Mechanistic CHF Modeling for Natural Circulation Applications in SMR

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Abstract

A mechanistic critical heat flux correlation has been developed for a wide range of operating conditions which include low mass fluxes of 540 to 890 kg/m²-s, high pressures of 12 to 13 MPa, and critical heat fluxes of 835 to 1,100 kW/m². Eleven experimental data points have been collected over these conditions to inform the development of the model using bundle geometry. Errors of within 15% have been obtained with the proposed model for predicting the critical heat flux value, location, and critical pin power for a non-uniform heat flux applied to a 2x2 bundle configuration.

Keywords: Japan-US seminar, two-phase, critical heat flux, CHF, boiling crisis

1. Introduction

The critical heat flux (CHF) is one of the key physical phenomena that limits the allowable linear power for a nuclear reactor core design under steady-state operating conditions and can be the limiting condition under transient conditions. The advanced light water reactor designs being considered for modular reactor plants normally involve core flow rates (either under natural circulation or forced circulation) at mass fluxes far below current reactor designs; i.e., estimated to be 100-1000 kg/m²-sec. At such flow conditions, the current CHF data is extremely sparse, and the available CHF correlations need to be scrutinized for applicability to small modular reactor (SMR) designs. Based on an open literature review, one finds that practically no data exists for rod bundles and only a sparse database for heated tubes (Chang et al. [1], Chen et al.[2], Mishima [3]). For these advanced designs to address the needed safety performance some fundamental understanding must be obtained for CHF under these conditions. Focus here will be given to the development of a mechanistic based critical heat flux correlation based on mass, momentum, and energy conservation equations with the goal to be applicable to flow velocities present in the proposed SMR designs.

2. Background

Various approaches have been employed in an effort to predict the onset of the critical heat flux, which can be categorized in the form of empirical correlations, look-up tables or mechanistic/semi-empirical models. Empirical correlations have been historically the focus of

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the development of CHF models for a specific design and prototypic flow conditions without modeling the physical mechanisms that occur during the boiling process. An example of this type of correlation is that developed by the Electric Power Research Institute (EPRI) which utilizes over 11,000 data points (Fighetti and Reddy [4-7]). Look-up tables provide the CHF value in terms of determined key parameters in tabular format typically including pressure, mass flux, and quality. The values presented are normalized to a single geometric configuration. Once the CHF value for the normalized condition is obtained, corrections are then applied for the specific geometry. Conditions that lie between table values are determined through interpolation. The most popular look-up tables developed are those by Groeneveld et al. [8-10]. These are utilized in many thermal-hydraulic system codes such as RELAP5, RELAP5-3D, and TRACE. They offer solutions that are not computationally expensive and provide the widest operating range of coverage. The applicability of look-tables, however, like empirical correlations, must be scrutinized carefully for the intended application. Look-up tables do not provide responses to actual physical mechanisms and thus trends may be missed when applied to conditions (e.g. geometric and flow fields/regimes) not represented by the table.

Mechanistic based CHF correlations are developed generally by postulating a trigger mechanism for the onset of the CHF condition and then attempts are made to model the phenomena that occur during this process. Balance equations are formulated to capture the phenomena and the trigger mechanism. The advantage of such a method is that the applicability of the model can be potentially expanded outside of the originally tested subset of experimental data as long as the mechanisms are appropriately captured. Two fundamental approaches have been taken in developing a DNB type CHF model as summarized by Katto [11]; they are the hydrodynamic model and the macro-layer dryout model. The hydrodynamic model postulates that a stability limit is reached with the ejection of vapor from the surface and the incoming reflood of liquid. Here the near wall fluid conditions are generally neglected. The macro-layer dryout model postulates that CHF occurs when the underlying liquid layer below a vapor mass is evaporated away. Examples of those two approaches are summarized below.

