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**PREDICTION OF THE ONSET OF SIGNIFICANT VOID IN
TRANSIENT SUBCOOLED FLOW BOILING (U)**

by
S. C. LEE, S. G. BANKOFF

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SAVANNAH RIVER TECHNOLOGY CENTER, AIKEN, SC 29808
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SRTC Perspective

D. A. Coutts

Senior Engineer

The coolant in the Savannah River Site (SRS) production nuclear reactor assemblies is circulated as a subcooled liquid under normal operating conditions. This coolant is evenly distributed throughout multiple annular flow channels with a uniform pressure profile across each coolant flow channel. During the postulated Loss of Coolant Accident (LOCA), which is initiated by a hypothetical guillotine pipe break, the coolant flow through the reactor assemblies is significantly reduced. The flow reduction and accompanying power reduction (after shutdown is initiated) occur in the first 1 to 2 seconds of the LOCA. This portion of the LOCA is referred to as the flow instability (FI) phase (1).

The reactor assemblies are subject to a type of FI known as a Ledinegg flow excursion. This type of instability is common in low pressure subcooled flow channels that exhibit a demand curve with a negative sloped region and a relative maximum and minimum as shown in Figure 1. The relative minimum is the result of the flow acceleration created by the change in fluid density, and an increase in the frictional pressure loss created by vapor nucleating on the heater surface and effectively increasing the flow channel roughness. The postulated LOCA can initiate such a flow excursion.

Assembly power limits are calculated for LOCA conditions using the FLOWTRAN-FI code. The code calculates a dimensionless ratio known as the Stanton number at each axial position in the channel through the course of the transient. This Stanton number was introduced by Saha and Zuber (2) to describe the onset of significant void (OSV) in channels where subcooled nucleate boiling is present.

$$(1) \quad St = \frac{\phi}{Gc_p(T_{sat} - T_b)}$$

where:

ϕ = local heat flux
 G = mass velocity
 c_p = liquid specific heat
 T_{sat} = saturation temperature
 T_b = bulk liquid temperature

The Stanton number will vary both with time and position in the assembly. The reactor operating power is limited to preclude Stanton numbers in excess of 0.00455 during the LOCA. This value is 70 percent of the 0.065 value proposed by Saha and Zuber (2). This difference is a safety margin that accounts for data and calculation uncertainties.

OSV and the Stanton number criterion are steady-state concepts. It is desirable to know the uncertainties associated with their application to a transient process. The Experimental Thermal-Hydraulics Group of the Nuclear Engineering Section contracted SGB Associates of Evanston, IL to study the transient FI process. This work has been completed. Table 1 presents a summary of the reports produced during this study.

References

1. *Action Plan to Upgrade Thermal-Hydraulic Limits for Savannah River Site Production Reactors (U)*. 30 June 1989. WSRC-RP-89-373.
2. Saha, P., and N. Zuber, 1974, Point of Net Vapor Generation and Vapor Void Fraction in Subcooled Boiling. In *Proceedings of the 5th International Heat Transfer Conference, Tokyo, Japan, September 3-7, 1974*, 175-79.

Figure 1. Demand curves for subcooled isothermal and diabatic systems

Table 1--Reports prepared as part of AX-1811457

Authors	Title	Document number
Bankoff, S. G., S. C. Lee, & A. Knaani	Literature Review on Forced-Convection Subcooled Boiling	WSRC-TR-91-98
Lee, S. C., & S. G. Bankoff	Stability of Forced Convection Subcooled Boiling	WSRC-TR-92-233
Dorra, H., S. C. Lee, and S. G. Bankoff	A Critical Review of Predictive Models for the Onset of Significant Void in Forced-Convection Subcooled Boiling	WSRC-TR-93-404
Lee, S. C., and S. G. Bankoff	Prediction of the Onset of Significant Void in Transient Subcooled Flow Boiling	WSRC-TR-93-405
Gehrke, Volker, & S. G. Bankoff	Stability of Forced-Convection Subcooled Boiling in Steady-State and Transient Annular Flow	WSRC-TR-93-406
Dorra, H., S. C. Lee, and S. G. Bankoff	Onset of Significant Void in Internal Flow, Forced Convection, Subcooled Boiling	WSRC-TR-93-407

Prediction of the Onset of Significant Void in Transient Subcooled Flow Boiling¹

S. C. Lee² and S. G. Bankoff
Chemical Engineering Department
Northwestern University
Evanston, IL 60208-3120

June 30, 1992

¹ Work supported by Savannah River Laboratory, Westinghouse Savannah River Co., Aiken, S.C.

