4. GENERAL FILM BOILING HEAT TRANSFER PREDICTION METHODS FOR ADVANCED WATER COOLED REACTORS

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NOMENCLATURE

A	area of surface;
C_p	specific heat;
C	concentration;
D	hydraulic diameter;
d	drop diameter;
F	empirical function;
G	mass flux;
g	acceleration of gravity;
h	enthalpy;
l	length
Nu	Nusselt number;
P	pressure;
Pr	Prandtl number;
q	heat flux;
Re	Reynolds number;
r	latent heat of evaporation;
S	velocity slip ratio; pitch of rod bundles;
T	temperature;
t	time;
V	specific volume;
X	mass quality;
X_a	actual quality;

Greek symbols

Π	perimeter;
ΔT_{min}	$\Delta T_{min} = T_{min} - T_s$;
α	heat transfer coefficient
ε	emissivity
λ	thermal conductivity;
λ_c	critical wave length;
φ	void fraction;
μ	dynamic viscosity;
ρ	density;
σ	surface tension;
σ_o	Stefan-Boltzman constant;
Θ	inclination angle in degrees

Subscripts

a	actual;
c	critical; critical heat flux
cr, CHF	critical heat flux;
e	equilibrium;
f	front of wetting;
g	gas (vapour);
h	hydraulic; heat;
ℓ	liquid;
min	minimum;
Q	quench;
s	saturation;
st	stabilisation;
sub	subcooled;
tot	total;
TB	transition boiling;
v	vapour;
vd	vapour-to-drop;
w	wall;
wv	wall-to-vapour;
wd	wall-to-drop;

ABREVIATIONS

AWR	advanced water reactor;
CHF	critical heat flux;
CRP	coordinating research program;
DFFB	dispersed film flow boiling;
ECC	emergency cooling of core;
IAFB	inverted annular film boiling;
LOCA	loss of coolant accident;
LWR	light water reactor
MFBT	minimum film boiling temperature;
MHF	minimum heat flux;
PDO	post-dryout heat transfer;
QF	quench front
RCM	research coordinating meeting;

4.1. INTRODUCTION

Post-CHF (or post-dryout) heat transfer is encountered when the surface temperature becomes too high to maintain a continuous liquid contact, and the surface becomes covered by a continuous or intermittent vapour blanket. Post-CHF heat transfer includes transition boiling, where intermittent wetting of the heated surface takes place, and film boiling, where the heated surface is too hot to permit liquid contact. The boundary between these post-CHF heat transfer modes is the minimum film boiling temperature, or T_{MFB}. Due to the poor heat transport properties of the vapour, high heated surface temperatures are often encountered during film boiling.

Although nuclear reactors normally operate at conditions where dryout does not occur, accidents can be postulated where dryout occurrence is possible. The most serious of the postulated accidents is thought to be the loss-of-coolant accident (LOCA) caused by a rupture in the primary coolant system. Accurate prediction of the consequences of a LOCA requires precise calculation of fuel-coolant heat transfer during (i) the blowdown phase (when the fuel channel is voided), and (ii) the subsequent emergency-core-cooling (ECC) phase. Although the time-in-dryout may be short, nevertheless this interval, when the primary mode of heat transfer is film boiling, can be of crucial importance in maintaining core integrity.

The post-CHF cladding temperature can be predicted from empirical correlation or from theoretical models. Since theoretical models are rather complex and the physical mechanisms on which they are based are not yet fully understood, predictions are usually based on empirical correlations. The main three methodologies considered by IPPE, AECL and CIAE have been presented in this chapter.

Film boiling heat transfer has been extensively investigated during the past 30 years. Excellent reviews may be found in text books by Tong (1965), Collier (1980), Delhaye et al. (1981), Stryikovitch et al. (1982), a handbook by Hetsroni (1982), and articles by Ganic et al. (1977), Mayinger (1978), Tong (1978), Sergeev (1978, 1987), Groeneveld and Snoek (1986), Groeneveld (1992), Yadigaroglu (1989), Sakurai (1990a), Andreoni and Yadigaroglu (1994) and in the proceedings of the 1st International Symposium on Fundamental Aspects of Post-CHF Heat Transfer (1984).

The objective of this chapter is to review and recommend film boiling prediction methods suitable for the assessment of LOCAs and other disruptive accidents in AWCRs and for implementation into systems codes such as RELAP, CATHARE, and CATHENA, as well as subchannel codes such as COBRA, ASSERT, and MIF. The requirements for this prediction method have been discussed in more detail in CRP RCM meetings and expert meetings (IAEA, 1994, 1996, 1997).

This chapter is subdivided as follows:

(i)   Section 4.2 discusses the mechanisms of the post-CHF heat transfer.
(ii)   Section 4.3 describes the film boiling data base in tubes and rod bundles.
(iii)   Section 4.4 provides an overview of the prediction methodology for film boiling heat transfer.
(iv)   Section 4.5 presents the recommended prediction methods for film boiling heat-transfer.

(v)   Section 4.6 discusses the film boiling prediction methodologies used in reactor safety codes
(vi)   Finally Section 4.7 provides final remarks related to the use of film boiling prediction methods in the thermal analysis of advanced water cooled reactors.

4.2. DESCRIPTION OF POST-CHF PHENOMENA

4.2.1. General

Post-CHF heat transfer is encountered when the surface temperature becomes too high to maintain a continuous liquid contact. As a result the heated surface becomes covered by a continuous vapour blanket as is the case in the film boiling regime, or an intermittent vapour blanket, as is the case in the transition boiling regime. The boundary between these post-CHF heat transfer modes is the minimum film boiling temperature, or T_{MFB}.

Post-CHF heat transfer is initiated as soon as the critical heat flux condition is exceeded; it persists until quenching or rewetting of the surface occurs. Depending on the particular scenario and flow conditions present, various heat transfer modes of the boiling curve of Figure 4.1 may be distributed along a heated surface, or a series of heat transfer modes can succeed each other in time at the same location as is the case during transients.

The occurrence of film boiling depends on surface temperature and flow conditions. Figure 4.2 is a three-dimensional representation of the variation of the heat flux with wall temperature and quality at constant mass flux and pressure, the so-called boiling surface concept described by Nelson (1975) and Collier (1980). The flow quality introduces a third dimension to the problem that was not present in pool boiling. This 3-D boiling surface or map shows the nucleate, transition and film boiling surfaces (regimes) as well as the critical and minimum heat flux lines for a given pressure and mass flux.

The post-CHF heat transfer modes in flow boiling can be classified as:

(i)   transition boiling (also referred to as "sputtering")
(ii)   inverted-annular film boiling (IAFB) associated with subcooled or low quality flow
(iii)   dispersed-flow film boiling (DFFB) associated with intermediate and high quality flow.

In the following sections concise descriptions of the mechanisms controlling these post-CHF heat transfer regimes will be presented.

4.2.2. Transition boiling

As the name implies, transition boiling is an intermediate boiling region. Berenson (1962) has provided a concise description of the transition boiling mechanism: "Transition boiling is a combination of unstable film boiling and unstable nucleate boiling alternately existing at any given location on a heating surface. The variation in heat transfer rate with temperature is primarily a result of a change in the fraction of time each boiling regime exists at a given location.

The transition boiling section of the boiling curve is bounded by the critical heat flux (Figure 4.3) and the minimum heat flux. The critical heat flux has been extensively studied

and can be predicted by a variety of correlations. The minimum heat flux has undergone less study; it is known to be affected by flow, pressure, fluid properties and heated surface properties and will be discussed in Section 4.2.3.

At surface temperatures in excess of the CHF temperature, the heated surface will be partially covered with unstable vapour patches, varying with space and time. Ellion (1954) studied forced convective transition boiling in subcooled water and observed frequent replacement of vapour patches by liquid. Although this may seem similar to transition pool boiling as described above, the introduction of the convective component will improve the film boiling component by reducing the vapour film thickness and changing the heat transfer mode, whether dry or wet, from free convection to forced convection. This will result in an increase in q_{min} and also can increase ΔT_{mfb} (if ΔT_{mfb} is hydrodynamically controlled). For low qualities and subcooled conditions the slope of the transition boiling is always negative, just as in pool boiling.

The amount of heat transfer in the transition boiling region is primarily governed by liquid-solid contact. At the critical heat flux point the contact-area (or time) fraction F is close to unity and, therefore, the liquid contact heat flux q_ℓ is close to the CHF. The value of F strongly decreases with increasing wall temperature. In the high quality region for example, most of the heat transferred during transition boiling will be due to droplet-wall interaction. Initially, at surface temperatures just in excess of the boiling crisis temperature, a significant fraction of the droplets will deposit on the heated surface but at higher wall superheats the vapour repulsion forces become significant in repelling most of the droplets before they can contact the heated surface. The repelled droplets will contribute to the heat transfer by disturbing the boundary layer sufficiently to enhance the heat transfer to the vapour.

The periodic contacts between liquid and heated surface in the transition boiling region of the boiling curve result in the formation of both large amounts of vapour, which forces liquid away from the surface, and creates an unstable vapour film or blanket. Because of this, the surface heat flux and the surface temperature can experience variations both with time, and position on a heater. However, the average heat transfer coefficient decreases as the temperature increases, because the time of contact between the liquid and the heater surface is decreased.

To gain a better understanding of the transition boiling mechanism, the phenomena occurring at the interface between fluid and a heated surface (i.e. the mechanism of fluid-solid contact including the frequency of this contact; heat transfer in the contact areas; time history of such contact) need to be considered. Comprehensive reviews of these phenomena have been presented by Kalinin et al. (1987) and Auracher (1987, 1990).

Transition boiling has received less attention than nucleate or film boiling. Only in recent years has the interest in this boiling regime increased because of its potential importance during a LOCA in a nuclear reactors. Overviews of the mechanisms and prediction methods for transition boiling have been provided by Bankoff and Mehra (1962), Groeneveld and Fung (1976), Auracher (1987, 1990), Winterton (1982), Groeneveld and Snoek (1986) and Johannsen (1991).

4.2.3. Minimum film boiling temperature

The minimum film boiling temperature (T_{MFB}) separates the high temperature region where inefficient film boiling or vapour cooling takes place, from the lower-temperature region, where much more efficient transition boiling occurs. It thus provides a limit to the application of transition boiling and film boiling correlations. Knowledge of the minimum film boiling temperature is particularly important in reactor safety assessments.

A large number of terms have been used for the minimum film boiling temperature or T_{MFB}. They include rewetting temperature, quench temperature, Leidenfrost temperature, film boiling collapse temperature and others etc.

During quenching of a surface (such as emergency core cooling), rewetting commences at the minimum film boiling temperature and, as a rule, rapidly proceeds until nucleate boiling is established at a much lower wall temperature. Predicting the minimum film boiling temperature as a function of the system parameters is thus very important since heat transfer coefficients on either side of the minimum film boiling temperature can differ by orders of magnitude. Generally, T_{MFB} is defined as the temperature at the minimum heat flux.

The T_{MFB} also represents a temperature boundary beyond which surface properties and surface conditions generally do not affect the heat transfer. Wettability or contact angle although important in nucleate and transition boiling, are not applicable in the film boiling regime, and conduction along the surface becomes less important when nucleate and film boiling no longer occur side-by-side.