Vapor injection in the bulk flow field from the wall region reduces the liquid velocity gradient near the wall causing the liquid to separate from the wall. This ‘blow-off’ of liquid near the wall results in a transition from nucleate to film boiling when the vapor flow from the surface reaches the critical velocity. This model was proposed by Kutateladze [12] and later confirmed by Zuber [13]. The near-wall bubble crowding based models postulate that the turbulent interchange between the bubbly layer and the core is the limiting mechanism for the trigger of the CHF condition. A balance between the outward vapor flow and the inward liquid flow is used to determine the void fraction in the bubbly sub-layer. The CHF is then predicted at a critical void fraction in the bubbly sublayer varying between 0.3 and 0.95 depending on the degree of subcooling, mass fluxes, and pressure. This model was proposed by Weisman and Pei [14] and Weisman and Ying [15]. The sublayer dryout models assume that a vapor blanket isolates the heated surface from being cooled from the bulk liquid leaving an isolated thin liquid sublayer near the wall. It then assumes that the CHF trigger occurs when the evaporation rate of

the liquid sublayer exceeds the liquid layer replenishment from the core region. This model is based on the improvement of the Weisman and Pei model by Lee and Mudawwar [16]. A summary of CHF correlations proposed by previous authors is provided in Table 1 with the mass flux and pressure range of which the model was developed.

Table 1. Critical heat flux model capability overview.

Model	Mass Flux		Pressure	
	(kg/m ² -s)		(Mpa)	
	Low	High	Low	High
Kutateladze (1948) [12]	atmospheric pool boiling			
Zuber (1959) [13]				
Tong (1968) [17]	1356.0	6781.0	6.89	13.79
Rogers et al.(1982) [18]	180.0	1200.0	0.16	0.16
EPRI (1982) [4-7]	271.0	5560.0	1.38	16.55
Weisman & Pei (1983) [14]	972.0	13611.0	2.00	20.20
Lee & Mudawwar (1988) [16]	1350.0	5200.0	6.89	15.76
Weber & Johannsen (1990a) [19]	25.0	200.0	0.11	1.00
Weber & Johannsen (1990b) [20]	10.8	301.4	0.11	1.20
Katto (1992) [21]	350.0	40600.0	0.10	20.00
Huang et al. (1993) [22]	25.0	500.0	0.10	1.20
Oh & Englert (1993) [23]	30.0	80.0	0.02	0.09
Liu et al. (2000) [24]	900.0	90000.0	0.10	19.25
Kwon et al. (2001) [25]	1000.0	18000.0	3.00	19.00
Kureta & Akimoto (2002) [26]	1000.0	2000.0	0.10	0.10
Groeneveld (1986) [8]	0.0	8000.0	0.10	20.00
Groeneveld (1996) [9]				
Groeneveld (2007) [10]				
Chung & No (2007) [27]	atmospheric pool boiling			
Wright et al. (2008) [28]	9.5	39	0.089	1.15

In addition to the proposed models and postulated mechanisms outlined here and listed in Table 1, much work has been done in effort to predict the onset of the critical heat flux condition in terms of empirical and look-up table CHF relations. Through a thorough survey for the various CHF relations in literature, in general, not many validated correlations exist for the low mass flux conditions (<1000 kg-m²/s) and the intermediate to higher pressure conditions. This demonstrates an area of need for further development.

It is apparent that many models exist in literature that aim to predict the onset of CHF. Many of which are developed for specific geometric and fluid conditions. Hall and Mudawwar [29, 30] compiled an assessment of CHF various databases and about 100 CHF correlations. When correlations were assessed against their applicable range root mean square (RMS) values from 6.4% to 662% were obtained with the total mean RMS error of 46%. When assessed outside their

applicable range the average RMS error of the tested correlations was >500%. This expresses the limits and lack of potential broad applicability of many developed models forcing one to express great caution when applying relations to a new design with differing geometry and operating conditions. It is strongly believed that the development of a mechanistic based critical heat correlation will mitigate this issue and be more suited to allow for the development of a more broadly applicable correlation by means of modeling the surface phenomena through conventional mass, momentum, and energy balance equations.

In this investigation, using the data base produced by Corradini and Greenwood [31], a mechanistic critical heat flux correlation is developed for a range of operating conditions which include low mass fluxes of 540 to 890 kg/m²-s, at pressures from 12 to 13 MPa, and critical heat fluxes of 835 to 1,100 kW/m². Eleven experimental data points are used over these conditions to inform the development of the model using bundle geometry. Comparisons with relevant data are carried out.

