac{p_{ex}}{p_c})^{0.8} Re^{0.5} Ja) \]  
\[ (1-4) \]

where

\[ f = 8 (D/D_o)^{0.32} Re^{0.6}, \quad Ja = \frac{c_p (T_{b} - T_{sub}) \rho_l}{h_{lg}} \frac{\rho_l}{\rho_g} \quad Re = \frac{\bar{m} D}{\eta_l (1 - \varepsilon)} \]

with \( D_o = 1.27 \times 10^{-2} \) m being \( \varepsilon \) (void fraction) evaluated using the Thom's correlation (Collier & Thome (1994), Thom et al. (1965)) (recommended ranges: \( p = 6.8 - 13.6 \) MPa; \( u = 0.68 - 5.9 \) m/s; void fraction at \( CHF < 0.35 \); \( D = 3 - 10 \) mm; \( L/D = 5 - 100 \)).

Tong-68, Tong (1968)

\[ \bar{q}_{CHF} = C \bar{m} h_{lg} \]  
\[ (1-5) \]

with \( C = 1.76 - 7.433 x_{ex} + 12.222 x_{ex}^2 \)

(recommended ranges: \( p > 7 \) MPa). The Tong-68 correlation can also be written as:

\[ Bo = \frac{C}{Re^{0.6}} \]

where \( Bo \) and \( Re \) are Boiling number and Reynolds number, respectively.

A modification of the Tong-68 correlation for pressure lower than 7.0 MPa has been proposed by Celata et al. (1994a):

\[ Bo = \frac{C'}{Re^{0.5}} \]  
\[ (1-6) \]

where:

\[ C' = (0.216 + 4.74 \times 10^{-2} p) \psi \quad (p \text{ in MPa}) \]
\[ \psi = 1 \quad \text{if} \quad x_{ex} < 0.1 \]
\[ \psi = 0.825 + 0.986 x_{out} \quad \text{if} \quad 0 > x_{ex} \geq 0.1 \]

(recommended ranges: \( p \leq 5.5 \) MPa; \( u = 2.2 - 40 \) m/s; \( \Delta T_{sub,ex} = 15 - 190 \) K; \( D = 0.3 - 15 \) mm).

In using the above reported correlations, two methods are generally followed: the so-called heat balance method, HBM, which requires an iterative procedure, and the so-called direct substitution method, DSM (Inasaka & Nariai (1986), Groeneveld et al. (1986)). The two methods lead to
different results and their use has been deeply debated in the recent past, with the possible conclusion that the HBM would give better results and should be therefore preferred, Theofanous (1996).

For binary mixtures (Collier & Thome (1994), Celata & Cumo (1996)) the CHF in subcooled flow boiling may be expressed as the sum of two terms: the first term \( \dot{q}_{\text{CHF},i} \) is the ideal value evaluated from the CHF value of the two components at the same pressure, velocity and subcooling (linear combination), and the second term \( \dot{q}_{\text{CHF,E}} \) is an additional CHF connected to the increasing in the CHF due to mass transfer effects. Thus, the final expression is given by:

\[
\dot{q}_{\text{CHF}} = \dot{q}_{\text{CHF},i} + \dot{q}_{\text{CHF,E}} = \dot{q}_{\text{CHF},i} (1 + C_{II})
\]  

(1.7)

where

\[
\dot{q}_{\text{CHF},i} = \left[ x \dot{q}_{\text{CHF},1} + (1-x) \dot{q}_{\text{CHF},2} \right]
\]

being 1 and 2 referred to the more and the less volatile component, respectively, and \( C_{II} \) the mole fraction of the more volatile component in the liquid phase.

Sterman et al. (1968) verified in their experiments that \( C_{II} \) varies between 0 and 0.8 and proposed an expression for \( C_{II} \), as:

\[
C_{II} = A \frac{|C_{2l-x}|^3}{Re_2} + B \frac{|C_{2l-x}|^{1.5}}{Re_2^{0.4}} \left[ \frac{T_{\text{sat},m} - T_{\text{sat},l}}{T_{\text{sat},l}} \right]
\]  

(1.8)

with

\[
A = 3.2 \times 10^5 \; \text{;} \quad B = 6.9
\]

being \( C_{2l} \) the mole fraction of the more volatile component in the vapour phase. This correlation was proved valid also for refrigerant mixture, Celata et al. (1994d).

Tolubinsky & Matorin (1973) gave the following expression of \( C_{II} \), as:

\[
C_{II} = 1.5 |C_{2l-x}|^{1.8} + 6.8 |C_{2l-x}| \left[ \frac{T_{\text{sat},m} - T_{\text{sat},l}}{T_{\text{sat},l}} \right]
\]  

(1.9)

Equation (1.9) is applicable to ethanol-water, acetone-water, ethanol-benzene and ethylene-glycol-water mixtures with a ± 20% error.

1.3 Available Models for the Prediction of Subcooled Flow Boiling CHF

As is known, correlations have the drawback to be not reliable outside the recommended ranges of application. In this respect, models may have the advantage to characterize not only the existing and developing data base, but also to predict CHF beyond the established data base. Recent reviews about CHF modelling were given by Katto (1994, 1995), Weisman (1992) and Celata (1997).

Major theoretical approaches to CHF can be categorized into five groups, according to the basic mechanism assumed by relative authors to be the main cause of the CHF occurrence.

(1) Liquid layer superheat limit model. The difficulty of heat transport through the bubbly layer causes a critical superheat in the liquid layer adjacent to the wall, giving rise to the occurrence of the CHF, Tong et al. (1965).
(2) Boundary layer separation model. This model is based on the assumption that an injection of vapour from the heated wall into the liquid stream causes a reduction in the velocity gradient close to the wall. Once the vapour effusion increases beyond a critical value, the consequent flow stagnation is assumed to originate the CHF (Kutateladze & Leontiev (1966), Tong (1966, 1975), Purcupile & Gouse (1972), Hancox & Nicoll (1973), Thorgerson et al. (1974)). The weak physical basis of the model has been demonstrated by the studies above reported (Fiori & Bergles (1970), van der Molen & Galjee (1978), Hino & Ueda (1985), Mattson et al. (1973)).

(3) Liquid flow blockage model. It is assumed that the CHF occurs when the liquid flow normal to the wall is blocked by the vapour flow. Bergelson (1980) considers a critical velocity raised by the instability of the vapour-liquid interface, while Smogalev (1981) considers the effect of the kinetic energy of vapour flow overcoming that of the counter motion of liquid.

(4) Vapour removal limit and near-wall bubble crowding model. It is assumed that the turbulent interchange between the bubbly layer and the bulk of the liquid may be the limiting mechanism leading to the CHF occurrence. The CHF occurs when bubble crowding near the heated wall prevents the bulk cold liquid from reaching the wall (Hebel et al. (1981), Weisman & Pei (1983), and Weisman & Ying (1983) postulate that the CHF occurs when the void fraction in the bubbly layer is calculated under the assumption of homogeneous two-phase flow in the bubbly layer in Weisman & Pei (1983), and using the slip model in Weisman & Ying (1983), just exceeds the critical value of 0.82. The void fraction in the bubbly layer is determined through the balance between the outward flow of vapour bubbles and the inward liquid flow at the bubbly layer-bulk liquid flow interface. Weisman & Ileslamolou model (1988) is an improvement of Weisman & Pei model, for subcooled exit conditions. A research work carried out by Styrikovich et al. (1970), showed that measured void fraction at the CHF ranges from as low as 0.3 to as high as 0.95, making the validity of the near-wall bubble crowding models questionable. In addition, the models are quite empirical in the determination of the turbulent exchange in the bubbly layer.

(5) Liquid sublayer dryout model. The model is based on the dryout of a thin liquid sublayer underneath a vapour blanket or elongated bubble, due to coalescent bubbles, flowing over the wall. (Lee & Mudawar (1988), Katto (1990), Celata et al. (1994c)).

At present, the liquid sublayer dryout theory is being received significant attention, is well developed, and is able to provide good predictions over a wide range of conditions. Lee & Mudawar (1988) are the first in developing and proposing a mechanistic model based on the liquid sublayer dryout theory, which was assessed for data at a pressure above 5.0 MPa. Following the same principles as Lee & Mudawar, Katto (1990a, 1990b) developed a generalised CHF model applicable to not only water but also non aqueous fluids (water, nitrogen, helium, R 11, R 12, and R 113). Then Katto extended his model so as to cover the CHF of water boiling at low pressure also, (Katto (1992)). Lee and Mudawar, and Katto models make use of empirical constants determined through the experimental data. This limits somehow the use of these models within the data base on which they are assessed. Further, the Katto model is applicable only to those cases where the local void fraction at the CHF in the near-wall bubbly layer is lower than 0.7. The most recent model developed in the frame of the liquid sublayer dryout theory was proposed by Celata et al. (1994c),
without making use of any empirical constant, yet being capable of predicting the CHF of water boiling in a wide range of conditions for the subcooled flow boiling, Celata et al. (1995b).