Two theories have been proposed for the analytical prediction of the minimum film boiling temperature. One theory says that the minimum temperature is a thermodynamic property of the fluid (i.e., maximum liquid temperature) and thus is primarily a function of pressure. The other theory suggests that rewetting commences due to hydrodynamic instabilities which depend on the velocities, densities, and viscosities of both phases as well as the surface tension at the liquid-vapour interface. During fast transitions, where insufficient time is available to fully develop the hydrodynamic forces, rewetting is expected to be thermodynamically controlled while for low flows and low pressures, where sufficient time is available and the volumetric expansion of the fluid near the wall is large, rewetting is more likely to be hydrodynamically controlled. Once rewetting has occurred locally, the rewetting front can then propagate at a rate which is primarily controlled by axial conduction. These theories can be modified to include the thermal properties of the surface.

There is no general consensus on the effect of the various system parameters on the minimum film boiling temperature under forced convective conditions. These effects are included in correlations for the minimum temperature which have been tabulated by Groeneveld and Snoek (1986).

4.2.4. Flow film boiling

4.2.4.1. General

Film boiling is generally defined as that mode of boiling heat transfer where only the vapour phase is in contact with the heated surface. The term film boiling was originally applied to pool boiling where the stagnant liquid was separated from the heated surface by a vapour film. The term has been used in forced convective boiling to refer to conditions

where the liquid does not contact the heated surface but is usually in one of the following forms:

(i)   a dispersed spray of droplets, normally encountered at void fractions in excess of 80% (liquid-deficient or dispersed flow film boiling regime),

(ii)   a continuous liquid core (surrounded by a vapour annulus which may contain entrained droplets) usually encountered at void fractions below 40% (inverted annular film boiling or IAFB regime)

(iii)   a transition between the above two cases, which can be in the form of an inverted slug flow for low to medium flow.

Figure 4.3 illustrates the above flow regimes. Of these, the dispersed flow film boiling (DFFB) regime is most commonly encountered and has been well studied. Its heated surface temperature is moderate while in the inverted annular and the inverted slug flow regimes, excessive surface temperatures are frequently encountered.

Radiation heat transfer, although unimportant in transition boiling, becomes increasingly important in film boiling, particularly at low flows, low void fractions and surface temperatures in excess of 700°C.

The main parameters controlling the film boiling heat transfer are: pressure, equilibrium quality (or subcooling), and mass flux. At low flows, strong non-equilibrium effects can be present which will need to be considered. In addition at locations just downstream of (or "just subsequent to" during fast transients) the CHF or quench occurrence, upstream/history effects are important. These effects frequently are not included in film boiling models [see also Gottula et al. (1985); Shiralkar et al. (1980); Kirillov et al. (1982)].

Due to the high surface temperatures frequently encountered during film boiling with water, studies using cryogenic and refrigerant fluids and pool boiling studies have been extensively employed to improve our understanding of film boiling and to extract parametric trends and derive correlations.

Reviews of the film boiling literature have been prepared separately for the higher quality DFFB regime [Mayinger (1978); Collier (1981); Groeneveld (1975a & 1977); Andreoni and Yadigaroglu (1994)]; the IAFB regime [Groeneveld (1984, 1992); Andreoni and Yadigaroglu (1974)] and for pool film boiling [Hsu (1972); Kalinin (1987)].

4.2.4.2. Inverted annular film boiling

IAFB refers to the film boiling type characterized by a vapour layer separating the continuous liquid core from the heated surface. Figure 4.3 (RHS) shows schematically the phase distribution during IAFB. IAFB resembles pool film boiling superficially, but the actual heat transfer mechanisms are considerably more complex.

In the inverted annular flow regime few entrained droplets are present while the bulk of the liquid is in the form of a continuous liquid core which may contain entrained bubbles. At dryout the continuous liquid core becomes separated from the wall by a low viscosity vapour layer which can accommodate steep velocity gradients. However, the velocity distribution across the liquid core is fairly uniform. Once a stable vapour blanket has formed, the heat is transferred from the wall to the vapour and subsequently from the vapour to the wavy liquid core. Initially, for very thin vapour films, heat transfer from the wall to the liquid is primarily by conduction across a laminar vapour film. When the vapour film thickness increases, turbulent flow will occur in the film, and the liquid-vapour interface becomes agitated. Heat transfer across the wavy vapour-liquid interface takes place by forced convection. This mode of heat transfer is much more efficient than the single-phase convective heat transfer between a smooth wall and the vapour; hence it is assumed that the bulk of the vapour is at or close to the liquid core temperature (i.e., saturation temperature). The low-viscosity, low-density vapour-flow experiences a higher acceleration than the dense core flow. This results in an increased velocity differential across the interface which may lead to liquid entrainment from the wavy interface. It may also lead to more interaction of the liquid core with the heated surface through dry collisions and will increase the turbulence level in the vapour annulus. The resulting increase in wall-vapour and wall-core heat transfer will lower the wall temperature; if the wall temperature drops below the minimum film boiling temperature rewetting may occur. Rewetting can also occur at higher temperatures if it is caused by a propagating rewetting front.

Modeling of IAFB requires proper relationships for the interfacial heat and momentum transfer between the superheated vapour blanket and the subcooled or saturated liquid core. The net interfacial heat transfer determines the rate of vapour generation and, therefore, the film thickness.

The heat transfer process in IAFB can be considered by the following heat flux components:

(i)   Convective heat transfer from the wall to vapor ($q_{w,v}$);
(ii)   Radiation heat transfer from the wall to liquid (q_{rad});
(iii)   Heat transfer from vapor to the vapor-liquid interface ($q_{v,i}$);
(iv)   Heat transfer from the vapor-liquid interface to the liquid core ($q_{i,l}$).

In the case of subcooled film boiling, the last heat flux component is used for both vaporization and reducing liquid subcooling. For saturated liquid, $q_{i,l}$ is used only for vaporization, thus increasing the vapor film thickness more rapidly.

A significant increase in heat transfer coefficient with an increase in liquid subcooling has generally been observed in pool film boiling and flow film boiling [e.g. see Groeneveld (1992)]. The effect of subcooling on the film boiling heat transfer coefficient may be explained as follows: heat is transferred primarily by conduction across a thin vapour film to the interface (convection and radiation may also be significant). Here a fraction of the heat received is used for heating up the liquid core, while the remainder is used for evaporation.

Higher subcoolings results in less evaporation, and hence a thinner vapour film, which consequently increases the heat transfer coefficient h. During tests on heated bodies immersed in water, Bradfield (1967) observed that subcooled film boiling with subcoolings less than 35°C resulted in a calmer interface with a wavelike motion compared to saturated boiling. Most experimental studies show an increase in h with an increase in X_e at the high mass velocities (G > 1000 kg/m^{2}s at P > 6 MPa [Stewart (1981); Laperriere and Groeneveld (1984)] and G > 100 kg/m^{2}s at P = 0.1 MPa, [Fung (1981)] although at times this increase in h may not be evident near zero qualities). At lower mass velocities, a decrease in h (= $q_w/(T_w-T_s)$) with an increase in X_e is frequently observed. The above effect is due to the gradual thickening of the vapour film with increasing X_e. This will increase the resistance to conduction heat transfer which may still be dominant at low G and X_e values. It also increases the convective heat transfer coefficient, defined as $h_c = q/T_w-T_v$). Since at low flows the vapour temperature T_v may rise significantly above saturation, the T_w may still increase despite the increase in h_c. At high mass velocities (G > 2000 kgm^{-2}s^{-1}) T_v is usually near saturation and h generally increases with X_e. With an increase in quality or void fraction the IAFB regime breaks up at void fraction of about 30-60% and the transition to the DFFB regime occurs.

The recent reviews on IAFB published by Groeneveld (1992), Johannsen (1991) and Hammouda (1996) include description or tabulations of new or modified models for IAFB heat transfer related to reflood heat transfer of water-cooled nuclear reactors. These reviews are based on publications by Analytis et al. (1987), Klyugel et al. (1986), Mosaad (1986), Hsu et al. (1986), Wang et al (1987,1988), Yan (1987), and Lee et al. (1987).

4.2.4.3. Slug flow film boiling

Slug flow film boiling is usually encountered at low flows and void fractions which are too high to maintain inverted annular film boiling but too low to maintain dispersed flow film boiling. In tubes, it is formed just downstream of the inverted annular flow regime when the liquid core breaks up into slugs of liquid in a vapour matrix. The prediction of the occurrence of slug flow during bottom flooding ECC is important because of the change in heat transfer rate long before the arrival of the quench front.

Several theories for the break-up of the IAFB regime have been proposed. Data of Chi (1967) suggest that the liquid core will break up into slugs which are equal in length to the most unstable wavelength of interfacial waves. Subcooling tends to stabilize the liquid-vapour interface, and thus inhibits the formation of slug flow. Smith (1976) assumes the location of slug flow to correspond to the point of minimum heat transfer coefficient in the film boiling region. In doing so, he is suggesting that if the vapour velocity is high enough to break up the liquid core, then it is also high enough to considerably improve the heat transfer coefficient. Kalinin (1969) observed another possible mechanism for the onset of slug flow in transient tests. Immediately after the introduction of liquid to their test section, the sudden increase in void due to vapour generation at the leading edge of the liquid caused a back pressure which decelerated the flow. The higher pressure and lower flow rate caused a decrease in vaporization and the flow surges forward. The cycle was repetitive with a liquid slug separating from the liquid core with each cycle.

4.2.4.4. Dispersed flow film boiling (DFFB)

The DFFB regime is characterized by the existence of discrete liquid drops entrained in a continuous vapor flow. This flow regime may be defined as dispersed flow film boiling, liquid deficient heat transfer, or mist flow. It is of importance in nuclear reactor cores for off-normal conditions such as the blowdown or ECCS phase of a LOCA, as well as in steam generators.

The DFFB regime usually occurs at void fractions in excess of 40%. No exact lower-bound value for the onset of DFFB is available as the transition from IAFB or slug flow film boiling is likely to be gradual. According to Levitan and Borevskiy (1989), the beginning of the dispersed regime is determined by the following correlation

$$X_{ad} = (2.7 \pm 0.3)\left(\frac{\rho_\ell \cdot \sigma}{G^2 \cdot d}\right)^{1/4}\left(\frac{\rho_v}{\rho_\ell}\right)^{1/3} \tag{4.1}$$

where

X_{ad} represents the onset of annular dispersed flow.

In the DFFB regime the vapour temperature is controlled by wall-vapour and vapour-droplet heat exchange. Due to the low superheat of the vapour near the dryout location or rewetting front the vapour droplet heat exchange is small and most of the heat transferred from the wall is used for superheating the vapour. At distances further downstream, however, an "equilibrium" vapour superheat can be reached, i.e., the amount of heat transferred from the wall to the vapour may approximately balance the amount of heat absorbed by the droplets (from the vapour) and used for evaporation of the droplets.

Near the heated surface the heat exchange between vapour and droplets is enhanced due to the temperature in the thermal boundary layer being well above that of the vapour core [Cumo and Farello (1967)]. If the temperature of the heated surface is below the minimum temperature, some wetting of the wall may occur resulting in an appreciable fraction of the droplets being evaporated [Wachters (1965)]. At temperatures above the minimum temperature only dry collisions can take place (collisions where a vapour blanket is always present between surface and droplet). Little heat transfer takes place to small droplets which resist deformation and bounce back soon following a dry collision [Wachters (1965); Bennett et al. (1967)]. However, the dry collisions disturb the boundary layer thus improving the wall-vapour heat transfer. Larger droplets are much more deformable and tend to spread considerably thus improving both the wall-vapour and vapour-droplet heat exchange [Cumo and Farello (1967); Wachters (1965)]. This spreading may lead to a breakup into many smaller droplets if the impact velocity is sufficiently high [McGinnis and Holman (1969)]. The vapour film thickness separating the stagnated droplets from the heated surface is difficult to estimate but must be greater than the mean free path of the vapour molecules in order to physically separate the liquid from the heated surface.

