1 of 1

FLOW REGIMES AND HEAT TRANSFER
IN VERTICAL NARROW ANNULI

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ABSTRACT

In shell side boiling heat exchangers narrow crevices that are formed between the tubes and the tube support structure provide areas for local thermal-hydraulic conditions which differ significantly from bulk fluid conditions. Understanding of the processes of boiling and dryout in flow restricted crevices can help in designing of tube support geometries to minimize the likelihood of tube support plate and tube corrosion observed in commercial power plant steam generators.

This paper describes a one dimensional thermal-hydraulic model of a vertical crevice between a tube and a support plate with cylindrical holes. The annulus formed by the support plate hole and an eccentrically located tube has been represented by vertical strips. The formation, growth and collapse of a steam bubble in each strip has been determined. Based on the bubble history, the flow regimes characterized by "isolated" bubbles, "coalesced" bubbles and liquid deficient regions have been defined.

INTRODUCTION

In shellside boiling heat exchangers narrow crevices formed between the tubes and the tube support plates provide areas for local thermal-hydraulic conditions which differ significantly from bulk fluid conditions. Flow starvation or "dryout" in these restricted flow passages play a predominant role in concentration of chemical species. Such concentration can lead to tube support plate or tubing corrosion problems, denting or cold leg thinning experienced in power plant boilers. Therefore, an understanding of the flow and the heat transfer processes in narrow crevices are essential for the description of the corrosion mechanism.

The literature on two-phase flow in narrow passages is quite limited, and only a few works deal with the effects of heating or boiling. One of the earlier works on the effects of space restriction on pool boiling was published by Ishibashi and Nishikawa (1969). In an experimental study of pool boiling in vertical concentric annuli around a 12 inch (304 mm) long electrically heated tube they observed three distinctively different boiling regimes. At radial gap sizes above 0.120 inch (3.0 mm), the boiling character was indistinguishable from unconfined boiling. At smaller gap sizes, large "coalesced" bubbles filled the crevice. A liquid deficient region occurred at gap sizes less than 0.022 inch (0.57 mm) at a heat flux of about 24300 Btu/hr-ft² (76800 W/m²). The heat transfer coefficient in the "coalesced" bubble regime in a 0.038 inch (0.97 mm) gap was a factor of 32 greater than that of pool boiling.

EPRI sponsored two studies of local heat transfer in prototypical tube support plate crevices of PWR steam generators. Bankoff, et al., (1982) measured improvements of up to 80% in crevices over the bare tube heat flux. Arpaci, et al., (1982) confirmed the existence of three flow regimes in narrow crevices. They also measured the form losses in concentric crevices using single phase liquid. Their measurements showed significantly higher pressure

losses than those predicted by theoretical correlations at crevice Reynolds numbers less than about 2000.

Bell and Bergelin (1957) studied pressure drop through annular concentric and eccentric crevices using single phase liquid flow. They found the laminar or viscous flow to exist up to a Reynolds number of 40. The flow was fully turbulent at a Reynolds number of 4000. Lowry and Kawaji (1988) studied upward flow of air and water through a narrow passage between parallel plates. They found flow patterns similar to those in larger pipes, but the transition lines between slug, churn, bubbly and annular flow regimes were modified. The pressure drop data correlated well with the friction coefficient multiplier of Lockhart-Martinelli.

It is evident that the character of flow and heat transfer in narrow annuli is significantly different from those in large passages or pipes. It is the object of this work to develop a simple analytical model for boiling two-phase flow in eccentric narrow crevices. A one dimensional model, while neglecting the minor influences of cross flows, allows description of the flow regimes and bubble dynamics efficiently.

THE ANALYTICAL MODEL

The presence of large, elongated "coalesced" bubbles in the flow of a two-phase fluid in a narrow annular gap has been observed in a number of studies. It has also been shown that in vertical geometries these bubbles follow nearly vertical paths [3] and that the effects of two-dimensionality are not significant. Thus the flow field in an eccentric annular crevice can be approximated by a circumferential summation of flows in discrete vertical elements. The flow in each element can be approximated by that between two parallel plates.

Figure 1 shows schematically the assumed flow structure at a circumferential location. Heating is provided by high temperature liquid flow in the tube. The four distinct regions shown here may not always be present. As the ratio of the heat flux to the flow rate decreases, first the "dryout" disappears, followed by the annular and the churn flow regimes and so on.