Briefly, to describe the Celata et al. model, let us consider the situation at the tube exit (locus of the CHF for axial uniform heating) approaching the CHF, that may be presumably that sketched in Fig. 1-8 (Celata et al. (1995a)): a thin vapour clot or blanket forms in the vicinity of the heated wall due to small bubbles coalescence, holding a liquid sublayer between the vapour clot and the wall surface. The occurrence of the CHF is determined by the evaporation of the liquid sublayer during the passage time of the blanket which insulates the liquid sublayer between the heating surface and the bulk of the liquid:

\[ \dot{q}_{CHF} = \frac{\delta \rho_l h_{fg}}{L_B} u_B \tag{1-10} \]

where \( \delta \) is the initial liquid layer thickness, \( \rho_l \) is the liquid layer density, \( L_B \) and \( u_B \) are the blanket length and velocity, respectively. The vapour blanket length, \( L_B \), is assumed to be given by the Helmholtz instability wavelength at the interface facing to the liquid sublayer. The vapour blanket velocity, \( u_B \), is evaluated considering the velocity distribution of the main stream in the tube under the assumption of homogeneous flow. The Celata et al. model considers the temperature distribution of the main stream in the tube under the assumption of homogeneous flow, determining the thickness \( s^* \) of the superheated layer (distance from the heated wall at which the liquid temperature is equal to the saturation value), beyond which vapour blanket cannot develop or exist due to subcooled conditions. Vapour blanket can develop and exist only in the near-wall region where the local liquid temperature is above the saturation value.

As the temperature distribution is linked to the inside tube wall temperature, this latter is obtained by equating the local cross-section average fluid temperature given by the coolant heat balance with that provided by the temperature profile. Then \( \delta \) can be determined as the difference between the superheated layer \( s^* \) (where the vapour clot can exist only, and as close as possible to the saturation line) and the vapour blanket thickness, \( D_B \). This latter is calculated from the Staub model (1968), under the assumption (common with the Lee & Mudawar model) that the circumferential growth of a vapour blanket is strongly limited by adjacent blankets and by the steep velocity gradient in case of high liquid velocity. It is therefore assumed that the equivalent diameter of each blanket (i.e., its thickness) may be approximated by the diameter of a bubble at the departure from the wall. In other words, it is assumed that departing bubbles may coalesce into a distorted blanket that stretches along the fluid flow direction (due to vapour generation by sublayer evaporation) and keeps almost a constant equivalent diameter (thickness). Equations used in the mathematical description of the Celata et al. model are reported in the Appendix. A comparison between Katto and Celata et al. models is reported in Fig. 1-9, for the data set published in Celata & Mariani (1993) (about 1900 data). The figure reports the percentage of data point calculated with a given error band (%). The Celata et al. model, unlike the other liquid sublayer dryout models, can be also used for peripheral non uniform heating simply by considering the total thermal power delivered to the fluid in the coolant heat balance for the calculation of the local average coolant temperature (Celata et al. (1995b)).
Fig. 1-8  Schematic of the liquid sublayer dryout theory, Celata et al. (1994c)
Fig. 1-9 Comparison between Katto (1990) and Celata et al. (1994c) models for the prediction of water subcooled flow boiling CHF

APPENDIX. CHF Calculation Procedure in the Celata et al. Model (1994c)

Input parameters $\dot{m}$, $p_{ex}$, $D$, $L$, $T_{in}$. Assume a value of $q_1$. Necessary physical properties are: $c_p$, $\lambda_i$, $\eta_i$, $h_i$, $p_i$, $p_g$. Where not specified, physical properties are calculated at the saturated state at $p_{ex}$.

\[
T_{in} + \frac{qA}{M c_{pl}} = \frac{5}{s^+(R)} T_{m1} + \frac{25}{s^+(R)} T_{m2} + \frac{s^+(R) - 30}{s^+(R)} T_{m3}
\]

where $c_{pl}$ is calculated at $(T_{m1} + T_{in})/2$ and $T_{m1}$, $T_{m2}$ and $T_{m3}$ are calculated from the temperature distributions:

\[
T_w - T = QPr \ s^+ \quad 0 \leq s^+ < 5
\]

\[
T_w - T = 5Q \left[ Pr + \ln \left( 1 + Pr \left( \frac{s^+}{5} - 1 \right) \right) \right] \quad 5 \leq s^+ < 30
\]
In the above temperature distribution equations, \( c_{pl} \) is calculated at saturated conditions at \( p_{ex} \), \( s^* \) is the non-dimensional distance from the wall, and \( u_T \) is the friction velocity. From the above calculation the wall temperature \( T_w \) is obtained. Using the above temperature distribution equations it is possible to calculate \( s^* \), that is the value of the distance from the heated wall, \( s \), at which the fluid temperature is equal to the saturation value at \( p_{ex} \). Calculation of \( D_B \):

\[
D_B = \frac{32}{f} \frac{\sigma f(\beta)}{m^2} \frac{1}{\sqrt{f}} = 1.14 - 2.0 \log \left( \frac{0.72 \sigma \rho_i}{f D m^2} + \frac{9.35}{\text{Re} \sqrt{f}} \right)
\]

where \( f(\beta) = 0.03 \). Calculation of \( \delta \) and \( C_D \)

\[
\delta = s^* - D_B
\]

\[
C_D = \frac{2}{3} \frac{D_B}{\left( \frac{\sigma}{g(\rho_i - \rho_g)} \right)^{0.5}}
\]

Calculation of \( u_B \) and \( L_B \) (linked each other) through an iterative procedure:

\[
u_B = \left( \frac{2 L_B \gamma (\rho_i - \rho_g)}{\rho_l C_D} \right)^{0.5} + 0.125 \left( \delta + \frac{D_B}{2} \right) \frac{f m^2}{\rho_l \eta_i}
\]

\[
u_B = \left( \frac{2 L_B \gamma (\rho_i - \rho_g)}{\rho_l C_D} \right)^{0.5} + 1.768 \sqrt{\frac{f}{\rho_l}} \frac{m}{\eta_i} \ln \left[ 0.354 \frac{m}{\eta_i} \sqrt{\frac{f}{\rho_l}} \left( \delta + \frac{D_B}{2} \right) \right] - 0.61
\]

\[
u_B = \left( \frac{2 L_B \gamma (\rho_i - \rho_g)}{\rho_l C_D} \right)^{0.5} + 0.884 \sqrt{\frac{f}{\rho_l}} \frac{m}{\eta_i} \ln \left[ 0.354 \frac{m}{\eta_i} \sqrt{\frac{f}{\rho_l}} \left( \delta + \frac{D_B}{2} \right) \right] + 2.2
\]

where \( L_B \) is given by

\[
L_B = \frac{2 \pi \sigma (\rho_g + \rho_i)}{\rho_g \rho_l u_B^2}
\]

Calculation of \( \dot{q}_2 \):

\[
\dot{q}_{CHF} = \frac{\rho_l \delta h_{lb}}{L_B} u_B
\]

The condition of critical heat flux, \( \dot{q}_{CHF} \), is reached when \( \dot{q}_1 = \dot{q}_2 \).
CHF in Saturated Flow Boiling

Forced convection saturated flow boiling involves a boiling liquid, whose average bulk temperature is at the saturation temperature, flowing over a surface exposed to a heat flux. The critical heat flux always occurs with a positive quality at the CHF. Generally speaking, under saturated conditions, we may have two different types of CHF: i) the DNB type, typically occurring at low quality conditions, and ii) the dryout type, which is encountered in high quality flow. Although the two different types of CHF are much different each other from the phenomenological point of view, this kind of classification is somewhat schematic, the threshold being very difficult to be established. As the quality at the CHF increases we gradually pass from DNB to dryout. An interesting simple method to identify a priori the CHF type has been recently given by Lombardi & Mazzola (1998).

Nonetheless, although DNB and dryout types of the CHF are associated with different mechanisms leading to the onset of thermal crisis, parametric trends of the CHF in saturated flow boiling may be more or less independent of the CHF mechanisms, and the general trends can be given for the CHF in saturated flow boiling.

2.1 Parametric Trends

The magnitude and the occurrence of the CHF are affected by many parameters such as thermal-hydraulic geometric and external parameters. Among thermal-hydraulic parameters we have subcooling, mass flux, pressure, while important geometry parameters are channel diameter, heated length, channel orientation, tube wall thickness. External parameters of interest are heat flux distribution and binary component fluids.

2.1.1 Influence of subcooling

For fixed mass flux \( \dot{m} \), tube length \( L \), and tube diameter \( D \), the CHF increases almost linearly with inlet subcooling, but the effect decreases with decreasing mass flux, as reported in Fig. 2-1, where data of Weatherhead (1963) are plotted. At a mass flux of 500 kg/m²s Moon et al. (1996) observed that the inlet subcooling effect on the CHF is very small, suggesting that it can be negligible at much lower mass fluxes (Mishima (1984), Chang et al. (1991)). If we plot the same data of Fig. 2-1 in terms of exit conditions, see Fig. 2-2, we find an interesting feature, which accounts for the inter-relation between exit quality and mass flux effects on the CHF. In the subcooled region \( (x < 0) \) the CHF increases as mass flux increases for a given exit quality \( x \). In the saturated region \( (x > 0) \) we may find a cross-over, and the CHF decreases with increased mass flux, for a given \( x \). It is therefore important to establish which variables are kept constant when considering the influence of a specific variable on the CHF, also specifying if we refer to inlet or exit condition.