Attempts to evaluate the direct heat flux to the droplets due to interaction with the heated surface have resulted in the postulation of many simplifying assumptions, e.g., Bailey (1972), Groeneveld (1972), Plummer et al. (1976). These assumptions may be questionable when applied to liquid deficient cooling. However, due to lack of direct measurement of droplet-wall interaction during forced-convective film boiling conditions no other approach can be taken.

The heat flux encountered during DFFB can be partitioned as follows:

(i)   Heat transfer from wall to liquid droplets which reach the thermal boundary layer without wetting the wall (dry collisions) - q_{wdd};
(ii)   Heat transfer from wall to liquid droplets which temporarily wet wall (wet collisions)- q_{wdw};
(iii) Convective heat transfer from wall to vapor - q_{wv};
(iv)   Convective heat transfer from steam to droplets in the vapor core - q_{vd};
(v)   Radiation heat transfer from wall to liquid droplets - q_{rad};
(vi)   Radiation heat transfer from wall to vapor - $q_{rad,v}$;

The most important unknown in DFFB is the thermal non-equilibrium or vapour superheat. The vapour superheat increases with heat flux (its main driving force) and decreases with interfacial area and interfacial drag. Both the interfacial area and the interfacial drag are dependent hydrodynamic parameters controlled by the dynamics of interfacial shear, droplet generation, break-up, and coalescence mechanisms, and evaporation history. There are basic difficulties in determining experimentally important parameters such as the interfacial drag coefficient. Since the spectrum of droplet sizes may vary from case to case, and the closure laws depend on droplet diameter, the formulation of universally valid closure laws is difficult. This has been investigated in more detail analytically and experimentally by Andreoni and Yadigaroglu (1991, 1991a, 1992), and Kirillov and Smogalev (1973).

4.3. FILM BOILING DATA BASE

4.3.1. General

Because of the importance of film boiling heat transfer and reactor accident analysis, there has been a significant interest in providing a good film boiling data base for reactor conditions of interest. The high CHF and generally low heat transfer coefficients in film boiling results in high surface temperatures and this restricts the range of conditions at which measurements are feasible under steady state conditions. Hence many of the earlier experimental data were obtained in cryogenics and refrigerants, in temperature controlled systems [Smith (1976); Ellion (1954)] or from transient tests [Newbold et al. (1976); Cheng and Ng (1976); Fung (1977)]. However, a novel approach has been developed at Chalk River for obtaining subcooled film boiling data [Groeneveld and Gardiner (1978)]. Using the so-called hot-patch technique steady-state subcooled and low-quality film boiling data can be obtained in a heat flux controlled system at heat flux levels well below the CHF. This approach has permitted a much more extensive study of film boiling especially at IAFB conditions [e.g. Stewart (1981); Fung (1981); LaPerriere and Groeneveld (1984); Gottula et al. (1985); Johannsen (1991)].

4.3.2. Tube and annuli

Tables 4.1 and 4.2 summarize the test conditions of film boiling data obtained in tubes and annuli, respectively. Although the coverage is extensive, there is still a scarcity of film boiling data at low pressures and low flows. Recent data obtained by CIAE have helped to resolve this lack of data [Chen and Chen (1998)].

4.3.3. Bundle

Table 4.3 summarizes the film boiling data available for rod bundles. Many other bundle data have been obtained but these are inaccessible because of their potential commercial value and because of licensing concerns. The film boiling bundle data base is more limited than the CHF bundle data base because of the higher temperatures which makes testing much more difficult. The hot patch approach, used successfully in tubes, cannot be used in bundles and this further restricts this data base.

4.4. OVERVIEW OF FILM BOILING PREDICTION METHODS

4.4.1. General

Accurate prediction of the wall temperature in the film boiling regime is of vital importance in accident analysis of the core and steam generators of advanced water cooled reactors. The following four methods for estimating the film boiling heat transfer are commonly used:

(i) Semi-theoretical equations for pool film boiling (Section 4.4.2);
(ii) Semi-theoretical models to predict flow film boiling. They are based on the appropriate constitutive equations, some of which are empirical in nature;
(iii) Purely empirical correlations for flow film boiling, which do not account for any of the physics, but instead assume a forced convective type correlation;
(iv) Phenomenological equations for flow film boiling, which account for the thermal non-equilibrium and attempt to predict the "true" vapour quality and the vapour temperature.

Because of the proliferation of film boiling prediction methods (there are currently over 20 film boiling models available and well over 50 correlations) tabular methods have recently been proposed. Tabular methods are well accepted for the prediction of CHF and are based more closely on experimental data. They will be discussed in Section 4.4.5.

4.4.2. Pool film boiling equations

4.4.2.1. Horizontal surfaces

Pool film boiling from a horizontal surface has been investigated for over 50 years, and can be reasonably well represented by analytical solutions. Most pool film boiling and low flow film boiling prediction methods [e.g., Bromley (1950); Borishanskiy (1959, 1964); Berenson (1961)] are of the following form:

$$\alpha = C \cdot \left[\frac{\lambda_v^3 \cdot r^* \cdot \rho_v (\rho_\ell - \rho_v)}{\mu_v \cdot DT} \right]^{1/4} \cdot F(U, l) + \alpha_{rad}(\varepsilon, T_W, T_S) \tag{4.2}$$

where

r^* is an equivalent latent heat and includes the effect of vapour superheat, sometimes expressed as $r^* = r + 0.5(C_p)_v \Delta T$. The velocity effect on the heat transfer coefficient is taken into account by the F-function. The symbol l represents either a characteristic length (e.g. diameter) of the surface or the critical wave length which is usually defined as:

$$l = \sqrt{\frac{\sigma}{g(\rho_\ell - \rho_v)}} \qquad (4.3)$$

Other relations for pool film boiling have been proposed by Epstein and Hauser (1980), Klimenko (1981), Dhir (1990), and Sakurai (1990a, 1990b). Table 4.4 gives the correlations for the film boiling heat transfer on horizontal surfaces in pool boiling based on the following dimensional groups

$$Nu = \alpha l / \lambda_v \qquad (4.4)$$

$$Ra = \frac{g \cdot l^3 \cdot (\rho_\ell - \rho_v)}{\rho_v \cdot v_v^2} \cdot Pr \qquad (4.5)$$

4.4.2.2. Vertical surfaces

Saturated pool film boiling on vertical surfaces has been investigated experimentally and theoretically by many researchers including Hsu, and Westwater (1960), Suryanarayana, and Merte (1972), Leonardo, and Sun (1976), Andersen (1976), Bui, and Dhir (1985). Frequently equations similar as those for horizontal surfaces are proposed for vertical surfaces; the main difference is usually in the constant C in front of the equation and the characteristic length. Sakurai (1990a, 1990b, 1990c) developed new equations for film boiling heat transfer on surfaces with different configurations. In particular, correlations for vertical plates and tubes, spheres and horizontal plates were derived by the same procedure as that used for horizontal cylinders. The latter was derived by slightly modifying the corresponding analytical solution to get agreement with the experimental results.

4.4.2.3. Downward-facing surfaces

Recent experiments by Kaljakin et al. (1995) on curved down-facing surfaces have demonstrated that in many cases the heat transfer coefficient prediction for pool film boiling or for low mass velocities can be based on the modified Bromley formula (1950) :

$$\alpha_x = \beta \sqrt{\frac{\lambda_v^3 \rho_v g \cdot (\rho_\ell - \rho_v) r^*}{\mu_v \Delta T \cdot x}} \qquad (4.6)$$

where

$\beta = 0.8 + 0.0022\Theta$; Θ is a surface inclination angle, in degrees;
$\Delta T = T_w - T_s$; is a characteristic length along the vessel surface, and
r^* is defined as in Equation 4.2.

Eq. 4.6 is reportedly valid for a pressure range of about 0.1 - 0.2 MPa.

4.4.3. Flow film boiling models

4.4.3.1. General

The first flow film boiling models were developed for the DFFB regime. In these models, all parameters were initially evaluated at the dryout location. It was assumed that heat transfer takes place in two steps: (i) from the heated surface to the vapour, and (ii) from the vapour to the droplets (see also Section 4.2.4.4). The models evaluate the axial gradients in droplet diameter, vapour and droplet velocity, and pressure, from the conservation equations. Using a heat balance, the vapour superheat was then evaluated. The wall temperature was finally found from the vapour temperature using a superheated-steam heat transfer correlation. Improvements to the original model have been made by including droplet-wall interaction, by permitting a gradual change in average droplet diameter due to the break-up of droplets, and by including vapour flashing for large pressure gradients.

Subsequent to the development of models for the DFFB regime, models have also been developed for the IAFB regime. They are basically unequal-velocity, unequal-temperature (UVUT) models which can account for the non-equilibrium in both the liquid and the vapour phase. Most of the models are based on empirical relationships to predict interfacial heat and momentum transfer. Advanced thermalhydraulic codes employ similar models to simulate the post-CHF region. Universal use of film boiling models is still limited because of unresolved uncertainties in interfacial heat transfer, interfacial friction and liquid-wall interactions, as well as the difficulty in modelling the effect of grid spacers.

4.4.3.2. IAFB regime

A large number of analytical models have been developed to simulate the IAFB conditions [e.g. Analytis and Yadigaroglu (1987); Kawaji and Banerjee (1987); Denham (1983); Seok and Chang (1990); Chan and Yadigaroglu (1980); Takenaka (1989); Analytis (1990); de Cachard (1995); Mosaad (1988), Mosaad and Johannsen (1989); Hammouda, Groeneveld, and Cheng (1996)]. The salient features of many of these models have been tabulated by Groeneveld (1992) and Hammouda (1996). Table 4.5 provides an overview of some of the current IAFB models. The majority is based on two-fluid models and employ some or all of the assumptions listed below:

(i)   at the quench front the liquid is subcooled and the vapour is saturated.
(ii)   vapour will become superheated at the down stream of the quench front;
(iii)   both the vapour and liquid phases at the interface are at saturation;
(iv)   the interfacial velocity is taken as the average of the vapour and liquid velocities;
(v)   there is no entrainment of vapour in a liquid core or of the liquid in the vapor film;
(vi)   the vapour film flow and the liquid core flow are both turbulent

The above assumptions clearly indicate differences from the classical Bromley-type analysis for pool film boiling, and there is no smooth transition between these two cases.

The main challenge in implementing IAFB models into two-fluid codes resides in the proper choice of the interfacial heat and momentum exchange correlations. Interfacial heat exchange enhancements may be due to turbulence in the film, violent vaporization at the quench front, liquid contacts with the wall near the quench front, upstream grid spacers or approaching quench front, and the effect of the developing boundary layer in the vapour film. The large amount of vapour that may be generated right at the quench front (release of the heat stored in the wall due to quenching) must also be taken into account. Reflooding experiments

clearly show an exponential decay of the heat transfer coefficient with distance from the quench front for a length extending some 20 or 30 cm above the quench front.

The constitutive relations employed are based on the simplifying assumptions. In general, there are too many adjustable parameters and assumptions made by different authors which results in a multitude of IAFB models. A part of the reason is the difficulty in verifying the proposed interfacial relationships with experimental-based values. Despite this, relatively good agreement was reported by the model developers between their model prediction and the experimental data, but no independent review of their models was ever made.

During high-subcooling film boiling the vapour film at the heated surface is very thin over most of the IAFB length. Here the prediction methods or models tend to overpredict the wall temperature, presumable because the conduction-controlled heat transfer across a very thin film was not properly accounted for.

