CHAPTER 12A

Ross L. Judd
Department of Mechanical Engineering
McMaster University
Hamilton, Ontario

SUMMARY

This chapter introduces the concepts of nucleate boiling and forced convective boiling heat transfer. Its purpose is to introduce the reader to the phenomena associated with the heat and mass transfer processes which occur in a nuclear reactor. While the information presented is not specific enough to predict the conditions which exist in any particular reactor design, the reference material which is cited is comprehensive enough that the reader should be able to predict these conditions himself. The subsequent chapters will build upon the material presented herein.

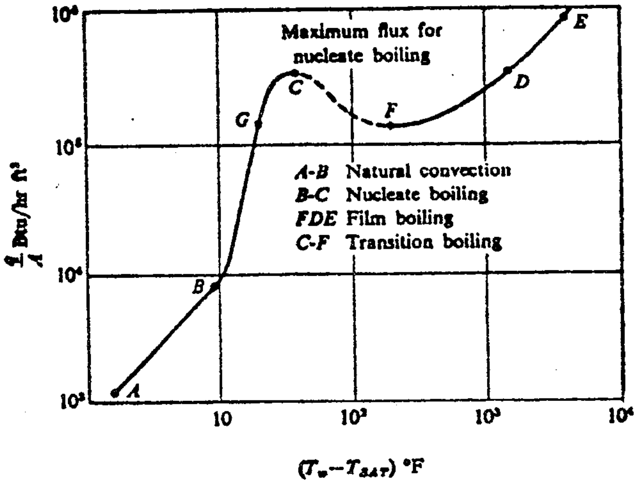
12A. HEAT TRANSFER AND FLOW PHENOMENA II

12A.1 Pool Boiling Phenomena

An understanding of pool boiling is fundamental to the design of steam generators and the study of two phase flow heat transfer. All of the phenomena which occur in two phase flow boiling occur in pool boiling as well but fewer independent variables need to be considered. Certain aspects of two phase flow heat transfer such as the role of surface condition which will ultimately have to be incorporated in any comprehensive two phase flow heat transfer model have not yet been addressed whereas a start has been made to quantify them in pool boiling. Furthermore, although the CANDU reactor design is based upon single phase forced convective cooling, pool boiling heat transfer considerations would play a significant role in the event of a loss of cooling accident since the processes of vapour bubble nucleation, formation and growth which had been suppressed under reactor operating conditions would become active in such a situation. The conditions which exist at the break strongly influence the conditions which pertain in the entire reactor system and consequently, an understanding of pool boiling is important in order to be able to predict the progress of the system from film boiling to nucleate boiling to single phase forced convection once again.

12A.1.1 Overview

The objective of pool boiling research is to be able to relate surface heat flux q/A to temperature difference $(T_w - T_\infty)$, fluid properties and surface condition without recourse to any prior knowledge of the boiling behaviour of the surface under consideration. Three distinctly different modes of boiling heat transfer are encountered as indicated in the diagram below of which different functional relationships between heat flux and temperature difference are found.



The diagram presents what is known as the "characteristic curve" for water at the saturation temperature boiling on a copper cylinder but it is representative of all boiling heat transfer results irrespective of fluid/surface combination. For relatively small values of superheat ($T_w - T_{sat}$), the heat flux q/A is predicted by the relationships for "natural convection" and operation occurs along curve A-B because there is no significant generation of vapour bubbles under these conditions even though the surface temperature is in excess of the saturation temperature. The temperature difference required for sufficient vapour bubble generation to occur that the characteristic curve would be affected depends upon the pressure at which the system is boiling and the type of fluid although the type and condition of the heater surface is significant as well since these factors determine the number of the active nucleation cavities at which the bubbles form.

Once the superheat has exceeded that corresponding to point B, the characteristic curve begins to rise more steeply and operation occurs along curve B-C which indicates a large increase in heat flux q/A with only moderate increase in superheat ($T_w - T_{sat}$). This process is known as "nucleate boiling" heat transfer because sufficient vapour bubbles are formed at discrete nucleation sites on the surface to promote large heat transfer rates. When the liquid temperature is at the saturation temperature, the vapour bubbles formed grow until the inertia and/or buoyancy forces overcome the surface forces holding the bubble to the heater surface at which point the bubble breaks free and rises in the liquid. When the liquid temperature is below the saturation temperature, sufficient heat will be exchanged between the vapour bubble and the liquid through which it is rising that the bubble will get smaller and may completely condense or collapse before arriving at the free surface. Consequently, both the superheat ($T_w - T_{sat}$) and the subcooling ($T_{sat} - T_\infty$) are important so that

$$q/A_T = \mathfrak{F}[(T_w - T_{sat}), (T_{sat} - T_\infty), \text{Fluid Properties, Surface Conditions}]$$

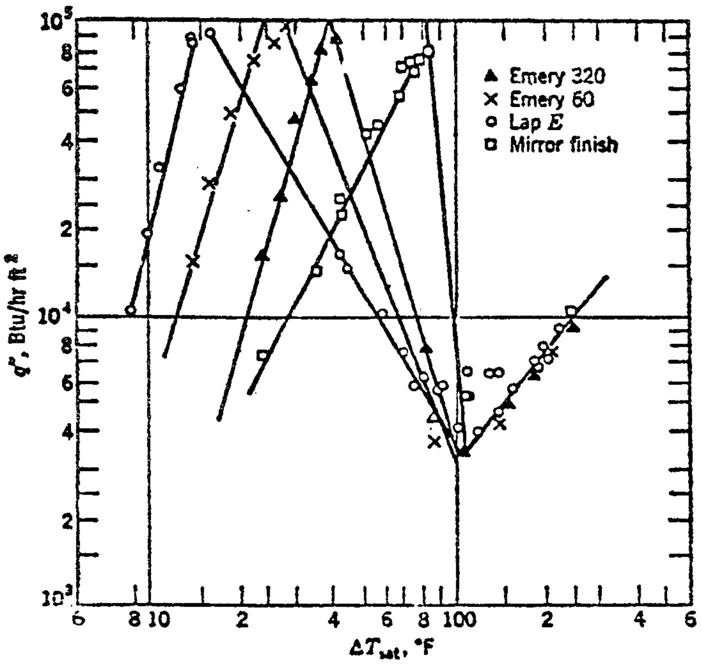
As the heat flux increases, the number of active nucleation sites becomes more numerous and the activity at the nucleation sites increases as well. Discrete vapour bubbles merge into vapour columns at each of the sites and then vapour columns merge to form large globules of vapour. Ultimately a situation is reached in the vicinity of point C where the surface is blanketted with vapour and circulation of the surrounding liquid is prevented so that the ability of the boiling process to remove heat is diminished. A limiting condition referred to as the "critical heat flux" arises such that any further increase in heat flux leads to a rapid excursion to a new operating condition in the vicinity of point D.

Even though the energy input to the heater is the same, the superheat increases several orders of magnitude and the heater surface often fails by melting so that "burnout" is said to have occurred.

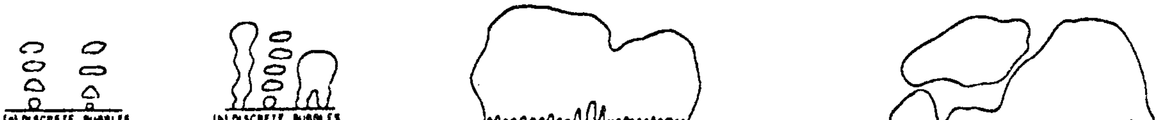
Should heater surface not fail, then operation along curve DE is possible. A continuous layer of vapour blankets the surface and although the interface between the liquid and vapour is in continuous motion because of the regular formation of vapour globules at discrete points, the liquid never contacts the surface. The process is known as "film boiling" and energy is transferred between the heater surface and the liquid by conduction and radiation. Should the heat flux now be reduced to that corresponding to point F, a sudden excursion back to nucleate boiling in the vicinity of point G may occur and further reductions would lead to operation along curve GB.

The dashed curve describes an unstable phenomenon known as "transition boiling". Operation along curve FC would be observed if the heater surface temperature were controlled rather than if the heat flux were controlled. In transition boiling, a combination of film boiling and nucleate boiling occurs. The fluid motion is very chaotic and operation under these conditions is avoided whenever possible.