2.1.2 Influence of mass flux

For fixed inlet conditions and geometry, the CHF increases with increasing mass flux. At low values of \( \dot{m} \), the CHF rises approximately linearly with \( \dot{m} \), but then rises much less rapidly for higher \( \dot{m} \) values. The effect of mass flux on the CHF depends on the pressure, being stronger at lower pressures. The influence of mass flux on the CHF for fixed exit conditions has been already outlined in the previous sections: the CHF increases with \( \dot{m} \) for \( x < 0 \), while decreases with \( \dot{m} \) for \( x > 0 \), being \( x \) the exit quality.
Fig. 2-1  Critical heat flux versus inlet subcooling, for different mass fluxes

Fig. 2-2  Critical heat flux versus exit quality, for different mass fluxes
2.1.3 **Influence of pressure**

The influence of the pressure on the CHF is very complex as indicated by Collier & Thome (1994), and reported in Fig. 2-3, where data of Alekseev et al. (1965) are plotted. In overall, for fixed inlet conditions the CHF increases with increasing pressure at low pressure, passes through a maximum, at around 3.0 MPa, and then decreases at higher pressures. Yin et al. (1988) experienced a secondary maximum at 19.0 MPa for \( \dot{m} = 2040 \text{ kg/m}^2\text{s} \) and inlet subcooling of 33 and 55 K. For fixed exit conditions, Moon et al. (1996) report a clearer trend than that for fixed inlet conditions. As the pressure increases the CHF sharply increases, passes a maximum, then gradually decreases. The pressure corresponding to the maximum CHF decreases as quality increases.

2.1.4 **Influence of diameter**

The effect of tube diameter on the CHF for fixed inlet and exit conditions is shown in Figs. 2-4 and 2-5, respectively. For fixed inlet conditions, the CHF increases with increasing tube diameter, the effect increasing with the inlet subcooling. For fixed exit conditions, the CHF is a decreasing function of tube diameter. It appears that the diameter effect strongly depends on the flow regime due to the difference in CHF mechanisms.

![Fig. 2-3 Influence of the pressure on the critical heat flux](image-url)
Fig. 2-4  Effect on tube diameter on the CHF for fixed inlet conditions

Fig. 2-5  Effect on tube diameter on the CHF for fixed exit conditions
2.1.5 Influence of heated length
For fixed inlet conditions there is a common evidence (Collier & Thome (1994), Hewitt (1982) and Chang et al. (1991)) that the CHF decreases with increasing heated length. For fixed exit conditions, from the interesting study of Moon et al. (1996), reported in Fig. 2-6, we may say that for short tubes the CHF decreases with the heated, while for heated lengths above a threshold the heated length effect would seem to disappear. The threshold length is a function of other system parameters.

2.1.6 Effect of channel orientation
Vertical downflow against upflow CHF studies have been performed among others by Papell et al. (1966) using liquid nitrogen, Kirby et al. (1967) using water, and Bertoni et al. (1976) using R-12. Generally speaking, downflow CHF was found to be 10-30% lower than upflow, buoyancy effects playing the main role in the reduction. The buoyancy effect was found to be an inverse function of pressure and subcooling, and was proved to be small if the liquid downflow velocity is significantly above the bubble rise velocity.

Among other, Becker (1971) found that the CHF for horizontal tubes results lower than that experienced for vertical upflow if the mass flux is lower than a critical value. This is because bubbles formed in the nucleate boiling regime move upwards due to gravity and concentrate in the upper region of the tube, thus causing a premature burnout, with respect to vertical upflow, as the void increases. Larger diameter tubes require larger critical mass fluxes to avoid the separation of the phases.

Cumo et al. (1978) carried out experiments using R-114 at different pipe inclinations between horizontal and vertical upflow conditions included. The tube inclination has a significant
influence on the CHF, which varies up to a factor of two passing from horizontal to upward vertical flow. Authors found that the buoyancy effect on the CHF may be neglected when the modified Froude number, as given by eq. (1-1), is greater than 5-7.

2.1.7 Influence of wall thickness
Relatively little information is available on the effect of wall thickness. As reported by Collier & Thome (1994), some experiments on the wall thickness effect were performed by Aladyev et al. (1961), Barnett (1963), Lee (1965), and Tippets (1962). Results are quite contradictory, as Aladyev et al. (1961) did not find any effect in the range 0.4 to 2.0 mm, Lee (1965) observed a 5% reduction as the tube wall thickness is decreased from 2.1 to 0.86 mm, and Tippets (1962) found up to 20% decrease as a 0.254 mm ribbon heater was replaced by a 0.152 mm thick ribbon.

2.1.8 Influence of heat flux distribution
The effect of the axial heat flux distribution has been investigated, for example, by Keeys et al. (1972) and by Cumo et al. (1980). Authors found a considerable difference in heat flux for burnout at a given quality for the uniform and non-uniform heating mode, noting that, with the non-uniform heating burnout can occur first up-stream of the end of the tube.

2.1.9 Influence of mixture composition
The effect of composition on the CHF in the case of binary mixtures has been studied by Auracher & Marroquin (1995), Celata et al. (1994d), and Mori et al. (1990). The composition of the binary mixture has little or no effect on the CHF for long tubes, i.e., \( L/D > 30 \). For shorter tubes, as also reported by Collier & Thome (1994), the CHF increases with the mixture composition, passes through a maximum, and then decreases, all with respect to the ideal linear behaviour between the values of the pure fluids, for same thermal hydraulic conditions.

2.2 Available Correlations for the Prediction of Saturated Flow Boiling CHF
For given fluid, thermal-hydraulic and geometric conditions, and for a given heat flux, axially uniform, experimental data are usually found to lie approximately on a single curve in a CHF versus burnout quality representation, being the CHF located at the end of the channel. This implies that the local quality conditions govern the magnitude of the CHF, and is termed as local conditions hypothesis.

We can plot the same data in terms of burnout quality and boiling length at burnout, this latter being the length between the location where the saturation condition is reached and the CHF location. The boiling length is easily obtained form a heat balance knowing heat flux, quality, mass flux and tube geometry. This type of plot can be regarded a indicating the possibility of some integral rather than local phenomenon.

Existing correlations are given in one of the two above reported forms and, for uniform heat flux, can be converted easily to the other, providing with equivalent results. When the heat flux is non-uniform, the two forms give quite different results, and this will be discussed later.

Referring the reader also to other sources collecting CHF correlations, such as Lee (1977), Katto (1986), Whalley (1987) and Collier & Thome (1994), some widely used correlations for uniform heat flux are reported hereunder, for which great care is recommended in their application. As usually such correlations are not based on a physical background, they should be regarded as mathematical interpolation for the data range they cover. Their use outside this range can give high inaccuracy in the prediction.
\[ \frac{\dot{q}_{\text{CHF}}}{\pi D L M h_{ig}} = \frac{a - x_{in}}{1 + \frac{b}{L}} \]  

(2-1)

where \( \dot{q}_{\text{CHF}} \) is the critical heat flux in kW/cm\(^2\), \( D \) and \( L \) are the tube internal diameter and length, respectively, in cm, \( M \) the mass flow rate in g/s, and

\[
a = \left( 1 - \frac{p}{p_c} \right) \left( \frac{\dot{m}}{\dot{m}_0} \right)^{0.33} \quad \dot{m}_0 = 100 \text{ g/cm}^2\text{s}
\]

\[
b = 0.315 \left( \frac{p_c}{p} - 1 \right)^{0.4} D_h^{1.4} \dot{m}
\]

being \( p_c \) the water critical pressure, and \( D_h \) the equivalent hydraulic diameter in cm (recommended ranges: \( p = 45 - 150 \text{ kg/cm}^2 \); 100 \((1 - p/p_c)^3 \leq \dot{m} \leq 400 \text{ g/cm}^2\text{s}; x_{in} \leq 0.2; D > 0.7 \text{ cm}; L = 20.3 - 267 \text{ cm}).

- W-3, Tong (1969)

\[
\frac{\dot{q}_{\text{CHF}}}{10^6} = \left[ (2.022 - 0.0004302 p) + (0.1722 - 0.0000984 p) \exp \left( \frac{(18.177 - 0.004129 p)x}{10} \right) \right] \\
\left[ (0.1484 - 1.596x + 0.1729x) \dot{m}/10^6 + 1.037 \right] (1.157 - 0.869 x) \left[ 0.2664 + 0.8357 \exp \left( -3.151 D_h \right) \right] \left[ 0.8258 + 0.000784 \left( h_{in} - h_{in} \right) \right]
\]

(2-2)

The heat flux \( \dot{q}_{\text{CHF}} \) is in Btu/(hr)(ft\(^2\)) (recommended range and units of the parameters are: \( p = 1000-2300 \text{ psia}; \dot{m} = 1.0 \times 10^6 - 5.0 \times 10^6 \text{ lb/(hr)(ft}^2\); \( D_h = 0.2 - 0.7 \text{ in}; x = -0.15 \text{ to } +0.15; h_{in} \geq 400 \text{ Btu/lb}; L = 110 - 144 \text{ in}; \text{heated perimeter/wetted perimeter} = 0.88 - 1.0).)  