4.4.3.3. DFFB regime

Significant non-equilibrium between the liquid and vapor phases is usually present in the DFFB regime, except for the high mass velocities. Mixture models are intrinsically not able to predict this non-equilibrium and hence the need for two-fluid models. As the interfacial heat transfer is easier to determine either experimentally or analytically for the DFFB regime vs. the IAFB regime, these models tend to be somewhat more accurate than those simulating IAFB.

As discussed in Section 4.2 the heat transfer in DFFB is a two-step process, i.e. (i) wall to vapour heat transfer and (ii) vapour to entrained droplets heat transfer. Enhancement of heat transfer due to the interaction of the droplets with the heated wall are usually small except for low wall superheats, near the T_{MFB} , where transition boiling effects become important.

At high mass velocities, the droplet size is small, the interfacial area is large and the interaction between the vapor and droplets is sufficiently intensive to keep the vapor temperature close to the saturation temperature. Here a Dittus-Boelter type equation, based on the volumetric flow rate and vapour properties, provides a reasonable estimate of the overall heat transfer coefficient, and an analytical model is not required.

A large number of models have been developed for the DFFB regime. The first DFFB models were developed for the liquid deficient regime by the UKAEA [Bennett (1967)] and MIT [Laverty and Rohsenow (1967)]. In these models, all parameters were initially evaluated at the dryout location. The models evaluated at the axial gradients in droplet diameter, vapour and drop velocity, and pressure, from the conservation equations. Using a heat balance, the vapour superheat was then evaluated. The wall temperature was finally found from the vapour temperature using a superheated-steam heat transfer correlation. Bailey (1972), Groeneveld (1972), and Plummer et al. (1976) have suggested improvements to the original model by including droplet-wall interaction, by permitting a gradual change in average droplet diameter due to the break-up of droplets, and by including vapour flashing for large pressure gradients. Additional expressions for the vapour generation rate have also been suggested by Saha (1980), and Jones and Zuber (1977).

The various models tend to have the same basic structure but differ in the choice of interfacial relationships and separate effects. The following variants have been used in the models:

(i)   droplet size: based on various Weber number criteria for the initial droplet size and for subsequent break-up; Weber number may be ignored; subsequent droplet break-up is often ignored

(ii)   droplet size distribution: various assumptions have been made, e.g., constant size, gaussian distribution

(iii) droplet drag force: depends on drag coefficient and assumed shape of the droplet

(iv)   interfacial heat transfer: depends on phase velocity differential: various equations are possible

(v)   droplet-wall heat transfer q_{dw}: this may be expressed by a separate heat flux $q_{dw} = 0$ or $q_{dw} = f(T_w-T_{SAT})$; may be ignored ($q_{dw} =0$) or may be incorporated by enhancement of the convective heat transfer.

Despite these variants the agreement between the predictions of most DFFB models is quite good at steady-state conditions, and medium flows and pressures ($G = 0.3$-6 $Mgm^{-2}s^{-1}$, $P = 5$-10 MPa).

Details of the models and the equations on which they are based may be found in Andreoni and Yadigaroglu (1994), Groeneveld and Snoek (1986), Chen and Cheng (1994), and Hammouda (1996). Table 4.6 provides an overview of the major features of the DFFB models.

4.4.4. Flow film boiling correlations

4.4.4.1. IAFB correlations

For the IAFB regime many equations have been proposed, including the classic Bromley (1950) equation for the vertical surface, the Ellion (1954) equation, the Hsu and Westwater (1960) equation, the modified Bromley equation for pool film boiling: [Leonard (1978); Hsu (1975)], and various other ones. Groeneveld (1984, 1992) later updated by Hammouda (1996) have tabulated the proposed equations for IAFB. None of the proposed prediction method appears to have a wide range of application as far as flow conditions is concerned or as far as geometry is concerned. Most are derived for tube flow or for pool boiling conditions and none has been derived for application in a bundle geometry equipped with rod spacing devices. Hence caution should be exercised before applying them to AWCRs.

4.4.4.2. DFFB correlations

4.4.4.2.1. Correlations based on equilibrium conditions

Most of the equilibrium-type equations for film boiling are variants of the single-phase Dittus-Boelter type correlation. These equations were empirically derived or simply assume that there is no non-equilibrium and hence use the same basic prediction method as for superheated steam except that the Reynolds number is usually based on the homogeneous (no slip) velocity. These equations usually have a very limited range of application, or are valid only for the high mass velocity regime where non-equilibrium effects are small. The most

common correlations of this type are tabulated in Table 4.7. Among these the Dougall Rohsenow (1963), the Miropolskiy (1963) and the Groeneveld (1973) equations are the more popular ones. The latter two are both based on Miropolskiy's Y-factor as defined in Table 4.7. This factor is particularly significant at lower pressures and qualities. Groeneveld optimized his coefficients and exponents based on a separate data base for tubes, annuli and bundles.

4.4.4.2.2. Phenomenological equations based on non-equilibrium conditions

Phenomenological equations attempt to predict the degree of non-equilibrium between the liquid and vapour phase. These equations are a compromise between the empirical correlations discussed in the previous section and the film boiling models described in Section 4.4.3. The phenomenological equations generally predict an equilibrium vapour superheat corresponding to fully developed flow and based on local equilibrium conditions. They generally do not require knowledge of upstream conditions, such as location of the quench front. An overview of the phenomenological film boiling equations is given in Table 4.8.

The non-equilibrium equations are based on film boiling data for water and have been developed by Groeneveld and Delorme (1976), Plummer et al. (1977), Chen et al. (1977, 1979), Saha (1980), Sergeev (1985a), Nishikawa (1986). Most of them use the of the Dittus-Boelter type equation e.g. Equation 4.7:

$$Nu_v = \frac{a \cdot D}{l_v} = a \cdot \mathrm{Re}_v^{\,b} \cdot \mathrm{Pr}_v^{\,c} \tag{4.7}$$

where

a, b, and c are constants and α is the two-phase heat transfer coefficient in a tube with an inside diameter D.

The equations are based on a vapour Reynolds number which is usually based on the actual quality X_a instead of the equilibrium quality X_e. Some of the equations [e.g. Plummer et al. (1977)] also permit slip to exist between the phases as shown in Equation 4.8

$$\mathrm{Re}_v = \frac{G \cdot D}{\mu_v} \left[X_a + S \cdot \left(1 - X_a\right) \frac{\rho_v}{\rho_\ell} \right] \tag{4.8}$$

where S is the slip ratio which in this case depends on the degree of non-equilibrium.

However most of the phenomenological equations are based on the assumption of homogeneous flow. Further details of the equations are provided in Table 4.8.

The main difference between the various phenomenological equations is primarily in the relation between the equilibrium quality X_e and the actual quality X_a. For example Groeneveld and Delorme (1976) recommended the following relationship:

$$[X_e / X_a] - \max(1, X_e) = \exp(-\tan\psi) \tag{4.9}$$

where

$$\psi = f(\text{Re}_{v,hom}, P, q, X_e)$$

The non-equilibrium correlation developed by Plummer et al. (1977) was based an expression for $(X_a - X_{do})/(X_e - X_{do}) = f(G)$ while Tong and Young (1974) expressed $X_a/X_e = f(X_e, G)$ and Chen et al. (1977) expressed $X_a/X_e = f(P, T_w)$. Plummer based his equation on data for water, nitrogen and freon-12 and takes into account the wall-to-drop heat transfer α_{wd} as well. The heat flux from the wall to vapor and from the wall to droplets is given as,

$$q = \alpha_{wv}\left(T_w - T_v\right) + \alpha_{wd}\left(T_w - T_s\right) \tag{4.10}$$

where the heat transfer coefficient to the vapour α_{wv} is given in Table 4.8 and the wall-to-droplet heat transfer coefficient α_{wd} is given as,

$$\alpha_{wd} = \frac{\lambda_v}{\delta_f}\left(1-\varphi\right)\exp\left(-2\frac{D}{L}\right) \tag{4.11}$$

and $\delta_f = 1.2 \cdot 10^{-4}$ and the void fraction φ is based on the actual quality.

Sergeev's method (1978, 1985a, 1985b, and 1987) evaluates the wall temperature and is valid for $G \leq 1000$ kg/m^2·s; $P = 3 \div 18$ MPa; $X > X_{cr}$; $\Delta T = T_w - Ts \leq 500^{\circ}$C. It is based on a known critical quality and the assumptions that:

(i)   the radiation heat transfer coefficient is small,
(ii)   the interaction of drops with a wall is insignificant
(iii)   the heat transfer coefficient can be found from a single-phase convection equation (e.g., see Section 4.5.4).

The relation between X_e and X_a can be found by solving the following differential equation:

$$\frac{dX_a}{dX_e} = C \cdot m \frac{\lambda_v}{\sigma \cdot \rho_\ell} \cdot \frac{G^2 \Pi_w}{\sum_j \Pi_{th}} \cdot X_a \left(1 - X_a\right)\left(\frac{X_e - X_a}{X_a}\right)^n \tag{4.12}$$

where

C is an empirical constant; $C = 1.5$ for tubes, rod bundles, and annuli at the PDO regime on two surfaces; $C = 3$ for annuli at the PDO regime on one surface. Besides, m and n are functions of pressure; Π_w and Π_{th} are the wetted and thermal channel perimeters. Eq. 4.12 can be integrated from X_{cr} (at $T_{va} \equiv T_s$) to the given channel section for given X_e. This method has been used for tubes, annuli (with a gap of 2 mm and more) and rod bundles (without heat transfer enhancement due to spacing devices)

4.4.5. Look-up tables for film boiling heat transfer in tubes

The high interest in film boiling heat transfer over the past 30 years has led to a proliferation of filmboiling models and prediction methods, many of them film-boiling-

regime specific, applicable only over the range of test conditions investigated by the individual investigator. Hence it has become increasingly more difficult to select film boiling prediction methods which can be used with confidence over a wide range of conditions and geometries as will be encountered in AWCRs. In addition, these prediction methods, particularly the models and phenomenological equations, are very time consuming even with the use of fast computers. This is because of (i) frequent iteration, (ii) the large number of equations involved and (iii) evaluation of many different fluid properties during each iteration.

To simplify the film boiling prediction process, and to make it more universally applicable, the film boiling table look-up method has been developed. This approach is similar to the CHF table look-up method, and is basically a methodology which is based on a combination of all available filmboiling data and predicted values covering a very wide range of conditions. It contains a tabulation of normalized heat transfer coefficients for fully developed film-boiling at discrete values of pressure, mass flux, quality, and heat flux. Because the world's film boiling data base still has significant gaps, particularly at conditions where experiments are difficult (i.e. high surface temperatures), the tables are based partially on extrapolation using the observed trends from the better film boiling models or correlations and of known asymptotic trends. Ideally the tabulated heat transfer coefficient should be based on the wall superheat with respect to the actual vapour temperature, but since this temperature is almost always difficult to evaluate, the equilibrium vapour temperature or the saturation temperature are usually used as reference temperatures.

The look-up table method for film boiling was first suggested by Groeneveld (1988) and has since been refined into an improved method [Leung et al. (1997)], based on over 15 000 film boiling data for a wider range of conditions. Leung's most recent look-up table is given in Appendix IV (Table IV.I), where the fully developed heat transfer coefficient with respect to to the equilibrium vapour temperature is tabulated for discrete values of mass velocities (0 to 7 $Mgm^{-2}s^{-1}$ in 12 steps), pressures (0.1 to 20 MPa in 14 steps) and quality (-0.2 to +1.2 in 11 steps) and heat flux (0.05 to 3 MWm^{-2} in 9 steps). In the development of this table the developing heat transfer coefficients close to the dryout point or quench point were not used, as these values depend on prior history which is different in accident scenarios (where film boiling prediction methods are most often applied) then in steady state conditions. This table was compared extensively with the data base and the rms error was 6.73% in surface temperature. The error and data distribution for Leung's table [Appendix IV (Table IV.II)] show significant gaps in the data base at low flows and medium pressures. Some of these gaps in the data have since been partially filled by the CIAE [Chen and Chen (1998)].