The experimental results of Berenson [1] depicted in the diagram demonstrate that surface finish (surface condition) has a marked effect on the boiling characteristic curve. The fact that surface conditions are very hard to duplicate and that there is no standard means of characterizing a boiling heat transfer surface is responsible for wide discrepancies in experimental results of different authors. Profilometer measurements enable surface conditions to be reproduced but these measurements cannot be correlated with boiling heat transfer performance.



What is really required is a knowledge of the number of active sites per unit area N/A_T, the frequency f with which vapour bubbles are emitted at each site and the quantity of heat transmitted per bubble Q_{Bubble} for each surface-fluid combination of interest. Statistical representation of these parameters is required since boiling is best described by statistical distributions. In addition, two distinctively different modes of nucleate boiling exist, the "isolated bubble mode" and the "coalescent bubble mode" respectively and the mechanisms promoting heat transfer will obviously be different in each case. Surprisingly, however, there is little difference in the quantity of heat transmitted per bubble. From these considerations


(a) DISCRETE BUBBLES
(b) DISCRETE BUBBLES, VAPOR COLUMNS, VAPOR MUSHROOMS
(c) VAPOR COLUMNS, LARGE VAPOR MUSHROOMS
(d) LARGE VAPOR MUSHROOMS, VAPOR PATCHES

$$q/A_T = \mathcal{F}\,[(T_w - T_{sat}),\ (T_{sat} - T_\infty),\ N/A_T,\ f,\ Q_{Bubble}]$$

where the parameters $(T_w - T_{sat})$ and $(T_{sat} - T_\infty)$ are not the only independent variables since obviously N/A_T, f and Q_{Bubble} are dependent on surface-fluid conditions. In order to gain insight into the phenomenon, it is essential to know something of the microscopic aspects such as the relationship of the number of active sites per unit area to the microroughness, the relationship of the period of bubble formation and the waiting period between bubble formations to the superheat and subcooling and the relationship of the heat transferred per bubble to bubble departure size and velocity which themselves are functions of superheat, subcooling and surface-fluid conditions.

12A.1.2 Nucleation Theory

The primary requirement for nucleation to occur is that the liquid be superheated. For a nucleus to grow into a bubble, its size must exceed that for thermodynamic equilibrium corresponding to the state of the liquid which for a spherical nucleus of radius r in a pure substance can be written

$$p_v - p_\ell = 2\sigma/r$$

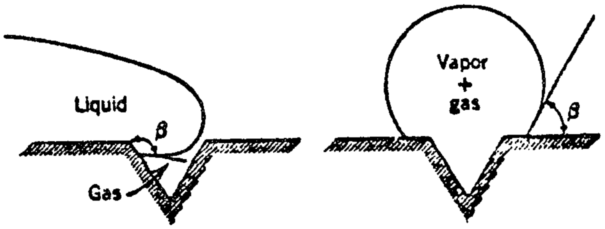
where σ is the surface tension at the interface of the vapour and liquid phases. For a liquid at pressure p_ℓ, the vapour pressure p_v of the superheated liquid near the wall can be related to the superheat $(T_v - T_{sat})$ by the Clausius Clapeyron equation

$$p_v - p_\ell = (T_v - T_{sat})\ \rho_v\ h_{fg}/T_{sat}$$

so that the equilibrium bubble size is given by the relationship

$$r = 2\sigma\ T_{sat}/\rho_v\ h_{fg}\ (T_v - T_{sat})$$

The process by which bubble nucleii are formed is depicted below.



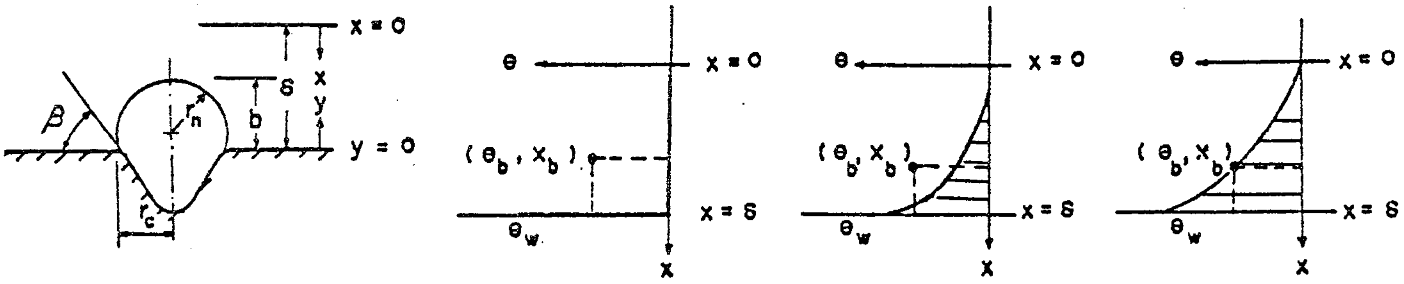
The angle β is called the contact angle. A fluid-surface combination with a large value of β (poorly wetted surface) has a better chance of trapping gas within the cavity by capillary action than a fluid-surface combination with a small value of β (well wetted surface).

Assuming that T_v may be assumed equal to T_w, it would appear that the wall superheat required for nucleation to occur at a nucleation site on a solid surface could be evaluated by

$$(T_w - T_{sat}) = (T_v - T_{sat}) = (2\sigma\, T_{sat}/\rho_v\, h_{fg})/r_c$$

where r_c is the cavity radius but such predictions do not agree with observed data primarily because the fluid adjacent to the heat transfer surface is not uniformly superheated. Griffith and Wallis [2] demonstrated this point conclusively. At one extreme, the equation predicts that $(T_v - T_{sat}) \to 0$ as $r_c \to \infty$ but no boiling can occur until some finite incipient superheat is achieved; at the other extreme, $(T_v - T_{sat}) \to \infty$ as $r_c \to 0$ and it is known that when cavities become very small, boiling is not possible no matter how great the superheat. Obviously, the theoretical model is too simplistic.

Hsu [3] solved the problem by adapting the model to accommodate the initiation of a bubble from a nucleus sitting at the mouth of a cavity. The nucleus is assumed to be formed from the residue of the preceding bubble which was trapped by the cavity. At the beginning of the cycle, relatively cool liquid which replaced the void left by the preceding bubble surrounds the nucleus. As time goes on the liquid is warmed up through transient conduction to the liquid modelled as a semi infinite medium and a thermal layer grows. It is hypothesized that the bubble nucleus will grow when the surrounding liquid is warmer than the vapour in the nucleus which has been assumed to be at the wall temperature. When the nucleus starts to grow, the waiting period ends. The sequence of events is depicted below.



The results of this model yield the relationship

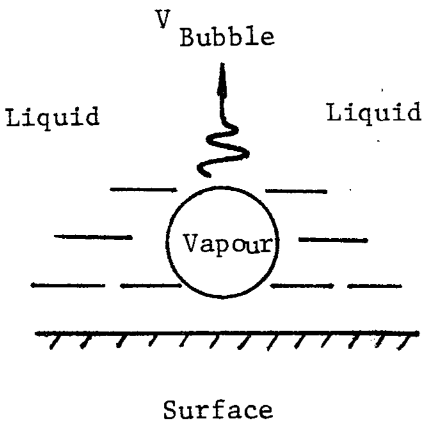
$$(T_w - T_{sat}) = (T_{sat} - T_\infty) + \frac{4\sigma\, T_{sat}}{\rho_v\, h_{fg}\, \delta}(1 + \cos\beta)$$

$$+ \sqrt{\left[(T_{sat} - T_\infty) + \frac{4\sigma\, T_{sat}}{\rho_v\, h_{fg}\, \delta}(1 + \cos\beta)\right] \frac{4\sigma\, T_{sat}}{\rho_v\, h_{fg}\, \delta}(1 + \cos\beta)}$$

to predict the superheat at which incipience occurs. Han and Griffith [4] obtained a similar result with a different analysis. Recent findings by Ibrahim and Judd [5] suggest that a further modification to incorporate a convective heat transfer term is required to properly accommodate subcooled boiling conditions.