² Visiting Scholar, Present address: Department of Mechanical Engineering, Yeungnam University, Gyongsan, Korea 712-749

ABSTRACT

A new model to predict the onset of significant void (OSV) in transient subcooled flow boiling has been developed. The model is based upon the influence on vapor bubble departure of the single-phase temperature profile. The steady-state result of the present model was compared to the experimental data of Whittle and Forgan [1] and Dougherty, et al. [2], showing an excellent agreement. The model was then employed in a transient analysis of OSV for vertical downwards turbulent flow to predict whether OSV takes place. The condition for OSV to occur in transient flow situations was also predicted by this model. Two modes for pressure gradient change inside the channel are considered in the present study: step change and ramp change. The calculations were made for various combinations of the flow operating condition and the mode of pressure drop change.

KEYWORDS: Onset of Significant Void, Bubble Departure, Transient Subcooled Boiling, Vertical, Parallel Plates,

1. INTRODUCTION

In a particular reference loss-of-flow accident, the flow through the channels decreases rapidly, owing to the decrease in the pressure difference across the channels resulting from the pipe break. The power also decreases, following the reactor trip. There is thus a race between the decreases in flowrate and power which determines the conditions within the channels, and ultimately the temperature of the fuel elements. Normally, the flow is all-liquid, but as the flow decreases, it is possible to have the onset of nucleate boiling (ONB). This corresponds to the appearance of the first bubbles on the walls. Since the bulk of the liquid at the point of appearance is subcooled, the bubbles usually do not immediately detach from the wall. As the flow decreases further, the point of ONB moves backwards along the wall surface, resulting in a formation of a bubble boundary layer in the vicinity of the wall. The appearance of the bubble boundary layer causes the pressure gradient to increase, owing to the increased apparent roughness of the wall. At some point, owing to the detachment of the individual bubbles or the instability of the bubble boundary layer, there is significant vapor generation and mixing of bubbles with the core liquid. This onset of significant void (OSV) generally signals the onset of flow instability (OFI) in a system with pressure-driven boundary conditions. This is because the sudden decrease in mean flow density results in an increased pressure gradient due to the combined effects of friction and acceleration. With pressure boundary conditions, rather than specified inlet flowrates, this can lead rapidly to OFI, and consequent dryout and overheating.

Current methods of predicting the onset of significant void, and of the subsequent flow instability, are based on steady-state, small-scale experiments on forced-convection, subcooled nucleate boiling (FCSNB). This is appropriate, since no experimental studies have come to our attention involving transient FCSNB. The current method of choice is to use the empirical correlation of Saha and Zuber [3], together with a calculation for the velocity and temperature of the bulk liquid as functions of position and time throughout the transient. However, a more mechanistic approach is needed in predicting OFI for the transient case, since the time of the transient is comparable to the residence time of a fluid particle in the channel, and also to the time delays

for the development of the thermal and velocity boundary layers at the wall.

The objective of the present study is to develop a mechanistic model for OSV and also to establish a criterion for OSV under transient flow conditions. Towards these aims, the hydrodynamic condition for the departure of an isolated vapor bubble in a vertical downwards flow between parallel plates was established from a force balance acting around the bubble. Then a transient analysis of the two-dimensional momentum and energy equations was carried out to obtain transient temperature profiles when the pressure gradient and the heat flux are time-dependent. Following the criterion suggested by Levy [4] and other investigators [5-7], the onset of significant void in transient vertical flow boiling was predicted. In addition, the present theory was compared with the Saha-Zuber correlation [3], which has been widely used in computer codes for nuclear safety analysis.

2. DESCRIPTION OF A MECHANISTIC MODEL FOR OSV

The present model aims at predicting the OSV heat flux in transient vertical downwards flow between parallel plates, provided that the operating pressure, the pressure drop and the liquid inlet temperature are prescribed. The basic concept of the present model is similar to that of previous models proposed by such investigators as Levy [4], Staub [5] and Rogers, et al. [6]. However, the present model is based upon a full two-dimensional analysis of fluid flow and heat transfer, whereas the previous models employed a one-dimensional heat transfer equation, although the temperature distribution obtained by Martinelli [8] was used to evaluate the liquid temperature at the top of the bubble.

Experimental evidence shows that the onset of significant void (OSV) precedes and is very close to OFI [2,9]. Furthermore, the OSV phenomenon is characterized by the departure of vapor bubbles from the heated surface. Thus, in the present model, attention is focused on the thermal and hydrodynamic conditions under which a bubble starts to leave the surface to predict OSV.

Two requirements are considered in the present model to establish the criterion for the departure of an isolated bubble. One is that the resultant of the forces tending to detach the bubble from the surface exceeds the resultant of the forces holding it on the surface at bubble departure. The other is a thermal condition, which states that the temperature at the top of the bubble should be at least equal to the saturation temperature for bubble departure.

The basic concept of the present model is described in detail elsewhere[10].

