CHAPTER 4

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

ABSTRACT

This chapter introduces the concepts of nucleate boiling heat transfer, forced convective boiling heat transfer, critical heat flux and burnout as well as post dryout heat transfer and rewetting. 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 upon consulting it. The subsequent chapters will build upon the material presented herein.

4. HEAT AND MASS TRANSFER AND TWO PHASE FLOW II

4.1 Pool Boiling Phenomena

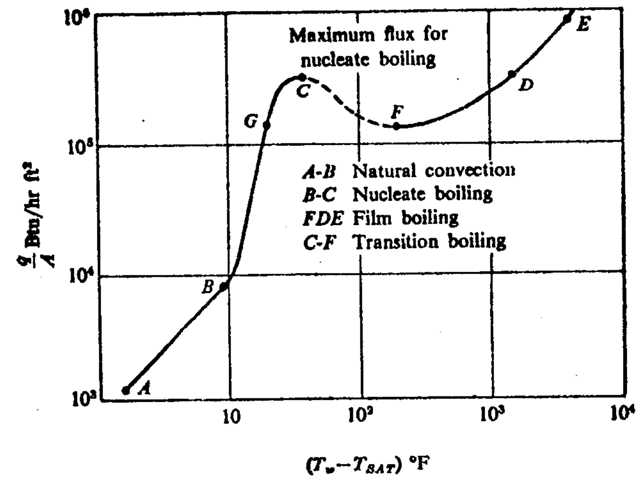
An understanding of pool boiling is fundamental to the design of steam generators and the study of two phase flow heat transfer. All of the phenomena which occur in two phase flow boiling occur in pool boiling as well but fewer independent variables need to be considered. Certain aspects of two phase 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 LOCA since the processes of vapour bubble nuucleation, formation and growth which had been suppressed under reactor operating conditions would become active in such a situation. The conditions which pertain 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.

4.1.1 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

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

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 at the right for which different functional relationships between heat flux and temperature difference pertain. 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_∞) are important so the preceding equation should be written

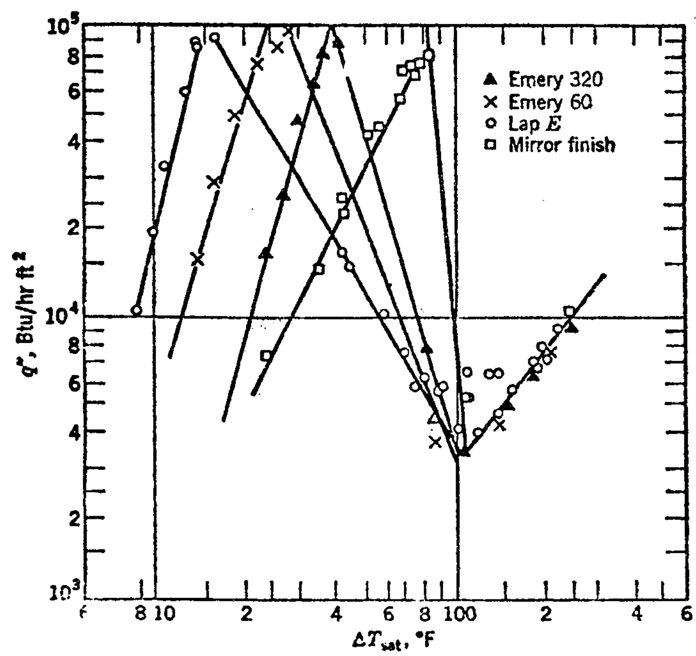
$$q/A = \mathfrak{F}[(T_w - T_{sat}), (T_{sat} - T_\infty), \text{Fluid Properties, Surface Conditions}]$$

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.