THE ISOLATED BUBBLE REGIME

This flow regime extends from the crevice inlet to the bottom of the first "coalesced" bubble as shown in Figure 2. The length, ℓ_1 , of this region can be calculated by applying the conservation equations for mass and energy to the control volume. The inlet velocity V_0 will be calculated iteratively to correspond to any given crevice pressure drop. For the time being it can be treated as one of the known quantities. The inlet void fraction is a known quantity.

FIGURE I
TWO-PHASE FLOW IN A VERTICAL CREVICE

FIGURE 2
ISOLATED BUBBLE REGION

The continuity equations for the vapor and the liquid can be written as

$$-\rho_g A_c V_o \alpha_o + \rho_g A_c V_{gc} \alpha_c - \dot{m}_{ifg} = 0 \quad (1)$$

$$-\rho_l A_c V_o (1-\alpha_o) + \rho_l A_c V_{lc} (1-\alpha_c) + \dot{m}_{ifg} = 0 \quad (2)$$

where $\dot{m}_{ifg}$ is the vapor generation rate in the isolated bubble regime and the subscript "c" refers to the boundary between this and the "coalesced" bubble regimes.

Neglecting heat transfer to the support plate, the vapor generation rate can be obtained from

$$\dot{m}_{ifg} = q_1'' y / h_{fg} \quad (3)$$

where y is the width of the vertical strip and h_{fg} is the latent heat of vaporization.

In the isolated bubble regime the boiling process is not much different than that on the open tube surfaces. Therefore, the correlation of Thom (1965) can be used for the secondary side heat transfer coefficient.

$$h_{s1} = 192.9 e^{P_s/630} (T_w - T_s), \quad P_s \text{ in psia, } T \text{ in } ^\circ\text{F} \quad (4)$$

The inside tube liquid temperature and the flow rate are known quantities.

The heat transfer coefficient can be obtained from the forced convection correlation of Dittus-Boelter

$$h_{p1} = \frac{k_{p1}}{D_i} 0.023 Re^{0.8} Pr^{0.4} \quad (5)$$

The overall heat transfer coefficient U_1 is then given by

$$U_1 = 1 / \left(\frac{D_o}{D_i h_p} + \frac{D_o}{2k_t} \ln \frac{D_o}{D_i} + \frac{1}{h_{s1}} \right) \quad (6)$$

Then, the region 1 heat flux is

$$q_1'' = U_1 (T_p - T_s) \quad (7)$$

The tube outside surface temperature, T_w , can be calculated from the combined resistances of the primary side and the tube wall.

$$U_{wi} = 1 / \left(\frac{D_o}{D_i h_{pl}} + \frac{D_o}{2k_t} \ln \frac{D_o}{D_i} \right) \quad (8)$$

$$T_{wi} = T_p - q''_i / U_{wi} \quad (9)$$

Equations (4) through (9) can be solved iteratively for q''_i , and T_{wi} .

We still need a criterion for transition from the isolated bubble to the "coalesced" bubble regime and a relationship between the vapor and the liquid velocities when this transition occurs. Radovicich and Moissis (1962) and Taitel, et al., (1980) recommend 0.3 for the critical void fraction and this value has been adopted here. At transition the velocity of the small bubbles can be assumed equal to that of the "coalesced" bubble. The velocity of a large bubble in slug flow was given by Nicklin, et al., (1962).

$$V_B = 1.2 V_L + 0.35 \sqrt{g D_h} \quad (10)$$

The 1.2 factor accounts for the velocity at the center of the pipe compared to the average. The second term is the velocity of a large bubble in stagnant liquid, and was derived by Davies and Taylor in 1949. It accounts for the effects of buoyancy. Maneri and Mendelson (1968) studied bubble rise in stagnant fluid contained in a two dimensional tank having a 0.25 inch (6.35 cm) spacing. The terminal bubble velocity was related to bubble size by the following relationship.

$$V_{B\infty} = [g (2r_e + \delta) / \pi]^{1/2} \quad (11)$$

where r_e is the equivalent radius defined in terms of the bubble volume by the relation

$$r_e = [V/\pi\delta]^{1/2} \quad (12)$$

In the present study the Maneri-Mendelson equation for relative velocity was used.

$$V_{gc} = 1.2 V_{LC} + V_{B\infty} \quad (13)$$

The liquid velocity at transition can now be calculated from Equations (1), (2) and (13).