3. Summary of Experimental Investigations

The University of Wisconsin has performed a set of tests that have aided in the correlation development efforts for the ranges of interest (Corradini and Greenwood [31]). A high-pressure forced flow test facility was constructed as shown in Figure 1. With these tests, information related to the CHF location, value and critical pin power were measured. Four heater rods, each capable of 100 kW max power, were installed in the 2.675 cm × 2.675 cm square test section. The reported experimental CHF data spans a system pressure ranging from 12.1 – 13.0 MPa and a total power applied to the test section of 160 – 225 kW. Experimental conditions are provided in Table 2. The active length of each heater rod is 200 cm with a diameter of 0.95 cm. The water mass flow rate through the test section varies from 540 kg/m²s to 890 kg/m²s. The rod power follows a cosine profile as provided in Figure 1.

Figure 1. University of Wisconsin test facility loop (left); test bundle configuration (right).

Table 2. Experimental data from Corradini and Greenwood (2014).

Test	Inlet Temperature (°C)	Pressure (MPa)	Mass Flux (kg/m ² -s)	Critical Rod Power (kW)	Critical Heat Flux (kW/m ²)	Normalized CHF Location from Inlet
1	254.33	12.55	585.68	46.06	856.70	0.731
2	251.19	12.3	740.34	51.03	950.87	0.731
3	250.61	12.37	740.97	52.00	968.44	0.731
4	251.05	12.11	889.43	55.98	1042.03	0.731
5	251.17	12.11	889.61	55.98	1041.97	0.731
6	275.29	13.02	542.19	39.98	861.90	0.667
7	273.41	12.90	547.46	40.67	877.82	0.667
8	271.56	12.47	701.66	45.00	837.67	0.731
9	271.12	12.47	702.05	45.00	969.24	0.667
10	265.52	12.16	860.17	50.01	1077.60	0.667
11	269.14	12.20	852.36	49.00	1056.04	0.667

4. Mechanistic CHF Model Development

The determining of a limiting surface heat flux for flow boiling involves the quantification of the separate heat transfer components that make up the total heat exchange. Focus here is given to the phenomena that occur very near the surface relevant to the scale of the bubble size, by relating the forces acting on a bubble growing on a heated surface within a flow field. The limiting heat removal point (the critical heat flux) is hypothesized to occur at the highest attainable boiling frequency at a nucleation site for a given nucleation site density. The highest attainable boiling frequency implies that at the same time a bubble departs from a nucleation site the next bubble immediately begins to grow. It is noted that this model does not depend on the coalescence of departed bubbles away from the surface; rather it focuses on the near-wall phenomenon. The far-wall phenomena (e.g. flow regime characteristics) are postulated to occur as a result of the near-wall behavior.

The total surface heat flux can be partitioned into the convective heat flux ($q''_{1\phi}$), evaporative (interfacial) heat flux (q''_l), transient conduction heat flux (q''_{trans}), and heat flux in the bubble sliding phase (q''_{slide}), as depicted in Figure 2.

Figure 2. Boiling heat transfer mechanisms at a heated surface.

It is assumed that the heat flux at the surface is high enough (due to the CHF condition) such that the bubble generation rate is continuous at a given nucleation site. This phenomenon was also considered by Zuber [13]. As the surface heat flux increases the number of activated sites increase up to a point. Additional increases in heat flux beyond this point only increases the boiling frequency (not the number of activated sites) until a continuous vapor column is formed. Furthermore, as the heat flux increases the contact time also decreases. Due to the high heat surface flux, it is assumed that the bubble departure from a cavity and the bubble lift-off from the surface are coincident (i.e. the bubble does not slide on the heated surface). This is mainly due to the high evaporative force of the bubble growth causing the bubble to be ejected away from the surface, comparing to the relative slow bulk fluid motion (shear lift force) in the low mass flux cases.

With these assumptions liquid is not allowed to occupy the region of the previous departed bubble nor is there a displacement of liquid along the surface where boiling does not occur due to bubble sliding. Thus, the transient conduction heat flux mechanism at the nucleation site and heat removal from sliding of a bubble along the surface are neglected and the total heat flux is reduced to Equation (1).

$$q''_{total} = q''_{1\phi} + q''_I. \quad (1)$$

Relations describing the evaporative interfacial energy transfer and the single phase heat transfer are developed in the following sections.

Bubble Departure Criterion

An extensive review of the forces acting on a growing bubble are provided by Thorncroft [32] for various boiling configurations. A similar force balance as that employed here was performed by Kandlikar [33]. For a vertically aligned flow channel the assumed forces acting perpendicular to a heated surface are the surface tension force, evaporative thrust force, and bubble shear lift force, outlined in Figure 3.

Figure 3. Bubble perpendicular force balance on a vertically aligned heated surface.

