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TECHNICAL NOTES

A phenomenologically based prediction of the critical heat flux in channels containing an unheated wall

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INTRODUCTION

IT IS GENERALLY recognized that the critical heat flux (CHF) in flow boiling is initiated by different phenomena at high and low qualities. When the fluid is subcooled or at low qualities, the bubbles along the wall coalesce to form a vapor film when CHF occurs. It is only fairly recently that useful phenomenological models of this process have appeared. One of the more widely applicable of the phenomenologically based approaches is that due to Weisman and Pei [1] which has been shown to hold for a variety of fluids. This note considers the application of the aforementioned model to channels with one unheated wall.

Weisman and Pei's model [1] is based on the existence of a bubbly layer adjacent to the heated wall. The CHF, $q''_{\rm CHF}$, is assumed to occur when the bubbles in this layer become so crowded that they agglomerate into a continuous film. It is further assumed that the rate of liquid transport to the bubbly layer, as determined by the turbulent interchange at the outer edge of the bubbly layer, is the limiting mechanism. By using a simple mass balance over the bubbly layer, it is found that

$$q''_{\rm CHF}/(h_{\rm fg}G\Psi i_{\rm b}) = (x_2-x_1)\left(\frac{h_{\rm f}-h_{\rm ld}}{h_{\rm l}-h_{\rm ld}}\right) \qquad (1)$$

where Ψ is the factor representing portion of turbulent velocity fluctuations capable of reading the wall (dimensionless), G the mass flux (mass/(area × time)), $h_{\rm f}$, $h_{\rm l}$ the enthalpy of saturated liquid and bulk liquid, respectively (energy/mass), $h_{\rm ld}$ the enthalpy of bulk liquid at the point of bubble departure (energy/mass), $h_{\rm fg}$ the heat of vaporization (energy/mass), $i_{\rm b}$ the turbulent intensity (dimensionless), and x_2, x_1 the weight fractions of vapor in the bubbly layer and core regions, respectively (dimensionless).

The calculation details needed to determine each of the foregoing quantities have been provided previously [1–3].

The original work of Weisman and Pei was limited to CHF computation in round tubes where the mean void fraction was at or below 0.6. Weisman and Ying [2] showed that the procedure could also be used for rod bundles. They subsequently showed [3] that the approach could be extended to compute CHF values at mean void fractions up to 0.8. At void fractions above 0.6, an effective quality replaced x_1 in equation (1). This effective quality is lower than the average quality due to the accumulation of voids in the central region of the tube.

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CRITICAL HEAT FLUX IN ROD BUNDLE CHANNELS CONTAINING AN UNHEATED WALL

The application of Weisman and Pei's model to a rod bundle channel with an unheated wall, and an average void fraction below 0.6, was considered previously [4]. In applying the approach to rod bundles, the predictions were determined for a tube having the same equivalent diameter, $D_{\rm e}$, as the rod bundle channel. Since one rod is unheated, $q''_{\rm av}$, the average heat flux producing the same enthalpy rise in the round tube as in the rod bundle channel, is below the hot-rod heat flux. When the bulk fluid enthalpy and quality, x_1, were evaluated using $q''_{\rm av}$, predictions from equation (1) were unsatisfactory. Reinterpretation of the results of Weisman *et al.* [4] indicates satisfactory predictions were obtained when the enthalpy of the fluid adjacent to the heated rod was taken as higher than the average enthalpy. We may rewrite Weisman *et al.*'s result as [4]

$$(q''_{\rm CHF})_{\rm hot\,surface}/(h_{\rm fg}G\Psi i_{\rm b}) = \left(\frac{h_{\rm f}-h_{\rm ld}}{h_{\rm l}-h_{\rm ld}}\right)(x_2-x'_1) \qquad (2)$$

where x'_1 is an effective quality adjacent to the bubbly layer. This may be determined using an effective heat flux, $q''_{\rm eff}$, given by

$$q''_{\rm eff} = q''_{\rm hot\,surface}/[(1-K)+K(p/p_{\rm h})] \qquad (3)$$

where K is the parameter which depends on geometry and heat flux, and $p, p_{\rm h}$ the total and heated perimeters, respectively. In all cases $0 < K < 1$ thus indicating an enthalpy gradient across the channel.

CRITICAL HEAT FLUX IN ANNULI WITH ONE UNHEATED WALL

Based upon the results for rod bundle channels with an unheated rod, one might expect that, in an annulus with one heated wall, the enthalpy of the fluid adjacent to the heated wall would be higher than the average enthalpy. However, this appears to be contradicted by the results of Wittekind [5]. Wittekind applied Weisman and Pei's approach [1] to concentric annuli with one or both walls heated. The annuli considered simulated the nuclear fuel for Hanford's N reactor and therefore were placed horizontally and contained a series of spacer ribs. Wittekind found that his data with either the inner or outer wall being heated could be predicted by Weisman and Pei's approach by:

(1) computing the core quality, x_1, and the core liquid enthalpy, $h_{\rm l}$, using the actual heat input;

(2) multiplying the computed value for the turbulent intensity, $i_{\rm b}$, by a constant, C, set to obtain agreement for a particular geometry.