Recently Kirillov et al.(1996) have taken parts of the Leung/Groeneveld table, experimental data and combined them with measurements and predictions from the Sergeev et al.(1985a) model, and added a gradual transition between. There heat transfer coefficients were tabulated for pressures of 0.1 to 20 MPa, mass flux values of 250 to 2000 $kgm^{-2}s^{-1}$, thermodynamic qualities from -0.2 to +2.2 in intervals of 0.1 and heat flux values of 0.2, 0.6 and 1.0 MWm^{-2} and is presented in Appendix V. Kirillov however defined his heat transfer coefficient based on a saturation temperature but extended his tabulated values up to thermodynamic qualities of 2.2, which corresponds to equilibrium bulk steam temperatures over 1000 °C at low pressures. This representation suppresses the effect of mass velocities, particularly at the highest qualities as can be seen in Appendix V. An example of the variation of the heat transfer coefficient is shown in Figure 4.4.

Chen and Chen (1994) measured film boiling at low flows and low to medium pressures, and noted the presence of strong inlet effects at these conditions. Subsequently Chen and Chen (1998) proposed a new method for predicting the film boiling heat transfer based on finding the Plummer (1976) non-equilibrium factor $K = (X_a - X_c)/(X_e - X_c)$ which is a function only of P, G, and X_c. The K value can be derived using the method of Appendix VI. This permits the vapour temperature to be found from iteration after which, using a pure steam heat transfer equation, the heat transfer coefficient and wall temperature can be found. The table is suitable for finding the heat transfer coefficient in the developing heat transfer region downstream of the CHF location. This method differs significantly from those discussed above as it requires also knowledge of the critical quality; as expected this will improve the prediction accuracy particularly for the low flow cases where developing non-equilibrium effects are significant (the Leung table look-up method does not predict the developing heat transfer, only fully developed heat transfer coefficients were used in its development). For low flow Chen's data and table, as presented in Appendix VI [Chen and Chen (1998)] make a significant contribution as they fill a gap in both the data base and in our understanding of the non-equilibrium effects during low flow film boiling. However in many cases the non-equilibrium is still an inferred value as actual vapour temperature measurements are difficult to measure and have only been obtained successfully over very limited conditions.

The above table prediction methods partially complement each other but can result in significant different predictions. These differences in predictions need to be resolved and work is in progress to wards this.

4.5. RECOMMENDED/MOST RECENT FILM BOILING PREDICTION METHODS

4.5.1. Pool film boiling

There is a general agreement that the modified Bromley equation for film boiling may be used for horizontal surfaces:

$$\alpha_{FB} = 0.62 \cdot \left[\frac{\lambda_v^3 \rho_v (\rho_\ell - \rho_v) \cdot r \cdot g}{\Delta T_s \cdot \mu_v} \cdot \frac{1}{2\pi} \sqrt{\frac{g(\rho_\ell - \rho_v)}{\sigma}} \right]^{1/4} \tag{4.13}$$

For vertical surfaces, changes to the constant in front of the equation and the characteristic length are required as indicated in Section 4.4.2.

4.5.2. Flow film boiling

4.5.2.1. Film boiling table

Because of the large number of film boiling methods presently available, it would be desirable to have more universal prediction method. The bundle look-up table method appears to be a more promising approach because of the following reasons:

(i)   simplicity
(ii)   correct asymptotic, and parametric trends.
(iii)   most universal method with the best overall fit to the fully developed film boiling data base
(iv)   with modifications now being introduced, it can be used to account for effects such as geometry, spacer devices etc.

A similar approach has recently been adopted for predicting the CHF in safety analysis [e.g. in RELAP and CATHARE (Section 3.6.1)]. The current look-up tables do not yet properly account for the developing flow effects, in particular as it is encountered during accident scenarios, but a combination of the approach proposed by Chen and Chen (1998) and an appropriate transformation from a time dependent heat transfer coefficient [e.g. $\alpha = f(t - t_c)$] to a length dependent heat transfer coefficient [$\alpha = f(z - z_c)$] is expected to resolve this shortcoming. Current work in progress will also account for the effects of upstream flow obstructions (such as grids or endplates), which are known to have a significant desuperheating effect.

The film boiling look-up table and other film-boiling prediction-methods are least reliable in areas where data are unavailable, and this is particularly true if strong non-equilibrium effects are present. At high flow this problem disappears and the equilibrium-type correlations will apply, i.e. the equations of Table 4.8 will apply but with the vapour temperature based on equilibrium conditions ($T_v = \min [T_{sat}, T_b]$).

The film boiling look-up table method has been used for the following applications

(i)   as a normalized database for validation of film boiling models;

(ii) as an alternative to film boiling models which cover only limited ranges of flow conditions.

For application in AWCR condition, correction factors may eventually be incorporated in the look-up table to account for the effects of the heat flux distribution and transmissions. They are not available at present. The mechanistic models may also be used to account for these effects [e.g.: Analytis and Yarigaroglu (1987); Analytis (1989, 1990) and Chen and Chen (1997)].

4.5.2.2. Inverted annular film boiling

The film-boiling prediction methods are least accurate for the IAFB regime. The data base coverage in IAFB is much more sparse compared to DFFB. Upstream effects, prior history effects and spacer effects will affect the heat transfer prediction. No properly validated method covering all conditions of interest is available for this regime. At the low flow end of the IAFB regime the pool boiling prediction method will provide a lower-bound prediction. The look-up table method for the IAFB regime is based both on experimental data (where available) and on the model of Hammouda and Groeneveld (1996).

4.5.2.3. Dispersed flow film boiling

High mass flux

The prediction accuracy for flow film boiling is most accurate at high mass-velocities $(G > \sim 3~\mathrm{Mgm^{-2}s^{-1}})$ where non-equilibrium effects are unimportant. For these conditions existing equations for heat transfer to superheated steam may be used. Section 4.5.4 presents some of these equations. The film boiling look-up tables (Appendix IV, Table IV.I and Appendix V) at high mass velocities are based primarily on single phase heat transfer equations.

Low mass flux

At low mass velocities non-equilibrium effects become significant and the prediction accuracy reduces. Also the effect spacers will complicate the prediction accuracy. Further work on the look-up table is required as the recent data of Chen and Chen (1994) has not yet been used in updating the AECL look-up table. The upstream history effect is also more important at these conditions as film boiling may never become fully developed; a method such as the one suggested by Chen and Chen (1998) may need to be combined or incorporated in the look-up table. No single validated prediction method is available covering all conditions of the low mass velocity DFFB regime.

4.5.3. Radiation heat transfer in film boiling

The radiation heat transfer coefficient is usually evaluated separately and added to the convection heat transfer coefficient, i.e.:

$$\alpha = \alpha_{conv} + \alpha_{rad} \tag{4.14}$$

It should be noted that the radiation heat transfer is particularly significant for the IAFB regime. In this case, the heat transfer at the wall-to-liquid radiation is expressed according to Siegel and Howell (1972) as,

$$\alpha_{rad} = 5.67 \cdot 10^{-8} \frac{\left(T_w + T_s\right)\left(T_w^2 + T_s^2\right)}{\dfrac{1}{\varepsilon_w} + \dfrac{1}{\varepsilon_\ell \sqrt{\varphi - 1}} - 1} \tag{4.15}$$

where:

T_w and T_s are the surface and the saturation temperatures, respectively in K; ε_w and ε_ℓ are respectively the emissivity of the heated surface and the liquid.

At $T_w \leq 700$ ^{o}C the radiation heat transfer is relatively small for the DFFB regime. Nevertheless, it is added to the convection heat transfer coefficient. The following simple two-gray-plane method may be used:

$$\alpha_{rad} = 5.67 \cdot 10^{-8} \frac{\left(T_w + T_s\right)\left(T_w^2 + T_s^2\right)}{\dfrac{1}{\varepsilon_w} + \dfrac{1}{\varepsilon_v} - 1} \tag{4.16}$$

The emissivity of the heated surface ε_w is dependent on both surface material and surface temperature. The surface emissivity is affected by oxidation, particularly for Zr with thin ZrO_2 coatings, while the vapour emissivity can be indirectly affected by the droplet concentration.

4.5.4. Correlations for single phase heat transfer to superheated steam

Single phase heat transfer to superheated steam is important as it provides an asymptotic value to the film boiling heat transfer for cases when the actual quality approached 1.0 . A number of tube-based correlations have been proposed; all of them are of the Dittus-Boelter type and give similar predictions. The following two equations are frequently used:

(i)   Miropolskiy (1975) equation, valid for P = 4 - 22 MPa, G = 0.4 - 2 Mgm^{-2}s^{-1} and ρ_w/ρ_v = 0.5 - 0.9, range of $Re = 10^5 - 2 \times 10^6$

$$Nu_v = \alpha_v \cdot D/\lambda_v = 0.028 Re_v^{0.8} \cdot Pr_v^{0.4}(\rho_w/\rho_v)^{1.15} \tag{4.17}$$

where

$$Re_v = G \cdot X \cdot D/\mu_v$$

(i)   Colborn equation:

$$Nu_v = 0.023 Re_v^{0.8} \cdot Pr_v^{0.4}(T_{va}/T_w)^{0.5} \tag{4.18}$$

4.5.5. Application to rod bundles

Virtually all film boiling prediction methods are based on correlations derived for tubes. Applying them to the prediction of fuel-bundle cladding temperatures is common practice; in doing so the following bundle-specific factors should be considered:

(i)   bundle enthalpy and flow imbalance
(ii)   heat transfer enhancement downstream of grids or spacers
(iii) adjacent wet surface or cold wall
(iv)   narrow gaps between elements
(v)   change in wall friction in dry portion of bundle (resulting in higher flow in dry subchannels)
(vi)   non-circular subchannel cross section shape
(vii) presence of axial dry-streaks in partially dry bundles

Reactor safety codes may account for some but not all of the above effects. Bundle enthalpy and flow imbalance can be evaluated using subchannel codes (see also Section 3.4.4) to predict the flow conditions in individual subchannels. The flow conditions, in turn, will permit the evaluation of the local CHF as described in Chapter 3. When the heat flux of a rod surface facing a given subchannel exceeds the local CHF, both the wall-fluid heat transfer coefficient and the wall friction factor will be reduced drastically. By keeping track of the circumferential drypatch fraction (CDF) and the axial drypatch length (ADL) for each rod facing each subchannel, the flow and enthalpy distribution as well as the distribution in film-boiling heat transfer coefficient can be evaluated. This will permit the evaluation of the fuel temperature distribution and the prediction of the extent of fuel melting. The above approach is being incorporated in some of the subchannel codes to permit a detailed prediction of the cladding temperature distribution.

As was noted also in Chapter 3, the change in geometry from tubes to bundles considerably complicates the thermalhydraulic analysis. Aside from the cross-section differences, the global and local effects of the grid or spacers on the wall heat transfer, quench behaviour and interface mass and energy transport are usually unknown, or at best are included via an empirical fix for each grid spacer configuration. In general (grid) spacers can have the following effects:

(viii) promotes rewetting downstream of the grid due to the larger turbulence level (i.e. encourages multiple quench fronts)
(ix)   acts as a cooling fin
(x)   causes desuperheating of the vapour
(xi)   results in an increase of interfacial area by breaking up the droplets or liquid core
(xii) homogenizes the flow

No satisfactory models are available to model the film boiling heat transfer in bundles equipped with grid spacers and hence most of the codes neglect the presence of the spacer grids. This is despite the fact that experimental studies by Yao et al. (1982), Yoder et al. (1983), Lee et al. (1984), Ihle et al.(1984) and others have demonstrated the beneficial effect of grid spacers, particularly during a reflooding phase where grid spacer can considerable reduce cladding temperatures.