$$V_{lc} = \frac{V_o [\rho_g \alpha_o + \rho_l (1 - \alpha_o)] - V_{B\infty} \rho_g \alpha_c}{1.2 \rho_g \alpha_c + \rho_l (1 - \alpha_c)} \quad (14)$$

Since $A_c = y\delta$ the length of the isolated bubble regime can be determined from Equation (3) and one of the continuity equations as

$$x_1 = \frac{\rho_g h_{fg} \delta}{q_1''} (V_{gc} \alpha_c - V_o \alpha_o) \quad (15)$$

This completes the definition of the isolated bubble regime and provides the starting conditons for the "coalesced" bubble regime discussed in the next section.

COALESCED BUBBLE REGIME

Figure 3 shows the bubble geometry where the bubble shape has been approximated by a right rectangular prism.

The bubble volume is given by

$$V_B = x_1 y_1 (z_2 - z_1) \quad (16)$$

The volume rate of change of the bubble can be obtained by differentiation of Equation (16) as

$$\frac{dV_B}{dt} = x_1 \left[\frac{dy_1}{dt} (z_2 - z_1) + y_1 \left(\frac{dz_2}{dt} - \frac{dz_1}{dt} \right) \right] + \frac{dx_1}{dt} y_1 (z_2 - z_1) \quad (17)$$

Except for the initial growth period dx_1/dt is relatively small and can be neglected. Relationships between the velocities dy_1/dt , dz_1/dt and dz_2/dt are not known. It is assumed that the expansion velocity in the z direction is K times that in the y direction. Furthermore, the lower surface of the bubble

FIGURE 3
BUBBLE GEOMETRY

expands in the negative z direction. Then Equation (13) yields the velocities at the upper and lower surfaces of the bubble after addition of the components due to expansion.

$$V_{BH} = \frac{dz_2}{dt} = 1.2 V_{L2} + V_{B\infty} + K \frac{dy_1}{dt} \quad (18)$$

$$V_{BL} = \frac{dz_1}{dt} = 1.2 V_{L1} + V_{B\infty} - K \frac{dy_1}{dt} \quad (19)$$

The value of K determines the shape of the bubble but has negligible effects on the overall results. The present results were obtained with a K value of 1.0. Larger values lead to elongated bubbles.

Equations (17), (18), and (19) can be solved simultaneously to give the bubble velocity in the circumferential direction.

$$\frac{dy_1}{dt} = \left[\frac{dV_B}{dt} - 1.2 x_1 y_1 (V_{L2} - V_{L1}) \right] / [x_1 (z_2 - z_1) + 2K x_1 y_1] \quad (20)$$

In general the liquid film on the tube is sufficiently thin so that heat transfer into the bubble can be represented by conduction through the film. A film heat transfer coefficient can be defined by

$$h_B = k_L / \epsilon \quad (21)$$

The heat transfer rate into the bubble can be calculated from an overall heat transfer coefficient U_B similar to that given by Equation (6).

$$\dot{q}_B = U_B (T_p - T_B) \quad (22)$$

The rate of evaporation underneath the bubble can be calculated by neglecting the superheating of the liquid in the film.

$$\dot{m}_B = \dot{q}_B / h_{fg} \quad (23)$$

where δ is the film thickness yet to be determined. The bubble volumetric expansion is simply given by

$$\frac{dV_B}{dt} = \dot{m}_B / \rho_B \quad (24)$$

The vapor density, ρ_B , within the bubble can be obtained by using the Maneri-Mendelson effective bubble radius given by Equation (12)

$$r_B = [x_1 y_1 (z_2 - z_1) / \pi \delta]^{1/2} \quad (25)$$

The excess pressure in the bubble must balance the surface tension forces and this is expressed by

$$P_B = P_s + 2\sigma / r_B \quad (26)$$

The linearized Clapeyron equation yields the bubble temperature.

$$T_B = T_s + T_s (P_B - P_s) \rho_B / h_{fg} \quad (27)$$

where the units of p_B and T are psf and degrees Rankine. For relatively large bubbles the properties inside the bubble are not significantly different than those at saturation.

