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AUTHOR(S): Ralph A. Nelson
Cetin Unal

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Los Alamos, New Mexico 87545

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A PHENOMENOLOGICAL MODEL OF THE THERMAL HYDRAULICS OF CONVECTIVE BOILING DURING THE QUENCHING OF HOT ROD BUNDLES PART I: THERMAL HYDRAULIC MODEL

by

Ralph Nelson and Cetin Unal

Los Alamos National Laboratory
Nuclear Technology and Engineering Division
Engineering and Safety Analysis Group
Los Alamos, NM 87545

ABSTRACT

In this paper, a phenomenological model of the thermal hydraulics of convective boiling in the post-critical-heat-flux (post-CHF) regime is developed and discussed. The model was implemented in the TRAC-PF1/MOD2 computer code (an advanced best-estimate computer program written for the analysis of pressurized water reactor systems). The model was built around the determination of flow regimes downstream of the quench front. The regimes were determined from the flow-regime map suggested by Ishii and his coworkers.

Heat transfer in the transition boiling region was formulated as a position-dependent model. The propagation of the CHF point was strongly dependent on the length of the transition boiling region. Wall-to-fluid film boiling heat transfer was considered to consist of two components: first, a wall-to-vapor convective heat-transfer portion and, second, a wall-to-liquid heat transfer representing near-wall effects. Each contribution was considered separately in each of the inverted annular flow (IAF) regimes. The interfacial heat transfer was also formulated as flow-regime dependent.

The interfacial drag coefficient model upstream of the CHF point was considered to be similar to flow through a roughened pipe. A free-

stream contribution was calculated using Ishii's bubbly flow model for either fully developed subcooled or saturated nucleate boiling. For the drag in the smooth IAF region, a simple smooth-tube correlation for the interfacial friction factor was used. The drag coefficient for the rough-wavy IAF was formulated in the same way as for the smooth IAF model except that the roughness parameter was assumed to be proportional to liquid droplet diameter entrained from the wavy interface. The drag coefficient in the highly dispersed flow regime considered the combined effects of the liquid droplets within the channel and a liquid film on wet unheated walls.

The heat-transfer and interfacial drag models used were based on the flow-regime map noted above with length averaging of the flow-regime length if more than one regime existed in a given hydraulic cell.

NOMENCLATURE

A:	Surface area
A_f:	Flow area
Ca:	Capillary number
C_i:	Interfacial drag coefficient
d:	Bubble or droplet diameter
D:	Diameter
f:	Friction factor
f_l:	Wall area fraction in contact with liquid (dimensionless)
F_s:	Sink function used in Webb-Chen correlation
F_u:	Fraction of unheated surface area in the core
g:	Gravitational acceleration
G:	Mass flux
h:	Heat-transfer coefficient
h':	Phasic wall-to-fluid heat-transfer coefficient
h_{fg}:	Latent heat of evaporation
h'_{fg}	Modified latent heat of evaporation
k:	Thermal conductivity
m:	Viscosity
P:	Pressure
Pe:	Perimeter
Pr:	Prandtl number
q:	Heat flux
q':	Phasic heat flux
Re:	Reynolds number
t:	Time
T:	Temperature
V:	Velocity
Vol:	Cell volume
W:	Weighting factor
X:	Quality
Y_b:	Bubble height
Z:	Axial elevation
α:	Void fraction
α_l:	Thermal diffusivity

Γ:	Vapor generation rate
δ:	Film thickness
$\Delta\rho=\rho l-\rho v$	Density difference
ε:	Roughness
ε_r:	Emissivity
λ:	Taylor wavelength
ρ:	Density
σ:	Surface tension
σ_r:	Stefan-Boltzmann constant
τ:	Shear stress

Subscripts

ac:	Actual
ag:	Agitated
b:	Bubble
Brom:	Bromley correlation
CHF:	Critical Heat Flux
c:	Core
cr:	Critical
dd:	Single droplet
d:	Droplets
Den:	Denham correlation
df:	Highly dispersed flow
fr:	Free stream
fls:	Flashing
fo:	Single phase
lf:	Liquid film
film:	Film boiling
g:	Gas
gap:	Gap between wall and liquid core
hom:	Homogeneous
h:.	Hydraulic
i,inv:	Interfacial, inverted annular flow
i,lf	Interfacial, liquid film
i,d	Interfacial, droplet
i,df:	Interfacial, highly dispersed flow

i,sb:	Interfacial, subcooled boiling
i,fr:	Interfacial, free stream
i,sm:	Interfacial, smooth IAF
i,rw:	Interfacial, rough-wavy IAF
i,dd	Interfacial, droplets
i,pa:	Interfacial, post-agitated IAF
i,cell:	Interfacial, finite-difference cell
inv:	Inverted annular flow
il:	Interface-to-liquid
ig:	Interface-to-gas
lc:	Liquid core
l:	Liquid
min:	Minimum film boiling
nuc:	Nucleate boiling
pa:	Post-agitated
r:	Relative
ra:	Radiation
rw:	Rough-wavy IAF
sat:	Saturation
sv:	Saturation temperature of vapor
sb:	Subcooled boiling
sm:	Smooth IAF
slab:	Heat structure
total:	Total
top:	Top of the finite-difference cell
tb:	Transition boiling
teo:	Theoretical
u:	Unheated
v:	Vapor
w:	Wall
wg:	Wall-to-gas
wl:	Wall-to-liquid
wi:	Wall-to-vapor/liquid interface
W-C	Webb-Chen Correlation

I. INTRODUCTION

Boiling systems normally operate in the nucleate boiling regime. However, as the heat flux from the heated wall increases, a point is reached where the heated wall can no longer sustain liquid contact. Such a situation is called the Critical Heat Flux (CHF), the burnout, or the dryout condition. Prediction of the wall heat transfer beyond the CHF location is an important aspect of nuclear reactor safety and the safety of other boiling systems, such as cryogenic systems, metallurgical processing, and steam generators.

Boiling beyond the CHF location can occur with several different flow patterns depending upon the flow quality and mass flux at the CHF point. Recently, a detailed study of the inverted annular flow (IAF) regime was reported by Ishii and his coworkers (1, 2, 3). A sketch of these regimes is given in Fig. 1a. If CHF occurs at low or negative (subcooled liquid) flow qualities, the flow pattern can be expected to be an inverted annular flow. In inverted annular flow, a liquid core is surrounded by an annular vapor film as shown in Fig. 1a. Further downstream, the liquid core may break up into an agitated region of slugs or large droplets, and later followed by a small-droplet dispersed flow region. At moderate to high qualities, the flow pattern may be thought of as dryout of a liquid film from the wall, which creates a small droplet dispersed flow regime downstream of the dryout (see Fig. 1b). In the post-CHF regime, the two-phase fluid may exist in a thermodynamic nonequilibrium state as confirmed experimentally by Nijhawan et al. (4) , Evans et al. (5), and Gottula et al. (6) for single tubes, and Unal et al. (7) and Loftus et al. (8) for rod bundles.

In the last two decades, a significant number of experimental and analytical studies have been published that report on the post-CHF boiling. Chen (9) recently summarized the state of the art in convective post-CHF heat transfer. He classified the studies into two major groups: local models and history-dependent models. Unal et al. (10) recently assessed some of the recommended local and history-dependent models against rod bundle data obtained at Lehigh University (11). He showed that a large amount of scatter exists between the predictions from published correlations and measured data.