- Bowring (1972)

\[
\dot{q}_{\text{CHF}} = \frac{A + 0.25 D \dot{m} (\Delta h_{\text{sub}})_{in}}{p' + L} \]  

(2-3)

\[
A = \frac{2.317 \left[ \frac{D \dot{m} h_{ig}}{4} \right] F_1}{1.0 + 0.0143 F_2 \dot{m} D^{1/2}}; \quad F = \frac{0.077 F_3 D \dot{m}}{1.0 + 0.347 F_4 \left( \dot{m}/1356 \right)^n}; \quad n = 2.0 - 0.00725 p
\]

where \( \dot{q}_{\text{CHF}} \) is the critical heat flux in W/m\(^2\), \((\Delta h_{\text{sub}})_{in}\) is the inlet subcooling expressed in J/kg, \( L \) is the tube length expressed in m, \( D \) is the internal tube diameter in m, \( \dot{m} \) the mass flux in kg/m\(^2\)s, \( h_{ig} \) is the latent heat of vaporization in J/kg, and \( p \) is the system pressure in bar. Parameters \( F_1, F_2, F_3, \) and \( F_4 \) are given by:

\( p' = p/69 \)

\( p' < 1 \)
\[
F_1 = \left( p'^{18.942} \exp[20.8 (1 - p')] \right) + 0.917, \quad F_2 = \left( p'^{1.316} \exp[2.444 (1 - p')] \right) + 0.309
\]

\[
F_3 = \left( p'^{17.092} \exp[16.658 (1 - p')] \right) + 0.667, \quad F_4 = \left( p'^{1.649} \right)
\]

\( p' > 1 \)

\[
F_1 = p'^{-0.368} \exp[0.648 (1 - p')] \quad ; \quad F_1 = p'^{-0.448} \exp[0.245 (1 - p')]
\]

\[
F_3 = p' 0.219 \quad ; \quad F_4 = p' 1.649
\]

(Recommended ranges: \( p = 2 - 190 \) bar; \( D = 0.002 - 0.045 \) m; \( L = 0.15 - 3.7 \) m; \( m = 136 - 18600 \) kg/m^2s).

- Katto & Ohno (1984)

a) In the case of \( \rho_g/\rho_l < 0.15 \)

\[
\dot{q}_{\text{CHF}} = \frac{C}{(l_b/D)} \left( \frac{\sigma \rho}{m^2 l_b} \right)^{0.043}
\]

(2-4)

\[
\dot{q}_{\text{CHF}} = 0.10 (\rho_g/\rho_l)^{0.133} \left( \frac{\sigma \rho}{m^2 l_b} \right)^{1/3} \left( \frac{1}{1 + 0.0031 l_b/D} \right)
\]

(2-5)

\[
\dot{q}_{\text{CHF}} = 0.098 (\rho_g/\rho_l)^{0.133} \left( \frac{\sigma \rho}{m^2 l_b} \right)^{0.433} \left( \frac{1}{l_b/D} \right)^{0.27} \left( \frac{1}{1 + 0.0031 l_b/D} \right)
\]

(2-6)

Where \( C \) is given as \( C = 0.25 \) for \( l_b/D < 50 \), \( C = 0.25 + 0.0009 [(l_b/D) - 50] \) for \( l_b/D = 50 - 150 \), and \( C = 0.34 \) for \( l_b/D > 150 \), being \( l_b \) the boiling length. Roughly speaking, eqs. (2-4) and (2-5) correspond to the CHF in annular flow, and eq. (2-6) to the CHF in froth or bubbly flow. With increasing \( \dot{m} \) (i.e., with decreasing \( \rho \rho_l/m^2 l_b \)), the above equations are employed in the order of the first, second, and third equation so as to connect the value of the CHF continuously.

b) In the case of \( \rho_g/\rho_l > 0.15 \)

\[
\dot{q}_{\text{CHF}} = C \left( \frac{\sigma \rho}{m^2 l_b} \right)^{0.043} (l_b/D)
\]

(2-7)

\[
\dot{q}_{\text{CHF}} = 0.234 (\rho_g/\rho_l)^{0.513} \left( \frac{\sigma \rho}{m^2 l_b} \right)^{0.433} (l_b/D)^{0.27} \left( \frac{1}{1 + 0.0031 l_b/D} \right)
\]

(2-8)

\[
\dot{q}_{\text{CHF}} = 0.0384 (\rho_g/\rho_l)^{0.6} \left( \frac{\sigma \rho}{m^2 l_b} \right)^{0.173} \left( \frac{1}{l_b/D} \right)^{0.27} \left( \frac{1}{1 + 0.28 (\sigma \rho/m^2 l_b)^{0.233} l_b/D} \right)
\]

(2-9)
where $C$ takes the same value as in eq. (2-4) (recommended ranges: $L = 0.01 - 8.8$ m; $D = 0.001 - 0.038$ m; $L/D = 5 - 880$; $p_c/p_l = 0.00005 - 0.41$; $(\sigma p_l m^2 L) = 3 \times 10^{-9} - 2 \times 10^{-2}$).

The Katto & Ohno (1984) correlation has been tested for water, ammonia, benzene, ethanol, helium, hydrogen, nitrogen, R12, R21, R22, R113, and potassium.

- Correlations for the CHF in binary mixtures

The above reported correlations have been developed for pure fluids such as water (CISE, W-3, and Bowring, 1972) or more fluids (Katto & Ohno, 1984). Much different is the case where we have to face with binary mixtures. An exhaustive description of the CHF in binary mixtures can be found in Collier & Thome (1994), while Celata et al. (1994d), Auracher & Marroquin (1995) and Celata & Cumo (1996) dealt specifically with refrigerant binary mixtures. Upon results obtained with mixtures of refrigerants and on the basis of the parametric trends described in 2.1.9, it is possible to say here that for short tubes, i.e., $L/D \leq 30$, the CHF can be calculated using the Tolubinsky & Matorin (1973) correlation, given by eqs. (1-7) and (1-9). For long tubes, i.e., $L/D > 30$, Celata et al. (1994d) found that the CISE correlation, proposed by Bertoletti et al. (1965), provides quite good results. Also the Katto & Ohno (1984) correlation may be directly applied to binary mixtures in long tubes, although the accuracy is less than the CISE correlation.

- Correction for axial non-uniform heat flux

For non-uniform heat flux single channels, Tong et al. (1966) recommends to use a shape factor $F_c$ so that:

$$
\dot{q}_{CHF,nu} = \frac{\dot{q}_{CHF,u}}{F_c}
$$

(2-10)

where subscript $nu$ indicates the non-uniform heating and subscript $u$ indicates uniform heating supply, and where $F_c$ is expressed as:

$$
F_c = \frac{C}{\dot{q}_{loc} \left[ 1 - \exp \left( -C l_{CHF,u} \right) \right]} \int_{l_{OB}}^{l_{CHF,nu}} q(z) \exp \left[ -C(l_{CHF,nu} - z) \right] dz
$$

(2-11)

with

$$
C = 0.44 \left( \frac{1 - x_{CHF,nu}}{\dot{m}/10^6} \right)^{0.9} \left( \frac{1.72}{\dot{m}/10^6} \right)^{-1/2} (in.-1)
$$

In eq. (2-11) $\dot{q}_{loc}$ is calculated using one of the available correlations for uniform heat flux, and $l_{CHF,nu}$ is the axial location at which the CHF occurs for non-uniform heat flux, in., $l_{CHF,u}$ is the axial location at which the CHF occurs for uniform heat flux, in., $l_{OB}$ is the axial location at which nucleate boiling begins, in., $x_{CHF,nu}$ is the quality at the CHF location under non-uniform heat flux, and $\dot{m}$ is in lb/(hr)(ft$^2$). The term $F_c$ is a memory effect parameter which accounts for the thermal history of the fluid along the tube. $F_c$ is small in the subcooled region and local heat flux
determines the boiling crisis. At high qualities, $C$ is small, the memory effect is high, and the average heat flux, or enthalpy rise, primarily determines the boiling crisis.

2.3 The Artificial Neural Network as a CHF Predictor

An advanced information processing technique such as artificial neural networks (ANNs) (Wasserman, 1989) might provide a valuable alternative to the current techniques for estimating the CHF, since there exists a large number of experimental data for the CHF. Yapo et al. (1992), Moon & Chang (1994), Moon et al. (1996), Mazzola (1997) applied the ANNs to the CHF prediction, showing promising results. An artificial neural network is composed of elements that are analogous to the elementary functions of biological neurons. ANNs have the characteristic of tolerance against experimental noise owing to the massive internal structure of the network. Also, it is easy to update the performance of the ANN for new experimental data. Although the ANNs do not require accurate information about physical phenomena, however, their main drawbacks are the loss of model transparency (black-box character) and the lack of any indicator for evaluating the accuracy and reliability of the ANN answer when never-seen patterns are presented. From applications to CHF of Moon et al. (1996) and Mazzola (1997), it appears, none the less, that the ANNs are able to predict CHF data within $\pm 20-25\%$ for most of data points, providing a consistent alternative method to empirical correlations.