4.6. APPLICATION TO FILM BOILING PREDICTION METHODS CODES

4.6.1. General

Most system analysis codes used in LWR and HWR safety analysis represent the core or bundle by an equivalent tube. Bundle specific effects as discussed in Section 4.5.4 above are frequently ignored. Also the axial node size used is frequently so large that it skips the transition boiling IAFB region. Thus the details of spatial variation of the heat flux cannot be considered properly, unless the size of the nodes is drastically reduced.

4.6.1.1. RELAP

Different geometrical configurations (more than 10) can be accommodated in the RELAP5 code. For each of these, various options for heat transfer modes and correlations are available. Here reference is made to the "default" geometry, that is a standard cylinder externally cooled, [see RELAP5 (1995)].

In the reference geometry, at least three types of flow patterns are distinguished, namely, inverted annular flow, slug flow and dispersed flow. The mechanisms of the wall-to-fluid heat transfer include conduction across a vapor film, convection to flowing vapor, convection between vapor and the droplets, and radiation across the vapor film.

For pool film boiling and IAFB conditions where forced convection is not important, the Bromley (1950) equation is basically adopted in this case. However, the Berenson (1961) wave length concept, was introduced in this equation, together with a factor to account for the void fraction effect, and a correction for the liquid subcooling proposed by Sudo and Murao (1975). For higher vapour velocities, the wall-vapour heat transfer coefficient is predicted using the well-known Dittus-Boelter type correlation for single phase heat transfer. The Analytis and Yadigaroglu (1987) model has also been implemented in RELAP5/MOD 2 and was reported to successfully predict reflooding transients [Analytis (1989, 1990)].

Radiation heat transfer will be evaluated using Sun's (1976) methodology by considering the radiative heat transfer between wall-to-liquid, wall-to-vapor, and vapor-to-liquid and their respective emissivities.

4.6.1.2. CATHARE

In the CATHARE code the heat transfer from the wall across the vapour film is predicted with either

(xiii) Bromley-type equation modified to account for the effect of subcooling,
(xiv) a pure heat conduction equation,
(xv) Dittus-Boelter type equation, used primarily at higher vapor velocities and void fractions, and
(xvi) natural convective equations at high void fraction and low velocities.

If in doubt which equation applies, the maximum predicted heat transfer coefficient should be used. Further details of the CATHARE equations can be found in Groeneveld (1982), Bestion (1990) and Groeneveld and Rousseau (1982). The radiation from wall to both phases is modeled using equations proposed by Deruaz and Petitpain (1976), which strictly speaking are applicable only for DFFB.

4.7. CONCLUSIONS AND FINAL REMARKS

1. The prediction of filmboiling heat transfer is much more complex than that of CHF. Aside from requiring a 4th parameter in the look-up table ("heat flux"), non-equilibrium effects should also be considered, especially in the region just down steam of the quench, near flow obstructions, and as low flows.

2. Current film boiling models and correlations appear to be flow regime specific. No single prediction method can currently provide a satisfactory prediction for both the IAFB and the DFFB regime.

3. All film boiling prediction methods are derived or validated based on data obtained in directly heated tubes. They have generally not been validated for bundle geometries experiencing severe transients. Effects such as differences in flow cross sections (subchannels vs. tubes) , presence of narrow gaps and cold walls are usually not accounted for.

4. Fuel bundles are equipped with bundle appendages (as in PHWRs) or grid spacers. These appendages have a CHF and heat transfer enhancing effect, as well as a desuperheating effect thus reducing the non-equilibrium. They also result in having multiple quench fronts. Current film boiling prediction methods usually ignore these important effects

5. The current proliferation of film boiling prediction methods, and their limited range of validity, has reinforced the need for universal prediction methods. Several such prediction methods are now under development.

6. Despite the ever increasing speed of computers, the evaluation of film boiling temperatures is still time-intensive requiring coarse nodalization. The main reasons for this are: (i) frequent iteration, (ii) the large number of equations involved and (iii) evaluation of many different fluid properties during each iteration. Table look-up methods vastly simplify this prediction process, and permit direct evaluation of the film boiling heat transfer coefficient.

7. Caution should be exercised when extrapolating steam heat-transport properties to high temperatures ($>1500°C$). In addition to the uncertainty in extrapolating to high temperatures, the dissociation of steam will also affect the steam properties.

8. The film boiling prediction methods discussed in this chapter were based primarily on steady state conditions. Transient can have a significant effect on film boiling. Aside from affecting the region over which film boiling will occur (by affecting the CHF) IAFB or pool film boiling can be destabilized, possibly resulting in a momentary return to transition boiling. Increases in heat transfer coefficient of 10-40 times have been recorded due to small pressure pulses or by passing through shock waves.

9. During the past three years progress has been made in developing look-up tables for film boiling heat transfer. This has been embodied in this chapter. No final recommendation for any specific prediction method for film boiling has been made as work on combining the most promising methods into a single, fully validated method is still in progress.

10. The table prediction methods discussed previously partially complement eachother but can result in significant different predictions. Further research to resolve these differences in prediction methods are currently in progress.

11. Table IV.I (see Appendix IV) and Table V.I (see Appendix V) contain values for filmboiling heat transfer for all heat flux values including those where the heat flux value is below the CHF but above the minimum heat flux. The data base for these table usually comes from the "hot path" type of experiments or from predictions of models.

12. The differences between the three main prediction methods for filmboiling heat transfer appear exaggerated as the reference temperatures of the heat transfer coefficients differ; a table based only on surface temperature will result in a convergence of these prediction methods.

13. The heat transfer coefficient a_s applied in Table 4.1 and Fig. 4.4 is referred to a temperature difference $(T_w - T_s)$. This results in seeming absence of the mass flux effect on the heat transfer intensity at high qualities. However, such definition of a_s is convenient for engineering calculations. It is preferred to use only the value T_s as the reference temperature because that allows to simplify considerably the prediction of the FFB heat transfer coefficient. It should be borne in mind that the recalculation between the values of a_s and a_v leads to $a_s/a_v = (T_w - T_v)(T_w - T_s)$ improper results and the distortion of function $a_s(G)$. During recalculation $a_s \rightarrow a_v$ we discovered that the effect is negligible and $a_s \, ^1 G^{0.8}$. The calculation was carried out by the method based on Miropolsky's work (1975) where it was found that $Nu \gg Re^{0.8}$ and $a_v \approx G^{0.8}$. It should be distinguished two FFB heat transfer regimes: 1) PDO (post-CHF) heat transfer, 2) before-CHF heat transfer. At present time it is not obvious yet whether the heat transfer correlations will be the same for both regimes or not. The FFB heat transfer prediction in a rod bundle is performed by both the CHF look-up table for bundles and the LUT for FFB in tubes with appropriate correction factors.

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TABLE 4.1. EXPERIMENTAL DATA ON FILM BOILING IN TUBES

Year	Reference	P, MPa	G, kg/m^{2}s	d, mm	L m	X	q, MW/m^2	T$_w$, ^{o}C	n, number of points	Notes
1950	Mc Adams	0.8 - 24	70 - 230	3.3			0.03 - 0.54			L/d=14.7 - 80
1960	Hemann	2.1 - 10.3	190 - 1070	2.5 - 8.4			0.16 - 0.92			L/d=36 - 100
1961	Collier	0.1 - 7.4	580 - 1380	4.3 - 61			0.16 - 0.41			L/d=35 - 170
1961	Parker	0.2	50 - 100	25.4	0.5	0.89 - 1.0	0.01 - 0.06			rewetting of wall
1961	Swenson	20.7	949 - 1356	14		0.08 - 0.98	0.297 - 0.581	379 - 499		
1963	Miropolskiy	3.9 - 21.6	398 - 2100	8	1.5	-2.43 - 3.42	0.07 - 2.33		5500	
1964	Bertoletti	7	1000 - 4000	5; 9		0.4 - 0.90	0.1 - 1.60			unstable temperature
1965	Bishop et al.	16.6 - 21.5	2000 - 3377	2.5 - 5.1		0.07 - 0.91	0.905 - 1.92	390 - 610		
1967	Bennet et al.	6.89	380 - 5180	12.6		0.229 - 1.48	0.383 - 2.07	454 - 840		
1967	Era et al.	6.89 - 7.28	1090 - 3020	6		0.456 - 1.24	0.20 - 1.65	295 - 630		
1967	Herkenrath et al.	14 - 20.5	693 - 3556	10 - 20		-0.117 - 1.32	0.253 - 1.666	374 - 592		
1967	Mueller	6.9	700 - 1000	15.7		0.62 - 1.0	0.5 - 0.85			
1967	Polomik	6.9	700 - 1350	15.7		0.8 - 1.0	0.55 - 1.10			
1969	Brevi	5	470 - 3000	6.5; 9.3		0.40 - 1.0	0.38 - 1.5			
1969	Kutcukcuoglu	1 - 3.3		7 - 14			0.03 - 0.57			L/d= 50, 150
1970	Lee	14 - 18	1000 - 4000		9; 13	0.3 - 0.7	0.3 - 1.4			heated by sodium
1971	Keeys	6.9	700 - 4100	12.7		0.1 5 - 0.90	0.8 - 1.5			cosine heat flux distribution

TABLE 4.1. EXPERIMENTAL DATA ON FILM BOILING IN TUBES (CONT'D)

Year	Reference	P, MPa	G, kg/m²s	d, mm	L, m	X	q, MW/m²	T_w, °C	n number of points	Notes
1972	Bailey	17.8	668 - 2690	12.7		0.391 - 0.95	372 - 454			U-tube
1973	Sutherland	6.9	24 - 175	38			0.016 - 0.063			L/d=120 - 220
1974	Grachev et al.	7 - 14	350 - 1000	11.12	2.1-9.0	0.35 - 1.3	0.05 - 0.3		414	heated by sodium
1975	Janssen et al.	0.683 - 7.07	16.6 - 1024	12.6		0.584 - 1.63	0.034 - 0.997	341 - 727		
1981	Fang	0.089 - 0.145	50 - 495	11.8 - 11.9		-0.026 -0.138	0.025 - 0.257	362-1148		
1982	Stewart, Groeneveld	1.94 - 9.05	114 - 2810	8.9	1.71	-0.12 - 0.736	0.064 - 0.459	306 - 780	1023	
1983	Becker et al.	2.98 - 20.1	4.96 - 3110	10 ÷ 24.7		-0.042 - 1.65	0.083 - 1.29	279 - 722		
1983	Borodin	8.2 - 8.34	1350 - 6870	8.9		0.133 - 1.07	0.90 - 2.7	378 - 720		
1983	Chen and Nijihawan	0.226 - 0.419	18.7 - 69.5	14.1		0.072 - 0.838	0.0027 - 0.088	229 - 648		
1983	Laperriere	3.95 - 9.63	962 - 4510	9		-0.119 -0.597	0.069 - 0.736	308 - 781		
1985	Gottula et al.	0.290 - 0.79	1.21 - 19.3	15.7		0.319 - 0.87	0.003 - 0.044	175 - 789		
1987	Remizov et al.	4.9 - 19.6	350 - 3000	10	1.5-10.2	0 - 2.48	0 - 1.28		37298	
1988	Chen, Fu, Chen	0.15 - 1.02	100 - 512	7; 12	0.99				38	
1988	Mosaad	0.11	100 - 500	9	0.28	-0.12 - 0			2100	
1988	Swinnerton et al.	0.2 - 1.92	200 - 1000	9.75	0.92	0 - 0.46	0.005 - 0.5		273	
1989	Chen Yu-Zhou et al.	0.41 - 6	47.6 - 1462	12	2.2	-0.05 - 0.24	0.028 - 0.260			
1996(b)	Chen and Chen	0.1-6.0	23-1462	6.8; 12	1.2-2.6	-0.05-1.36	0.015-0.49		3568	