When these published correlations are used in computer codes such as TRAC (12) and RELAP (13), this disagreement frequently becomes more profound because the codes are transient codes while most of the available models and correlations were developed from steady-state experiments. Another reason for the disagree-

ment is that most of the models were developed from data that were missing information on the companion phenomena required to describe the process. For example, heat-transfer correlations are developed without information on the hydraulic aspects of the experiments. Similarly, hydraulic correlations are frequently derived from adiabatic experiments. Therefore, the majority of these models, both heat-transfer and hydrodynamic, report correlations and data which cannot be separated into the phasic components. The data obtained by Unal et al. (7) and Evans et al. (5) are examples of this situation. It includes only the heat-transfer information, wall and vapor temperatures, heat flux, etc., while information on void fraction and phasic velocities is not available. Naturally, most of the heat-transfer models developed from such data bases used the assumption of homogeneous flow. Other researchers put emphasis on the measurement of hydrodynamic parameters such as velocity and void fraction, performing their tests on adiabatic test rigs. Models developed from this kind of data must use other models for heat transfer or assume no influence, due to their omission from the experimental results.

Thermal hydraulic computer codes such as TRAC solve the mass, momentum, and energy equations for each phase. They require constitutive relations to determine mass, momentum, and heat-transfer interchange between the phases and between both heated or unheated structures and the phases. Since phasic constitutive relations are generally not available except under certain special conditions, code developers are forced to infer these phasic relationships based upon limited information available from the data they are analyzing. This is frequently done by modifying existing models and combining these modified models to represent the different phenomena for the required phasic contributions.

To develop more accurate models, information on heat transfer must be used with the best available hydrodynamic data. If such a model is used in the large computer codes, the prediction of post-CHF heat transfer can be improved. Until recently, there was limited information on the flow patterns of the post-CHF region. In this paper, a post-CHF model for large-scale computer codes is developed based on the hydrodynamic information recently obtained by Ishii and his coworkers (1, 2, 3). This model is developed for the TRAC-PF1/MOD2 (14) computer code based on the IAF and annular flow maps shown in Fig. 1. In this case, the model suggested by Ishii for the IAF regimes shown in Fig. 1a collapses into the annular flow model shown in Fig. 1b. The developed model uses separate heat-transfer and interfacial drag models for each IAF regime encountered in Ishii's study. The formulation of the thermal hydraulic heat-transfer model is explained in section II. Assessment of

the model with Winfrith steady-state post-CHF tube data (15), transient Lehigh rod bundle data (11), and Cylindrical Core Test Facility (CCTF) rod bundle data (16) is presented a companion paper.

II. DESCRIPTION OF THE THERMAL HYDRAULIC MODEL

The model discussed below is developed for the TRAC-PF1/MOD2 computer code to model the thermal hydraulics in a core undergoing reflood. The TRAC program is an advanced best-estimate computer program for analyzing light-water reactor (LWR) accidents. The TRAC series of codes formulate the fluid dynamics using the six equation, two-fluid, nonequilibrium model with a staggered difference scheme. The mass, momentum, and energy equations are available in three-dimensional form in the vessel component and in one-dimensional form in other available components. A detailed description of the capabilities of the code and the numerical solution method is available in Ref. 14.

The field equations used in the TRAC-PF1/MOD2 require closure relationships to represent the wall heat transfer (wall-to-liquid and wall-to-vapor), the interfacial heat transfer, the wall shear (wall-to-liquid and wall-to-vapor) , the net vaporization rate, and state equations. The TRAC-PF1/MOD2 code, as well as other TRAC series codes, invokes a quasi-steady approach to the heat-transfer coupling between the wall and the fluid as well as the other closure relations for interfacial heat transfer , interfacial drag, and wall-to-fluid drag. This quasi-steady approach assumes detailed knowledge of the local fluid parameters and ignores time dependencies of the closure quantities themselves. This implies the time rate of change in the closure relationships become infinite and the time constants are zero. This approach has the advantages of being reasonably simple and applicable to a wide range of problems.

A. Post-CHF Flow Regimes

The flow regimes downstream of the quench front are determined from the map suggested by Ishii and his coworkers (1, 2, 3). Ishii and DeJarlais (1, 2) performed visualization experiments of IAF in the central channel of a heated double quartz tube. A summary of their qualitative results is depicted in Fig. 1a. The inverted annular region was initiated using concentric injection nozzles with liquid injected from an inner nozzle surrounded by a vapor annulus. Both motion and

still pictures were taken to identify the flow-regime characteristics. In the region directly downstream of the nozzles, a smooth liquid core was observed. This was followed by wave development on the liquid core's surface. The wavelengths were on the order of 10 mm, with droplets being sheared from the wave crests. Also observed (although not shown in the figure) was a thin, highly agitated annular sheet of liquid near the heated wall. Above this region, an agitated slug/churn region was observed. Droplets (3-mm-diam) swept past the slugs. The slugs were deformed into multiple ligaments and were eventually broken up. In the dispersed region, the droplets were evaporating and desuperheating the steam. Several droplet sizes were observed from the agitated liquid annulus (0.05 mm), from the wave crests (0.2 mm), and from the slug breakup (0.6 to 3 mm).

Obot and Ishii (3) extended the work of Ishii and DeJarlais (1, 2) and developed the flow-regime transition criteria. The final results were developed in terms of the capillary number and the length above the quench front (indicated in Fig. 1 and listed in Table 1). The relatively large droplets observed downstream of the agitated region become smaller as the void fraction increases further downstream. We have further refined the dispersed flow to consist of two regions. First, a dispersed flow regime with large droplets, and second, a highly dispersed flow regime

TABLE 1
POST-CHF FLOW-REGIME TRANSITION CRITERIA

Regime	Ishii's Corr.	Implementation in TRAC	
Smooth IAF	$\frac{Z}{D} = 60\ Ca^{1/2}$	$\frac{Z}{D} = 60\ Ca^{1/2}$,	$0.05 < \alpha < .3$
Rough-Wavy IAF	$\frac{Z}{D} = 295\ Ca^{1/2}$	$\frac{Z}{D} = 295\ Ca^{1/2}$,	$0.3 < \alpha < 0.4$
Agitated IAF	$\frac{Z}{D} = 595\ Ca^{1/2}$	$\frac{Z}{D} = 595\ Ca^{1/2}$,	$0.4 < \alpha < 0.75$
Dispersed IAF (post-agitated IAF)			$0.75 < \alpha < 0.98$
Highly dispersed IAF			$\alpha > 0.98$

with fine droplets. The highly dispersed flow regime was assumed to occur when the cell void fraction was greater than 98%. Additional constraints based on the void fraction were also introduced to force the IAF regimes to occur within certain void fraction ranges (see Table 1). This was required because transient calculations realize situations not present in the steady-state experiments used in model development. For example, the early part of a prediction with liquid flowing into an empty tube may, based upon the capillary number for the liquid velocity alone, indicate flow-regime lengths downstream at locations where the liquid has not yet had time to move. Table 1 shows Ishii's original post-CHF flow-regime correlations and the modifications made for application to the TRAC model.

Since the capillary number is negative for downflow conditions, motion of the flow-regime positions is determined by liquid velocity movement back toward the quench front location.

B. Partitioning of the Constitutive Relations for Wall-to-Fluid and Interfacial Heat Transfer

The TRAC codes partition the total energy transferred from the wall to a nonequilibrium (both mechanical and thermal) two-phase mixture into the components going into the respective phases. This division is required by the two-fluid model to determine the sensible heat present in each phase. The solution of the conduction problem associated with each structure present in the fluid, however, uses the total energy transferred to the phases. Thus, the total wall heat flux consists of the two phasic components (wall-to-vapor and wall-to-liquid) and is given by

$$q_{total} = q'_{wl} + q'_{wg} = f_l\, h'_{wl} (T_w - T_l) + (1 - f_l)\, h'_{wg} (T_w - T_g) , \qquad (1)$$

where h'_{wl} and h'_{wg} are the separate phasic heat-transfer coefficients (HTCs), not yet defined by experiments. The measurement and modeling of the liquid/solid contact fraction of the total area, f_l, is very limited for forced convection. Therefore, the effect of the phasic wall/liquid contact area is assumed to be included within both phasic models and the weighting factors used to combine the separate correlations.