2.4 The Tabular Method for the Prediction of Saturated Flow Boiling CHF

Another interesting method for the prediction of the CHF in saturated flow boiling is that proposed first by Doroshchuk et al. (1975) which consists in a series of standard tables of CHF values as a function of the local bulk mean water condition and for various pressures and mass fluxes for a fixed tube diameter of 8 mm. Correction factors for tube length and for tube diameters other than 8 mm must be used. The latest updating of these look-up table, (Groeneveld et al. 1996) consists of 22946 data points covering the range 0.1 to 20.0 MPa, up to 8.0 Mg/m$^2$s and -0.5 to 1.0 for discrete values of pressure, mass flux and CHF quality, respectively. For tube diameters other than 8 mm, the CHF is given by the approximate equation:

$$\dot{q}_{CHF} = \dot{q}_{CHF, 8\, mm} \left( \frac{D}{0.008} \right)^k$$

(2-12)

being $k = -1/2$ the best parameter found by Groeneveld et al. (1996) in the range of tube diameter from 3 to 25 mm. Other researchers propose $k = -1/3$, such as Smith (1986) and Groeneveld et al. (1986).

The CHF look-up table method has become a widely accepted prediction technique. It has the following advantages over correlations or semi-analytical CHF models: i) accurate prediction; ii) the widest range of applications; iii) ease of use (no fluid properties are needed); iv) ease of updating; and v) correct parametric and asymptotic trends. Main drawbacks are the complexity of their use in a computer code with respect to a correlation, providing more or less the same accuracy.

2.5 Available Models for the Prediction of Saturated Flow Boiling CHF

The main advantage of mechanistic methods, is that, as they are based on the physical mechanisms leading to the CHF, in principle their validity should not be confined to the range of the available experimental data on which they are assessed. The models should be only linked to the range of validity of the mechanisms identified, which should result of much more general application. As a matter of fact, sometimes some models for the mathematical description of bubble dynamics, rely on empirical constants or correlations which restrict their general validity.
As a model is strictly linked to the mechanisms which can be responsible of the CHF occurrence, it is necessary a grouping of existing models in DNB and dryout models.

2.5.1 DNB type critical heat flux

Different CHF mechanisms have been postulated for the DNB type thermal crisis, in order to develop reliable correlations or predicting methods for the CHF calculation, or to identify possible methods to avoid the CHF occurrence. Typically, for low quality flow, the flow regime consists in an agglomeration of vapour in the near-wall region, and a prevailing presence of liquid in the centre of the channel. The governing heat transfer mechanisms is the bubble growing and detachment at the wall, and their migration in the liquid bulk. Among the many mechanisms proposed, see, for instance, detailed reviews by Tong & Hewitt (1972), Hewitt (1980), Weisman (1992), and Katto (1994), those which appear to be somehow established experimentally are the following:

a) Hot spot formation under a growing bubble. As observed by Kirby et al. (1967), a dry patch forms between the growing bubble and the nucleation cavity as the micro-layer of liquid under the bubble evaporates. The dry patch may be rewetted at the bubble departure and the process can go on. Before the rewetting of the dry patch, the wall temperature rises due to the heat transfer deterioration. However, if the dry temperature exceeds a critical temperature (often called Leidenfrost temperature), then rewetting does not happen readily, thus causing local overheating and hence burnout. A schematic of this mechanism is drawn in Fig. 2-7.

b) Near-wall bubble crowding model. Tong et al. (1966) first started from the idea that a bubble boundary layer takes place on the surface and vapour generated by boiling at the heated wall must leave the near-wall region through this two-phase boundary layer. Burnout occurs when vapour escape through the layer is prevented because of a critical crowding of the boundary layer with bubbles. More recently, Hebel et al. (1981) and Weisman & Pei (1983) and Weisman & Ying (1983) assumed that the turbulent interchange between the bubbly layer and the bulk of the liquid may be the limiting mechanism leading to the CHF occurring. CHF occurs when bubble crowding near the heated wall prevents the bulk cold liquid from reaching the wall. This mechanism is discussed in more detail below.

c) Dryout under a slug or vapour clot. Fiori & Bergles (1968, 1970) observed that in plug flow, the thin liquid film around the large bubble may dry out causing burnout. Alternatively, a stationary vapour clot can form on the heated wall, being a thin liquid film present between the clot and the wall. In this case the local drying out of the film causes wall overheating and then burnout. A schematization of this mechanism is shown in Fig. 2-8

d) Liquid sublayer dryout theory. This mechanism has been already discussed for the understanding of the CHF in subcooled flow boiling, chapter 1.3. The Lee & Mudawwar (1988) liquid sublayer dryout model was developed for subcooled flow boiling, on the basis of the Helmholtz instability at the microlayer/vapour interface as trigger condition for microlayer dryout. Such a model has been extended to low-quality flow by Lin et al. (1989) under pressurized water reactor conditions. Basically, the main improvements of Lee & Mudawwar's model include the following: 1) The homogeneous two-phase flow model is assumed to be suitable for high-pressure, high-mass flux conditions. Fluid properties are calculated using the effective homogeneous flow rather than single-phase fluid properties. 2) The liquid enthalpy flowing into the microlayer is assumed to be independent of bulk
subcooling and is approximated by the saturated liquid enthalpy for maintaining the local boiling.

Among the models listed above, it is interesting to give few details on the Weisman & Pei (1983) model, above described in b), which is currently the only theoretically based CHF prediction procedure that has been shown to give good accuracy with fluids other than water, especially with refrigerants.

The Weisman & Pei (1983) model, the schematic of which is drawn in Fig. 2-9, assumes that: a) During low-quality boiling, the bubbly layer builds up along the channel until it fills the region near the wall where the turbulent eddies are too small to transport bubbles radially. At the CHF site, the bubbly layer is assumed to be at this maximum thickness. b) CHF occurs when the volume fraction of steam in the bubbly layer just exceeds the volume fraction (critical void fraction) at which an array of slightly flattened ellipsoidal bubbles can be maintained without significant contact between the bubbles. c) The volume fraction of steam in the bubbly layer is determined by a balance between the outward flow of vapor and the inward flow of liquid at the bubbly layer-core interface.

![Fig. 2-7 Schematic of the hot spot formation under a growing bubble model, Kirby et al. (1967)](image-url)
Fig. 2-8  Schematic of the dryout under a slug or vapour clot model, Fiori & Bergles (1968, 1970)

Fig. 2-9  Schematic of the near-wall bubble crowding model, Weisman & Pei (1983)
Considering a bubbly layer control volume, they can write the total mass balance on the bubbly layer taking into account the total flow rate from core to bubbly layer, which must be equal to the total flow rate from bubbly layer to core plus the axial flow in and out of the bubbly layer control volume. From a simple mass balance over the bubbly layer they obtain:

\[
\dot{m} = \frac{Q_{CHF}}{x_2 - x_1}
\]

where \( \dot{m} \) represents the mass flow rate into the bubbly layer. This mass flow rate is determined by the turbulent velocity fluctuations at the bubbly layer edge. The distance from the edge of the bubbly layer to the wall is taken as the distance at which the size of the turbulent eddies is \( k \) times the average bubble diameter. Only a fraction of the turbulent velocity fluctuations produced are assumed to be effective in reaching the wall. The effective velocity fluctuations are those in which the velocity exceeds the average vapour velocity away...
from the wall produced by the vapour being generated at the wall. The quantities \( x_1 \) and \( x_2 \) represent the vapour qualities in the core region and bubbly layer, respectively, at the CHF (these are actual values and not thermodynamic equilibrium qualities).

The factor \( F \) represents the fraction of the heat flux producing vapour that enters the core region, given by the ratio between the difference of the enthalpy of saturated liquid and that at bubble detachment point, and the difference between the enthalpy of liquid at given axial location and that at bubble detachment point. The occurrence of the CHF is for that quality in the bubbly layer that corresponds to the maximum void fraction that is possible in a bubbly layer of independent bubbles just prior agglomeration. For slightly flattened elliptically shaped bubbles with a length-to-diameter ratio of 3/1, this void fraction is estimated as 0.82.

2.5.2 Dryout type critical heat flux

This type of CHF mechanism consists in the gradual depletion of the liquid film wetting the heating wall, until the liquid film flow rate is zero and consequent drying of the wall. It is evident that the dryout type is linked to the annular flow regime in convective flow boiling, as reported in the sketch of Fig. 2-10. Observations of transparent test sections and flow pattern maps show that, for most CHF cases where we have an exit quality greater than 10\%, the flow pattern is annular. And this is probably the most frequent situation in steam generation apparatuses.

Many studies have suggested that the CHF may occur when the liquid film flow rate goes to zero due to the combined effects of: i) liquid droplet entrainment from the liquid film, produced by the gas flow in the core (droplets are mainly entrained from liquid waves on liquid film surface); ii) liquid droplet deposition on the liquid film (some droplets initially entrained by the gas flow hit the liquid film and are captured); and iii) evaporation of the liquid film because of the heat flux delivered from the wall.

The first evidence showing that dryout occurs at the point where the film flow rate becomes zero was due to the measurement of the film flow rate at the end of a heated channel as a function of power input to the channel, performed by Hewitt et al. (1963, 1965) and detailed in Hewitt & Hall-Taylor (1970). The results are drawn in Fig. 2-11, where it is possible to observe that the critical heat flux point occurs at the power delivered to the fluid for which the film flow rate at the tube outlet is zero. More exactly, the occurrence of dryout should happen when the liquid film flow rate becomes smaller than the minimum value which is necessary to wet the whole heating wall, and the liquid film breaks. Also the so-called cold patch experiments by Bennet et al. (1967) represent a further evidence of this CHF mechanism.