12A.1.3 Nucleate Boiling

Numerous models have been advanced to predict the nucleate boiling heat transfer rates but none of them is capable of completely explaining the heat transfer rates observed under various boiling conditions. The problem is that the mechanisms projected to explain the effect of bubble formation and growth upon the rate of heat transfer are too simplistic. Judd and Merte [6] reviewed these mechanisms and compared the predictions of the models based upon them with their experimental results.



The simplest explanation of the action of the bubbles with respect to the transfer of heat is that a bubble transports one bubble volume of latent heat from the surface as it departs or collapses. This explanation was originally advanced by Jakob [7] who was unable to verify it for lack of the necessary measurements. The model is expressed mathematically by the relationship

$$q/A_T = \rho_v\, h_{fg}\, V_{Bubble}\, (N/A_T) f$$

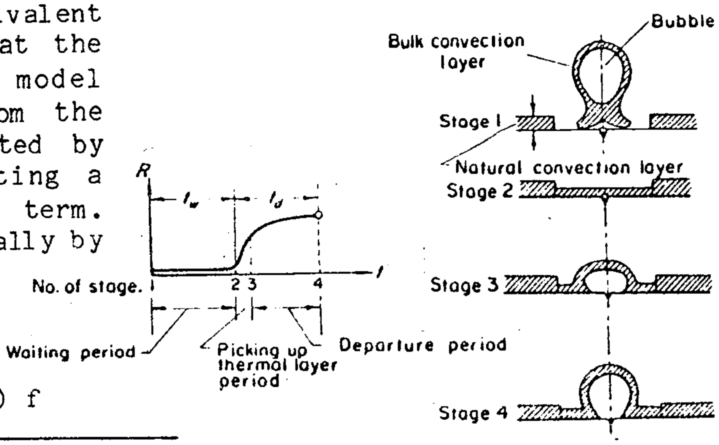
Rohsenow and Clark [8] analysed the results of Gunther and Kreith [9] for water boiling on a surface and refuted this model pointing out that latent heat transport could only account for the transfer of a few percent of the impressed heat transfer rate. More recently, Rallis and Jawurek [10] have lent support to this model with their experimental results for water boiling on a wire. It seems that the surface geometry has a significant influence on the latent heat transport model.

Han and Griffith [11] formulated a model in which the enthalpy transported by a single bubble was equated to the superheat enthalpy associated with an approximately cylindrical volume of twice the departure diameter and height equivalent to the thermal layer thickness at the instant of departure. The model accounted for heat transfer from the portions of the surface unaffected by nucleate boiling by incorporating a natural convection heat transfer term. The model is expressed mathematically by the relationship

$$q/A_T = \frac{1}{2} \rho_\ell C_\ell D_d^2 \delta_d$$

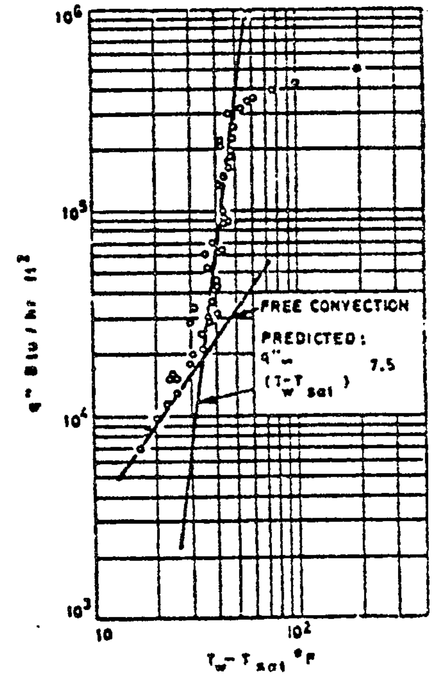
$$\left[4 - \frac{1}{3}(1-\delta/\delta_d)\right] (T_w - T_\infty) (N/A_T) f$$

where $\delta = \sqrt{\pi \kappa_\ell \tau_g}$ and $\delta_d = \sqrt{\pi \kappa_\ell (\tau_g + \tau_w)}$. This model depends upon a knowledge of τ_g and τ_w but adequately predicts nucleate boiling heat transfer of water under isolated bubble conditions when these parameters are known.



Subsequently, Mikic and Rohsenow [12] modified this procedure to incorporate the relationship $N/A_T \sim 1/r_c^m$. The departure diameter and frequency were obtained from empirical correlations so that cavity size distribution was the only input which the model required. The data of Gaertner and Westwater [13] who presented active cavity density results as a function of superheat determined the exponent m and hence enabled the nucleate boiling heat flux to be predicted.

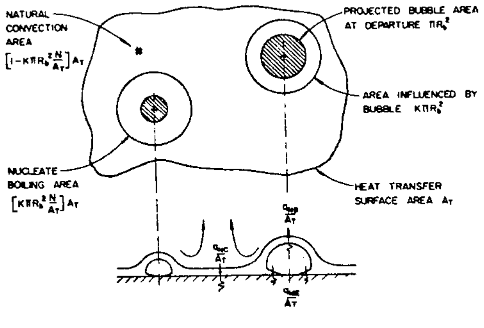
As the diagram at the right indicates, the heat fluxes predicted by the Mikic and Rohsenow model are in excellent agreement with those measured by Gaertner and Westwater. The model was also capable of predicting the heat flux-superheat characteristics at different pressures once the exponent m had been determined by fitting data obtained at atmosphere pressure.



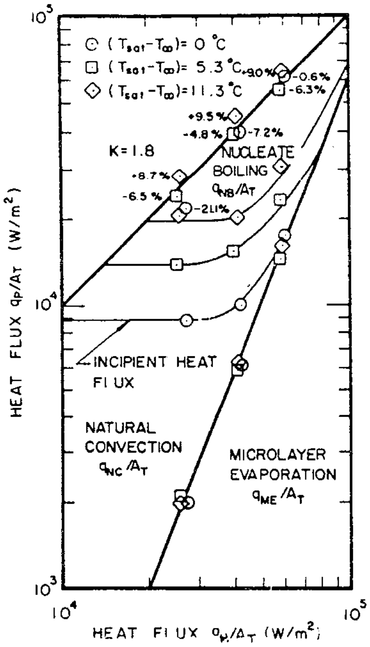
More recently, Judd and Hwang [14] presented a comprehensive model for nucleate pool boiling heat transfer which included the effect of microlayer evaporation as well. Using a similar approach to represent the nucleate boiling heat transfer component within the model depicted in the diagram below, the authors were able to show that a relationship of the form

$$q/A_T = q_{ME}/A_T + q_{NC}/A_{NC}[1-K\pi R_b^2 N/A_T]A_T$$

$$+ q_{NB}/A_{NB}[K\pi R_b^2 N/A_T]A_T$$



which includes natural convection heat transfer between the nucleation sites and microlayer evaporation heat transfer at the base of the bubbles accounted for all of the impressed heat flux in their experiments with dichloromethane boiling on a glass surface as indicated in the diagram at the right. In this relationship, K which is a parameter greater than unity relating the area of influence around a nucleation site from which energy is transported by nucleate boiling to the projected bubble area at departure was the only unknown since all of the other parameters were obtained by experimental measurement. In a similar fashion, Fath and Judd [15] showed that this type of nucleate boiling heat transfer model was able to accommodate variations in system pressure as well.



12A.1.4 Critical Heat Flux

Experimental evidence has shown that as heat flux is increased, more nucleation sites become active and the frequency of bubble emission increases. Gradually, streams of individual bubbles forming at a nucleation site begin to coalesce into vapour columns and as the nucleation sites become more numerous, the columns agglomerate into large vapour masses which begin to interfere with the circulation of liquid near the surface. Ultimately the ability of the boiling phenomenon to remove heat is impaired and a condition is reached where an incremental increase in heat flux results in a pronounced increase in the surface temperature. It is not conclusively established whether the cause of the peak nucleate boiling heat flux is vapour blanketting of the surface or interruption of liquid circulation but the result is an excursion to a higher temperature condition where burnout may occur. Rohsenow and Griffith [16] were able to correlate their data with a relationship of the form

$$q_{crit}/A_T = 143\ \rho_v\ h_{fg}\left[\frac{\rho_\ell - \rho_v}{\rho_v}\right]^{0.6}\ \left[\frac{a}{g}\right]^{0.25}$$

while Zuber [17] correlated his data with a relationship of the form

$$q_{crit}/A_T = 0.131\ \rho_v\ h_{fg}\left[\frac{\sigma(\rho_\ell - \rho_v)g}{\rho_v^{\,2}}\right]^{0.25}\ \left[\frac{\rho_\ell}{\rho_\ell + \rho_v}\right]^{0.50}$$

while Kutateladze [18] advanced the theoretical relationship

$$q_{crit}/A_T = \frac{\pi}{24}\ \rho_v\ h_{fg}\left[\frac{\sigma(\rho_\ell - \rho_v)g}{\rho_v^{\,2}}\right]^{0.25}$$

These relationships predict critical heat flux for plane surfaces and consequently more specific information such as that provided by Sun and Lienhard [19], Lienhard and Dhir [20] and Lienhard, Dhir and Riherd [21] should be consulted when the geometry is different.