The interfacial heat-transfer rate is calculated by combining the volume-averaged liquid-side and vapor-side heat-transfer rates. The interface is always assumed to be at the saturation temperature corresponding to the partial steam

pressure. The rate of mass transfer between phases is determined from a simple thermal-energy jump relationship given by

$$\Gamma = \frac{-(q_{il} + q_{ig})}{h_{fg\,Vol}} = \frac{-\left[h_{il} A_{il} (T_{sv} - T_l) + h_{ig} A_{ig} (T_{sv} - T_g)\right]}{h_{fg\,Vol}} \quad . \tag{2}$$

C. Correlation Selection and Modification

Wherever possible, correlations known to apply to a given regime for a particular closure quantity were used. Frequently, however, the original correlation could not be applied directly but had to be modified. For those cases, we tried to use the "kernel" or "functional" dependence of the original correlation and modify only its magnitude by use of a multiplier. When no correlations were available for given regimes, we tried to define known bounding regimes and use a weighting function between the known regimes to represent the unknown quantities.

III. WALL HEAT TRANSFER

The wall-to-fluid HTCs in the nucleate boiling region are calculated using the Chen correlation as discussed in Ref. 14. While interfacial heat transfer and drag for nucleate boiling will be discussed in greater detail, wall heat transfer will not be discussed in detail due to its straightforward nature. The following subsections describe the HTC correlations used in the wall-to-fluid heat-transfer model in the post-CHF region.

A. Transition Boiling

The transition boiling regime spans the boiling surface between CHF and minimum film boiling. In earlier TRAC codes (TRAC-PF1/MOD1, Ref. 12), transition boiling was thought to be a combination of both nucleate boiling (wet-wall) and film boiling (dry-wall) heat transfer. A weighting factor representing the fraction of wet versus dry surface that was dependent upon wall temperature was applied to both the CHF and minimum film boiling heat flux. Transition boiling was assumed to occur if the wall temperature was between T_{CHF} and T_{min}.

This modeling approach does not depend upon axial position. Instead, it depends upon the local wall temperature at any position downstream of the CHF point. It has been observed that very different results for wall temperature history

and precursory cooling can be obtained if the axial heat-structure node size is changed from large to small, or vice versa (17). We determined that this node size sensitivity arises because the only limiting factor within the code in such a local temperature formulation is that imposed by the numerics associated with axial conduction. In particular, no limit is applied to the spatial dimension over which a given convective heat-transfer process, such as transition boiling, must apply. The local temperature transition boiling formulation will allow the axial temperature distribution to grow sharper as the node size is decreased until a size smaller than that required to properly model the axial conduction effect is reached. The formulation has no way of knowing if this is the proper spatial temperature distribution or not. Nothing is present within the local wall temperature formulation of transition boiling to prevent the "collapse in the axial direction" of the convective transition boiling process to that thermal distribution limit imposed by axial conduction. While some flow states exist where this collapse of transition boiling to the axial conduction limit is the correct representation of the convective process, in general this collapse is not correct.

As seen in many experimental studies (5, 6, 7), the extension of the transition boiling downstream of a CHF point in forced convective flow depends upon the thermal-hydraulic conditions at the CHF point. Thus, to eliminate difficulties associated with nodalization, an axial-history-dependent transition boiling model is developed.

Typical conditions for a post-CHF convective flow are illustrated in Fig. 2. Figure 2 shows Ishii's inverted annular flow regime map with the postulated axial wall heat flux profile. The wall heat flux at the CHF point is significantly higher than the heat flux of the film boiling regime. The transition boiling heat flux is limited by a maximum of q_{CHF} and a minimum of q_{film}. It was assumed, in the current model, that the total transition boiling heat flux exponentially decreased with the axial distance from the CHF location. The total transition boiling heat flux is given by

$$q_{tb} = q_{CHF}\, e^{[-B\,(Z-Z_{CHF})]}. \qquad (3)$$

The determination of the coefficient B is not straightforward and should ultimately consider all possible post-CHF flow conditions and wall material and thickness combinations that are of interest. For now, we propose that three

hydrodynamic flow parameters—the capillary number, the vapor Reynolds number, and the void fraction at the CHF point—should be considered in determining B.

Ishii's flow-regime map indicates that the length of each IAF regime is proportional with the square root of capillary number defined at the CHF point. Thus, the IAF flow regimes extend higher when the liquid velocity increases at the CHF point for a given pressure. We assume that the length of transition boiling should exhibit the same trend; higher liquid velocities at the CHF point should extend the transition boiling region further downstream. Therefore, the first dimensionless parameter in determining B is the capillary number at CHF. B is assumed to be proportional to the inverse square root of capillary number ($B = \text{Constant} \times Ca^{-1/2}$). The proportionality constant was found to vary with vapor Reynolds number (defined at the CHF point) from 16 to 10 when the vapor flow changed from laminar to turbulent.

For higher void fractions, the flow regimes downstream of the CHF point are expected be annular transition and dispersed flow, respectively. We believe that the transition boiling region should occur in a relatively short region for these high-void flow conditions and should diminish when the void fraction goes to unity. Thus, the transition boiling region is forced to decrease with increasing void fraction if the void fraction at CHF is between 0.8 to 0.995.

To better understand this formulation, we can first note that the length of the transition boiling region can be determined from Eq. (3) to be

$$Z_{tb} - Z_{CHF} = \frac{-\ln\left(\frac{q_{film}}{q_{CHF}}\right)}{\text{Constant } Ca_l^{-0.5}} \tag{4}$$

for the case where $\alpha < 0.8$. Dividing Eq. (4) by the diameter yields

$$\frac{Z_{tb} - Z_{CHF}}{D} = \text{Constant } Ca^{1/2}, \tag{5}$$

which is the same type of formulation developed by Ishii (3) for his flow-regime map.

So far we have discussed the transition boiling region for upflow conditions. In computer code calculations as well as in integral experiments, the vapor and/or

liquid can flow downward. In these situations, the capillary and Reynolds numbers become negative. As with Ishii's flow map, the transition boiling model discussed above is not valid under downflow conditions. Until data are available for the flow regime as well as other thermal hydraulic behavior in downflows, the coefficient B is assumed to be either a constant or void-fraction-dependent, as shown in Table 2.

In steady-state post-CHF tests using hot patches (5, 6, 7), the wall temperature profile showed a sharp increase following the CHF point. We believe that the transition boiling regime for such tests should be limited to a very short region. Thus, the coefficient B (see Table 2) was multiplied by a constant to ensure that the length of the transition region was very small in calculations including hot patches.