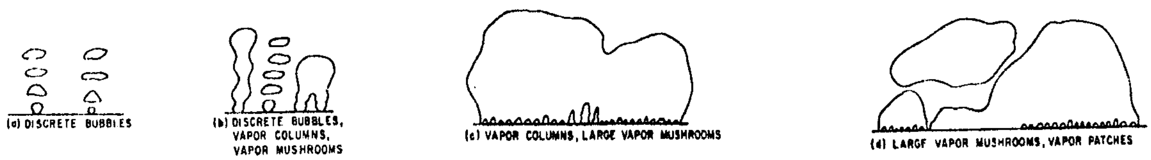
Should heater surface not fail, then operation along curve DE is possible. A continuous layer of vapour blankets the surface and although the interface between the liquid and vapour is in continuous motion because of the regular formation of vapour globules at discrete points, the liquid never contacts 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 the fact 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.



However, there is little difference in the quantity of heat transmitted per bubble. From these considerations

$$q/A = J[(T_w - T_{sat}),\ (T_{sat} - T_\infty),\ N/A_T,\ f,\ q_{Bubble}]$$

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.

4.1.2 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

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

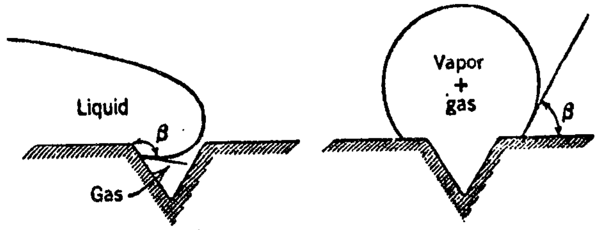
where σ is the surface tension at the interface of the vapour and liquid phases. For a liquid at pressure p_ℓ, the vapour pressure p_v of the superheated liquid near the wall can be related to the superheat $(T_v - T_{sat})$ by the Clausius Clapeyron equation

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

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.



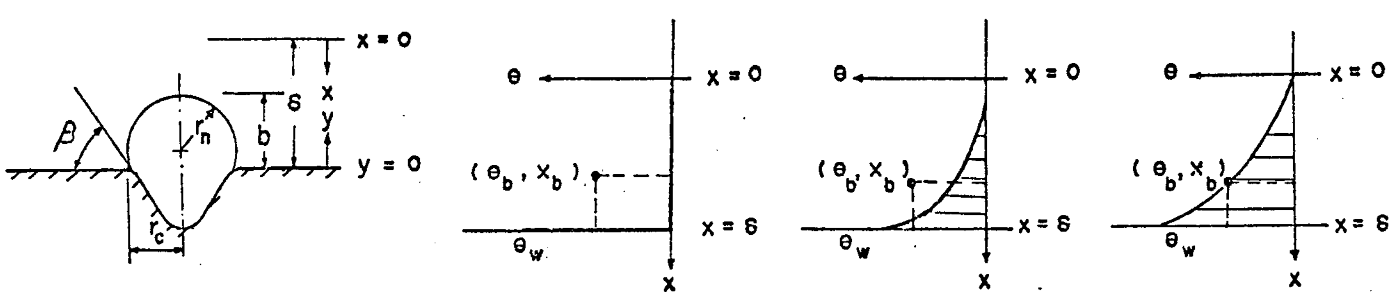
The angle β is called the contact angle. A fluid-surface combination with a large value of β has a better chance of trapping gas within the cavity by capillary action than a fluid-surface combination with a small value of β. In the former case, the surface is "non wetted" whereas in the latter case, the surface is "totally wetted".

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.

Hsu [3] solved the problem by formulating a model to explain the initiation of a bubble from a nucleus sitting at the mouth of a cavity. This nucleus is assumed to be formed from vapour from 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.

4.1.3 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 treat transfer are too simplistic. Judd and Merte [5] 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 one bubble transports one bubble volume of latent heat from the surface as it departs or collapses. This explanation was originally advanced by Jakob [6] who was unable to verify it for lack of the necessary measurements. The model is expressed mathematically by the relationship

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

Rohsenow and Clark [7] analysed the results of Gunther and Kreith [8] 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 [9] have lent support to this model with their experimental results.