12A.1.5 Transition Boiling

In situations where heater surface temperature is varied rather than surface heat flux, it is possible to encounter a phenomenon characterized by violent motion of liquid and vapour known as transition boiling. Liquid rushes toward the surface but as soon as it makes contact, vapour is formed and it is thrust back. Because of the turbulent nature of transition boiling, operation in this regime is avoided. No adequate theory exists to predict transition boiling heat flux.

12A.1.6 Film Boiling

Film boiling is the term which refers to the transfer of heat through a stable film of vapour. This phenomenon can be modelled mathematically and solutions for the film boiling heat flux can be obtained for a variety of surface configurations in various orientations. Bromley [22] derived the original relationship for the average heat transfer coefficient for a vertical plate of height L

$$\bar{\alpha} = 0.943 \left[\frac{k_v^3 \rho_v h'_{fg} (\rho_\ell - \rho_v) g}{\mu_v L (T_w - T_{sat})}\right]^{0.25}$$

where

$$h'_{fg} = h_{fg}\left[1 + 0.5\left(\frac{C_v (T_w - T_{sat})}{h_{fg}}\right)\right]$$

Breen and Westwater [23] adapted this solution for horizontal tubes of diameter D obtaining the relationship

$$\bar{\alpha} = (0.59 + 0.069 \frac{\lambda_c}{D}) \left[\frac{k_v^3 \rho_v h'_{fg} (\rho_\ell - \rho_v) g}{\mu_v \lambda_c (T_w - T_{sat})}\right]^{0.25}$$

where the minimum wavelength for a Taylor instability λ_c is given by

$$\lambda_c = 2\pi \left[\frac{g_c \sigma}{g(\rho_\ell - \rho_v)}\right]^{0.5}$$

Film boiling from a flat horizontal surface was investigated by Berenson [24] who presented the relationship

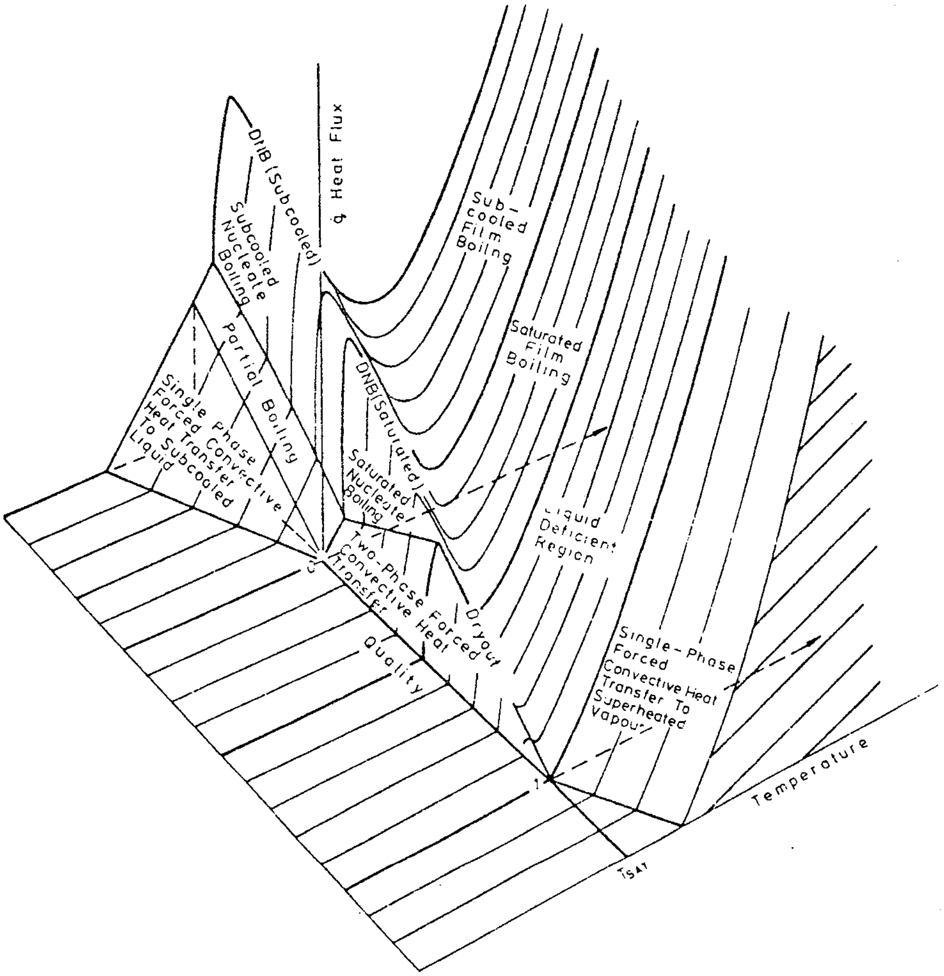
$$\bar{\alpha} = 0.67 \left[\frac{k_v^3 \rho_v h'_{fg} (\rho_\ell - \rho_v) g}{\mu_v \lambda_c (T_w - T_{sat})}\right]^{0.25}$$

Since the surface temperature is generally quite high in film boiling, radiation heat transfer is generally taken into account by combining film boiling with radiation heat transfer coefficients given by the relationship

$$\bar{\alpha} = \sigma\varepsilon \left[\frac{(T_w^4 - T_{sat}^4)}{(T_w - T_{sat})} \right]$$

12A.2 Forced Convective Boiling Phenomena

The pool boiling phenomenon and the forced convective boiling phenomenon may be considered to be related through the quality of the liquid/vapour mixture in the vicinity of the boiling heat transfer surface. Even though a considerable number of bubbles may be present in a pool of boiling liquid, the heat transfer medium is essentially saturated liquid (0% quality) whereas the heat transfer medium in a channel in which boiling is occuring may have any level of quality up to



and including saturated vapour (100% quality). The quality governs the mechanisms of heat transfer which are active and in a very real way determines the ability of the heat transfer medium to remove heat at the boiling heat transfer surface.

The figure above which has been abstracted from reference [25] depicts the "boiling surface", a plot of heat flux versus temperature and quality that illustrates the relationship between pool boiling and forced convective boiling. The intersection of the surface with the heat flux/temperature plane at 0% quality should be recognized as the "characteristic curve" previously discussed. Notice how the boiling regimes diminish in significance until they vanish completely at 100% quality. Each of the heat transfer regimes identified in the figure will be discussed at some length in the sections which follow.