The mass flow rate at any cross section is equal to the mass entering the element, $\dot{m}_0$, for the steady state conditions assumed here.

$$\dot{m}_0 = V_0 y \delta [\rho_L (1 - \alpha_0) + \rho_g \alpha_0] \quad (28)$$

The continuity applied to sections $z = z_1$ and $z = z_2$ yields

$$V_{L1} = (\dot{m}_0 - V_{BL} y \delta \rho_g \alpha_1) / [y \delta \rho_L (1 - \alpha_1)] \quad (29)$$

$$V_{L2} = (\dot{m}_0 - V_{BH} y \delta \rho_g \alpha_2) / [y \delta \rho_L (1 - \alpha_2)] \quad (30)$$

The void fractions α_1 and α_2 can be determined from the conservation of liquid mass between the inlet and the cross sections at $z = z_1$ and $z = z_2$.

$$\alpha_1 = 1 - [\rho_l y \rho V_0 (1 - \alpha_0) - \dot{m}_{Lfg}] / (\rho_l y \delta V_{l1}) \quad (31)$$

$$\alpha_2 = 1 - [\rho_l y \delta V_0 (1 - \alpha_0) - \dot{m}_{Hfg}] / (\rho_l y \delta V_{l2}) \quad (32)$$

In Equations (31) and (32) $\dot{m}_{Lfg}$ is the rate of evaporation from the inlet to $z = z_1$ and $\dot{m}_{Hfg}$ is that from the inlet to $z = z_2$.

The vapor generation rate in this region is the sum of those underneath each "coalesced" bubble and within the liquid slugs separating them. In theory, the evaporation rate can be calculated using the elapsed time between the birth and the collapse of "coalesced" bubbles. Since this information is not available until a complete solution has been obtained, the above method would require an iterative procedure. A practical method is offered by the empirical correlation of Ishibashi and Nishikawa (1969).

$$h_{CB} = \frac{k_l}{\delta} 200 \left(\frac{\bar{D}}{F\delta} \right)^{-2/3} Pr^{-2/3} \left(\frac{\rho_l}{\rho_g} \right)^{-1/2} \quad (33)$$

$$\text{where } F = 1.365 \times 10^{-9} q_{CB}'' \delta^{-3/2} Pr^{1.627} \left(\frac{\rho_l}{\rho_g} \right)^{1.085} \quad (34)$$

In Equation (33) " $\bar{D}$ " is the thermal diffusion constant in water. In Equation (34) q_{CB}'' must have the original units, kcal/hr-m². The above equations can be used in an iterative procedure to solved for q_{CB}'' .

The heat transfer rate from the inlet to $z = z_1$ and the vaporization rates can now be calculated for each strip.

$$\dot{q}_L = q_1'' x_1 y + q_{CB}'' (z_1 - x_1) y \quad (35)$$

$$\text{and, } \dot{m}_{Lfg} = \dot{q}_L / h_{fg} \quad (36)$$

$$\dot{m}_{Hfg} = \dot{m}_{Lfg} + \dot{m}_B \quad (37)$$

The above equations can be solved iteratively for V_{BH} , V_{BL} , V_{l1} , V_{l2} , α_1 and α_2 . The film thickness under the bubble was calculated by an iterative solution of

$$A_f = [\rho_l y_1 \delta (1-\alpha_2)(V_{BH} - V_{l2}) - 2 \dot{m}_B] / [\rho_l (V_{BL} + V_f)] \quad (38)$$

and

$$V_f = \frac{1}{12} \frac{A_f^2}{y_1^2} \frac{\rho_l}{\mu_l} g \quad (39)$$

Equation (38) gives the film flow area and results from a mass balance for the liquid relative to the moving bubble. The average liquid velocity in the film, V_f , was obtained from a force balance.

The description of the "coalesced" bubble regime is complete except for a criterion for transition to churn flow. For turbulent flow of gas and liquid in round tubes Taitel, et al., (1980) obtained an equation for this transition.

$$(V_g^S/V^S)_{cr.} = l_B/l + \alpha (1-l_B/l) \frac{1}{1-(1-\alpha)/6} \quad (40)$$

where l is the sum of the bubble and the slug lengths and the superscript "s" designates superficial velocity. V^S is the total superficial velocity, $V^S = V_g^S + V_l^S$. Clearly when l_B/l becomes 1.0 the slug length is zero and the continuity of the liquid is destroyed. In practice the transition occurs sooner. Taitel, et al., suggest $l_B/l = 0.8$ as an approximation. Also, the critical bubble packing criterion of $\alpha = 0.3$ can be used in Equation (40).