The first attempt to use an annular flow model for the prediction of dryout is due to Whalley et al. (1974), while the model has been recently updated by Govan et al. (1988) and by Hewitt and Govan (1989). For the complexity of the model description, the reader is referred to the original sources, while a brief review will be given here. Figure 2-12 shows the postulated mechanisms, in which dryout occurs when the liquid film flow rate falls smoothly to zero as a result of entrainment and evaporation. A mass balance, which also accounts for deposition, gives:
Fig. 2-11 Measurement of the film flow rate at the end of a heated channel as a function of power input to the channel, Hewitt et al. (1963, 1965)

Fig. 2-12 Schematic of the annular flow model, Whalley et al. (1974)
Where $\dot{m}_{lf}$ is the liquid film mass flux, $DR$ the deposition rate, and $ER$ the entrainment rate. In order to integrate this equation, it is required:

i) a value for $\dot{m}_{lf}$ at the start of annular flow. Typically, it is assumed that at the start of annular flow $x_I = 0.01$ and $\dot{m}_{lf} = 0.99 \dot{m}_l$. Govan (1984) found that the predicted CHF was sensitive to $\dot{m}_{lf}$ but not to $x_I$. However, very little information exists on the transition to annular flow in a boiling channel.

ii) a means to calculate the entrainment rate $ER$. Whalley et al. (1974) expressed this as a function of surface tension, interfacial shear and liquid film thickness. Govan (1984) tried using various entrainment correlations but found that the CHF predictions were not greatly affected, mainly because the entrainment becomes small as dryout is approached.

iii) a means to calculate the deposition rate $DR$. Whalley et al. (1974) assumed a simple proportionality between $DR$ and the droplet concentration in the gas core, the constant of proportionality depending on surface tension. Govan (1984) found that the predicted CHF is sensitive to $DR$.

This mechanism of dryout is widely accepted though there is some debate about the details. Anyway, recent updatings by Govan et al. (1988) and Hewitt & Govan (1989) demonstrated that comparison with 5300 CHF data points shows a mean error of -9.7% with a standard deviation of 16%, provided the CHF mechanism is dryout, for a wide range of fluids.

3 Post-CHF Heat Transfer

Post-CHF heat transfer is of interest in all cases where the CHF condition can be reached or exceeded and the heating wall temperature is still low in comparison with the melting temperature or that value for which the wall material failure may happen. Heat transfer knowledge in these areas is required in many engineering applications such as in the design of once-through steam generators (where complete evaporation of the feedwater occurs), or of very high pressure recirculation boilers (where the CHF levels are low). The thermal-hydraulic design of pressurized water reactors has also called for an intensive investigation of heat transfer rates beyond the CHF point for transient and accident analyses.

Main heat transfer regimes in post-CHF heat transfer are film boiling and liquid deficient region. Film boiling typically occurs after the CHF in subcooled flow boiling, with low-quality CHF or in pool boiling. A schematic representation of such a heat transfer regime is given in Fig. 3-1. The liquid deficient region or dispersed flow boiling, which occurs after the high-quality CHF is schematically drawn in Fig. 3-2.

3.1 Film Boiling

In pool boiling or after the subcooled flow boiling CHF we may have the occurrence of the film boiling heat transfer regime once the CHF has been exceeded. The heat is transferred by conduction through the vapour film, and evaporation takes place at the liquid-vapour interface. Nucleation is absent and, in general, the problem may be simply treated as an analogy to filmwise condensation. Many theoretical solutions can be obtained for horizontal and vertical flat surface, and also inside and outside tubes under both laminar and turbulent conditions with and without interfacial stress. The simplest solution may be obtained for laminar flow and linear temperature distribution. For a flat vertical surface the local heat transfer coefficient is given by:

$$\frac{d\dot{m}_{lf}}{dz} = \frac{4}{D} (DR - ER - \frac{\dot{q}}{h_{lg}})$$  (2.14)

Where $\dot{m}_{lf}$ is the liquid film mass flux, $DR$ the deposition rate, and $ER$ the entrainment rate. In order to integrate this equation, it is required:
\[ \alpha(z) = C \left[ \frac{\lambda_g^3 \rho_g (\rho_l - \rho_g) g h_l g}{z \Delta T \eta} \right]^{1/4} \]  

(3-1)

where \( C \) is dependent on boundary conditions; for zero interfacial stress we have \( C = 0.707 \), while for zero interfacial velocity we have \( C = 0.5 \). For film boiling outside a cylinder of diameter \( D \) we have \( C = 0.62 \) and eq. (3-1) calculated for \( z = D \).

Wallis & Collier (from Collier & Thome (1994)) for turbulent flow in the vapour film found (vertical flat surface):

\[ \frac{\alpha(z)}{\lambda_g} = 0.056 \text{Re}_g^{0.2} [\text{PrGr}^*]^{1/3} \]  

(3-2)

where:

\[ \text{Gr}^* = \frac{z^3 g \rho_g (\rho_l - \rho_g)}{\eta^2} \]

Fung et al. (1979) developed a model which covers both the laminar and the turbulent flow.
Although eq. (3-1) gives good predictions in some cases (see Fig. 3-3, where the Costigan et al. (1984) data for water in an 8 mm diameter vertical tube are compared with theoretical predictions), the vapour film is not smooth in reality (Dougall & Rohsenow (1963)), and more refined equations are therefore necessary for a better physical description of the phenomenon (Bailey (1971), Denham (1984)). Further experimental evidences (Bromley et al. (1953), Motte & Bromley (1957), Liu et al. (1992), Papell (1970, 1971), Newbold et al. (1976)) can be summarized as follows: classical laminar film boiling may be a valid approximation up to 5 cm downstream of the CHF front; the heat transfer coefficient is an increasing function of the velocity and a decreasing function of the channel diameter (for film boiling inside and on tubes); the heat transfer coefficient in downflow is generally lower (up to 3-4 times) than in upflow. Information on hydrocarbons can be found in Glickstein & Whitesides (1967).
3.2  Heat Transfer in the Liquid Deficient Region

This heat transfer regime is sketched in Fig. 3-2, and its knowledge is important in the design of high-pressure once-through steam generators and recirculation boilers. Experimental data for steam-water mixtures, up to 25 MPa, have been produced in the past (Schmidt (1959), Swenson et al. (1961), Herkenrath et al. (1967), Bahr et al. (1969)). The liquid deficient region heat transfer in circular bends has been recently experimented (Lautenschlager & Mayinger (1986), Wang & Mayinger (1995)), together with the use of refrigerants (Lautenschlager & Mayinger (1986), Wang & Mayinger (1995), Nishikawa et al. (1986), Obot & Ishii (1988), Yoo & France (1996)). Kefer et al. (1989) studied the post-CHF heat transfer in inclined evaporator tubes, while Burdunin et al. (1987) and Unal et al. (1988) investigated complex geometries.

Three types of predictive tools have been adopted for the calculation of the heat transfer coefficient (generally through wall temperature calculation), as reviewed by Groeneveld (1972), and Wang & Weisman (1983):

a)  empirical correlations (no theoretical background behind, but only functional equations between the heat transfer coefficient and independent variables);

b)  correlations which take into account the thermodynamic non-equilibrium and calculate the true vapour quality and temperature; and

c)  theoretical or semi-theoretical models.
3.2.1 Empirical correlations

Many empirical correlations have been proposed for the calculation of the heat transfer coefficient, mostly based on modifications of the well-known Dittus-Boelter type equation for liquid single-phase flow. None of them takes into account non-equilibrium effects. One of the most accurate among available correlations is that proposed by Groeneveld (1973):

\[
Nu_g = a \left\{ Re_g \left[ x + \frac{\rho_g}{\rho_l} (1 - x) \right] \right\}^b Pr_{g,w}^c Y^d \quad (3-3)
\]

where:

\[
Y = 1 - 0.1 \left( \frac{\rho_l}{\rho_g} - 1 \right)^{0.4} (1 - x)^{0.4}
\]

For tubes \( a = 1.091 \times 10^{-3} \); \( b = 0.989 \); \( c = 1.41 \); and \( d = -1.15 \), while for annuli \( a = 5.2 \times 10^{-2} \); \( b = 0.688 \); \( c = 1.26 \); and \( d = -1.5 \). The range of data on which correlations are based is reported in Tab. 3-1. Improvements of eq. (3-3) have been given by Slaughterback et al. (1973a, 1973b).

3.2.2 Correlations accounting for thermodynamic non-equilibrium

These correlations account for thermodynamic non-equilibrium. Theoretically, two extreme conditions would be possible, i.e.:

a) all the heat is transferred to liquid drops until their complete evaporation (complete equilibrium, hypothesis valid for very high pressure, nearly critical, and mass flux > 3000 kg/m²s);
b) all the heat is transferred to the vapour phase, causing its superheating (complete non-equilibrium, hypothesis acceptable for low pressure and low flow rate).