Forster and Grief [10] proposed the vapour liquid exchange model to explain the action of the bubbles with respect to the transfer of heat. The postulate of this model is that each bubble departing or collapsing at the heat transfer surface exchanges a volume of liquid at the surface temperature for a volume of liquid at the bulk temperature. This model is expressed mathematically by the relationship

$$q_{NB}/A_T = \rho_\ell\, C_\ell\, V_{Bubble}\, (T_w - T_\infty)\, (N/A_T)\, f$$

The implication of this model is that subcooling should greatly influence the rate of heat transfer since

$$(T_w - T_\infty) = (T_w - T_{sat}) + (T_w - T_\infty)$$

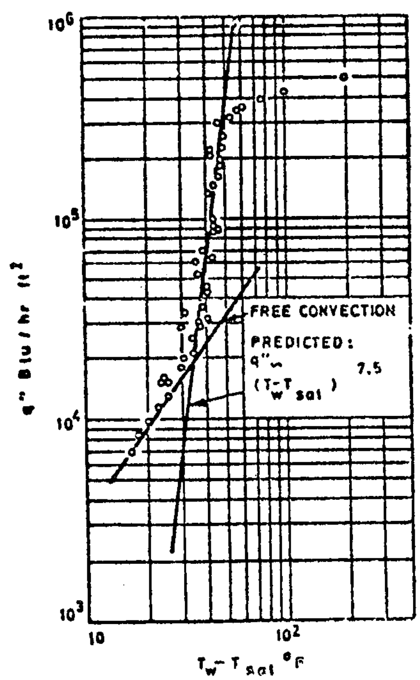
but it is a matter of experience that subcooling has hardly any effect at all on the rate of heat transfer. Most likely subcooling effects V_{Bubble}, N/A_T and f in the opposite sense to that in which it effects $(T_w - T_\infty)$.

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 connection heat transfer term. The model is expressed mathematically by the relationship

$$q_{NB}/A_T = \frac{1}{2}\rho_\ell C_\ell D_d^2 \delta_d \left[4 - \frac{1}{3}(1-\delta/\delta_d)\right] (T_w - T_\infty)(N/A_T) f$$

where $\delta = \sqrt{\pi \alpha_\ell \tau_g}$ and $\delta_d = \sqrt{\pi \alpha_\ell (\tau_g + \tau_w)}$. 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.



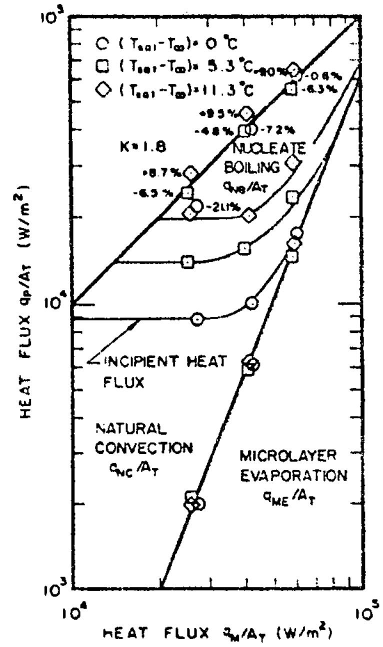
As the diagram at the right indicates, the heat fluxes predicted by the Mikic and Rohsenow model are in excellent agreement with those measured by Gaertner and Westwater. The model was also capable of predicting the heat flux-superheat characteristics at different pressures once the exponent m had been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

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

$$q/A_T = 4.35 + 10^8 \left(\frac{N}{A_T}\right) f\, \bar{V}_{ME} + 184\, (T_w - T_\infty)^{4/3} \left[1 - K\pi R_b^2 (N/A_T)\right]$$

$$+ 1543\, R_b^2 \sqrt{f} \left(\frac{N}{A_T}\right)(T_w - T_\infty)$$

which included 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.



4.1.4 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 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

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

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

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

but these correlations are based upon limited data and consequently more specific information such as that provided by Sun and Lienhard [18], Lienhard and Dhir [19] and Lienhard, Dhir and Riherd [20] has to be consulted when heater geometry and size become important. More will be said later about the peak nucleate boiling heat flux in connection with burnout and rewetting.

4.1.5 Transition Boiling

If 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.