TABLE 2
COEFFICIENT B AS A FUNCTION OF REYNOLDS NUMBER AND VOID FRACTION

$$B = \text{Cons.}\ Ca_l^{-0.5} \qquad \text{for } \alpha < 0.8,$$

$$B = e^{\left(\ln(\text{Cons.}\ Ca_l^{-0.5}) + \left(\frac{\alpha - 0.80}{0.995 - 0.8}\right)^{1.5} (7.601 - \ln(\text{Cons.})\ Ca_l^{-0.5})\right)} \qquad \text{for } 0.80 < \alpha < 0.995,$$

$$B = 2000 \qquad \text{for } 0.995 < \alpha$$

where

$$\text{Cons.} = 16 \quad \text{if} \quad Re_v < 2000$$

$$\text{Cons.} = 10 \quad \text{if} \quad Re_v > 2000$$

$$Ca_l = \left(\frac{V_l \mu_l}{\sigma}\right)_{CHF}$$

$$Re_v = \left(\frac{\alpha \rho_v D_h V_v}{\mu_v}\right)_{CHF}$$

To partition the total transition boiling heat flux into its phasic components, the gas phase HTC, h_{wg}, was evaluated by the Webb-Chen correlation (18). This correlation is explained in section III.B. Once h_{wg} is evaluated, the wall-to-vapor heat flux, q_{wv}, can be calculated. Then h_{wl} is found by dividing the difference between the total transition boiling heat flux, q_{tb}, and the heat flux from wall to vapor, q_{wv}, by the driving force of (T_w-T_l).

Several tests are made to determine if the transition boiling regime exists. These tests are defined by three basic considerations as follows: 1) Has CHF occurred? 2) Are we far enough downstream that film boiling must exist? and 3) Is the void fraction too high? The first consideration was relatively simple, and determined that CHF had been exceeded if $T_w > T_{CHF}$, using temperature, or if $q_{nucleate\ boiling} > q_{CHF}$, using heat flux. The second consideration was made to save computional time and involved several steps. In the first step, the distance past the CHF point was considered, with transition boiling being possible if $Z\text{-}Z_{CHF} < Z_{tb,max}$. Then, if transition boiling was possible based on this distance criterion, both the transition boiling and film boiling heat fluxes were evaluated. Finally, if q_{tb} was greater than q_{film}, transition boiling was determined to have occurred at that particular Z. This heat flux comparison leads to the definition/determination of Z_{tb}.

The third consideration involved a test on void fraction to ensure continuity in the high-void region as the flow becomes single-phase vapor. If $\alpha > 0.995$ (defining the single-phase vapor flow), q_{tb} was reevaluated to ensure a smooth transition between the flow regimes (see Ref. 19). The wall-to-vapor HTC, h_{wg}, was also interpolated to ensure a smooth transition between flow regimes if $\alpha > 0.995$.

B. Film Boiling

The film boiling heat-transfer regime incorporated several different correlations to describe the HTCs in each of the IAF regimes. The film boiling regime was assumed to occur when $T_W > T_{CHF}$ and $Z - Z_{CHF} > Z_{tb}$. The wall-to-liquid and wall-to-vapor HTCs, h_{wl} and h_{wv}, were treated separately for each of the individual IAF regimes. Figure 3 shows the selection logic for the HTCs in the film boiling regimes. In this figure, the trend of each correlation in each IAF regime is shown. While the weighting functions are shown as linear in Fig. 3, this is only a conceptual representation, with the true weighting frequently being non-linear. The following paragraphs discuss the selected HTC correlations in each of the IAF regimes. Table 3 summarizes the correlations used in film boiling regime.

TABLE 3
WALL-TO-FLUID HEAT-TRANSFER CORRELATIONS IN FILM BOILING

Regime	Wall-to-Liquid	Wall-to-Vapor
Smooth IAF	$h_{Den} + h_{ra}$	not used
Rough-Wavy IAF	$h_{Den}\left(\frac{Z_{slab}-Z_{sm}}{Z_{rw}-Z_{sm}}\right)^{0.5} + h_{Brom}\left(\frac{Z_{slab}-Z_{sm}}{Z_{rw}-Z_{sm}}\right)^{0.9} + h_{ra}$	$1.2\ h_{W\text{-}C}$
Agitated and Post-Agitated IAF	$h_{Brom}\left(F(Re_{v,ag})\left(\frac{0.75-\alpha}{0.75-\alpha_2}\right)^{F_1(Re_{v,ag})}\right) + h_{ra}$	$1.2\ h_{W\text{-}C}$
Highly Dispersed IAF	not used	$1.2\ h_{W\text{-}C}$

$$h_{Den} = 0.4472\ k_v \left[\frac{g\,(\rho_l-\rho_v)}{\mu_v V_v}\right]^{\frac{1}{2}}$$

$$h_{Brom} = 0.62\left(\frac{\rho_g k_g^3(\rho_l-\rho_g)\,g\,h_{fg}'}{\mu_g(T_w-T_{sat})\lambda}\right)^{0.25}$$

$$h_{ra} = (0.9999-\alpha)\,\sigma_r\,\varepsilon_r\,\frac{(T_w^4-T_{sat}^4)}{(T_w-T_l)}$$

$$F(Re_{v,ag}) = -0.09567 + 4.8644\ 10^{-4} Re_{v,ag}$$

$$F_1(Re_{v,ag}) = 6.18 - 9.37\ 10^{-3} Re_{v,ag} + 5.38\ 10^{-6}\ Re^2_{v,ag} - 1.03\ 10^{-9}\ Re^3_{v,ag}$$

$$h_{W\text{-}C} = \frac{f}{2} G\ X_a\ C_{p_{vf}}\ Pr_{vf}^{-2/3} (1 + F_s) \left(1 + 0.8 \left(\frac{Z}{D_h}\right)^{-1}\right), \quad F_s = 250 \left(\frac{P}{P_{cr}}\right)^{0.69} \left(\frac{1-X_{ac}}{X_{ac}}\right)^{0.49} Re_v^{-0.55}$$

$$Re_v = \left(\frac{\alpha\ \rho_v D_h V_v}{\mu_v}\right)_{ag}^{\left(1+(1-0.95)\left(\frac{P-2\ 10^5}{10\ 10^5 - 2\ 10^5}\right)\right)}$$

In the smooth IAF regime, the total heat generated by the wall was assumed to be transferred directly to the liquid interface across the vapor film surrounding the liquid core. Denham (20) recently developed an expression for heat transfer in the IAF. The derivation of that correlation was similar to that of the Bromley correlation except that the vapor film thickness was obtained from a force balance on vapor film. By dividing the thermal conductivity of vapor by the vapor film thickness, the wall-to-liquid HTC is determined. Since the total heat generated by the wall was assumed to be transferred directly to the liquid, no heat transfer to the vapor phase was assumed to occur. The heat transfer by radiation from wall-to-liquid was also considered and added to the wall-to-liquid HTC for all film boiling regimes.

In the flow regimes downstream of smooth IAF, the wall was assumed to be cooled by a combination of vapor and liquid. The Webb-Chen (18) correlation was used to obtain the wall-to-vapor HTC in the remainder of the flow regimes downstream of smooth IAF. The Webb-Chen correlation was developed from a non-equilibrium data base for single tubes. It was based upon the momentum-transfer analogy and considered possible entrance-region effects and the effect of entrained liquid droplets.

The Webb-Chen correlation alone cannot result in the correct prediction of the heat transfer in the IAF regimes downstream of the smooth IAF. Ishii (1, 2, 3) experimentally observed the existence of a fine sheet of liquid drops/ligaments between the liquid core of the IAF regime and the wall. It is clear from Ishii's study that the hydrodynamic behavior in the rough-wavy and agitated IAFs shows a unique characteristic. Interfacial surface area is increased significantly and liquid exists near the wall, possibly in momentary contact with the wall. As a result, the heat-transfer mechanism in these flow regimes is significantly enhanced. While the heat-transfer aspects of their flows were not measured by Ishii and coworkers, the

existence of a posttransition boiling region (near region) immediately downstream of the CHF point was found by Unal et al. (21). They indicate that the evaporation of liquid is very efficient in the posttransition boiling region, so the measured vapor temperature is close to the saturation temperature of the fluid. They hypothesize that the heat-transfer enhancement could be due to liquid/wall direct contact heat transfer. Another mechanism which might explain this efficient process is the increase in local turbulence near the wall due to the existence of liquid droplets (22, 23).