2.1 Hydrodynamic Condition of bubble departure

Consider a vapor bubble attached to the vertical wall in forced-convection flow boiling. The bubble is assumed to be a sphere. Following Levy [2] and other investigators [5-7], it is assumed that the forces parallel to the surface are critical in determining the condition of bubble departure. Thus only three major forces will be considered here. These are the drag force, the surface tension force, and the buoyancy force.

Based upon the force balance around the bubble, the condition for the bubble detachment

can be written as follows:

$$F_D - F_B - F_S \geq 0 \quad (1)$$

where the drag force, F_D , is the component acting to detach the bubble, while the buoyancy force, F_B , and the surface tension force, F_S , are the components tending to hold it on the surface. The buoyancy force is given by:

$$F_B = C_B(\rho_f - \rho_g)g\left(\frac{4}{3}\pi R_b^3\right) \quad (2)$$

where C_B is a proportionality constant and may be determined empirically. Similarly, the surface tension force, F_S , can be expressed as follows:

$$F_S = C_S\sigma(2\pi R_b) \quad (3)$$

where C_S is a proportionality constant. The drag force is proportional to the projected area of the bubble and the dynamic pressure of the liquid flow, and thus can be written as:

$$F_D = C_D\left(\frac{1}{2}\rho_f u_m^2\right)(\pi R_b^2) \quad (4)$$

Substituting Equations (2), (3), and (4) into Equation (1) and converting it into dimensionless form, one obtains:

$$K_1(R_b^*)^3 - R_b^* + K_2 \leq 0 \quad (5)$$

where the dimensionless bubble radius, R_b^* , and the dimensionless parameters, K_1 and K_2 , are given by:

$$R_b^* = \frac{R_b}{H} \quad (6)$$

$$K_1 = \frac{8}{3}\left(\frac{C_B}{C_D}\right)\left[\frac{(\rho_f - \rho_g)gH}{\rho_f u_m^2}\right] \quad (7)$$

$$K_2 = \left(\frac{C_S}{C_D}\right)\left(\frac{4\sigma}{\rho_f u_m^2 H}\right) \quad (8)$$

Thus, if $4K_1K_2 \ll 1$, the dimensionless bubble radius when the bubble starts to depart from the heated wall can be approximated as follows:

$$R_b^* = \frac{R_b}{H} \simeq K_2 \quad (9)$$

Thus, the dimensionless bubble diameter at the bubble departure is inversely proportional to the Weber number:

$$\frac{D_b}{H} = \left(\frac{C_S}{C_D}\right) \left(\frac{8\sigma}{\rho_f u_m^2 H}\right) \quad (10)$$

The evaluation of the coefficients, C_S and C_D , in Equation (10) was made in a previous paper [10]. As mentioned in the earlier paper, the drag coefficient, C_D , is due to Avdeev [11]:

$$C_D = [5.84 + 16.0 \log(0.93 + 0.065 K^{0.25})]^{-2} \quad (11)$$

where the dimensionless parameter, K , is defined by:

$$K = \frac{\rho_f u_m C_{pf} (2H) (T_{sat} - T_f)}{k_f (T_w - T_{sat})} \quad (12)$$

The surface tension coefficient, C_S , was proposed in the previous paper [10] as follows:

$$C_S = 0.00048 [u_m / (g\sigma/\rho_f)^{1/4}]^{1.28} \quad (13)$$

where the reference velocity was taken as $(g\sigma/\rho_f)^{1/4}$ which has emerged from the condition that $4K_1 K_2 \ll 1$.

Physically, the condition of $4K_1 K_2 \ll 1$ means that the buoyancy force is negligibly small compared to the surface tension and the drag forces. The experimental evidence of Whittle and Forgan [1], that the flow direction in vertical channel, whether up-flow or down-flow, is not important for the initiation of the onset of flow instability, also confirms the negligibility of the buoyancy force. This result agrees with the argument of Levy [4] that the buoyancy force exerted on the vapor bubble is negligible in forced-convection boiling flow.

2.2 Transient Analysis of Temperature Profile

Consider a vertical downwards turbulent flow between heated parallel plates. The channel length is L and the channel depth is $2H$. Initially, the flow has a fully-developed velocity profile corresponding to a specified constant pressure gradient, which is a solution of Equation (14) when the inertia term vanishes. The initial temperature profile is also a solution of the steady-state energy equation corresponding to a specified constant heat-flux boundary condition. When $t > 0$,

a new pressure gradient and a new heat flux, both of which are in general time-dependent, are imposed as boundary conditions. The momentum equation of the flow can be written as follows:

$$\frac{\partial u(y, t)}{\partial t} = -\frac{1}{\rho_f} \frac{dp}{dx}(t) + \frac{\partial}{\partial y}[(\nu + \epsilon_M) \frac{\partial u(y, t)}{\partial y}] \quad (14)$$

with the boundary conditions:

$$u = 0 \text{ at } y = 0; \frac{\partial u}{\partial y} = 0 \text{ at } y = H$$

and the initial condition:

$$u = u_0(y) \text{ when } t = 0$$

where the pressure term includes the hydrostatic pressure as well as the static pressure. The energy equation may be written as:

$$\frac{\partial T(x, y, t)}{\partial t} + u(y, t) \frac{\partial T(x, y, t)}{\partial x} = \frac{\partial}{\partial y}[(\alpha + \epsilon_H) \frac{\partial T(x, y, t)}{\partial y}] \quad (15)$$

with the boundary conditions:

$$-k \frac{\partial T}{\partial y} = q(t) \text{ at } y = 0; \frac{\partial T}{\partial y} = 0 \text{ at } y = H; T = T_i \text{ at } x = 0$$

and the initial condition:

$$T = T_0(x, y) \text{ when } t = 0$$

In Equations (14) and (15) ϵ_M and ϵ_H denote the eddy viscosity and the eddy thermal diffusivity, respectively. Several models for the eddy viscosity have been proposed in the past. A modified Reichardt model suggested by Wilson and Medwell [12] is used in the present study. The original model for the eddy viscosity proposed by Reichardt [13] is valid only for regions away from the wall. However, the wall effect was taken into account by multiplying by the van Driest damping factor [14] in this modified model:

$$\epsilon_M = 0.133 H u^* [0.5 + (\frac{H-y}{H})^2][1.0 - (\frac{H-y}{H})^2][1.0 - \exp(-\frac{y}{A})] \quad (16)$$

where u^* is the friction velocity and A is defined by:

$$A = A^+ (\frac{\nu}{u^*}) \quad (17)$$

In Equation (17), A^+ is the damping constant and is taken to be 40, following Kawamura [15]. For the eddy thermal diffusivity, White [16] have recommended that the turbulent Prandtl number

be taken as a constant.

$$Pr_t = \frac{\epsilon_M}{\epsilon_H} \simeq 0.9 \quad (18)$$

The governing equations are converted into dimensionless form using the following dimensionless variables:

$$U = \frac{u}{u_{m0}} \quad (19)$$

$$X = \frac{x}{L} \quad (20)$$

$$Y = \frac{y}{H} \quad (21)$$

$$\tau = \frac{t}{(H^2/\nu_0)} \quad (22)$$

$$P = \frac{p}{\frac{1}{2}\rho_f u_{m0}^2} \quad (23)$$

$$\theta = \frac{T - T_i}{T_{sat} - T_i} \quad (24)$$

where u_{m0} is the initial mean velocity.

The dimensionless momentum equation becomes:

$$\frac{\partial U}{\partial \tau} = -\frac{u_{m0} H^2}{2\nu_0 L} \left(\frac{\partial P}{\partial X} \right) + \left(\frac{\nu}{\nu_0} + \epsilon_M^+ \right) \frac{\partial^2 U}{\partial Y^2} + \left[\frac{\partial}{\partial Y} \left(\frac{\nu}{\nu_0} + \epsilon_M^+ \right) \right] \frac{\partial U}{\partial Y} \quad (25)$$

with the dimensionless boundary conditions:

$$U = 0 \text{ at } Y = 0; \quad \frac{\partial U}{\partial Y} = 0 \text{ at } Y = 1$$

and the dimensionless initial condition:

$$U = U_0(Y) \text{ when } \tau = 0$$

In Equation (25), the dimensionless eddy viscosity, ϵ_M^+ , is defined by:

$$\epsilon_M^+ = \frac{\epsilon_M}{\nu_0} \quad (26)$$

Similarly, the dimensionless energy equation can be written as:

$$\frac{\partial \theta}{\partial \tau} + Re_0 \left(\frac{H}{4L} \right) U \frac{\partial \theta}{\partial X} = \frac{1}{Pr_0} \left[\left(\frac{\alpha}{\alpha_0} + \epsilon_H^+ \right) \frac{\partial^2 \theta}{\partial Y^2} + \frac{\partial}{\partial Y} \left(\frac{\alpha}{\alpha_0} + \epsilon_H^+ \right) \left(\frac{\partial \theta}{\partial Y} \right) \right] \quad (27)$$

with the dimensionless boundary conditions:

$$-\frac{\partial \theta}{\partial Y} = q^* \text{ at } Y = 0; \quad \frac{\partial \theta}{\partial Y} = 0 \text{ at } Y = 1; \quad \theta = 0 \text{ at } X = 0$$

and the dimensionless initial condition:

$$\theta = \theta_0 \text{ when } \tau = 0$$

where q^* and θ_0 are defined by, respectively:

$$q^* = \frac{q}{(k_f/H)(T_{sat} - T_i)} \quad (28)$$

$$\theta_0 = \frac{T_0 - T_i}{T_{sat} - T_i} \quad (29)$$

In Equation (27), the dimensionless eddy thermal diffusivity, the Reynolds number and the Prandtl number are defined as, respectively:

$$\epsilon_H^+ = \frac{\epsilon_H}{\alpha_0} = \frac{1}{0.9} Pr_0 \epsilon_M^+ \quad (30)$$

$$Re_0 = \frac{4Hu_{mo}}{\nu_0} \quad (31)$$

$$Pr_0 = \frac{\nu_0}{\alpha_0} \quad (32)$$

The solution of the dimensionless momentum and energy equations will be obtained using a finite difference method. Both equations are put into a finite difference form using the Crank-Nicolson method. Central differences in the Y direction and backward differences in the X direction are employed for the convective term of Equation (27).