As generally real situations will be in between, we may think to split the heat flux in two components:

\[
q_{tot} = q_g + q_l \quad (3-4)
\]

where \( q_g \) is the component of the heat flux delivered to the vapour (which raises its temperature) and \( q_l \) is the heat flux absorbed by liquid drops (which causes their evaporation). Usually, correlations provide an evaluation of:

\[
\varepsilon = \frac{q_l}{q_{tot}} \quad (3-5)
\]

through which it is possible to obtain the vapour and wall temperature with thermodynamic calculations. Such correlations have been proposed by a variety of investigators (Plummer et al. (1977), Groeneveld & Delorme (1976), Jones & Zuber (1977), Chen et al. (1977)) and that proposed by Plummer et al. (1977) is reported here:

\[
\varepsilon = C_1 \ln \left[ G \left( \frac{D_h}{\rho_g \sigma} \right)^{0.5} (1 - x_{CHF})^2 \right] + C_2 \quad (3-6)
\]
where $D_h$ is the hydraulic diameter, and the constants $C_1$ and $C_2$ have been given by authors for nitrogen, water, and R 12. More recently Yoo & France (1996) have proposed $C_1$ and $C_2$ for R 113, showing that the parameter $C_2$ could be correlated using the molecular weight. $C_1$ and $C_2$ values are given in Tab. 3-2, while Fig. 3-4 shows the prediction of experimental data obtained using eq. (3-6).

Nishikawa et al. (1986) proposed also a method based on a non-dimensional parameter representing the ratio of the heat capacitance of the vapour flow to the thermal conductance from the vapour to the liquid droplets. Such a parameter was described as a function of non-dimensional thermodynamic parameters. Prediction of experimental data with the Nishikawa et al. correlation is shown in Fig. 3-5.

### 3.2.3 Theoretical models

Many theoretical models have been proposed with different levels of complexity (Groeneveld (1972), Chen et al. (1977), Bennett et al. (1968), Illoeje et al. (1974), Ganic & Rohsenow (1976), Moose & Ganic (1982), Whalley et al. (1982), Hein & Köhler (1984), Kirillov et al. (1987), Yagov et al. (1987) Rohsenow (1988)), accounting in a more or less detailed way, for the various paths by which heat is transferred from the heating surface to the bulk vapour phase. Namely, models should account for: a) the heat transferred to liquid droplets impacting on the wall; b) the heat transferred to liquid droplets entering the thermal boundary layer without wetting the surface; c) the heat transferred from the heating surface to the vapour bulk by convection; d) the heat transferred from the vapour bulk to suspended droplets in the vapour core by convection; e) the heat transferred from the heating surface to liquid droplets by radiation; and f) the heat
transferred from the surface to the vapour bulk by radiation. Nonetheless, following the starting assumption, not all of the above mechanisms are generally considered in the proposed models. Because of the complexity of the general mathematical description of existing models, the reader is referred to original papers reported in the bibliography.

<table>
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<tr>
<th>Flow direction</th>
<th>Geometry</th>
<th>Annulus</th>
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</thead>
<tbody>
<tr>
<td>Dh, cm</td>
<td>Vertical and horizontal</td>
<td>Vertical</td>
</tr>
<tr>
<td>p, MPa</td>
<td>0.25 to 2.5</td>
<td>0.15 to 0.63</td>
</tr>
<tr>
<td>m, kg/m²s</td>
<td>6.8 to 21.5</td>
<td>3.4 to 10.0</td>
</tr>
<tr>
<td>x, fraction by weight</td>
<td>0.1 to 0.9</td>
<td>0.1 to 0.9</td>
</tr>
<tr>
<td>q, kW/m²</td>
<td>120 to 2100</td>
<td>450 to 2250</td>
</tr>
<tr>
<td>Nuₜ</td>
<td>95 to 1770</td>
<td>160 to 640</td>
</tr>
<tr>
<td>Reₜ (x + (1-x)pᵥ/pₜ)</td>
<td>6.6 10⁴ to 1.3 10⁶</td>
<td>1.0 10⁵ to 3.9 10⁵</td>
</tr>
<tr>
<td>Prₔₜ, w</td>
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<td>0.91 to 1.22</td>
</tr>
<tr>
<td>Y</td>
<td>0.706 to 0.976</td>
<td>0.61 to 0.963</td>
</tr>
</tbody>
</table>

Table 3-2 Constant for Plummer et al. correlation (1977)

<table>
<thead>
<tr>
<th>Fluid</th>
<th>C1</th>
<th>C2</th>
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<tr>
<td>Nitrogen</td>
<td>0.082</td>
<td>0.290</td>
</tr>
<tr>
<td>Water</td>
<td>0.07</td>
<td>0.400</td>
</tr>
<tr>
<td>R 12</td>
<td>0.078</td>
<td>0.255</td>
</tr>
<tr>
<td>R 113(a)</td>
<td>0.078(a)</td>
<td>0.13(a)</td>
</tr>
</tbody>
</table>

(a) values given by Yoo & France (1996)
Fig. 3-5  Prediction of wall temperature in post-CHF heat transfer using Nishikawa et al. correlation (1986)
4 Augmentation of CHF and Post-CHF Heat Transfer

In the thermal-hydraulic design of a heat exchanger, a steam generator, or a thermal equipment where the critical heat flux (CHF) is the limiting parameter or where the designer has to face with post-CHF heat transfer, it can be necessary to obtain a higher CHF value or a better post-CHF heat transfer coefficient than that allowed by the process thermodynamic and geometry conditions. It is therefore necessary to make use of enhancement techniques in order to have a higher CHF or a higher post-CHF heat transfer rate, similarly to what pursued in single and two-phase flow heat transfer (before the thermal crisis) (Thome (1990) and Bergles (1992)).

4.1 CHF Enhancement Techniques

Recent reviews of CHF enhancement techniques have been given by Boyd (1985a) and by Celata (1996). Among the possible techniques, we may have passive devices, such as swirl flow (twisted tapes and helically coiled wires), extended surfaces (hypervapotron), and helical coiled tubes, and active techniques, such as electrical fields, pressure wave generation and tangential injection. Only passive techniques will be discussed here; for active techniques see Boyd (1985a).

4.1.1 Swirl flow

Swirl flow is obtained using twisted tapes or helically coiled wires inside the flow channel to induce secondary radial and circumferential velocity components in the fluid, in order to obtain a better heat transfer rate and therefore a higher CHF value. A considerable increase in the pressure drop with respect to smooth tubes is observed, in general, with the use of swirl flow promoters. The use of twisted tapes as swirl flow promoters in the augmentation of the CHF in subcooled flow boiling has been studied by Gambill & Greene (1958), Gambill et al. (1961), Nariai et al. (1991), Cardella et al. (1992), and Achilli et al. (1993). An increase in the CHF typically by a factor of 2 over that for tubes without twisted tapes was generally obtained. Results by Gambill et al. (1961) and by Nariai et al. (1991) are plotted in Fig. 4-1, where the ratio between the CHF obtained with the twisted tape and the value obtained with the smooth tube is reported versus pressure for different values of the non-dimensional centrifugal acceleration, ϑ:

\[ \vartheta = \frac{a_t}{g} = \frac{\pi^2 \mu^2}{2 g D TTR} \]

(4-1)

where \( TTR \) is the twisted tape ratio; \( \vartheta \) is defined as the ratio between the tangential centrifugal acceleration (due to the twisted tape) and the standard gravitational acceleration. The thermal efficiency of the twisted tape decreases as pressure increases and becomes insignificant when pressure is above 2.0 MPa. This effect is probably due to the presence of a gap between the wall and the twisted tape. In fact, the clearance allows steam trapping in the tube-tape gap (which is an increasing function of pressure) which may results in premature CHF. Cardella et al. (1992) and Achilli et al. (1993) did not find any effect of the system pressure on the thermal efficiency of the twisted tape. A correlation for the prediction of the CHF with twisted tapes has been given by Nariai et al. (1992):

\[ \frac{\dot{q}_{CHF,t}}{\dot{q}_{CHF,st}} = (1 + 10^{-2} \vartheta \exp\{(-10^{-6} p)^2\})^{1/6} \]

(4-2)
with \( \dot{\varphi} \) given by eq. (4-1). Also eq. (1-4) can be used to predict the CHF with twisted tapes, using the resultant water velocity at the inner tube wall \( u_r \), in place of \( u \), as given by Schlosser et al. (19):

\[
\frac{u_r}{u} = \left( \frac{1 + \pi^2}{4 (TTR)^2} \right)^{1/2}
\]  

(4-3)

The Celata et al. model (1994, 1995) presented in section 1.3 can be also used to predict the CHF in subcooled flow boiling with twisted tapes.

The use of twisted tapes to enhance the CHF under saturated flow boiling conditions has been recently investigated by Lee et al. (1995). Authors found that in the low-quality region the effect of the twisted tape on the CHF is negligible. In the middle-quality region, the CHF of the twisted tape inserted tube increased with mass velocity, which was contrary to the trend observed for the empty one. Besides, the CHF was found to increase by insertion of the twisted tape except for cases of very small flow rate and large twist ratio. The clearance effect was weak as compared to the subcooled region. Finally, in the high-quality region, the CHF decreased with exit quality, the decreasing rate being slower with the twisted tape than without. Also, the CHF enhancement was most remarkable in this region. A correlation for the prediction of the CHF with twisted tape in saturated flow boiling has been proposed by Jensen (1985):

![Swirl flow CHF data using twisted tapes, Gambill et al. (1961), Nariai et al. (1992)](image-url)
\[
\frac{\dot{Q}_{H}}{\dot{Q}_{CHF}} = (4.597 + 0.09254 \, (TTR) + 0.004154 \, (TTR)^2 \, (-)^\cdot7012 \, + \, 0.09012 \, \ln \theta )
\]

with \(\theta\) given by eq. (4-1). The critical power \(\dot{Q}_{CHF}\) can be obtained using a suitable correlation or model.