Although either of these theories can predict heat-transfer enhancement in the IAF regimes downstream of smooth IAF, it is not clear that only one of two is the responsible mechanism for the overall heat-transfer enhancement. It is more likely that both heat-transfer mechanisms can exist. In our model, we name this heat-transfer mechanism the "near-wall liquid" effect. We assumed that it starts at the beginning of the rough-wavy IAF and gradually increases with increasing axial distance until the agitated IAF region. In agitated flow, we postulate that the near-wall liquid effect is at a maximum due to high turbulence and some possible liquid/wall contact. Downstream of the agitated region, this effect gradually decreases and finally becomes negligible in highly dispersed flow (21).

We induce this near-wall liquid effect through the wall-to-liquid HTC, h_{wl}. One can argue that this effect is due to either liquid/wall contact or turbulence enhancement where, much like Denham's model, the transport of wall energy into the vapor and then into the liquid is short-circuited due to the extremely fast and efficient transport process. There is no mechanistic model to predict this contribution. Therefore, we selected the modified Bromley correlation (24) as the "kernel" for the model. Our initial assessment attempts indicated that the magnitude of the near-wall effect for different mass and heat fluxes could not be predicted correctly by the Bromley correlation alone since this correlation depends upon only the pressure and wall temperature, a point raised by Denham several years ago. The results indicated that the near-wall effects tend to increase with increasing mass flux at a given heat flux and inlet subcooling. This is consistent with the experimental findings reported by Unal et al. (see Fig. 9 of Ref. 21) who indicated that their transition region extended further downstream with an increase in vapor flux.

Thus, for the Bromley correlation, we introduce a multiplier which depends on the vapor Reynolds number. The functional form of the multiplier was found by matching the measured wall and vapor temperatures to five of the Winfrith steady-state post-CHF tests. It was found to be a linear function of vapor Reynolds

number defined at the agitated IAF. The functional form listed in Table 3 gives a multiplication factor varying between 0.2 and 1.0 for Reynolds number varying between 500 and 2300. With this modification, low-pressure data were predicted reasonably well while higher-pressure data showed an underprediction of the measured wall temperatures. This underprediction arises because the Reynolds number becomes much higher than 2300 due to the increase in vapor density with pressure. The multiplication factor always becomes 1, resulting in high wall-to-liquid HTCs at higher pressures. Therefore, a pressure-dependent exponent was introduced and applied to the Reynolds number as summarized in Table 3. This is obviously an area where phenomenological modeling is needed in the feature.

As noted earlier, the near-wall liquid effects were forced to be diminished gradually with increasing axial distance downstream of the agitated IAF. This was done by introducing the weighting function listed in Table 3. The exponent of the weighting function was also found to be function of vapor Reynolds number at the beginning of the agitated IAF.

If the liquid is subcooled in the film boiling regime, an additional HTC, h_{gam}, is calculated and used to separate the latent heat of evaporation effect from the sensible heat effect. As Denham (20) indicated, the interface of the subcooled liquid becomes saturated due to condensation. Thus, the heat transfer from saturated interface to subcooled liquid core, q_{il}, was expressed as the conduction solution of a cylinder with a change in the surface temperature. The time required by his expression can be calculated as the ratio of node size to liquid velocity. The q_{il} becomes

$$q_{il} = k_l \sqrt{\frac{V_l}{\pi \alpha_l \left(Z_{top} - Z_{tb}\right)}} \frac{V_l D_h^2}{V_l D_h^2 + 15 \alpha_l \left(Z_{top} - Z_{tb}\right)} \left(T_{sat} - T_l\right) . \qquad (6)$$

Finally, h_{gam} can be calculated as the total wall-to-liquid heat transfer, less q_{il} divided by the driving force of $(T_w - T_l)$. This h_{gam} is calculated in all IAF regimes except highly dispersed flow.

IV. INTERFACIAL HEAT TRANSFER

To estimate the heat and mass transfer rates between phases, the interfacial surface areas and the vapor-to-interface and liquid-to-interface HTCs are required

(see Eq. 2). The term q_{il} in Eq. 2 accounts for the sensible heat transferred to or from the interface where the thermal energy is converted to or released as latent heat. The interfacial heat-transfer model treats evaporation and flashing separately. Consequently, two different liquid-side heat-transfer factors exist in the current model: one for flashing and another for evaporation. Evaporation occurs if $T_{sv} < T_l < T_{sat}$ and flashing occurs if $T_l > T_{sat}$. The interfacial surface area and heat-transfer coefficients for evaporation, condensation, and flashing are defined for each of the IAF regimes and other flow regimes. In addition to the post-CHF regimes, we will limit ourselves to consideration of the nucleate boiling region. The selected correlations for determining liquid- and vapor-side HTCs and interfacial areas are summarized below.

In the nucleate boiling region, the liquid-side HTC is calculated by using either the Chen-Mayinger or Wittaker correlations (see Ref. 14). The vapor-side HTC was assumed to be constant, 1000 $W/m^{2-\circ}C$, which was chosen to be large enough to keep the vapor near saturation. The interfacial surface area in the nucleate boiling regime is calculated depending upon the flow regime. For bubbly flow, a simple expression was used for the bubble diameter as suggested by Ishii (25). In annular mist flow, the interfacial surface areas due to liquid film on the wall and liquid droplets are calculated separately and then combined. The droplet diameter, the entrainment fraction, and other parameters are calculated using models developed by Ishii and his coworkers (see Ref. 14).

In the nucleate boiling regime, if the liquid temperature is higher than the saturation temperature, $T_l > T_{sat}$, flashing of the liquid is allowed. To determine the liquid-side HTC, a simple flashing model is used and the interfacial surface areas are determined as discussed above. The heat-transfer coefficient due to flashing was calculated using the kinetic theory of evaporation from a liquid surface (26). This theoretical maximum evaporation rate was converted to an equivalent HTC. The coefficient of 0.04 suggested by Hsu and Graham, to predict the evaporation rate in experimental studies was modified for each of the individual flow regimes in nucleate and film boiling. A modified coefficient of 0.002 was used for the possible nucleate boiling regimes. The HTC for nucleate boiling is given by

$$h_{fls,nuc} = 0.002\, h_{fls,teo} = 0.002 \times 0.01857 \frac{\rho_v h_{fg}^2}{T_{sat}^{1.5}} . \qquad (7)$$

In the smooth, rough-wavy, agitated, and post-agitated IAFs, th interfacial area was calculated by adding the surface area of the liquid core and the surface area of the bubbles in the liquid core if they exist. In post-CHF experiments (11, 21), the measured vapor superheat was not significantly high in the region near the CHF location. The axial vapor temperature profile showed an S-shaped profile; low in the region near to CHF and high in the far region. This S-shaped vapor temperature profile was encountered in relatively high-void-fraction dispersed-flow conditions (21) as compared to IAF. Thus, we would expect very little vapor superheat in the smooth, rough-wavy, and agitated IAFs. For this reason, the vapor-to-interface HTC in these flow regimes, h_{ig}, was assumed to be relatively high. A constant value of 3000 W/m^2-°C was used to ensure that the vapor was not superheated significantly. If the liquid is subcooled, the rate of heat transfer is represented by h_{gam} as discussed in the previous section. When the liquid is superheated, the flashing HTC in the inverted annular flows (the smooth, rough-wavy, agitated, and post-agitated flows), $h_{fls,inv}$, is obtained by multiplying $h_{fls,teo}$ with a constant of 0.02 ($h_{fls,inv} = 0.02\ h_{fls,teo}$).