A stretching node technique is employed to reduce the number of nodes in Y direction. This can be written as follows:

$$Y_{i+1} - Y_i = k_n(Y_i - Y_{i-1}) \quad (33)$$

where the stretching coefficient, k_n , is generally taken to be between 1.07 and 1.10. This stretching technique greatly reduces the number of nodes without losing accuracy of the solution, in particular, in the viscous sublayer. The first node for the momentum equation is taken at the fluid in the vicinity of the solid surface, whereas it is taken on the solid wall for the energy equation.

The change of the fluid properties as a function of time is taken into consideration by evaluating them at the mean of the bulk liquid inlet and outlet temperature at each previous time

step. It is thought that the error due to evaluating the fluid properties in this manner is not significant.

2.3 Prediction of OSV in transient flow

For vertical downwards turbulent flow between heated parallel plates, the liquid has initially a fully-developed velocity profile, $u_o(y)$, corresponding to an initial pressure gradient and a temperature profile, $T_o(x, y)$, resulting from an initial heat flux. When $t > 0$, a new pressure gradient is imposed inside the channel, resulting in a decrease of the liquid mean velocity. It can be a step change, or a time-dependent change, as a ramp change. The heat flux is assumed to be kept constant. Due to the decrease of the mean velocity, the liquid mean temperature will rise. It is of concern to predict whether OSV will take place before a new steady-state will be attained under specific geometrical and operating conditions. Because OSV takes place first at the bottom of the channel, where the fluid mean temperature is always maximum, attention is focused on the local condition at the bottom for OSV.

The transient dimensionless velocity and temperature profiles are calculated from Equations (25) and (27). The bubble diameter when the bubble starts to depart from the heated wall is calculated from Equation (10). The use of Equation (10) in the transient analysis may be possible if it is assumed that the maximum bubble diameter changes slowly during the transient flow compared to the bubble lifetime. In calculating the bubble diameter, the fluid properties evaluated at the local liquid mean temperature should be used. The distance from the wall at which the saturation temperature is reached is determined from the transient liquid temperature profile at the bottom of the channel. At each time step, the magnitude of the bubble diameter is compared to that of the distance to the point of the saturation temperature. When the temperature at the top of the bubble reaches the saturation temperature, OSV is thought to occur. If the distance to the saturation temperature is always smaller than the bubble diameter at each time step until a new steady-state is attained, one may consider that OSV will never take place under that particular condition.

3. COMPARISON

Comparisons of the present theory with the available data and other model were discussed in detail in the previous paper [10]. Only steady-state data are available at the present time and thus was compared with the steady-state result of the present model. The steady-state result was obtained by solving the liquid temperature profiles from the steady-state momentum and energy equations together with the hydrodynamic condition of bubble departure, that is, Equation (10). The results of comparison of the present theory with the data of Whittle and Forgan [1] and the data of Dougherty, et al.[2] showed, in the previous paper [10], that the present theory is in excellent agreement with both data sets. Most data were within error bounds of $\pm 10\%$.

In addition, a comparison of the present theory was made with the empirical correlation suggested by Saha and Zuber [3]. The Saha-Zuber correlation is very simple and convenient, and has been widely used in computer codes for nuclear safety analysis. However, recent experiments [2,9] indicate that the discrepancy between the data and the correlation appears to be considerable, with data scatter beyond the error margins described by Saha and Zuber [3] originally. The comparison was done by plotting the numerical data calculated from the present theory in the (Pe, St) plane. It was indicated that the present numerical data are fairly well fitted with the correlation [10]. However, the present theory hinted that, contrary to the correlation, the Stanton number is not solely related to the Peclet number, showing multiple values of the Stanton number depending upon the geometrical size at a single value of the Peclet number. The result may explain the large error margins in the Saha-Zuber correlation, and also the scatter beyond the specified error margins. More details in comparisons can be found in [10].

4. COMPUTATIONAL RESULTS

Numerical calculations for predicting the transient OSV in vertical downwards flow between parallel plates were made for the following dimensions:

channel height: 0.00685(m)

channel length: 3.8(m)

inlet temperature: 25($^{\circ}$ C)

saturation temperature: 125($^{\circ}$ C)

pressure gradient: step or ramp change from 55.0 to 24.0(kPa/m)

The dimensions were obtained from actual operating conditions.