For this analysis only the forces perpendicular to the surface are assumed to participate in the departure of the bubble. The buoyancy force is neglected since it is assumed to be negligible compared to the evaporative thrust force near the CHF condition. The virtual mass force is also omitted at the moment of bubble departure as there is no acceleration to the surrounding liquid. The horizontal forces are assumed to be equal and opposite at the moment of bubble departure. The forces parallel to the heated surface may come into play with higher mass fluxes because under these conditions the dynamic and static contact angles may play a bigger role in the overall bubble force balance. At the moment of bubble departure the force that drives bubble away from the surface exceeds the holding force. The bubble thrust force (F_{evap}) due to phase change and the lift force (F_{sL}) propel a bubble away from the surface, whereas the surface tension force (F_{σ}) holds bubble to the cavity size. With the stated assumptions and simplifications, the bubble departure criterion at CHF is given by Equation (2).

$$F_{\sigma,y} \leq F_{evap} + F_{sL}. \quad (2)$$

The evaporative thrust can be obtained from the bubble growth rate due to the interfacial mass transfer across the bubble interface. This evaporation thrust is proportional to the heat flux and the bubble size (interfacial area). Due to symmetry and neglecting of advancing versus receding contacting angles of the bubble, the growth forces in the circumferential directions parallel to the heated surface negate each other. In the direction perpendicular to the surface there is a thrust force projected above and away from the interface of the bubble due to mass exchange and expansion across the bubble interface area. For a bubble with mass m_b , interface velocity $v_{b,i}$, and an interface heat flux q''_i , the perpendicular thrust force is written as shown in Equation (3).

$$F_{evap} = v_{b,i} \frac{dm_b}{dt} = \left(\frac{q_i''}{h_{fg}} \right)^2 \frac{\pi(a^2 + H_b^2)}{\rho_v}, \quad (3)$$

where a is the vapor-solid contact area radius and H_b is the bubble height from the surface to the bubble cap, both of which vary with time as the bubble grows.

The bubble shear lift force results from the entrainment phenomena on a bubble from the bulk fluid movement parallel to the surface. The result of adding this force is to aid in bubble removal from the surface and will thus result in a decrease in the maximum heat flux required to remove a bubble from the surface. With the increase of this force the evaporative potential will decrease but there will be a counter increase in the convective heat flux. The lift force of a bubble with radius r_b , as derived by Klausner et al. [34], is expressed as:

$$F_{sL} = 0.5C_L \rho_l u_l (y')^2 \pi r_b^2, \quad (4)$$

where y' is the bubble center of mass distance from the wall approximated here as $y' \approx H_b / 2$. The bubbles are assumed to be of a small size, thus the distance from the center of the bubble to wall is assumed to be small with $y^+ \leq 5$, and the fluid velocity profile has a linear distribution, given by:

$$u_l(y) = \frac{\tau_w}{\mu_l} y \quad (5)$$

$$\tau_w = \frac{\lambda}{8} \rho_l j_l^2, \quad \lambda = \begin{cases} \frac{64}{Re}, & Re < 2320 \\ \frac{0.3164}{Re^{0.25}}, & 4 \times 10^3 < Re < 10^5 \\ 0.0032 + 0.221 Re^{-0.237}, & 10^5 < Re < 2 \times 10^6 \end{cases} \quad (6)$$

The corresponding lift coefficient, with the assumption of viscous sublayer, can be simplified from Klausner et al. [34]:

$$C_L = 3.877 [Re_b^{-2} + 0.014003]^{-\frac{1}{4}}. \quad (7)$$

The surface tension force describes the force along the bubble interface contact line with the surface. This forces works to keep the bubble interface attached to the surface. The force acting normal to the heated surface with a contact angle θ is given by Equation (8).

$$F_\sigma = 2\pi\sigma a \sin(\theta) \quad (8)$$

Combining the results of all the individual forces derived gives a net force balance. The bubble departure criterion is readily expressed by Equation (9).

$$2\pi\sigma a \sin(\theta) \leq \left(\frac{q_i''}{h_{fg}} \right)^2 \frac{\pi(a^2 + H_b^2)}{\rho_v} + \frac{1}{2} C_L \rho_l u_l^2 |_{H/2} \pi r_b^2 \quad (9)$$