For highly dispersed flow, the interfacial area consists of two components: liquid film on cold walls and liquid droplets. The vapor-to-interface HTC, h_{ig}, was obtained from the correlation for the rate of vapor generation in dispersed flow suggested by Unal et al. (10). This correlation was modified and converted to a heat-transfer coefficient. In both highly dispersed flow and other IAF flows, if there is a spacer located in a finite-difference cell, h_{ig} is assumed to be 10^6 W/m^2-°C to simulate the enhanced interfacial heat transfer. For subcooled liquid conditions, the rate of heat transfer is considered to be h_{gam} as discussed above. The flashing HTC in highly dispersed IAF, $h_{fls,df}$, is obtained by multiplying $h_{fls,teo}$ with a constant of 0.02 ($h_{fls,df} = 0.02\ h_{fls,teo}$).

Depending upon the location of a given cell with respect to the IAF elevations, the appropriate HTCs are determined using weighting factors (W_{sb}, W_{df}, W_{inv}). These weighting factors ar based upon cell-length averaging. The logic of the weighting factors is summarized in Ref. 19. The final HTCs were obtained by the following equation:

$$h = h_{sb}\, W_{sb} + h_{df}\, W_{df} + h_{inv}\, W_{inv}\ , \tag{8}$$

where h represents either h_{ig} or h_{il}/h_{fls}.

V. INTERFACIAL DRAG COEFFICIENTS

Models for the interfacial drag coefficients in a reflooding core were developed based on the inverted annular flow map shown in Fig. 1. For each flow regime, a separate interfacial drag model was developed. As noted earlier, since Ishii's flow map does not consider transient effects and the resulting availability of liquid, void fraction was used to limit the potential IAF regimes. This void effect is also present in lower liquid flow rate situations, where the liquid flow is insufficient to produce the IAF configuration. Thus, models for each of the flow regimes were defined in the flow regime-void fraction plane, as shown in Fig. 4 and summarized in Table 4. Three void fraction regions were identified: a) the low void fraction region characterized by void fractions less than 0.75, b) the high void fraction region characterized by void fractions higher than 0.98, and c) the intermediate void fraction region between the high- and low-void regions. In the following subsections, the interfacial drag models for each IAF regime are presented from the bottom to the top of the channel in the following order: subcooled and saturated nucleate boiling, smooth inverted annular, rough-wavy inverted annular, agitated, highly dispersed, and dispersed (or post-agitated) flow.

A. Subcooled and Saturated Nucleate Boiling Interfacial Drag Model

Subcooled nucleate boiling is characterized by two regions: 1) the partial-boiling region in which the bubbles remain attached to the wall, and 2) the fully developed boiling region where bubbles enter the free stream (27). In the partial-boiling region, the wall is sufficiently hot to cause a layer of superheated liquid to exist near the heated surface and cause bubble formation. The bulk fluid remains subcooled, hence the term "subcooled boiling." As shown in Fig. 5, the dominant forces on a bubble are buoyancy, drag, and surface tension. The film of bubbles attached to the wall in subcooled boiling can be considered to be a film of vapor instead of individual bubbles. At the interface the surface is rough, or dimpled. The vapor is moving relative to the liquid and is considered to be a continuous phase. Thus, the vapor can be pictured as flowing around a roughened liquid core with a diameter very similar to the diameter of the channel. By performing a steady-state force balance on the channel and using the vapor momentum equation and the

TABLE 4
INTERFACIAL DRAG MODELS

Flow Regime	Characteristic Dimensions or Number	Void Fraction or	Drag Coefficient Friction Factor
Subcooled	$d_b=2\left[\frac{\sigma}{g\Delta\rho}\right]^{\frac{1}{2}}$	$\alpha_w=\frac{\Pi}{6}\frac{Y_b}{D_h}$	$C_{i,sb}=\frac{2\rho g}{D_h}\left[1.14-2.0\log_{10}\left(\frac{\varepsilon}{D_h}\right)\right]^{-2}$
Nucleate	$Y_b=C\left[\frac{\sigma D_h}{\frac{0.01197G^2}{2\rho_l}}\right]^{\frac{1}{2}}$		$C_{i,fr}=\alpha\frac{\alpha_{fr}}{d_b}f(\alpha)\,\zeta_\rho\frac{(C_1V_v-C_0V_l)^2}{(V_v-V_l)^2}$
	$\varepsilon=0.01Y_b$	$\alpha_w=\frac{\Pi}{6}\frac{Y_b}{D_h}[0.2(T_{sat}-T_l)]$, $T_l\geq T_{sat}$	
Boiling		$\alpha_{fr}=\alpha_g-\alpha_w$	$f(\alpha)=\left[\frac{1+17.67\,(1-\alpha_{fr})^{1.3}}{18.76\,(1-\alpha_{fr})^{1.5}}\right]^2$

if $T_l \geq T_{sat}$

if $\alpha_w > 0$ $\quad C_{i,sb}=\frac{C_{i,sb}V_l^2+0.00175\,C_{i,fr}V_r^2}{(\alpha_w+\alpha_{fr})\,V_r^2}$

if $\alpha_w \leq 0$ $\quad C_{i,sb}=\frac{0.00175\,C_{i,fr}}{\alpha_{fr}}$

if $T_l < T_{sat}$ $\quad C_{i,sb}=\frac{2\rho g}{D_h}\left[1.14-2.0\log_{10}\left(\frac{\varepsilon}{D_h}\right)\right]^{-2}$

Smooth IAF

$$Re_g = \frac{(D_h - D_c)\alpha_g \rho_v V_v}{2 \mu_v}$$

$$f_{i,sm} = \frac{16}{Re_g}$$

or $f_{i,sm} = 0.079 Re_g^{-0.25}$

$$C_{i,sm} = 2\zeta_g f_{i,sm} \frac{(1-\alpha_g)^{1/2}}{D_h}$$

$$\varepsilon = 80 \times 1.838 \left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]^{1/2} N_{\mu g}^{1/3}$$

$$C_{i,rw} = 2\rho_g f_{i,rw} \frac{(1-\alpha_g)^{1/2}}{D_h}$$

Rough-Wavy IAF

if $\varepsilon/D_h < 1$

$$f_{rw} = \left[1.14 - 2.0 \log_{10}\left(\frac{\varepsilon}{D_h}\right)\right]^{-2},$$

if $\varepsilon/D_h > 1$

$$f_{rw} = 0.7694$$

$$d_d = 1.838 \left[\frac{\sigma}{g(\rho_l - \rho_v)}\right]^{1/2} N_{\mu g}^{1/3}$$

$$C_{i,dd} = \frac{0.75\, \alpha_{dd}\, \rho_g C_d}{d_d}$$

$$N_{\mu g} = \left[\frac{\mu_g}{\rho_g \sigma \sqrt{\frac{\sigma}{g(\rho_l - \rho_g)}}}\right]^{1/2}$$

$$C_d = \frac{24(1 + 0.1\, Re_d^{0.75})}{Re_d}$$

$$Re_d = \frac{\rho_v\, d_d (V_v - V_d)(1-\alpha_{dd})^{2.5}}{\mu_v}$$

$$C_{i,f} = \frac{2\, \rho_g f_{i,f}}{D_h}$$

Highly Dispersed

$$V_d = V_v - 2.462 \left[\frac{(\rho_l - \rho_v)\, g\, d_d}{2\, \rho_v}\right]^{1/2}$$

$$f_{i,f} = 0.005\,(1 + 75\, \alpha_f)$$

$$\delta_f = \frac{0.0025\,\rho_l V_g^2}{g\,\rho_l \frac{0.75\,\rho_g V_g^2}{D_h}} \qquad C_{i,df} = \frac{C_{i,dd}V_r^2 + C_{i,f}V_v^2}{\left[V_v - \frac{\alpha_{dd}V_d}{(1-\alpha_g)}\right]^2}$$