4.1 Typical velocity and temperature calculations

The mean velocity and the exit bulk temperature when the heat flux is kept at a constant value ($q = 2.0\text{ MW/m}^2$) are shown in Figures 1 and 2, respectively. The 1-D calculation results are also shown in the figures. The 1-D model is described in the Appendix. It is seen that a new steady-state is attained after some time. The differences between 1-D and 2-D models seem to be less than 5 % for each parameter. It turns out that the energy equation itself for the 1-D and 2-D models gives almost no difference. Thus, the difference between 1-D and 2-D models in Figure 2 is attributable to the mean velocity difference shown in Figure 1.

Figures 3 and 4 show typical velocity and temperature profiles at the outlet during transient change of flow.

4.2 Prediction of transient OSV at constant heat flux

Predictions of OSV in transient flow have been made for various operating conditions. Figures 5 and 6 show the mean velocity and the exit bulk temperature as a function of time for several heat fluxes after a step change of pressure gradient. The geometric and operating conditions are before. At $t = 0$, a step change of pressure gradient from 55 to 24(kPa/m) was made, while the heat flux was maintained at a constant value. The mean velocity decreases, and the bulk temperature increases. Figure 5 shows slight changes of the velocity variation depending on the magnitude of the heat flux added from both plates. It is shown in Figure 6 that OFI

will always take place for heat fluxes greater than $2.2(MW/m^2)$. According to the steady-state calculation, the OSV heat flux for initial pressure gradient, $55(kPa/m)$ will be $3.34(MW/m^2)$. Thus, this value can be regarded as the maximum possible heat flux to avoid OSV under steady-state operating conditions. Figure 7 indicates the transient effect of the OSV numerical data for various heat fluxes. This could be obtained by comparing the transient OSV heat flux with the steady-state OSV heat flux with the same mean velocity as that for the transient case. In the figure, the difference between the transient and steady-state heat fluxes represents the transient term in Equation (15). Thus, at initial and final states, the transient effect will be insignificant as shown in Figure 7. The maximum ratio of the heat flux contributed to the transient term to the total heat flux would be approximately 15 %. This ratio should be compensated when steady-state correlations of OSV are used in predicting OSV in transient flow and heat transfer.

Typical transient OSV predictions for ramp changes in pressure gradient are shown in Figures 8 - 11. The results were obtained for a constant heat flux of 3.0 and $2.4(MW/m^2)$ with six different ramp times. In the figure, $t_r = 0.0$ corresponds to a step change.

It is interesting to note that the outlet subcooling appears to be almost identical regardless of the mode of pressure gradient change and the heat flux. This observation is not in conflict with the finding that the local Stanton number slightly decreases as the Peclet number increases, as mentioned in the previous paper [10]. As seen in the result of the previous paper [10], the outlet subcooling is relatively insensitive for the high velocity range when OSV takes place. For the particular transient flow dealt with here, the mean velocity at which OSV occurs lies between only $7(m/s)$ and $9(m/s)$. As will be seen later, the Stanton number is a weak function of the Peclet number in this range of mean velocity.

4.3 Simple OSV equation applicable to 1-D computer code

The present theory is based upon a two-dimensional flow and heat transfer analysis. There may be difficulty in its application to computer codes for nuclear safety analysis, which are, in general, based upon one-dimensional flow and heat transfer analysis. Here a single simple equation is suggested to give a criterion for OSV in transient flow under the present geometric

and operating conditions. The Stanton number calculated for transient OSV in the present study can be approximated by the following equation:

$$St = 0.076 Pe^{-0.2} \quad (34)$$

This equation predicts lower Stanton numbers compared to the Saha-Zuber correlation for Peclet numbers between 500,000 and 900,000, as shown in Figure 12. Thus, the present theory is more conservative than the Saha-Zuber correlation.

5. DISCUSSION AND CONCLUSIONS

The present theory was derived under the assumption that the buoyancy force is negligibly small. This condition leads to a criterion for a lower limit of the mean velocity for the present theory. This can be calculated from the condition that $4K_1K_2 \ll 1$. Using Equations (7) and (8), one obtains:

$$u_m \gg \left[\frac{128}{3} \frac{C_B}{C_D} \frac{C_S}{C_D} \frac{(\rho_l - \rho_g) g \sigma}{\rho_l} \right]^{1/4} \quad (35)$$

Thus, it will be approximately 0.71 (m/s) at an operating pressure of 0.1 (MPa) assuming that C_B/C_D is $O(1)$ and C_S/C_D is $O(1)$, when $4K_1K_2 \approx 0.1$. For the velocity range smaller than this lower limit, the evaluation of C_B should be made before the present theory is applied. No physical evaluation can be found on the value of C_B in the literature. However, it is expected to be $O(1)$.