12A.2.1 Effect of Flow on Boiling Heat Transfer Rates

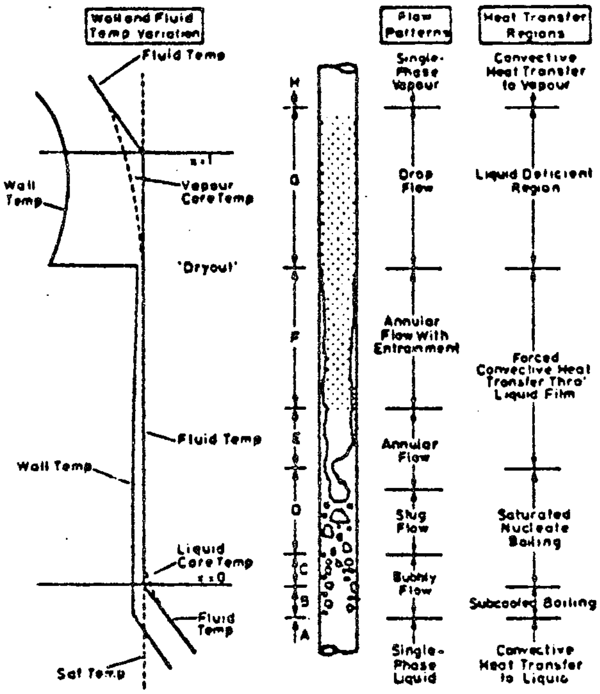
It would seem reasonable that the rates of boiling heat transfer would be affected when a velocity field was superimposed, either by agitating the fluid over the pool boiling surface or by imposing an internal flow on the boiling phenomenon occuring inside a flow passage, because of the pronounced effect which velocity has on non-boiling heat transfer phenomena. Such is not the case. Numerous experimental investigations have demonstrated that the agitation induced by the flowing liquid is ineffectual in promoting boiling heat transfer although the rate at which heat is transported by forced convection is decidedly influenced. Consequently, the mass flowrate is a more important parameter than the flow velocity in forced convective boiling. This is especially true for boiling heat transfer in a flow passage and since this configuration is of particular concern in reactor thermohydraulics, it will be discussed in some detail in the sections which follow.

12A.2.2 Regimes of Heat Transfer in a Vertical Heated Tube

In order to be able to predict the heat transfer processes occurring in the channels between reactor fuel elements, it is essential to have an understanding of the flow regimes through which the coolant flowing under pressure will pass. The traditional approach of discussing the flow regimes and heat transfer phenomena occurring in a uniformly heated vertical tube with subcooled fluid at the inlet will be used to introduce this topic as this configuration is discussed extensively in the literature [25], [26] while at the same time the example is illustrative of the phenomena which might be encountered in a vertical fuel channel. In the discussion which follows, reference is made to the thermodynamic mass quality x of the liquid vapour mixture at distance z which is defined as

$$x(z) = \left[\frac{h(z) - h_{\ell}}{h_{fg}}\right]$$

According to this relationship, x may have values less than zero and greater than unity which have no practical significance other than to signify that the fluid is subcooled or superheated respectively.

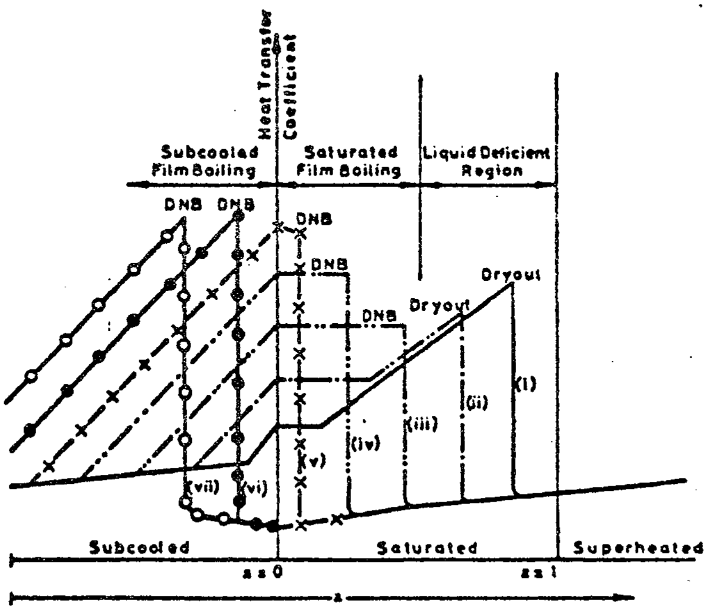


The illustration above predicts the various heat transfer regimes which will be traversed in turn as the coolant flows upward. At the lower end of the tube, "single phase convective heat transfer" occurs (Region A) until the wall temperature reaches the saturation temperature. "Subcooled nucleate boiling heat transfer" follows in which the wall temperature increases above the saturation temperature (Region B) and persists until the liquid core temperature achieves the saturation condition (x = 0). Because the wall temperature is sufficiently in excess of the saturation temperature, preexisting vapour nucleii have been activated and vapour bubbles are forming at the surface. However, the liquid core temperature is below the saturation temperature and the bubbles collapse in the core as they are carried downstream.

At some further distance along the tube, the liquid attains the saturation temperature, and "saturated nucleate boiling heat transfer" ensues (Region C). Considerably more nucleation sites have been activated and the boiling at the surface is much more vigorous. Even through the liquid at the centerline is subcooled to a certain extent, the bulk mixed liquid temperature is saturated and for this reason the bubbles which detach from the surface no longer collapse so that the mass quality of the liquid vapour mixture increases. Eventually a point is reached where the vapour bubbles at the centerline begin to coalesce after which heat is transferred by "forced convection heat transfer through a liquid film". Evaporation supercedes boiling as the liquid film thickness diminishes (Region D/Region E/Region F) while the flow pattern progressively changes from slug flow to annular flow to annular flow with entrainment. This process is often referred to as "two phase forced convective heat transfer."

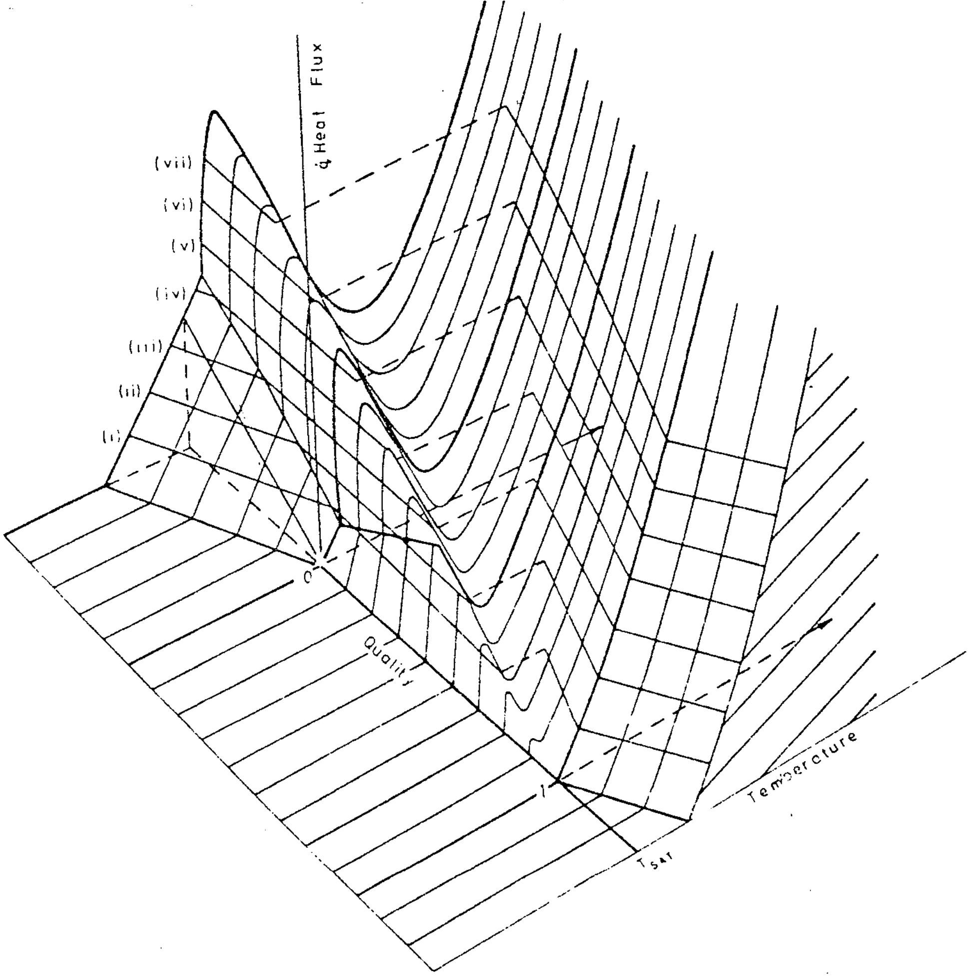
If the heated tube is long enough, the liquid film eventually evaporates completely. This phenomenon is known as "dry out" and is characterized by a drastic rise in wall temperature. Heat is transferred to a mixture of liquid droplets and vapour and the process is referred to as "liquid deficient heat transfer" (Region G). Ultimately, the droplets evaporate completely (x = 1) and the "convective heat transfer to vapour" region is entered in which both surface temperature and liquid temperature increase markedly with length.