4.1.6 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 [21] derived the following relationship for the average heat transfer coefficient for a vertical plate of height L

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

where

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

Breen and Westwater [22] adapted this solution for horizontal tubes of diameter D obtaining the relationship

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

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 [23] who presented the relationship

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

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

$$\bar{h}_t = \bar{h} + \bar{h}_r$$

where

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

4.2 Forced Convective Boiling Phenomena

4.2.1 Effect of Flow on Boiling Heat Transfer Rates

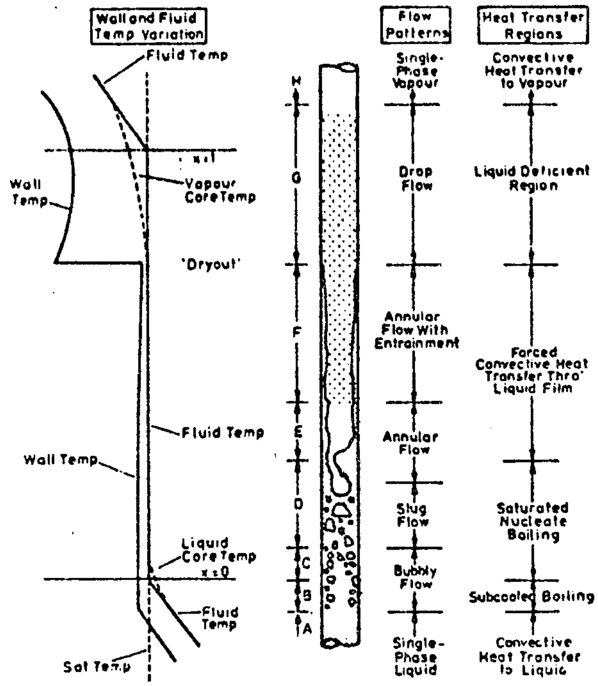
It would seem reasonable that the rates of boiling heat transfer would be affected when a velocity field was superimposed, either by agitating the fluid over a pool boiling surface or by imposing an internal flow on the boiling phenomenon occurring inside a flow passage, because of the pronounced effect 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 heat transfer although the rate at which vapour bubbles are removed from the boiling surface can be significant. 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.

4.2.2 Regions 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 [24], [25] while at the same time the problem is representative 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

$$x(z) = \frac{h(z) - h_\ell}{h_{fg}}$$

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 fluid is subcooled or superheated respectively.

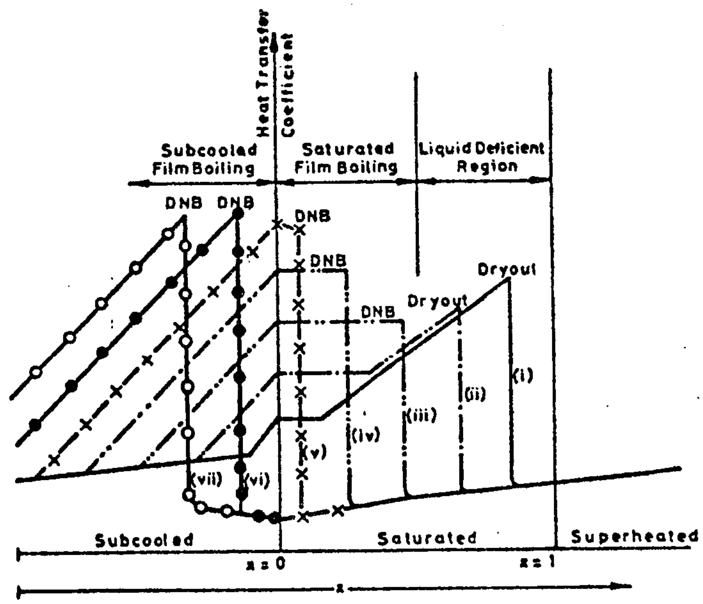


The illustration predicts the various heat transfer regimes which will be traversed in turn as the coolant flows upward. At the lower end of the tube, "single phase convective heat transfer" occurs (Region A) until the wall temperature reaches the saturation temperature. "Subcooled nucleate boiling heat 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.