It is interesting to note in Figure 12 that the present theory predicts that $St \propto Pe^{-0.20}$. This means that, from the definition of the Stanton number, $Nu \propto Re^{0.80}$, considering the heat transfer from the bubble boundary to the liquid core. This result is exactly identical to the Dittus-Boelter equation for the heat transfer coefficient in single-phase flow.

The fact that the transient calculations all show the onset of significant void, and hence of flow instability, at the same exit bulk temperature for various power levels is rather remarkable. One physical interpretation is that the departure of bubbles from the wall is condensation-controlled. With highly-subcooled bulk liquid bubbles grow and collapse at the wall. As the bulk liquid warms up, the collapse time becomes considerably longer than the growth time, because the thermal boundary layer around the bubble thickens during the collapse phase. At some point the collapse becomes very slow, and the bubble remains close to its maximum radius for a relatively long time, which allows the lift forces to detach it from the wall.

As shown in Figure 12 the fact that the Stanton number, as defined by Saha and Zuber [3], remains nearly constant is not in contradiction with the observation concerning the steady-state result mentioned in the previous paper [10]. With the specific step pressure change, the bulk velocity changes from about 9 (m/s) to only about 7 (m/s) for OSV, while the wall heat flux

remains constant. If the bulk liquid subcooling is also a constant at OSV, the Stanton number, $St = h/GC_p(T_{sat} - T_d)$, also is approximately constant at this high end of the Peclet number range. However, at lower Peclet numbers, the variation in the Stanton number is very much marked.

This is still a highly empirical model, although it contains much more physics than previous models. More experimental information over a wide range, particularly with flow and bubble visualization, is needed to check the details of this model, and to make necessary modifications. Nevertheless, it is encouraging that such a wide range of data can, at the present time, be fitted closely with the choice of a single empirical parameter.

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NOMENCLATURE

A	parameter defined by Equation (17)
A_c	cross-sectional area (m^2)
A_s	surface area (m^2)
A^+	damping constant
C	coefficient
C_p	specific heat (J/kgK)
D	diameter (m)
D_h	hydraulic diameter (m)
F	force (N)
f	friction factor
G	mass velocity (kg/m^2s)
g	acceleration due to gravity (m/s^2)
H	distance between the wall and the centerline (m)
K	dimensionless parameter
k	thermal conductivity (W/mK)
L	channel length (m)
n	exponent
Nu	Nusselt number
P	dimensionless pressure
p	pressure (Pa)
Pe	Peclet number
Pr	Prandtl number
Pr_t	turbulent Prandtl number
Q	heat flux ratio
q	heat flux (W/m^2)

R	radius (m)
Re	Reynolds number
St	Stanton number
T	temperature (K)
t	time (s)
T_d	bulk temperature at OSV (K)
T_i	inlet temperature (K)
T_o	outlet temperature (K)
U	dimensionless velocity
U_m	dimensionless bulk velocity
u	velocity in x direction (m/s)
u_m	bulk or mean velocity in x direction (m/s)
u^*	friction velocity (m/s)
X	dimensionless coordinate
x	coordinate
Y	dimensionless coordinate
y	coordinate

Greek Symbols

α	thermal diffusivity (m^2/s)
ϵ_H	eddy thermal diffusivity (m^2/s)
ϵ_M	eddy viscosity (m^2/s)
θ	dimensionless temperature
ν	kinematic viscosity (m^2/s)
ρ	density (kg/m^3)
σ	surface tension (N/m)