These diverse heat transfer processes can best be put in perspective by a discussion of the variation of heat transfer coefficient (the ratio of heat flux to wall temperature/fluid temperature difference) with quality (distance along the tube axis). This interdependence is shown in the figure at the right in which increasing levels of heat flux are depicted by curves (i) through (vii). Each of the curves shows similar behaviour. Heat transfer coefficient increases with quality in the subcooled boiling region because the wall temperature/fluid temperature difference decreases linearly with length up to the point where the bulk fluid attains saturation temperature. Depending upon whether the heat flux is high or low, either saturated nucleate boiling or liquid film evaporation procedes from this point. If saturated nucleate boiling occurs initially, the heat transfer coefficient remains more or less constant because the wall temperature/fluid temperature difference hardly varies but eventually a sudden transition to saturated film boiling occurs. If liquid film evaporation occurs initially, the series of events depicted in the preceding illustration occur as indicated with a steady increases in heat transfer coefficient until complete evaporation of the film occurs after which liquid deficient heat transfer takes over.

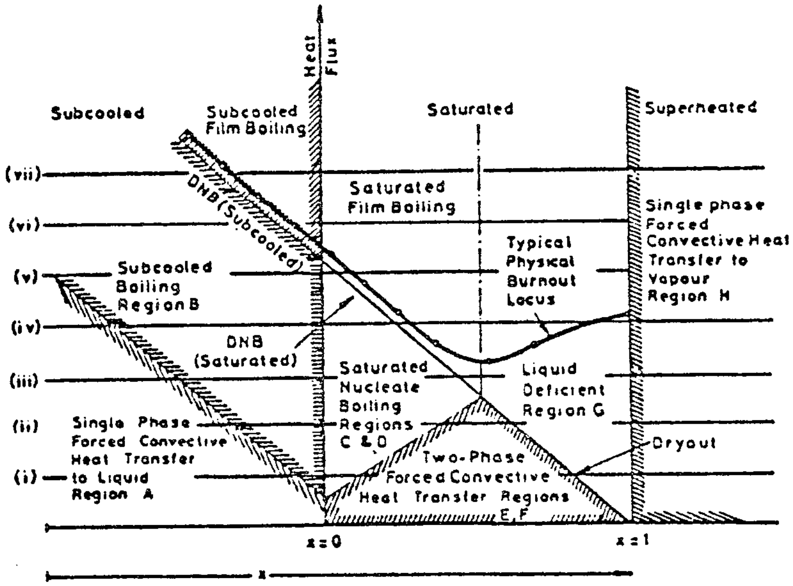


The transition from saturated nucleate boiling to saturated film boiling designated "departure from nucleate boiling" (DNB) is essentially the same phenomenon as the "critical heat flux phenomenon" (CHF) in pool boiling while the transition from forced convection heat transfer through a liquid film to liquid deficient heat transfer is known as "dryout" (DO). In either case, a marked reduction in heat transfer coefficient occurs although the value remains finite and actually increases with distance along the tube. Further insight into these processes can be gained by considering the "boiling surface" below in which the same levels of heat flux (i) through (vii) have been identified.

From this diagram, it may be appreciated that the low heat flux level curve (i) is representative of the events occurring in the uniformly heated vertical tube discussed above in as much as the single phase convective heat transfer to subcooled liquid, subcooled nucleate boiling heat transfer, saturated nucleate boiling heat transfer, two phase forced convective heat transfer, liquid deficient heat transfer and single phase convective heat transfer to superheated vapour regimes are encountered in turn. As heat flux increases, the relative significance of these various heat transfer regimes changes; curve (iv) shows that the two phase convective heat transfer regime is no longer encountered and that a transition from saturated nucleate boiling to saturated film boiling occurs before the liquid deficient heat transfer regime is entered. At high heat flux levels, curve (vii) shows that the saturated nucleate boiling regime is no longer encountered and that a transition from subcooled nucleate boiling to subcooled film boiling occurs before the flow phenomenon passes through the saturated film boiling and liquid deficient heat transfer regimes. The course of events is determined by the heat flux and initial conditions.



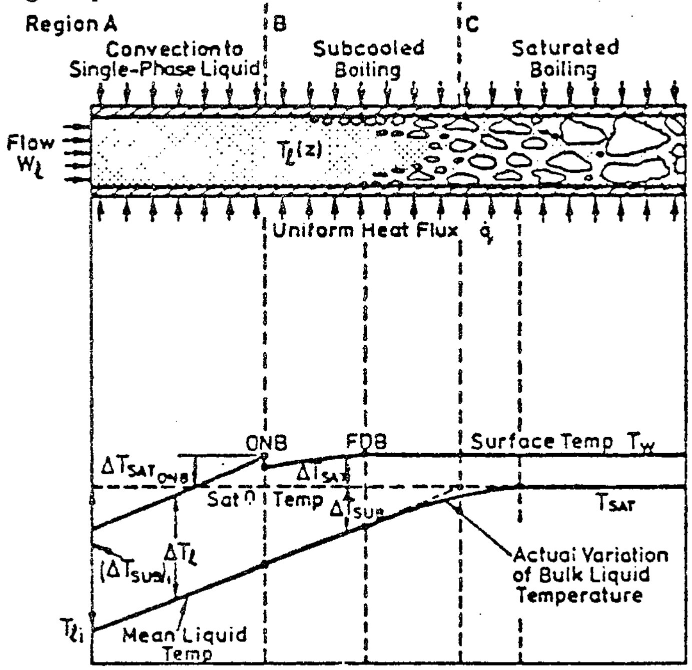
The diagram below presenting the projection of the "boiling surface" on the heat flux/quality plane enables the heat transfer processes described above to be appreciated from a different perspective.



12A.2.3 Subcooled Boiling

12A.2.3.1 Single-Phase Liquid Heat Transfer

The figure below shows the flow pattern and the variation of surface and liquid temperatures in the regions designated A, B and C. In the "single phase forced convective heat transfer to liquid region" the heat transfer relationships are well established. Typically, if the flow is fully developed and turbulent, the Dittus Boelter [27]



equation

$$\frac{\bar{\alpha} D}{k_\ell} = 0.023\left(\frac{VD}{\nu_\ell}\right)^{0.8}\left(\frac{\mu_\ell\ C_\ell}{k_\ell}\right)^{1/3}$$

will predict heat transfer coefficients at the surface satisfactorily.

12A.2.3.2 Onset of Subcooled Nucleate Boiling

The transition from forced convective heat transfer to subcooled nucleate boiling designated ONB has been investigated by Bergeles and Rohsenow [28], Davis and Anderson [29] and Frost and Dzakowich [30]. The most generally applicable relationship attributed to the latter authors may be written

$$(T_w - T_{sat})_{ONB} = \left[\frac{8\sigma\ T_{sat}}{\rho_v\ h_{fg}}\left(\frac{q/A_T}{k_\ell}\right)\right]^{0.5}\ Pr_\ell$$

The predictions of this expression are in good agreement with experimental results when there is a sufficiently wide range of "active cavity sizes" available on the heating surface. This condition is usually met by the manufactured surfaces most frequently encountered but if this is not the case, then the equation above may be considered as predicting a lower bound.

12A.2.3.3 Fully Developed Subcooled Boiling

The phenomenon which is encountered after the onset of nucleate boiling is known as "partial boiling" because patches of vigorous nucleation and bubble formation are interspersed with regions of forced convection heat transfer. However, the number of active sites increases with length along the tube because the surface temperature is increasing and ultimately the whole surface is covered with active nucleation sites and "fully developed" subcooled boiling ensues. When this occurs, velocity and subcooling no longer have a strong influence on the rate of boiling heat transfer as Rohsenow and Clark [31] determined experimentally. The relationships of pool boiling heat transfer apply, the most renowned being that developed by Rohsenow [32].

$$\left[\frac{C_\ell(T_w - T_{sat})}{h_{fg}}\right] = C_{sf}\left[\frac{q/A_T}{\mu_\ell\ h_{fg}}\left(\frac{\sigma}{g(\rho_\ell - \rho_v)}\right)^{0.5}\right]^{0.33}\left(\frac{\mu_\ell\ C_\ell}{k_\ell}\right)^{1.7}$$

The coefficient C_{sf} incorporates all of the surface effects and extensive tables of this parameter may be found in the literature for different surface/fluid combinations. For water, the exponent 1.7 should be changed to 1.0. Other researchers, most notably Jens and Lottes [33] and Thom [34] have investigated this phenomenon and presented empirical correlations for water which include the effect of system pressure as well.