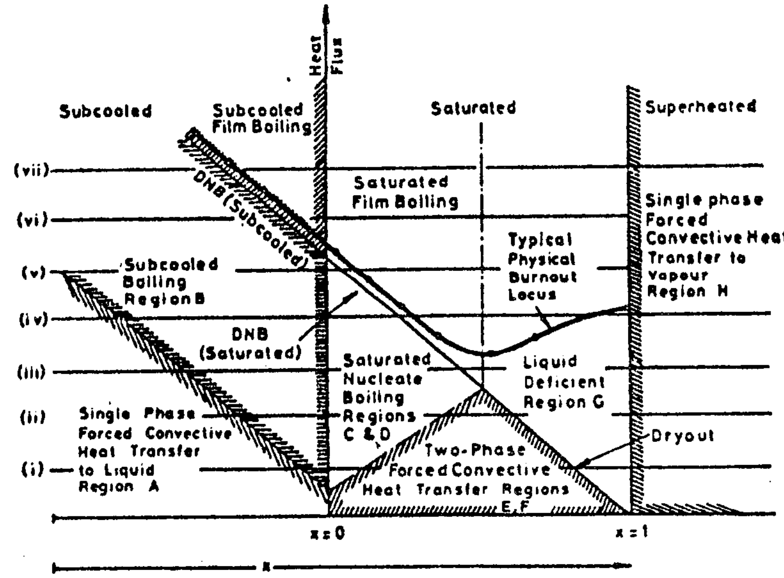
At some further distance along the tube, the liquid attains the saturation temperature, and "saturated nucleate boiling heat transfer" ensues (Region C). Considerably more nucleation sites have been activated and the boiling at the surface is much more vigorous. Even through the liquid at the centerline 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.



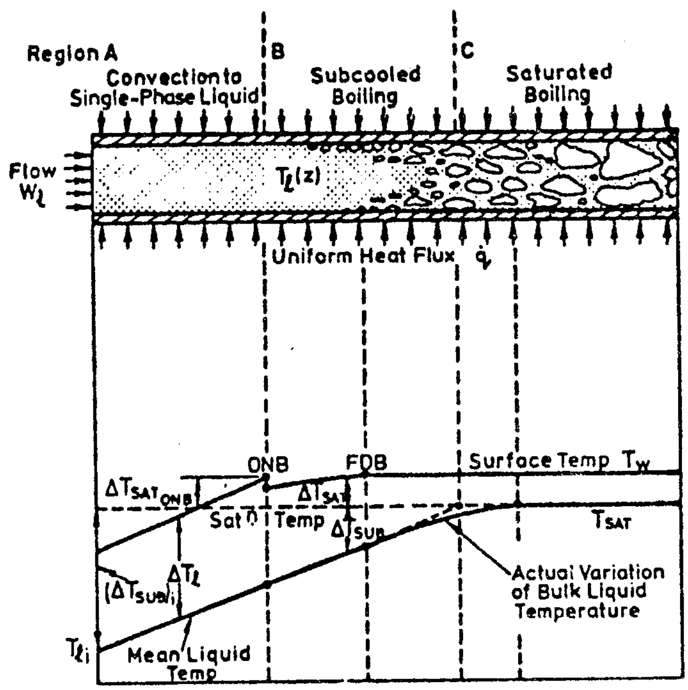
The transition from saturated nucleate boiling to saturated film boiling designated "departure from nucleate boiling" (DNB) is essentially the same phenomenon as the "critical heat flux phenomenon" (CHF) in pool boiling. The transition from forced convection heat transfer through a liquid film to liquid deficient heat transfer is known as "dryout". 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 heat flux/quality curve presented below in which the same levels of heat flux (i) through (vii) have been identified



4.2.3 Subcooled Boiling

4.2.3.1 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, Band 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 [26] equation

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

will predict heat transfer coefficients at the surface satisfactorily.

4.2.3.2 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 [27], Davis and Anderson [28] and Frost and Dzakowich [29]. 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.

4.2.3.3 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 [30] determined experimentally. The relationships of pool boiling heat transfer apply, the most renowned being that developed by Rohsenow [31].

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

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 [32] and Thom [33] have investigated this phenomenon and presented empirical correlations for water which include the effect of system pressure as well.

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. 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.