The ratio of the superficial gas velocity to the total superficial velocity at the top of the bubble ($z = z_2$) is given by

$$V_g^S/V^S = \alpha_2 V_{BH} / [\alpha_2 V_{BH} + (1-\alpha_2) V_{l2}] \quad (41)$$

When the ratio of the superficial velocities given by Equation (41) equals or exceeds the critical value given by Equation (40) the bubble is assumed to burst and the churn flow regime begins, defining the length of the "coalesced" bubble region l_2 (shown Figure 1).

Equations (18), (19), (20), (22), (23), (24), (28), (29), and (30) can be cast in the form of first order differential equations and integrated with respect to z , similar to an initial value problem. The integration proceeds from an assumed bubble size at $z = z_1$, until the bubble to churn transition criterion given by Equation (40) is satisfied.

CHURN-ANNULAR FLOW REGIME

This regime was assumed to exist between the end of the "coalesced" bubble regime and the point of complete evaporation. The heat transfer was modeled using the Dengler-Addoms correlation for annular flow [16].

$$h_3 = 3.5 h_{l0} (1/X_{tt})^{0.5} \quad (42)$$

where X_{tt} is the Lockhart-Martinelli parameter and h_{l0} is the forced convection heat transfer coefficient for liquid only flow given as follows:

$$\frac{1}{X_{tt}} = \left(\frac{\dot{m}_g}{\dot{m}_l} \right)^{0.9} \left(\frac{\rho_l}{\rho_g} \right)^{0.5} \left(\frac{\mu_g}{\mu_l} \right)^{0.1} \quad (43)$$

$$h_{l0} = 0.023 \frac{k_l}{D_h} \left(\frac{D_h \rho_l V_{l0}}{\mu_l} \right)^{0.8} Pr^{0.4} \quad (44)$$

The values for the flow rates and the liquid velocity were calculated by averaging those at the region boundaries.

The heat flux is given by

$$q_3'' = U_3 (T_p - T_s) \quad (45)$$

where the overall heat transfer coefficient is calculated from Equation (6) with h_s replaced by h_3 .

Defining $\dot{m}_{l3}$ as the liquid flow rate at the bubble to churn transition point, the area in this region is found by

$$A_{s3} = y (z_3 - z_2) = \dot{m}_{l3} h_{fg} / q_3'' \quad (46)$$

where z_1 is the position of the top of the "coalesced" bubble when transition to churn flow occurs. z_2 is the point of "dryout" if it falls within the crevice height.

PRESSURE LOSSES

The pressure losses were calculated for the four regions separately. The total pressure loss for a given vertical element was the sum of these plus the losses at the crevice inlet and the exit. The inlet velocity, V_0 , was calculated for each element using a Newton-Raphson iteration method [15], to produce the same axial pressure drop around the circumference.

The mixture momentum equation for two-phase flow can be written as follows [17].

$$\begin{aligned} & - (1-\alpha_1) \rho_L V_{L1}^2 + (1-\alpha_2) \rho_L V_{L2}^2 - \alpha_1 \rho_g V_{g1}^2 + \alpha_2 \rho_g V_{g2}^2 \\ & = - \frac{dP}{dz} \Delta z g_c - \left[\left(1 - \frac{\alpha_1 + \alpha_2}{2}\right) \rho_L + \frac{\alpha_1 + \alpha_2}{2} \rho_g \right] g \Delta z \end{aligned}$$

$$-F_w/A_c = \text{Losses} \quad (47)$$

The friction term can be calculated using the Lockhart-Martinelli two-phase friction multiplier which has been shown to be an adequate predictor of frictional losses in a narrow rectangular channel by Lowry and Kawaji (1988).

Using a linear approximation between the two region boundaries the friction force can be written as

$$F_w = \frac{1}{16} (F_{m1} + F_{m2}) f_{L0} A_w \rho_L V_{L0}^2 \quad (48)$$

where F_{m1} and F_{m2} are the values of the friction multiplier at the region boundaries, f_{L0} is the friction coefficient and V_{L0} is the velocity both evaluated assuming that the liquid flows at the total mass flow rate. The liquid only friction coefficient in the laminar flow range was calculated using [18]

$$f_{L0} = 96/R_e \quad (49)$$

$$\text{where } R_e = 2\rho V_{L0} \delta/\mu \quad (50)$$

In the transitional flow regime the Blasius' formula was used.