τ shear stress or dimensionless time

Subscript

B buoyancy

b bubble

D drag

f liquid

g gas

i i -th node in Y -direction

o initial state

S surface tension

sat saturation

w wall

Superscript

$\circ$ dimensionless

$+$ dimensionless

REFERENCES

- [1] R.H. Whittle and R. Forgan, Correlation for the minima in the pressure drop versus flow-rate curves for subcooled water flowing in narrow heated channels, *Nuclear Engineering and Design* 6 (1967) 89-99.
- [2] T. Dougherty, C. Fighetti, G. Reddt, B. Yang, T. Jafri, E. McAssey and Z. Qureshi, Flow boiling in vertical down-flow, *Proceedings of 10th Int. Heat Transfer Conf.* 6 (1990) 9-14.
- [3] P. Saha and N. Zuber, Point of net vapor generation and vapor void fraction in subcooled boiling, *Proceedings of Fifth International Heat Transfer Conference* 4 (1974) 175-179.
- [4] S. Levy, Forced convection subcooled boiling: prediction of vapor volumetric fraction, *Int. J. Heat Mass Transfer* 10 (1967) 951-965.
- [5] F.W. Staub, The void fraction in subcooled boiling: prediction of the initial point of net vapor generation. *J. Heat Transfer* 90 (1968) 151-157.
- [6] J.T. Rogers, M. Salcudean, Z. Abdullah, D. Mcleod and D. Poirier, The onset of significant void in up-flow boiling of water at low pressure and velocities, *Int. J. Heat Mass Transfer* 30 (1987) 2247-2260.
- [7] R.H.S. Winterton, Flow boiling: prediction of bubble departure, *Int. J. Heat Mass Transfer* 27 (1984) 1422-1424.
- [8] R.C. Martinelli, Heat transfer to molten metals, *Trans. ASME* 6 (1947) 941-947.
- [9] B.S. Johnston, Subcooled boiling of downward flow in a vertical annulus, *Savannah River Laboratory, DPST-88-891* (1988).
- [10] S.C. Lee and S.G. Bankoff, Theoretical prediction of the onset of flow instability in vertical flow between parallel plates, to be presented in *National Heat Transfer Conf.*, San Diego, CA (August 1992).
- [11] A.A. Avdeev, Application of the Reynolds analogy to the investigation of surface boiling in conditions of forced motion, *High Temperature* 24 (1986) 100-108.

[12] N.W. Wilson and J.O. Medwell, An analysis of the developing turbulent hydrodynamic and thermal boundary layers in an internally heated annulus, J. Heat Transfer 93 (1971) 25-32.

[13] H. Reichardt, Vollständige darstellung der turbulenten geschwindigkeitsverteilung in glatten leitungen, Z. Angew. Math. Mech. 31 (1951) 208.

[14] E.R. van Driest, On turbulent flow near a wall, Heat Transfer and Fluid Mechanics Institute Symposium, Paper No. 12, University of California, Los Angeles, CA (1955).

[15] H. Kawamura, Experimental and analytical study of transient heat transfer for turbulent flow in a circular tube, Int. J. Heat Mass Transfer 20 (1977) 443-450.

[16] F.M. White, Heat and Mass Transfer (Addison-Wesley, Massachusetts, 1988) pp. 315-332.

APPENDIX: ONE-DIMENSIONAL FLOW AND HEAT TRANSFER ANALYSIS

The momentum equation for unsteady one-dimensional flow can be written as follows:

$$\rho \frac{du_m}{dt} = \frac{dp}{dz}(t) - \frac{\tau_w}{H} \quad (36)$$

and the initial condition becomes:

$$u_m = u_{m0} \text{ when } t = 0$$

In the equation, the wall shear stress, τ_w , can be defined by:

$$\tau_w = \frac{1}{8} f \rho u_m^2 \quad (37)$$

and the friction factor will be given by:

$$f = C Re^n \quad (38)$$

where $C = 0.3164$ and $n = -0.25$ are given for the Blasius equation and $C = 0.1840$ and $n = -0.20$ for the White equation [16].

The energy equation can be written as follows:

$$\rho C_p \left[\frac{\partial T}{\partial t} + u_m(t) \frac{\partial T}{\partial z} \right] = \frac{q(t)}{H} \quad (39)$$

The boundary condition is

$$T = T_i \text{ at } z = 0$$

and the initial condition becomes:

$$T = T_0(z) \text{ when } t = 0$$

The governing equations will be converted into dimensionless form using the following dimensionless variables:

$$U_m = \frac{u_m}{u_{m0}} \quad (40)$$

$$X = \frac{z}{L} \quad (41)$$

$$\tau = \frac{t}{(H^2/\nu_0)} \quad (42)$$

$$P = \frac{p}{\frac{1}{2} \rho_f u_{m0}^2} \quad (43)$$

$$\theta = \frac{T - T_i}{T_{sat} - T_i} \quad (44)$$

Then, the dimensionless momentum equation becomes:

$$\frac{dU_m}{d\tau} + \frac{C}{32} \left(\frac{\nu}{\nu_o} \right)^{-n} Re_o^{n+1} U_m^{2+n} = - \left(\frac{H}{8L} \right) Re_o \left(\frac{dP}{dx} \right) \quad (45)$$

where the Reynolds number at the initial state, Re_o , is defined by:

$$Re_o = \frac{4H u_{mo}}{\nu_o} \quad (46)$$

and the dimensionless initial condition:

$$U_m = 1 \text{ when } \tau = 0$$

The dimensionless energy equation can be expressed by:

$$\frac{\partial \theta}{\partial \tau} + \left(\frac{H}{4L} \right) Re_o U_m(\tau) \frac{\partial \theta}{\partial X} = \frac{1}{4} Re_o St_o Q(\tau) \quad (47)$$

where the Stanton number at the initial state, St_o , is defined by:

$$St_o = \frac{q_o}{G_o C_p (T_{sat} - T_i)} \quad (48)$$

and the heat flux ratio, Q , is:

$$Q(\tau) = \frac{q(t)}{q_o} \quad (49)$$

Both dimensionless equations can be solved numerically.

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