12A.2.4 Saturated Boiling

12A.2.4.1 Saturated Nucleate Boiling Region

The previous discussion of subcooled nucleate boiling applies to saturated nucleate boiling as well, the only difference being that the bulk liquid temperature is now the saturation temperature and no longer varies with distance along the tube. As indicated in the preceding diagram, the wall temperature and bulk temperature vary smoothly and continuously at the apparent boundary separating subcooled and saturated boiling regions at x = 0. The rates of saturated nucleate boiling heat transfer turn out to be independent of velocity and quality and the heat transfer coefficient is constant because the wall temperature/bulk temperature difference is invariant along the length of the tube.

12A.2.4.2 Nucleate Boiling/Two Phase Forced Convection Transition

In order for nucleate boiling to occur, it is required that the wall temperature exceed the critical value for a specified heat flux. This was the criterion used to establish the relationship to predict the conditions under which the onset of nucleate boiling would occur and as was pointed out, it is valid provided that a complete range of active cavity sizes is available on the heating surface. Should the heat transfer coefficient change in some way so that this condition were no longer satisfied, then nucleate boiling would cease. This explains why it is that for low enough levels of heat flux, nucleate boiling does not occur at all whereas at higher levels of heat flux a transition occurs from "saturated nucleate boiling" to "two phase forced convective heat transfer". The equation defining the interface between these regions is given by the relationship

$$(q/A_T)_{OTPFC} = \left[\frac{8\sigma\, T_{sat}}{\rho_v\, h_{fg}}\,\bar{\alpha}_{TP}^{2}\right]^{0.5}$$

where $\bar{\alpha}_{TP}$ is the two phase heat transfer coefficient in the absence of nucleation to be discussed below.

Analogous to the onset of nucleate boiling criterion, the predictions of this relationship should be considered as representing a lower bound. Experimentation has indicated that the value of heat flux at which the transition is predicted to occur is lower than that at which the transition actually occurred, most likely because the range of active cavities available for nucleation on the surface used during the test was in some way restricted.

12A.2.5. Two Phase Forced Convection

The two phase forced convection most often encountered in channels is characterized by the annular flow pattern in which vapour flows down the passage whose walls are covered with a continuous film of liquid.

Heat is transferred by conduction through the liquid film while vapour is generated continuously at the liquid/vapour interface. Extremely high heat transfer coefficients are encountered in this flow regime, even higher than those encountered in nucleate boiling and it is for this reason that a transition from nucleate boiling to two phase forced convection occurs whenever the conditions are appropriate.

The original work in quantifying two phase forced convection heat transfer began with Martinelli [35] who correlated his experimental results with relationships of the form

$$\frac{\bar{\alpha}_{TP}}{\bar{\alpha}_{\ell o}} = \mathcal{F}\left(\frac{1}{X}\right) \qquad X = \left(\frac{\Delta p_\ell}{\Delta p_v}\right)^{0.5}$$

where $\bar{\alpha}_{\ell o}$ is the single phase liquid heat transfer coefficient based upon the assumption that the total mass flowrate of liquid and vapour flowed through the channel as liquid and X is the Martinelli parameter. Empirical relationships were proposed by Dengler and Addoms [36], Guerrieri and Talty [37], Bennett et al. [38] and Schrock and Grossman [39] [40]. One of the best correlations was that proposed by Chen [41] which assumed that both nucleate boiling and forced convection were occurring in differing amounts at different locations throughout the two phase forced convective region and attempted to predict the two phase heat transfer coefficient by combining the contributions of the two mechanisms in accordance with

$$\bar{\alpha}_{TP} = \bar{\alpha}_{NB} + \bar{\alpha}_{FC}$$

The convective contribution was given by the relationship

$$\bar{\alpha}_{FC} = 0.023 \frac{k_\ell}{D} \left[\frac{\dot{m}(1-x)D}{\mu_\ell}\right]^{0.8} \left(\frac{\mu_\ell C_\ell}{k_\ell}\right)^{0.4} F$$

and the nucleate boiling contribution was given by the relationship

$$\bar{\alpha}_{NB} = 0.00122 \left[\frac{k_\ell^{0.79} C_\ell^{0.45} \rho_\ell^{0.49}}{\sigma^{0.5} \mu_\ell^{0.29} \rho_v^{0.24} h_{fg}^{0.24}}\right] (T_w - T_{sat})^{0.24} \Delta p_{sat}^{0.75} S$$

where F and S may be determined by evaluating the relationships

$$F = 1 \text{ for } 1/X \leqslant 0.1$$

$$F = 2.35 \; (1/X + 0.213)^{0.736} \text{ for } 1/X > 0.1$$

$$S = 1/(1 + 2.53 \times 10^{-6} \, Re_\ell^{1.17})$$

This correlation fits the available experimental data very well.

References

1. Berenson, P.J., "Experiments on Pool Boiling Heat Transfer", International Journal of Heat and Mass Transfer, 5, 985-999, 1962.

2. Griffith, P. and Wallis, "The Role of Surface Conditions in Nucleate Boiling", Chemical Engineering Progress Symposium Series, 56, 49-63, 1960.

3. Hsu, Y.Y., "On the Size Range of Active Nucleation Cavities on a Heating Surface", Journal of Heat Transfer, 84, 206-216, 1962.

4. Han, C.Y. and Griffith, P., "The Mechanism of Heat Transfer in Nucleate Pool Boiling - Part I Bubble Initiation, Growth and Departure", International Journal of Heat and Mass Transfer, 8, 887-904, 1965.

5. Ibrahim, E. and Judd, R. L., "An Experimental Investigation of the Effect of Subcooling on Bubble Growth and Waiting Time in Nucleate Boiling", Journal of Heat Transfer, Accepted for Publication, 1984.

6. Judd, R.L. and Merte, H., "Evaluation of Nucleate Boiling Heat Flux Predictions at Varying Levels of Subcooling and Acceleration", International Journal of Heat and Mass Transfer, 15, 1075-1096, 1972.

7. Jakob, M. and Fritz, W., "Versuche uber den Verdampsfungsvorgang", Forschung auf dem Gebiete des Ingenierwesens 2, 435-447, 1931.

8. Rohsenow, W.M. and Clark, J.A., "A Study of the Mechanism of Boiling Heat Transfer", Transactions of the American Society of Mechanical Engineers, 73, 609-620, 1951.

9. Gunther, F.C. and Kreith, F., "Photographic Study of Bubble Formation in Heat Transfer to Subcooled Water", Proceedings of the Heat Transfer and Fluid Mechanics Institute, 1949.

10. Rallis, C.J. and Jawurek, H.H., "Latent Heat Transport in Saturated Nucleate Boiling", International Journal of Heat and Mass Transfer, 7, 1051-1068, 1964.

11. Han, C.Y. and Griffith, P., "The Mechanism of Heat Transfer in Nucleate Boiling - Part II The Heat Flux - Temperature Difference Relationship", International Journal of Heat and Mass Transfer, 8, 905 - 914, 1954.

12. Mikic, B. and Rohsenow, W.M., "A New Correlation of Pool Boiling Data Including the Effect of Heating Surface Characteristics", Journal of Heat Transfer, 91, 245-250, 1969.

13. Gaertner, R. and Westwater, J.M., "Population of Active Sites in Nucleate Boiling Heat Transfer", Chemical Engineering Progress Symposium Series, 56, 39-48, 1960.

14. Judd, R.L. and Hwang, K.S., "A Comprehensive Model for Nucleate Pool Boiling Heat Transfer Including Microlayer Evaporation", Journal of Heat Transfer, 98, 623-629, 1976.