TABLE 4.2. EXPERIMENTAL DATA ON FILM BOILING IN ANNULI

Year	Reference	P, Mpa	G, kg/m^{2}s	d_i, mm	L, m	X	q, MW/m^2	T_w, ^{o}C	n number of points	Notes
1961	Polomik	5.5 - 9.7	1000 - 2560			0.15 - 1.0	0.6 - 2.2			d_e = 1.52; 3.05
1964	Bennet	3.5 - 6.9	700 - 2700			0.2 - 1.0	0.6 - 1.8			
1967	Era	7	800 - 3800			0.3 - 1.0	0.13 - 1.0			d_e = 2; 5 spacers
1969	Groeneveld	4.1 - 8.3	1350 - 4100				0.5 - 1.4			two heated sections separated by unheated section
1971	Polomik	6.9	350 - 2700			0.15 - 0.65	0.75 - 2.3			d_e = 3.3 spacers
1971	Era	5	600 - 2200			0.2 - 0.9	0.2 - 0.6			d_e = 3 uniformly and nonuniformly heated
1980	OKB Gidropress Report No 431-0-047	1.5 - 15.9	8.9 - 148	9.1 d_o = 15.5	3.24	0.5 - 1.96	0.03 - 0.275		1154	

TABLE 4.3. EXPERIMENTAL DATA ON FILM BOILING IN ROD BUNDLES

Year	Reference	P, MPa	G, kg/m^2s	d_r, mm	S, mm	L, m	n, number of rods	X	q, MW/m^2	n number of points	Notes
1963	Matzner	6.9	700 - 2700				19	0.17 - 0.60	0.8 - 2.35		d_e = 8.3 mm, mainly stable temperature
1964	Hench	4.1 - 9.7	390 - 2700				2	0.2 - 0.9	0.45 - 1.9		d_e = 10.3 mm
1965	Kunsemiller	4.1 - 9.7	390 - 1350				3	0.3 - 0.7	0.55 - 1.0		d_e = 11.2 mm
1966	Adorni	5 - 5.5	800 - 3800				7	0.2 - 0.9	0.2 - 1.5		mainly stable temperature
1968	Matzner	3.4 - 8.3	700 - 1400				19	0.23 - 0.38			d_e = 6.7 mm segmented bundle
1970	Groeneveld et al.	6.3	1100 - 2200	15.2	16.2	0.5	3	0.3 - 0.6	0.033 - 1.16	160	inpile test trefoil
1971	Mc Pherson	10.9 - 2.17	700 - 4100				28	0.28 - 0.53	0.6 - 1.45		d_e = 7.8 mm mainly stable temperature
1973	Groeneveld and Mc Pherson	6.8 - 10.2	630 - 1350	13.8	14.8 - 15.8	0.5	36	$0.35 \div 1$	0.08 - 1.2		inpile test $T_w = 650^\circ C$
1976	OKB Gidropress Report No 213-0-084	1 - 6	130 - 700	9.1		1.75	7	0.6 - 1.24	0.1 - 0.35	301	d_e = 2.5 mm spacers

TABLE 4.4. CORRELATIONS FOR FILM BOILING HEAT TRANSFER ON HORIZONTAL SURFACES IN POOL BOILING

References	Correlations	Notes
1	2	3
Chang 1959	$Nu = 0.295\left(Ra \cdot \dfrac{r^*}{C_p \cdot \Delta T}\right)^{1/3}$ (1)	Laminar flow in vapor film; Ra, r^*, and l according to Eqs. 2, 3 and 5.
Berenson 1961	$Nu = 0.672\left(Ra \cdot \dfrac{r^*}{C_p \cdot \Delta T}\right)^{1/4}$ (2)	Laminar flow in vapor film; Ra, r^*, and l according to Eqs. 2, 3 and 5.
Frederking et al. 1966	$Nu = 0.20\left(Ra \cdot \dfrac{r^*}{C_p \cdot \Delta T}\right)^{1/3}$ (3)	Turbulent film boiling; Ra, r^*, and l according to Eqs. 2, 3 and 5.
Hamill, Baumeister 1967	$Nu = 0.648\left(Ra \cdot \dfrac{r^*}{C_p \cdot \Delta T}\right)^{1/4}$ (4)	Turbulent film boiling; Ra, r^*, and l according to Eqs. 2, 3 and 5.
Clark 1968	$Nu = 0.012\left(Ra \cdot \dfrac{r^*}{C_p \cdot \Delta T}\right)^{1/4}$ (5)	Turbulent film boiling; Ra, r^*, and l according to Eqs. 2, 3 and 5.
Lao, 1970	$Nu = 185\,Pr \cdot \left(\dfrac{r}{C_p \cdot \Delta T}\right)^{-0.09}$ (6) $l = 2\pi\sqrt{\left[6\sigma / g\left(\rho_\ell - \rho_v\right)\right]}$ (7)	Turbulent film boiling; Ra, r^*, according to Eqs. 2 and 5.

TABLE 4.4. CORRELATIONS FOR FILM BOILING HEAT TRANSFER ON HORIZONTAL SURFACES IN POOL BOILING (CONT'D)

1	2	3
Klimenko 1981	$Nu = 0.19 Ar^{1/3} \Pr^{1/3} \cdot f_1 \left(\dfrac{r}{C_{pv} \cdot \Delta T} \right)$ (8) Here $Ar = \dfrac{g \cdot l^3 \cdot (r_\ell - r_v)}{r_v \cdot n_v^2} < 10^8$ (9) $f_1 \left(\dfrac{r}{C_{pv} \cdot \Delta T} \right) = \begin{cases} 1 & \text{at } \dfrac{r}{C_{pv} \cdot \Delta T} < 1.4 \\[2em] 0.89 \cdot \left(\dfrac{r}{C_{pv} \cdot \Delta T} \right)^{1/3} & \text{at } \dfrac{r}{C_{pv} \cdot \Delta T} > 1.4 \end{cases}$ (10)	Laminar flow in vapor film; l according to Eq. 3
Klimenko 1981	$Nu = 0.0086 Ar^{1/2} \Pr^{1/3} \cdot f_2 \left(\dfrac{r}{C_{pv} \cdot \Delta T} \right)$ (11) Here $Ar \geq 0.8$	Turbulent film boiling

$$f_2\left(\frac{r}{C_{pv} \cdot \Delta T}\right) = \begin{cases} 1 & \text{at } \dfrac{r}{C_{pv}\Delta T} < 2 \\ 0.71\left(\dfrac{r}{C_{pv} \cdot \Delta T}\right)^{1/2} & \text{at } \dfrac{r}{C_{pv}\Delta T} > 2 \end{cases} \quad (12)$$

TABLE 4.4. CORRELATIONS FOR FILM BOILING HEAT TRANSFER ON HORIZONTAL SURFACES IN POOL BOILING (CONT'D)

1	2	3
Granovsky et al 1992	$Nu = 0.031 \cdot (\lg A)^{3.5} \boldsymbol{j}^{2/3}$ (13) $$A = \frac{(Ar)^*}{\left(\dfrac{C_p \cdot \Delta T}{r^* \cdot \mathrm{Pr}}\right)^2 + \dfrac{C_p \cdot \Delta T}{r^* \cdot \mathrm{Pr}}}$$ (14) $$(Ar)^* = \frac{g \cdot l^3 \cdot \boldsymbol{r}_v (\boldsymbol{r}_\ell - \boldsymbol{r}_v)}{\boldsymbol{m}_v^2}$$ (15) $\varphi = \alpha_{film}/(\alpha_{film} + \alpha_{rad})$ (16)	Turbulent film boiling; r* and l according to Eq. 2 and 3.

TABLE 4.5. SUMMARY OF IAFB MODELS

CHARACTERISTICS	REFERENCES							
	Analytis, Yadigaroglu (1987)	Avdeev (1986)	Chen Yu (1986)	Mosaad, Johannsen (1989)	Fung, Groeneveld (1982)	Groeneveld, Chen, Hammomuda (1991)	Hsu et al (1982)	Klugel, Kabanov (1986)
1. Dimensional (1D – one-dimensional; 2D - two-dimensional)	1D	1D	1D	1D	1D	1D	2D	1D
2. Flow Structure (h – homogeneous; t – two fluids)	t	t	t	t	t	t	t	t
3. Vapour Generation								
3.1 from liquid surface	+	+	+	+	+	+	+	+
3.2 evaporation of drops in a vapour film	–	–	+	–	–	–	–	+
3.3 evaporation of drops on a wall	–	–	–	–	–	–	–	+
3.4 wall-liquid interaction	–	–	–	–	–	–	+	–
4. Vapour Film								
4.1 flow regime (l – laminar; t – turbulent)	t	t	t	t	l, t	l, t	t	l, t
4.2 presence of drops	–	–	–	–	–	–	–	+
4.3 boundary of liquid (s – smooth; w – wavy)	w	s	s	w	s	s	w	s
4.4 Radiation through a vapour film	+	–	+	+	–	+	+	+
5. Central Flow								
5.1 1 – one phase flow; 2 – two-phase flow	1	1	1	1	1	1	1	2
5.2 flow regime (l – laminar; t – turbulent)	l, t	l, t	t	t	t	t	t	t
6. Accuracy by author (%)		20-25		RMS 12			11	
7. Verification								
7.1 Pressure, MPa	1	1-20	<1	0.1-8	0.1		2; 4	
7.2 Velocity, m/s	0.025 - 0.17	0.85 - 73	0.1 - 10	0.1 - 10	0.2 - 0.3			
7.3 Subcooling, K	< 70	5-200		20-60	< 20			

TABLE 4.6. SUMMARY OF DFFB MODELS

	CHARACTERISTICS	Andreani, Yadigaroglu (1991)	Barzoni, Martini (1982)	Chen Yu (1994)	Chen, Ozkaynak, Sundaram (1979)	Forslund, Rohsenow (1968)	Ganich, Rohsenow (1977)	Groeneveld, Delorme (1976)	Jones, Zuber (1977)	Kudriavtzev et al. (1987)
1	2	3	4	5	6	7	8	9	10	11
1.	Dimensional (1D – one-dimensional; 2D - two-dimensional; 3D - three-dimensional)	2D, 3D	1D	1D	1D	1D	1D	1D	1D	2D
2.	Flow Structure (h – homogeneous; dv - drops + vapour)	dv	h	dv	dv	dv	dv	h	dv	h
3.	Scheme of Heat Transfer*	II	I	II	II	III	III	I	II	II
4.	Effects									
4.1	Deposition of drops	+	–	–	–	+	+	–	–	–
4.2	Spectrum of drops	+	–	–	–	–	+	–	–	–
4.3	Effect of drops on transport properties of medium	+	–	+	–	–	–	–	–	+
4.4	Slip	+	–	–	–	–	+	+	–	–
4.5	Radiation	–	–	+	–	–	–	–	–	–
5.	Accuracy by author (%)		RMS 12.3		15			RMS 6.93		
6.	Verification									
6.1	Pressure, MPa			0.1-6			0.7 - 21.5			
6.2	Mass Flux, kg/m^2·s			24 - 1000				130 - 5200		
6.3	Quality			0.05 - 1.4				0.08 - 1.6		