IAF

$$\alpha_f = \frac{Pe\,\delta_f\,F_u}{A}$$

$$\alpha_{lf} = 5\left[\frac{\alpha_{df} - \alpha_g}{\alpha_{df} - \alpha_{ag}}\right]^{0.5} F_u(1-\alpha_g)$$

$$\alpha_{dd} = 1 - \alpha_{lf} - \alpha_g$$

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definition of the shear stress, the interfacial drag coefficient for partial-subcooled boiling, $C_{i,sb}$ can be expressed as

$$C_{i,sb} = \frac{2\rho_g f_{sm}}{D_h} . \qquad (9)$$

The Colebrook friction factor for turbulent flow and a completely rough zone is used as f_{sm} in Eq. 9. The roughness parameter required by the Colebrook friction factor is estimated to be the time-average height of the growing bubbles (28). We assumed that the roughness, ε, is 1% of the bubble height, Y_b. As given by Collier (27), the bubble height can be found by assuming that it is proportional to the bubble radius and can be obtained from a force balance equation between the buoyancy, drag, and surface tension forces (see Ref. 27, p. 182).

When the liquid is saturated, superheated, or when fully developed sub-cooled nucleate boiling is occuring, bubbles also exist in the free stream. The interfacial drag coefficient due to bubbles in the free stream is needed. To obtain this quantity, the portion of the bubbles on the wall and those in the free stream must be defined (see Table 4). Assuming that the bubbles attached to the wall are packed in a square grid, Collier (27) indicated that the void fraction at the wall, α_w, can be expressed in terms of the bubble height and the hydraulic diameter. Once α_w is known, the void fraction of the bubbles traveling in the free stream, α_{fr}, can be found by substracting α_{fr} from the total void fraction, α_g.

The free-stream drag coefficient is obtained from the drag force expression given for bubble flows by Ishii (25) as

$$C_{i,fr} = \frac{\alpha_{fr}}{2\left[\frac{\sigma}{g\Delta\rho}\right]^{\frac{1}{2}}} \left[\frac{1 + 17.67\left(1 - \alpha_{fr}\right)^{1.3}}{18.76\left(1 - \alpha_{fr}\right)^{1.5}}\right]^2 \rho_l \frac{\left(C_1 V_v - C_0 V_l\right)^2}{\left(V_v - V_l\right)^2} . \quad (10)$$

The total interfacial drag coefficient for fully developed nucleate boiling regime is obtained by a momentum-weighted equation given by

$$C_{i,sb} = \frac{C_{i,sb} V_l^2 + 0.00175 C_{i,fr} V_r^2}{\left(\alpha_w + \alpha_{fr}\right) V_r^2} . \quad (11)$$

This interfacial drag model is used when the cell void fraction is less than 0.5.

For void fractions greater than 0.98, the interfacial drag coefficient is calculated by the interfacial drag model for the annular mist flow regime as given in Ref. 14. For void fractions greater than 0.5 but less than 0.98, a liquid-void-cubic weighting is used and is given by

$$C_{i,sb} = C_{i,sb}\, 0.125\,(1 - \alpha)^3 . \quad (12)$$

This type liquid-void dependence is suggested by Ishii for the slug regime. We apply it to the total drag coefficient as given Eq. 11 to maintain continuity. If $C_{i,sb}$ calculated by Eq. 11 is less than the value of the drag coefficient in annular mist flow, the annular mist flow interfacial drag coefficient is used.

B. Smooth IAF Interfacial Drag Model

Using a steady-state force balance, the vapor momentum equation, and the definition of the shear stress, the interfacial drag coefficient for smooth IAF is found to be (see Ref. 19 for details)

$$C_{i,sm} = 2\rho_g f_{i,sm} \frac{(1 - \alpha_g)^{1/2}}{D_h} . \quad (13)$$

In this region, the interface is assumed to be smooth. Therefore, a simple smooth-tube correlation for the interfacial friction factor (28) is used for both laminar and turbulent flows.

The interfacial drag coefficient is further redefined based upon the cell void fraction, as illustrated in Fig. 4. If the cell resides in the low void fraction region, no adjustment is made. If the cell is located in the high void fraction region, the interfacial drag coefficient is assumed to be equal to that of the highly dispersed flow. In the transition between the high and low void fraction regions, the following weighting based upon the void fraction is used:

$$C_{i,sm} = C_{i,sm} + \left(C_{i,df} - C_{i,sm}\right) (4.348\,\alpha - 3.261)^{0.7} \quad (14)$$

C. Rough-Wavy IAF Interfacial Drag Model

This regime is similar to the smooth IAF case. However, the interfacial friction is now increased due to the presence of waves and the shearing of droplets from the wave crests. The formulation of the interfacial drag coefficient for rough-wavy IAF is similar to that of smooth IAF. To express the friction factor, $f_{i,rw}$, the turbulent rough-pipe correlation suggested by Colebrook (28) is used. The wavy vapor-liquid interface is considered to represent the pipe roughness. The roughness in Colebrook's friction factor is assumed to be proportional to the diameter of liquid droplets entrained from the wavy interface. The diameter of the liquid droplets is calculated using Ishii's equation for small droplets (25) with a proportionality constant of 80. If $\varepsilon / D_h > 1$, then a constant of 0.77 is used for $f_{i,rw}$. The interfacial drag coefficient is further weighted based upon the cell void fraction as done before for the smooth IAF regime (replacing $C_{i,sm}$ with $C_{i,rw}$ in Eq. 14).

D. Agitated IAF Interfacial Drag Model

This region is characterized by large liquid slugs of a diameter sometimes approaching the liquid core diameter in the rough-wavy region. The breakup of the slugs into smaller pieces and droplets eventually occurs. The region is quite chaotic. It should be similar to the churn-turbulent regime in adiabatic flows that is used to

characterize the transition region between slug flow and annular mist. The interfacial drag should again be dominated by the mechanism of droplet entrainment and breakup of the inverted annular liquid core. For this region, the same correlation as applied to the rough-wavy region is used.

E. Highly Dispersed Flow Interfacial Drag Model

The dispersed region is composed of droplets flowing up the channel. The droplet diameter is calculated using Ishii's dispersed flow droplet diameter correlation (25) given in terms of the Laplace and viscosity numbers. Ishii and Chawla (29) also gave the interfacial drag coefficient obtained by performing a separate force balance on the droplet that is in highly dispersed flow. Their expression requires the droplet velocity. An estimate of the droplet velocity, obtained from a separate momentum balance, is given by

$$V_d = V_v - 2.462 \left[\frac{(\rho_l - \rho_v) g d_d}{2 \rho_v} \right]^{1/2}, \tag{15}$$

assuming that the droplet drag coefficient is 0.44 (Ref. 29).

In numerous experimental studies and in all power reactor geometries, some of the structural surfaces are unheated due to the presence of control rods and structures at the periphery of the core. Thus, a liquid film may establish itself on these cold surfaces, adding significantly to the overall liquid fraction at the top of the core. A typical pressurized water reactor (PWR) core contains 45,548 heated rods of 10.73-mm diam and 3860 control rods of 13.8-mm diam. Thus, the unheated surface of the control rods alone accounts for approximately 10% of the total rod heat-transfer surface area. From geometric considerations for typical reactor hardware, one can calculate the local volumetric liquid fraction to vary from 0.6% to 11% for liquid film thicknesses varying from 0.2 to 3.0 μm. Thus, the liquid contained in this film cannot be neglected, since it can represent a significant portion of the total liquid fraction in the high-void region.