Solving for the interfacial heat flux (q_i''), the limiting evaporative component of the heat flux from a single bubble can be expressed as:

$$q_i'' = \left(\frac{2\pi\sigma a \sin(\theta) - \frac{1}{2} C_L \rho_l (u_l|_{(H_b/2)})^2 \pi r_b^2}{\pi (a^2 + H_b^2) / (\rho_v h_{fg}^2)} \right)^{1/2} \quad (10)$$

To quantify the total heat flux in Equation (1) at the CHF condition, the convective heat transfer from the heated surface without an activated cavity should be addressed. The fractional boiling area (f_{boil}) is determined from the nucleation site area density (N_a), the average bubble size ($\bar{A}_b$), and the total heated area (A_h).

$$f_{boil} = N_a A_h (\bar{A}_b / A_h) = N_a \bar{A}_b \quad (11)$$

The total contribution to the convective heat flux is then given by:

$$f_{conv} = 1 - f_{boil} \quad (12)$$

With the standard Dittus-Boelter single-phase convective heat transfer coefficient (h_{conv}), the total wall heat flux at CHF can then be expressed as:

$$q_{CHF}'' = \bar{h}_{conv} (T_w - T_{bulk}) (1 - f_{boil}) + \left(\frac{2\pi\sigma a \sin(\theta) - \frac{1}{2} C_L \rho_l (u_l|_{(H_b/2)})^2 \pi r_b^2}{\pi (a^2 + H_b^2) / (\rho_v h_{fg}^2)} \right)^{1/2} f_{boil} \quad (13)$$

This equation will give the maximum potential heat transfer for flow boiling and will be compared to experimental work in the following section after closure relations are defined.

Closure Relations

The model developed is fundamentally mechanistic based; nevertheless, a set of constitutive relations are necessary for closure of the model. It is noted that the selection of a specific closure relations will impact the accuracy of the final results. It is therefore left to the reader to select more appropriate models (if available) based on judgment and experience. The relations chosen here have been shown to provide initial good agreement for the final desired results at the conditions investigated. The remaining parameters to be defined requiring closure are the nucleation site cavity size, bubble-surface contact angle, bubble-surface contact radius, bubble radius, wall superheat, nucleation site density, and convective heat flux.

The bubble contact radius is assumed to be equal to an effective cavity size, and as soon the bubble emerges from the cavity lift-off of the bubble from the surface occurs due to the large heat fluxes. The bubble radius (r_b) is taken to be an effective radius calculated by the effective contact radius where the bubble radius is geometrically calculated from the cavity size and

contact angle. The bubble radius is used in combination with the contact angle to determine the bubble height as:

$$H_b = r_b (1 + \cos(\theta)) \quad (14)$$

It is noted that a distribution of cavity sizes is expected with the total availability of cavities increasing with increasing wall superheat. This can be demonstrated by the use of the Clausius-Clapeyron relation. The net integrated number of nucleation sites increases with increasing wall superheat conditions. Nevertheless, the large cavities dominate the evaporation process at the critical heat flux point.

The wall superheat is defined as $(T_w - T_{sat})$ and can be approximated with the use of the Clausius-Clapeyron relationship. This method is employed since detailed surface temperature information as well as the surface properties are not known in experiments. As the system pressure increases, latent heat and surface tension decrease, and finer bubbles may be activated, depending on the surface material and physical configuration, which may need further scrutiny for the modeling of mechanisms behind the process.

The wall superheat represents the range of possible activation sizes and the maximum temperature a nucleating bubble can reach before heat transfer ceases. The thermal boundary layer presents another limit for bubble growth. If the bubble grows beyond the point in the thermal boundary where the boundary layer temperature is equal to the bubble internal temperature a secondary path for heat rejection exists. Heat at this point will be transferred from the bubble cap to the bulk. This will limit the bubble growth rate and size. To account for these two effects two independent variables are chosen to represent the effective cavity size in a functional form. These parameters are system pressure and inlet mass flow rate. It is noted that mass flow rate could be replaced by velocity, mass flux, Reynolds number, non-dimensional velocity, and etc.