19. 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.

20. 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.

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

22. 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.

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

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

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

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

28. 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.

29. 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.

30. 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.

31. 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.

32. 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.

33. 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.

34. 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.

35. 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.

32. 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.

37. 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.

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

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

40. 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.

41. Hewitt, G.F., "Critical Heat Flux in Flow Boiling", Sixth International Heat Transfer Conference, Keynote Lecture, 1978.

42. Bertonni, R., Cipriani, R. and Cumo, M., "Up-flow and Down-flow Burnout", CNEN Report CNEN/RT/ING (76) 24, 1976.

43. Lienhard, J.H. and Eichhorn, R., "Peak Boiling Heat Flux on Cylinders in a Cross Flow", International Journal of Heat and Mass Transfer, 19, 1135-1141, 1976.

44. Hewitt, G.F., Kersey, H.A. and Collier, J.C., "The Correlation of Critical Heat Flux for the vertical Flow of Water in Uniformly Heated Tubes, Annuli and Rod Bundles", Atomic Energy Research Establishment Report, AERE-R-5590, 1969.

45. Whalley, P.B., Hutchinson, P. and Hewitt, G.F., "The Calculation of Critical Heat Flux in Forced Convective Boiling", Heat Transfer, 4, 290-294, 1974.

46. Merilo, M., "Critical Heat Flux Experiments in a Vertical and Horizontal Tube with Both Freon and Water as Coolant", Nuclear Engineering Design, 44, 1-16, 1977.

47. Merilo, M., "Fluid-to-Fluid Modelling and Correlation of Flow Boiling Crisis in Horizontal Tubes", International Journal of Multiphase Flow, 5, 313-325, 1979.

48. Ahmad, S.Y., Nickerson, J.R. and Midvidy, W.I., "Critical Heat Flux Experiments in a Horizontal 37 Element Bundle Cooled by Water and Freon", Seventh International Heat Transfer Conference, 1982.

49. Groeneveld, D.C., "Post-dryout Heat Transfer at Reactor Operating Conditions", Atomic Energy of Canada Limited Report, AECL-4513, 1973.

50. Slaughterback, D.C., Vesely, W.E., Ybarrondo, L.J., Condie, K.G. and Mattson, R.J., "Statistical Regression Analysis of Experimental Data for Flow Film Boiling Heat Transfer", ASME AIChE Heat Transfer Conference, 1973.

51. Cumo, M. and Urbani, G.C., "Post-burnout Heat Transfer (Attainable Precision Limits of the Measured Coefficient)", CNEN Report CNEN/RI/ING (74)24, 1974.

52. Mattson, R.J., Condie, K.G., Bengston, S.J. and Obenchain, C.F., "Regression Analysis of Post-CHF Flow Boiling Data", Fifth International Heat Transfer Conference, Paper B3.8, 1974.

53. Yamanouchi, A., "Effect of Core Spray Cooling in Transient State After Loss-of-Coolant Accident", Journal of Nuclear Science and Technology, 5, 547-558, 1968.

54. Elias, E. and Yadigaroglu, G., "A General One-Dimensional Model for Conduction Controlled Rewetting of a Surface", Nuclear Engineering and Design, 42, 185-194, 1977.

55. Salcudean, M. and Rahman, M., "An Analytical Solution for the Heat Conduction During the Rewetting of Hot Horizontal Channels", Transactions of the Canadian Society for Mechanical Engineers, 6, 106-114, 1980-81.

56. Chan, A.M.C. and Banerjee, S., "Refilling and Rewetting of a Hot Horizontal Tube - Part II: Structure of a Two Fluid Model", Journal of Heat Transfer, 103, 287-292, 1981.

57. Chan, A.M.C. and Banerjee, S., "Refilling and Rewetting of a Hot Horizontal Tube - Part III: Application of a Two-Fluid Model to Analyse Rewetting", Journal of Heat Transfer, 103, 653-659, 1981.

58. Chan, A.M.C. and Banerjee, S., "Refilling and Rewetting of a Hot Horizontal Tube - Part I: Experiments", Journal of Heat Transfer, 103, 281-286, 1981.