* Scheme of Heat Transfer
I - heat transfer wall to vapour
II - I + wall to droplet
III - I + II + wall to drops

TABLE 4.6. SUMMARY OF DFFB MODELS (CONT'D)

	CHARACTERISTICS	Marinov, Kabanov (1977)	Moose, Ganic (1982)	Nishikawa et al. (1982, 1981)	Saha (1980)	Sergeev (1976)	Tong, Young (1974)	Varone, Rohsenow (1986)	Vojtek (1982)	Welb, Chen.J. (1986)	Whalley et al. (1982)	Yoder, Rohsenow (1983)
1	2	12	13	14	15	16	17	18	19	20	21	22
1.	Dimensional (1D – one-dimensional; 2D - two-dimensional; 3D - three-dimensional)	1D	1D	1D	1D	1D	1D	1D	1D	1D	1D	1D
2.	Flow Structure (h – homogeneous; dv - drops + vapour)	h	dv	dv	dv	dv	dv	dv	h	dv	dv	dv
3.	Scheme of Heat Transfer*	II	III	III	III	II	II	III	II	II	II	II
4.	Effects											
4.1	Deposition of drops	−	+	+	−	−	−	+	+	−	−	−
4.2	Spectrum of drops	−	+	−	−	−	+	−	−	−	+	+
4.3	Effect of drops on transport properties of medium	−	−	−	−	−	−	+	−	+	−	−
4.4	Slip	+	−	−	−	−	+	−	−	−	−	+
4.5	Radiation	−	−	+	−	+	−	−	−	−	−	−
5.	Accuracy by author (%)			20		RMS 10				24	30 - 60	
6.	Verification											
6.1	Pressure, MPa					1-18			3-12	0.1-7		
6.2	Mass Flux, kg/m^2·s			400 - 1600		100 - 1500			300 - 1400	12 - 100		
6.3	Quality								0.3 - 0.1	0 - 0.99		

* Scheme of Heat Transfer I - heat transfer wall to vapour
           II - I + wall to droplet
           III - I + II + wall to drops

TABLE 4.7. EMPIRICAL FLOW FILM BOILING HEAT TRANSFER CORRELATIONS

References	Correlations	

TABLE 4.7. EMPIRICAL FLOW FILM BOILING HEAT TRANSFER CORRELATIONS

References	Correlations	Ranges of Parameters		
		P MPa	G kg/m²·s	X
Collier 1962	$q \cdot [d^{0.2}/(G \cdot X)^{0.8}] = c_0[(T_w - T_s)]^m$ (1) where $c_0 = [\exp(0.01665 \cdot G)]/389$; $m = 1.284 \div 0.00312 \cdot G$; $T_w - T_s < 200\,^0C$; [G]-kg/m²·s; [d]-m; [q]-kW/m2; [T]-K or ^0C;	7.03	>1000	0.15-1
Collier 1962	$q \cdot [d^{0.2}/(G \cdot X)^{0.8}] = 0.018(T_w - T_s)^{0.921}$ (2) where $T_w - T_s < 200\,^0C$; [G]-kg/ m²·s; [d]-m; [q]-kW/m2; [T]-K or ^0C;	7.03	<10^6	0.15-1
Swenson et al. 1961	$Nu_w = 0.076\{Re_w[X + (\rho_v/\rho_\ell)(1\text{-}X)](\rho_w/\rho_v)\}^{0.8}Pr_w^{0.4}$ (3)	20.6	945-1350	
Miropolskiy 1963	$Nu = 0.023Re_v^{0.8}Pr_w^{0.8}[X + (\rho_v/\rho_\ell)(1\text{-}X)]^{0.8} \cdot y$ (4) Where $y = 1-0.1[(\rho_v/\rho_\ell) - 1)](1 - X)^{0.4}$; $Nu = \alpha_s \cdot d/\lambda_v$; $Re_v = G \cdot d/\mu_v$; $0.23 \le q \le 1.16 MW/m^2$; $8 \le d \le 24$ mm;	3.9-21.6	800-4550	
Dougall 1963	$Nu = 0.0203\{Re[X + (\rho_v/\rho_\ell)(1\text{-}X)]\}^{0.8}Pr^{0.4}$ (5)	<3.5	1660-3650	<0.5
Bishop et al. 1964	$Nu_w = 0.098\{Re_w(\rho_\ell/\rho_v)[X + (\rho_v/\rho_\ell)(1 - X)]\}^{0.8}Pr_w^{0.83}(\rho_v/\rho_\ell)^{0.5}$ (6)	16.8-21.9	1350-3400	0.1-1

TABLE 4.7. EMPIRICAL FLOW FILM BOILING HEAT TRANSFER CORRELATIONS (CONT'D)

References	Correlations	P MPa	G kg/m^2·s	X
Bishop et al. 1964	$Nu_v = 0.055\{Re_v(\rho_\ell/\rho_v)[X + (\rho_v/\rho_\ell)(1-X)]\}^{0.82}Pr_w{}^{0.96}(\rho_v/\rho_\ell)^{0.35}$ $(1 + 26.9 \cdot d/L)$ (7)	16.8-21.9	350-3400	0.1-1
Bishop et al. 1965	$Nu_\ell = 0.0193\,Re_\ell^{0.8}\,Pr_\ell^{1.23}\,(\rho_v/\rho_\ell)^{0.068}\,[X + (\rho_v/\rho_\ell)(1-X)]^{0.68}$ (8)	4.08-21.9	700-3140	0.07-1
Bishop et al 1965	$Nu_\ell = 0.033\,Re_\ell^{0.8}\,Pr_\ell^{1.25}\,(\rho_v/\rho_\ell)^{0.197}[X + (\rho_v/\rho_\ell)(1-X)]^{0.738}$ (9)	4.08-21.9	700-3140	0.07-1
Tong 1965	$Nu_w = 0.005(d \cdot G/\mu_w)^{0.8}Pr_v{}^{0.5}$ (10)	>700	>14	<0.1
Quin 1966	$Nu_v = 0.023\{Re_v[X + (\rho_v/\rho_\ell)(1-X)]\}^{0.8}Pr_w{}^{0.4}(\mu_v/\mu_\ell)^{0.14}$ (11)	6.9	1150	0.72-0.79
Kon'kov et al. 1967	$Nu = 0.019 \cdot \{Re_v[X + (\rho_v/\rho_\ell)(1-X)]\}^{0.8}Pr_w$ (12) where $0.29 \leq q \leq 0.87\ MW/m^2$; $d = 8\ mm$;	2.94-19.6	500-4000	-
Henkenrath et al. 1967	$Nu_w = 0.06\{Re_w[X + (\rho_v/\rho_\ell)(1-X)]\,(\rho_v/\rho_\ell)\,Pr_w\}^{0.8}(G/1000)^{0.4}(P/P_{cr})^{2.7}$ (13)	14.2-22.3	750-4100	0.1-1
Brevi et al. 1969	$Nu_\ell = 0.0089(Re_\ell\,X/\varphi)^{0.84}\,Pr_\ell{}^{0.333}\,[(1 - X_{cr})/(X - X_{cr})]^{0.124}$ (14) where φ - void fraction.	5.06	500-3000	0.4-1
Lee 1970	$T_w - T_s = 1.915 \cdot \left[\dfrac{q}{G\left(X + \dfrac{1-X}{4.15}\right)}\right]^2$ (15)	14.2-18.2	1000-4000	0.30-0.75
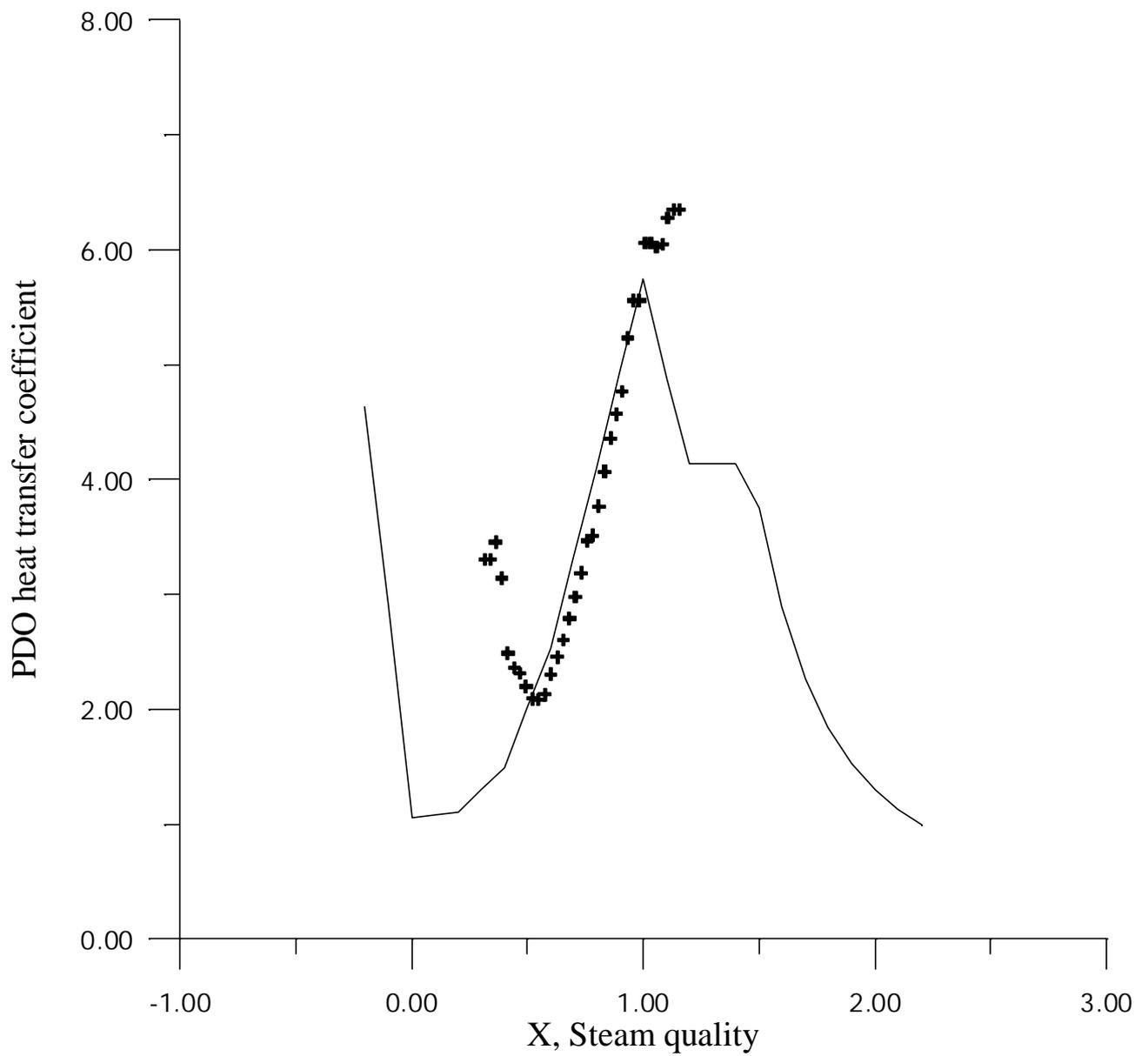
FIG. 4.4. PDO heat transfer coefficient as a function of steam quality; P=16 MPa, G=1000 kg/(m²s), q=0.6 MW/m²; the line presents the look-up table-1999 values, the pluses are experimental points.