$$f_{LO} = 0.316/R_e^n \quad (51)$$

The two-phase friction multiplier was calculated using the Chisholm's correlation [19] given as follows:

$$F_m = 1 + (\lambda^2 - 1) \{ \gamma [x(1-x)] \frac{2-n}{2} + x^2 - n \} \quad (52)$$

where λ^2 is the ratio of gas only to liquid only frictional pressure gradient and γ is given by

$$\gamma = (C \lambda - 2^{2-n} + 2)/(\lambda^2 + 1) \quad (53)$$

The value of the exponent n in the above equations is 0.25. The value of C in Equation (53) depends on the flow regime. Lowry and Kawaji used $C=14$ to correlate their data. For turbulent gas and laminar liquid flow regime, the recommended value of $C = 12$ was used here.

The momentum equation can now be rewritten explicitly for the pressure difference between the region boundaries

$$\begin{aligned} P_1 - P_2 = & \rho_L [(1-\alpha_2)V_{L2}^2 - (1-\alpha_1)V_{L1}^2]/g_C + \rho_G (\alpha_2 V_{G2}^2 - \alpha_1 V_{G1}^2)/g_C \\ & + [(1 - \frac{\alpha_1 + \alpha_2}{2}) \rho_L + \frac{\alpha_1 + \alpha_2}{2} \rho_G] \frac{g}{g_C} \Delta z \\ & + \frac{1}{8} (F_{m1} + F_{m2}) f_{LO} \rho_L V_{LO}^2 \Delta Z/\delta - \text{Losses} \end{aligned} \quad (54)$$

Equation (54) was applied to the four regions in the crevice separately with sections 1 and 2 representing the region boundaries.

The contraction loss at the crevice inlet and the expansion loss at the crevice exit were calculated by applying the Chisholm's correlation to the liquid only losses at these locations. The liquid only losses were calculated using the following equations.

$$\text{Inlet Loss} = \frac{1}{2} \rho_l K_C V_{l0}^2 \quad (55)$$

$$\text{Exit Loss} = \frac{1}{2} \rho_l K_E V_{l0}^2 \quad (56)$$

The contraction loss coefficient at the inlet, K_C , was taken as 0.67 [18]. The expansion loss coefficient at the exit, K_E , was obtained by interpolation of the data of Arpaci, et al., (1982).

The Chisholm's correlation requires the exit void fraction, α_e , and the vapor velocity V_{ge} . If complete evaporation occurs, $\alpha_e = 1.0$. If coalesced bubble region extends to the crevice exit all exit quantities are directly calculated. In cases where the churn-annular flow regime extends to the crevice exit, α_e , V_{le} and V_{ge} were calculated assuming that the slip ratio remained constant in this region.

RESULTS AND DISCUSSION

The above analysis was programmed in FORTRAN language and solved using a CDC-7600 computer. The average computation time, using 10 degree circumferential increments, was approximately 3 seconds. The following results are typical for an eccentric tube in a support plate thickness of 1.0 inch (25.4 mm). The minimum distance between the tube and the support plate was chosen to be 0.001 inch (0.025 mm). The crevice pressure drop was 0.06 psi (413.4 Pa). The crevice inlet flow was saturated at pressures in the 350 psia (2413 kPa) to 500 psia (3447 kPa) range, with 0.0 void fraction.

Figure 4 shows the crevice flow velocities at a pool boiling heat flux of 57070 Btu/hr-ft² (179880 W/m²). The peaks in the exit velocities occurs at the circumferential edge of the "dryout" region. This is simply due to the fact that on the left side of the peak the exit to inlet velocity ratio remains constant and equal to the liquid to vapor density ratio while the inlet velocity increases with increasing gap size. Beyond this point the exit velocity decreases, due to a monotonic decrease in the vapor fraction, faster than the increase in the inlet velocity. When complete "dryout" occurs the liquid velocity at the exit is fictitious and corresponds to the last computed liquid velocity in the vertical element.

The predicted large velocity gradient at the edge of the dry region may cause a flow circulation and backflow of liquid into the dry region thus closing it and giving it the appearance of a large vapor bubble. This liquid must be blown off periodically to relieve the pressure within the dry region. This may explain the bubble bursts observed by Arpaci, et al., (1982) with a frequency of about 100 per second.