Williams (30) has shown that the vapor velocities generated in the core during reflood are at about the correct value to cause countercurrent flow limitation. Thus, the drag and gravity forces balance, creating a "hanging" film condition.

Figure 6 depicts this condition in a top view. The overall drag coefficient must consider both the droplets and the film. It is not possible to do this directly with the two-fluid approximation since only one liquid field is assumed.

Pasamehmetoglu (see Ref. 14) derived the amount of liquid which can hang on the wall using a force balance on a differential liquid element that includes the forces due to gravity and interfacial shear. The film void fraction can then be obtained from geometrical consideration and is given by $\alpha_{lf} = Pe\delta_{lf}F_u/A$. The film void fraction, α_{lf}, can also be limited by the total liquid void fraction, α_l, available to be deposited on the unheated surface area. If the liquid and the unheated surface area are homogeneously distributed within a control volume, the amount of liquid might be considered to be $F_u(1-\alpha_g)$. However, with cross flow between subchannels and the ability of unheated surfaces to collect liquid and hold it, the amount of liquid "hanging" on an unheated wall may become greater than $F_u(1-\alpha_g)$. This effect is included in the following relationship, which has been developed using results from CCTF run 14:

$$\alpha_{lf} = 5\left(\frac{\alpha_g - \alpha_{ag}}{\alpha_{df} - \alpha_{ag}}\right)^{0.5} F_u\left(1 - \alpha_g\right) . \tag{16}$$

The first part of Eq. 16 is a weighting factor that allows the limiting amount of liquid deposited on the cold wall to be an amount greater than the homogeneous fraction of the unheated surface area. We assumed that the weighting becomes unimportant when the void fraction is less than α_{ag}. The weighting also allows the maximum fraction of the liquid to be accumulated in the dispersed flow regime.

The droplet void fraction becomes $\alpha_{dd}=1 - \alpha_{lf} - \alpha_g$. Using the modified Wallis relation (31) for the interfacial friction factor between the liquid film and the vapor, the interfacial drag on the film can be estimated by

$$C_{i,lf} = \frac{2\,\alpha_g\left[0.005\left(1 + 75\,\alpha_{lf}\right)\right]}{D_h} . \tag{17}$$

As will be discussed in the companion paper to this paper, $C_{i,lf}$ and $C_{i,dd}$ were multiplied by 0.5 and 0.15, respectively, to predict available pressure drop data.

The total weighted, averaged interfacial drag coefficient for this regime is expressed by an average momentum balance equation between the liquid and gas.

Assuming that the film velocity is small compared to the vapor velocity and that the total liquid velocity can be expressed in terms of weighted droplet velocity, the interfacial drag coefficient becomes

$$C_{i,df} = \frac{C_{i,dd} V_r^2 + C_{i,lf} V_v^2}{\left[V_v - \frac{\alpha_{dd} V_d}{(1 - \alpha_g)} \right]^2} \tag{18}$$

F. Dispersed (Post-Agitated) Flow Interfacial Drag Model

This region is located downstream of the agitated IAF and extends until the highly dispersed flow regime, where the droplets become smaller. In this region, the interfacial drag coefficient is obtained by performing a void fraction weighting using the interfacial drag coefficients in the rough-wavy and highly dispersed flow regimes. The interfacial drag coefficient is calculated by the following:

$$C_{i,pa} = 1.5\, C_{i,rw} + \left[C_{i,df} - 1.5\, C_{i,rw} \right] \left[\frac{\alpha - \alpha_{ag}}{\alpha_{df} - \alpha_{ag}} \right]^{0.5} ,$$

if

$$C_{i,df} < 1.5\, C_{i,rw}$$

or (19)

$$C_{i,pa} = 1.5\, C_{i,rw} ,$$

if

$$C_{i,df} \geq 1.5\, C_{i,rw} .$$

In Eq. 19, α_{ag} is selected to be the minimum of calculated α_{ag} and the upper limit imposed on the agitated regime as noted in Table 1.

G. Combinations of the Individual Drag Models for a Control Volume

When multiple regimes occur in a given hydro-cell, the interfacial drag models developed for each region of the core during reflood are length averaged to determine the cell average drag. The overall interfacial drag coefficient for such a situation is given by

$$C_{i,cell} = W_{sb}\, C_{i,sb} + 2.5W_{sm}\, C_{i,sm} + 1.5W_{rw}\, C_{i,rw} + W_{pa}\, C_{i,pa} + W_{df}\, C_{i,df} \quad . \tag{20}$$

For each IAF regime, a linear weighting is used in the above equation (W_{sb}, W_{sm}, W_{rw}, W_{pa}, W_{df}) which is based upon axial distance. Details of this weighting are available in Ref. 19.

VI. SUMMARY

The reflood model discussed above has been developed and implemented into the TRAC computer code to improve our ability to predict post-CHF transient convective heat transfer in a reflooding core as well as in other transients experiencing post-CHF conditions. The model was built around the flow-regime map of Ishii and his coworkers but required some modification to account for transient situations. The necessary closure relations, which include interfacial heat transfer, interfacial drag, and wall heat transfer, were then discussed relative to the flow map. Wherever possible, correlations known to apply to those regimes were used. Frequently, however, the original correlations could not be applied directly but had to be modified. For those cases, we tried to use the "kernel" or "functional" dependence of the original correlation and modify only its magnitude by use of a multiplier. When no correlations were available for given regions, we tried to define the bounding regions and use a weighting function between the known regimes to represent the unknown quantities. While we would like to say that the need to employ either of these modification methods was minimal, the opposite is true. We still find it necessary to infer a number of the closure relationships needed.

In terms of future experiments and model development efforts, we would identify the following elements as needing significant work:

a) Transient effects within the flow-regime map should be determined. These effects occur in many integral test facilities even under forced feed conditions.

b) The current formulation of the axial-distance-dependent transition boiling model is quite empirical. We did not attempt to develop a detailed mechanistic model due to the lack of the necessary experimental data and the limitations imposed by our program objectives. However, in this paper, we did attempt to find the dimensionless numbers and a functional form that properly describe the phenomenon in the hope that this information can be useful for experimentalists as well as other researchers. The determination of the coefficient B used in the transition boiling model obviously needs further work. Work in determining B might be adapted from either "quench front velocity" models or "quench temperature" models. Certainly all these factors are interrelated.

c) Our current estimation of heat-transfer enhancement due to near-wall liquid effects is also empirical. Again, we did not attempt to develop a detailed mechanistic model due to the lack of necessary experimental data and the limitations imposed by our program objectives. It has been apparent that detailed modeling in this area has been needed for a number of years; yet our ability to separate the phasic components has kept modeling efforts in this region very limited.

d) Interfacial heat transfer resulting in phase change is an area where large uncertainties exist. Not only are applicable HTCs limited, but models and data for the interfacial area are also minimal.

e) Ishii's work on modeling interfacial drag coefficients probably makes this the best-understood set of closure relationships. Yet it is lacking much in terms of specifying how to model the oscillating flows which frequently occur, even under simple forced reflooding conditions.

f) The wet unheated wall model needs additional work. It is a difficult process to represent using only a two-field code, since the process itself is a three-field phenomenon.

g) The grid spacer model is a very simple model which needs further improvement.

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Figure Captions

Fig. 1: Sketch of IAF regimes; a) Low-quality CHF; b) High-quality CHF.

Fig. 2: Typical axial wall heat flux profile for IAF.

Fig. 3: Illustration of HTC selection logic.

Fig. 4: The interfacial drag coefficient model selection logic in the IAF regime-void fraction plane.

Fig. 5: Bubble attached to wall in subcooled boiling (from Ref. 27).

Fig. 6: Proposed scheme for the dispersed region in reactor geometry.

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