Based on the data of the University of Wisconsin, a calibration was performed to determine the effective active nucleation cavity size at the CHF point. The following form was developed to relate the effective nucleation cavity size and the system pressure:

$$r_c [\text{microns}] = 2.151G^{-0.09458} - 0.005594P_{sys} [\text{bar}]^{-0.969} \quad (15)$$

The nucleation site density as a function of wall superheat was fit by Basu et al. [35] based on experimental data. Most data was correlated to within a 40% error band for nucleation boiling. The corresponding correlations are:

$$\begin{aligned} N_a &= 0.34(1 - \cos(\theta))\Delta T_w^{2.0}, & \Delta T_w < 15^\circ \text{C} \\ N_a &= 3.4 \times 10^{-5}(1 - \cos(\theta))\Delta T_w^{5.3}, & \Delta T_w \geq 15^\circ \text{C} \end{aligned} \quad (16)$$

The nucleation site density is used to determine the boiling fraction of the entire heated surface area.

With all those closure relations, the critical heat flux specified in Equation (13) as well as the CHF location can be calculated with the local flow conditions, using the simple Dittus-Boelter

convective heat transfer coefficient and a contact angle of 45° . For determining the critical heat flux as well as the CHF location, an algorithm using MATLAB© was developed based on the simple homogeneous equilibrium model for two-phase flow analysis. The results are compared to experimental data in the following section.

5. Model Evaluation

With specific channel geometry and operating conditions, including inlet pressure, inlet temperature, and the liquid mass flow rate, the calculation proceeds for a normalized heat flux profile and an initial guess of the pin power. From the known heat flux profile the one-dimensional energy equation is solved to determine the axially dependent fluid properties. At the moment the channel pressure is assumed constant and equal to the inlet defined pressure.

The effective active cavity size is then determined with the calibration curve, Equation (15), dependent on system pressure and mass flux. The wall temperature is then determined with Clausius-Clapeyron relation. With the heated surface temperature known the near wall thermal fluid properties are calculated. The wall superheat and contact angle determines the nucleation site density, which is then used to estimate the surface boiling heat flux fraction and the convective heat flux fraction. Thereafter, the individual terms of the force balance equation are then solved and the maximum boiling heat flux calculated along with the single phase convective heat flux. The boiling and convective heat fluxes are then partitioned based on the available boiling and convective fractional areas, leading to the initial prediction of the critical heat flux for each axial node, as shown in Figure 4, iteration 1.

From the minimum distance between the calculated critical heat flux and the initial guessed heat flux profile, the pin power is adjusted and the process is repeated. Convergence is achieved when the operating model heat flux curve meets the critical heat flux profile curve, as shown in iteration 2, 5 and 6 in Figure 4. The contact point of the curves presents the critical heat flux value, critical pin power, and the critical heat flux location for the specified conditions.

Figure 4. Convergence process for determine critical heat flux.

All the test data from University of Wisconsin were processed against the new model prediction, and the results are summarized in Figure 5 through Figure 7. As shown in Figure 5, excellent agreement exists between the experimental critical rod power and that predicted by the proposed model.

The proposed model allows for calculation of the critical heat flux location, where the pin power profile meets the calculated critical heat flux curve. Figure 6 shows the results of the critical heat flux locations. The model results are typically within +10% of the data, showing that the model tends to over-predict the critical heat flux location.

The local critical heat flux is shown for comparison in Figure 7. The model results are typically within 15% of the reported data showing that model tends to under-predict the critical heat flux values. This is consistent with the over-prediction of the CHF location.

Figure 5. Model critical pin power comparison to experimental data.

Figure 6. Model critical heat flux location comparison to experimental data.

Figure 7. Model critical heat flux value comparison to experimental data.

6. Conclusion and Future Effort

A mechanistic critical heat flux model has been developed and assessed against high-pressure data under various flow conditions. Although the current dataset evaluated lacked information in other pressure regions ($P < 12$ MPa); the CHF model allows for determination of the critical heat value, the critical heat flux location, and the critical pin power for a given set of power profiles and test conditions with satisfactory performance.

Future work should focus on key areas to improve and understand model performance. A parameter sensitivity analysis of input effects on the critical heat flux needs to be performed. This will provide insight on the areas of focus for model improvement. Based on the sensitivity results it will be determined if a more appropriate nucleation site size and density treatment

exists or is needed. The correlation test database should be expanded to include that of the 2006 Groeneveld look-up [10] and/or the EPRI database (Fighetti and Reddy [4-7]). This will help in broadening the coverage in other pressure ranges and provide insight to the applicable domain of the proposed model. A key dependence of the model is the treatment of the effective cavity size at CHF. This is treated as functional form (Equation (15)) which needs to be validated against experimental data near the CHF condition.

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