FIGURE 4

CREVICE FLOW VELOCITIES AT A HEAT FLUX OF 57070.
BTU/HR FT²

The corresponding flow regime boundaries are shown in Figure 5. The "coalesced" bubble regime correlation of Ishibashi and Nishikawa (1969) gave values from 81200 Btu/hr-ft² (255900 W/m²) at the minimum gap to 67700 Btu/hr-ft² (213400 W/m²) at the maximum gap. The respective average values through the thin film were 76900 Btu/hr-ft² (242400 W/m²) and 77100 Btu/hr-ft² (243000 W/m²). Effects of a reduction in heat flux to 10040 Btu/hr-ft² (31640 W/m²) has been shown in Figure 6. The "coalesced" bubble regime heat fluxes were 18600 Btu/hr-ft² (58600 W/m²) to 12100 Btu/hr-ft² (38140 W/m²). Those underneath the bubble were 19600 Btu/hr-ft² (61800 W/m²) and 14800 Btu/hr-ft² (46650 W/m²).

These results indicate that the correlation of Ishibashi and Nishikawa (1969) adequately represent the experimental findings of Bankoff, et al., (1982). The average increase in crevice heat fluxes over that of the bare tube are 53% and 30% at 10040 and 57070 Btu/hr-ft² (31640 and 179880 W/m²) respectively. Furthermore, the heat flux to a "coalesced" bubble through the liquid film are slightly higher than those calculated from the Ishibashi and Nishikawa correlation in most cases.

In Figures 5 and 6 the size of the radial gap at the "dry" region boundary indicates the minimum separation of the tube and the support plate necessary to eliminate the "dryout" for these cases. They are approximately 0.004 inch (0.10 mm) and 0.002 inch (0.05 mm) respectively. An increase in the crevice pressure drop or a decrease in the crevice length would lead to slightly larger values.

CONCLUSIONS

A one dimensional thermal-hydraulic model for vertical tube support plate crevices in shellside boiling heat exchangers has been developed. The model permits the prediction of flow regimes in narrow annuli. The flow rates of liquid and vapor along the crevice are two of the predicted variables which allow the calculation of chemical species concentration as a function of distance from the inlet. Corrosion of support plate and tubing can then be evaluated based on the predicted concentration gradients at a large number of sites with a minimum expenditure of computer resources.

FIGURE 5

CREVICE FLOW REGIMES AT A HEAT FLUX OF 57070.
BTU/HR FT²

FIGURE 6

CREVICE FLOW REGIMES AT A HEAT FLUX OF 10040.
BTU/HR FT²

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Nomenclature

A	Area
B	Body Force
$\bar{D}$	Thermal Diffusion Coefficient
D	Diameter
f	Friction Coefficient
F	Bubble Frequency
F_w	Frictional Force
F_m	Friction Multiplier
g	Gravitational Acceleration
g_c	Gravitational Constant
h_{fg}	Latent Heat of Vaporization
h	Heat Transfer Coefficient
k	Thermal Conductivity
K	Constant
Ln	Natural Logarithm
l	Axial Length
$\dot{m}$	Mass Flow Rate
$\dot{m}_{fg}$	Vapor Generation Rate
P	Pressure
Pr	Prandtl Number
q''	Heat Flux
$\dot{q}$	Heat Transfer Rate
r	Radius
Re	Reynolds Number
t	Time
T	Temperature
U	Overall Heat Transfer Coefficient
V	Volume
V	Velocity
x	Coordinate Along the Crevice Width
X_{tt}	Lockhart-Martinelli Parameter

y	Coordinate Along the Crevice Circumference
z	Coordinate in the Vertical Direction

Greek

α	Void Fraction
δ	Crevice Radial Gap
ϵ	Thickness of Liquid Film Underneath the Bubble
ρ	Density
θ	Angle Measured from the Point of Minimum Crevice Gap
σ	Surface Tension
μ	Absolute Viscosity
τ	Viscous Stress
x	Steam Quality

Subscript

O	Crevice Inlet
1, 2, 3, 4	Flow Regime Designations
B	Bubble
c	Cross Section, Contraction
CB	Coalesced Bubble
e	Expansion, Exit
f	Film
g	Vapor, Gas
H	High
i	Inside, Interface
l	Liquid
lo	Liquid Only
L	Low
o	Outside

P	Primary
s	Saturation Value
t	Tube
w	Wall
∞	Terminal

Superscript

.	Time Derivative
s	Superficial

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