15. Judd, R.L. and Fath, H.S., "Influence of System Pressure on Microlayer Evaporation Heat Transfer", Journal of Heat Transfer, 100, 49-55, 1978.

16. Rohsenow, W.M., and Griffith, P., "Correlation of Maximum Heat Transfer Data for Boiling of Saturated Liquids", Chemical Engineering Progress Symposium Series 52, 47-49, 1956.

17. Zuber, N., Tribus, M. and Westwater, J.W., "The Hydrodynamic Crisis in Pool Boiling of Saturated and Subcooled Liquids", International Developments in Heat Transfer, American Society of Mechanical Engineers, 230-235, 1961.

18. Kutateladze, S. S., "Boiling Heat Transfer", International Journal of Heat and Mass Transfer, 4, 1961.

19. Sun, K.H. and Lienhard, J.H., "The Peak Pool Boiling Heat Flux on Horizontal Cylinders", International Journal of Heat and Mass Transfer, 13, 1425-1439, 1970.

20. Lienhard, J.H. and Dhir, V.K., "Hydrodynamic Prediction of Peak Pool Boiling Heat Fluxes from Finite Bodies", Journal of Heat Transfer, 95, 152-158, 1973.

21. Lienhard, J.H., Dhir, V.K. and Riherd, D.M., "Peak Pool Boiling Heat Flux Measurements on Finite Horizontal Flat Plates", Journal of Heat Transfer, 95, 477-482, 1973.

22. Bromley, C.A., "Heat Transfer in Stable Film Boiling", Chemical Engineering Progress Symposium Series, 46, 221-227, 1950.

23. Breen, B.P. and Westwater, J.W., "Effect of Diameter of Horizontal Tubes on Film Boiling Heat Transfer", Chemical Engineering Progress Symposium Series, 58, 67-72, 1962.

24. Berenson, P.J., "Transition Boiling Heat Transfer from a Horizontal Surface", Journal of Heat Transfer, 83, 351-358, 1961.

25. Bergeles, A.E., Collier, J.G., Delhaye, J.M., Hewitt, G.F. and Mayinger, F., "Two-Phase Flow and Heat Transfer in the Power and Process Industries", Hemisphere Publishing Corporation, 1981.

26. Collier, J.G., "Convective Boiling and Condensation", McGraw Hill Book Company (UK) Limited, 1972.

27. Boelter, L.M.K., Cherry, V.H., Johnson, H.A. and Martinelli, R.C., "Heat Transfer Notes", McGraw Hill Book Company, 1965.

28. Bergeles, A.E. and Rohsenow, W.M., "The Determination of Forced Convection Surface Boiling Heat Transfer", Journal of Heat Transfer, 86, 365-372, 1964.

29. Davis, E.J. and Anderson, G.H., "The Incipience of Nucleate Boiling in Forced Convection Flow", American Institute of Chemical Engineers Journal, 12, 774-780, 1966.

30. Frost, W. and Dzakowich, G.S., "An Extension of the Method of Predicting Incipient Boiling on Commercially Finished Surfaces", ASME AIChE Heat Transfer Conference, Paper 67 - HT - 61, 1967.

31. Rohsenow, W.M. and Clark, J.A., "Heat Transfer and Pressure Drop Data for High Heat Flux Densities To Water at High Subcritical Pressure", Proceedings of the Heat Transfer and Fluid Mechanics Institute, 1951.

32. Rohsenow, W.M., "A Method of Correlating Heat Transfer Data for Surface Boiling of Liquids", American Society of Mechanical Engineers Transactions, 74, 969-975, 1952.

33. Jens, W.H. and Lottes, P.A., "Analysis of Heat Transfer Burnout, Pressure Drop and Density Data for High Pressure Water", Argonne National Laboratory Report, ANL - 4627, 1951.

34. Thom, J.R.S., Waller, W.M., Fallon, T.A., and Reising, G.F.S., "Boiling in Subcooled Water During Flow Up Heated Tubes or Annuli", Institute of Mechanical Engineers Symposium on Boiling Heat Transfer in Steam Generating Units and Heat Exchangers, Paper 6, 1965.

35. Martinelli, R.C. and Nelson, D.B., "Prediction of Pressure Drop During Forced Circulation Boiling of Water", Transactions of the American Society of Mechanical Engineers, 695-702, 1948.

36. Dengler, C.E. and Addoms, J.N., "Heat Transfer Mechanism for Vapourization of Water in a Vertical Tube", Chemical Engineering Progress Symposium Series, 52, 95-103, 1956.

37. Guerrieri, S.A. and Talty, R.D., "A Study of Heat Transfer to Organic Liquids in Single Tube Natural Circulation Vertical Tube Boilers", Chemical Engineering Progress Symposium Series, 52, 69-77, 1956.

38. Bennett, J.A.R., Collier, J.G., Pratt, H.R.C. and Thornton, J.D., "Heat Transfer to Two-Phase Gas-Liquid Systems", Atomic Energy Research Establishment Report, AERE-R-3519, 1961.

39. Schrock, V.E. and Grossman, L.M., "Forced Convection Boiling Studies", University of California Report 73308-UCZ21821, 1952.

40. Schrock, V.E. and Grossman, L.M., "Forced Convective Boiling in Tubes", Nuclear Science and Engineering, 12, 474-480, 1962.

41. Chen, J.C., "Correlation for Boiling Heat Transfer to Saturated Liquids in Convective Flow", Industrial Engineering Chemistry, Process Design and Development, 5, 322-327, 1966.

Nomenclature

Roman Letters

a	acceleration, m/s^2
C_ℓ	liquid specific heat, J/kg K
D	tube diameter, m
D_d	bubble departure diameter, m
f	bubble departure frequency, m
g	gravitational acceleration, m/s^2
g_c	gravitational constant, 1 kg m/Ns^2
$h(z)$	mixture enthalpy at position z, J/kg
h_{fg}	latent heat of vapourization, J/kg
h_ℓ	liquid enthalpy at tube inlet, J/kg
k_ℓ	liquid thermal conductivity, W/m^oC
k_v	vapour thermal conductivity, W/m^oC
K	influence area/projected area ratio, dimensionless
L	heater length, m
$\dot{m}$	mass flowrate, kg/s
N/A_T	active site density, $sites/m^2$
p_ℓ	liquid pressure, N/m^2
p_v	vapour pressure, N/m^2
Δp_ℓ	liquid pressure drop, N/m^2
Δp_v	vapour pressure drop, N/m^2
Pr_ℓ	liquid Prandtl number, dimensionless
q/A_T	overall surface heat flux, W/m^2
q_{crit}/A_T	critical surface heat flux, W/m^2
q_{ME}/A_T	microlayer evaporation heat flux, W/m^2
q_{NB}/A_{NB}	nucleate boiling heat flux, W/m^2

q_{NC}/A_{NC}	natural convection heat flux, W/m^2
Q_{Bubble}	quantity of heat transported per bubble, J
r	radius, m
r_c	cavity radius, m
r_n	nucleus radius, m
R_b	bubble departure radius, m
T_{sat}	saturation temperature, ^{o}C
(T_w-T_{sat})	wall superheat, ^{o}C
$(T_{sat}-T_{\infty})$	liquid subcooling, ^{o}C
(T_v-T_{sat})	vapour superheat, ^{o}C
V	velocity, m/s
X	Martinelli parameter, dimensionless
$X(z)$	mixture quality at position z, dimensionless
z	position from tube inlet, m

Greek Letters

$\bar{\alpha}$	heat transfer coefficient, $W/m^2\ ^{o}C$
β	contact angle, dimensionless
δ	thickness, m
ε	surface emissivity, dimensionless
κ_{ℓ}	liquid thermal diffusivity, m/s^2
λ_c	critical wavelength, m
μ_{ℓ}	liquid dynamic viscosity, N_s/m^2
μ_v	vapour dynamic viscosity, N_s/m^2
ν_{ℓ}	liquid kinematic viscosity, N_s/m^2
ν_v	vapour kinematic viscosity, N_s/m^2
ρ_{ℓ}	liquid density, kg/m^3

ρ_v	vapour density, kg/m^3
σ	surface tension, N/m
τ_g	growth time, s
τ_w	waiting time, s