Helically coiled wires as swirl flow promoters have been used by Celata et al. (1994b) for CHF in subcooled flow boiling. Authors used wires of spring steel having a diameter of 0.5, 0.7 and 1.0 mm and a pitch from 1.5 to 20.0 mm in 8.0 mm I.D. tubes. Results are presented in Fig. 4-2, where an increase in the CHF up to 50% using a 1.0 mm wire at 3.5 MPa can be observed. Contrarily to the twisted tape performance, where the increase in the thermal efficiency and the associate pressure drop increase are strictly inter-related, the thermal efficiency of wires is practically independent of the wire pitch, while pressure drop is inversely related to it. This latter can therefore be properly reduced decreasing the pitch, without affecting the thermal performance. The effect of the pressure on the wires efficiency is observed to be negative, in the sense that at a pressure of 5.0 MPa it drops to only 30%.

4.1.2 Extended surfaces
Kovalevev (1976) investigated three different fin design and obtained up to a factor of 10 increase in the CHF for low velocity (0.021 to 0.14 m/s) subcooled flow in an annulus. A thorough review on the use of fins has been given by Boyd (1985a).

A very intriguing technique using fins, but placed perpendicular to the fluid flow (in subcooled flow boiling), is the so-called hypervapotron technique. From a physical viewpoint, the hypervapotron effect consists of the following succession of events. The liquid inside two adjacent fins of high conductivity material and in contact with the heated wall starts boiling while the fluid bulk outside the fins is under subcooled conditions. Once the slot is full of steam, this latter undergoes a quick condensation in the subcooled liquid bulk, emptying the slots and making their replenishment with cold liquid easier. The heated wall is rewetted until the wall temperature during the uncovered phase is below the Leidenfrost temperature. The base of the fin is allowed to operate at a temperature greater than the CHF temperature while the remaining portion operates near the temperature for the onset of stable nucleate boiling. This continuous boiling and condensation sequence (frequency between 10 and 40 Hz) allows to get a high CHF, essentially on the basis of the transport of the latent heat extracted from the heated wall during boiling and transferred to the coolant outside the fins during condensation. Cattadori et al. (1993) obtained a maximum CHF of 29.4 MW/m². A typical picture from visualized tests is reported in Fig. 4-3.

4.1.3 Helically coiled tubes
Use of helically coiled tubes to get higher CHF values has been experienced by various researchers, such as Jensen & Bergles (1981), Berthoud & Jayanti (1990), Kaji et al. (1995) among the others. In the coiled tube, the liquid film thickness distribution is nonuniform around the tube circumference. But, due to the secondary flow caused by centrifugal forces, the entrainment rate of liquid droplets from the inside to the outside of the coil is large and the liquid film flows around the circumference. This may cause the dryout quality and therefore the critical heat flux to increase in the coiled tube.
Fig. 4-2 Swirl flow CHF data using helically coiled wires, Celata et al. (1994b)
4.2 Post CHF Heat Transfer Enhancement Techniques

Swirl flow promoters such as twisted tapes are also an effective enhancement technique for the augmentation of heat transfer in the post-CHF region of two-phase flow. Here the mechanism for the augmentation includes the effect of the radial velocity concentrating liquid, from the center of the flow stream, at the heat transfer surface. The first experiment of swirl flow post-CHF heat transfer was conducted by Bergles et al. (1971). They proposed a correlation, which is very complex, for the heat transfer coefficient:

\[
\alpha = C \left( \frac{T_h}{T_w} \right)^{0.32} \left[ 1 + 0.25 \frac{Gr_g^{0.5}}{Re_g} \right]^{1/4} \left[ \frac{(1-x)(6/\pi)}{x(\rho/\rho_g - 1) + 1} \right]^{2/3} \left( \frac{\pi}{4} \right) Z
\]

where

\[
Re_g = \frac{\dot{m}D}{\eta_g} \left[ x + (1-x)\left( \frac{\rho_g}{\rho_l} \right) \right]
\]

\[
C = 0.021 \left[ 1 + \frac{0.035 \pi^2}{D(TTR)^2 (1 + \pi^2/4(TTR)^2)} \right]
\]

\[
h_{lg}^* = h_{lg} \left[ 1 + \frac{0.35 c_{lg} (T_w - T_{sat})}{h_{lg}} \right]^{-3}
\]

\(Gr_g\) is the Grashof number of the gas phase, and \(Z\) is an empirically determined constant related to droplet size, while \(D\) is in feet. For film boiling of nitrogen at low mass velocity and at a reduced pressure of about 0.045, a value of \(Z = 7\) gave satisfactory results.

The post-CHF heat transfer in helical coiled tubes has been studied by Chen & Zhou (1986). Here the heat transfer coefficient in the helical coiled tube is higher than in the straight tube, also because of the action of the secondary flow and the deposition of liquid droplets in the vapour core which continuously we the place of dryout.
Fig. 4-3 Visualization of the hypervapotron effect, Cattadori et al. (1993)
Nomenclature

\begin{itemize}
\item \( A \) \hspace{1em} \text{heated surface}
\item \( a_t \) \hspace{1em} \text{centrifugal acceleration}
\item \( C_D \) \hspace{1em} \text{drag coefficient}
\item \( CHF \) \hspace{1em} \text{critical heat flux}
\item \( c_p \) \hspace{1em} \text{specific heat}
\item \( D \) \hspace{1em} \text{diameter}
\item \( D_h \) \hspace{1em} \text{equivalent hydraulic diameter}
\item \( DR \) \hspace{1em} \text{deposition rate}
\item \( ER \) \hspace{1em} \text{entrainment rate}
\item \( f \) \hspace{1em} \text{friction factor}
\item \( F_c \) \hspace{1em} \text{shape factor}
\item \( g \) \hspace{1em} \text{gravitational acceleration}
\item \( h, \Delta h \) \hspace{1em} \text{enthalpy, enthalpy difference}
\item \( h_{lg} \) \hspace{1em} \text{latent heat of vaporization}
\item \( I.D., O.D. \) \hspace{1em} \text{inlet and outlet diameter in annulus}
\item \( L \) \hspace{1em} \text{length, heated length}
\item \( l_b \) \hspace{1em} \text{boiling length}
\item \( M \) \hspace{1em} \text{mass flow rate}
\item \( \dot{m} \) \hspace{1em} \text{mass flux}
\item \( \dot{m}_{lf} \) \hspace{1em} \text{liquid film mass flux}
\item \( p \) \hspace{1em} \text{pressure}
\item \( p_c \) \hspace{1em} \text{critical pressure}
\item \( \dot{Q} \) \hspace{1em} \text{critical power}
\item \( \dot{q} \) \hspace{1em} \text{heat flux}
\item \( R \) \hspace{1em} \text{radius}
\item \( s^* \) \hspace{1em} \text{superheated layer}
\item \( T, \Delta T \) \hspace{1em} \text{temperature, temperature difference}
\item \( TTR \) \hspace{1em} \text{twisted tape ratio}
\item \( u \) \hspace{1em} \text{velocity}
\item \( u_r \) \hspace{1em} \text{resultant water velocity, with twisted tape}
\item \( u_T \) \hspace{1em} \text{friction velocity}
\item \( x \) \hspace{1em} \text{steam quality}
\item \( X \) \hspace{1em} \text{mixture composition in the liquid phase}
\item \( Y \) \hspace{1em} \text{mixture composition in the vapour phase}
\item \( z \) \hspace{1em} \text{axial distance}
\end{itemize}

Non-dimensional numbers

\begin{itemize}
\item \( Bo \) \hspace{1em} \text{Boiling number}
\item \( Fr \) \hspace{1em} \text{Froude number}
\item \( Gr \) \hspace{1em} \text{Grashof number}
\item \( Ja \) \hspace{1em} \text{Jakob number}
\item \( Nu \) \hspace{1em} \text{Nusselt number}
\item \( Pr \) \hspace{1em} \text{Prandtl number}
\item \( Re \) \hspace{1em} \text{Reynolds number}
\end{itemize}
Greek symbols

$\alpha$  heat transfer coefficient
$\beta$  contact angle
$\delta$  liquid layer thickness
$\varepsilon$  void fraction
$\phi$  angle
$\eta$  dynamic viscosity
$\dot{\phi}$  centrifugal acceleration (non dimensional)
$\lambda$  thermal conductivity
$\theta$  radial acceleration
$\rho$  density
$\sigma$  surface tension

Subscripts

$B$  pertinent to the blanket
$b$  bulk
$cal$  calculated
$CHF$  critical heat flux
$ex$  exit condition
$exp$  experimental
$g$  vapor
$in$  inlet
$l$  liquid, fluid
$m$  mean value
$nu$  non uniform
$sat$  saturated value
$st$  smooth tube
$sub$  subcooled condition
$tw$  twisted tape
$u$  uniform
$w$  wall
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