The first of these conditions is equivalent to using equations (2) and (3) with $K = 1.0$ and indicates no enthalpy gradient across the channel.

There may be some differences between horizontal and vertical flow at void fractions above 0.6. Wittekind [5] found that the model substantially underpredicted his experimental data taken at 103 bar when the quality exceeded approximately 0.13 (homogeneous void fraction $\simeq 0.65$).

The present study applies Weisman and Pei's CHF model to data at low qualities in simple, vertically placed, concentric annuli without the presence of spacer ribs. The study attempts to determine: (a) whether an enthalpy gradient needs to be considered for this geometry; (b) the appropriate modifications to Weisman and Pei's model as a function of annulus geometry; and (c) whether use of the Ying and Weisman high void correction [3] is sufficient to eliminate the underprediction of CHF at high void fractions in vertical annuli.

The data considered included those compiled by Barnett [6] for water in annuli with a uniformly heated inner rod (unheated outer rod). In addition, the data obtained by Judd and Wilson [7] for water in uniformly heated annuli were also considered. This latter data set included data with either the inner rod or the outer rod heated. The 285 data points analyzed fell within the following range:

$68 \leqslant P \leqslant 155$ bar
$0.62 \leqslant L \leqslant 2.75$ m
$3.1 \times 10^6 \leqslant G \leqslant 2.8 \times 10^7$ kg m^{-2}h^{-1}
$0.46 \leqslant D_e \leqslant 1.27$ cm
$-0.1 \leqslant x_{exit} \leqslant 0.16$.

In the initial analysis of the data, it was assumed that the annulus could be treated as a round tube having the same equivalent diameter but that there might be an enthalpy gradient across the annulus. Equations (2) and (3) were therefore used with several values of K. The accuracy of the results obtained were measured by defining the ratio 'R' as

$$R = (\text{predicted CHF})/(\text{observed CHF}). \quad (4)$$

The best results were obtained with $K = 1.0$. This corresponds to Wittekind's [5] use of the average heat flux and indicates no significant enthalpy gradient across the annulus. However, the results were only moderately satisfactory. While the mean value of R, $\mu(R)$, was 0.99, the standard deviation of R, $\sigma(R)$, was 0.13.

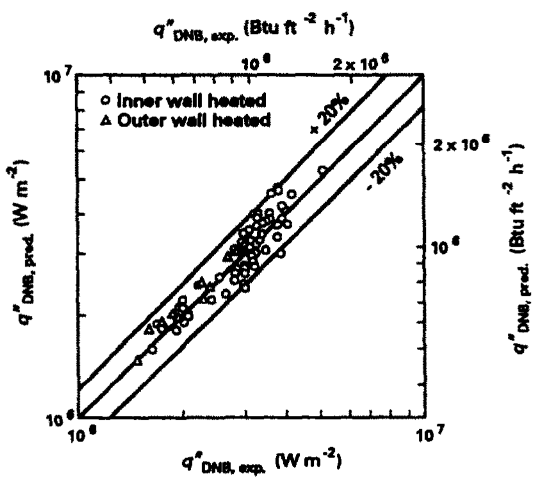

FIG. 1. Comparison of experimental CHF values with predictions obtained by recommended procedure.

Examination of the data showed that variation with annulus geometry was a major cause of the discrepancies between predictions and observations. When the ratio of outer diameter, D_o, to inner diameter, D_i, was 1.2, the predictions were 17% above observations. For $D_o/D_i > 2.0$, predictions were 16% below observations. This discrepancy is not surprising since the initial prediction procedure assumed the constitutive equation for turbulent intensity in round tubes, i_b, applies directly to annuli. This is almost certainly not the case. Reichardt's equation [8] for the variation of eddy diffusivity across an annular region depends on the diameter ratio. Since the eddy diffusivity at a given location is proportional to i_b, this implies that i_b is also dependent on D_o/D_i. However, when Reichardt's eddy diffusivity equation is applied to the prediction of single phase heat transfer [9], it is found that the heat transfer coefficient varies only with $(D_o/D_i)^{0.2}$. This variation is much smaller than seen in the recommended prediction procedure for heat transfer in annuli [10]. The recommended prediction shows the rate of heat transfer varying with $(D_o/D_i)^{0.53}$. In view of this, it was concluded that the actual D_o/D_i effect is greater than that predicted by Reichardt's equation [8]. Therefore, it was assumed that the turbulent intensity at the edge of the bubbly layer in an annulus, $(i_b)_a$, was equal to the turbulent intensity at the same distance from the wall in a round tube, i_b, multiplied by a correction factor that varied linearly with D_o/D_i. That is

$$i_b/(i_b)_a = [a' + b'(D_o/D_i)] \quad (5)$$

where D_o, D_i are the outer and inner diameters, respectively. By trial and error, the best values of the constants were found to be: $a' = 1.68$, $b' = 0.41$. After this change, a better prediction resulted with $\mu(R) = 1.01$ and $\sigma(R) = 0.11$.

Since the turbulent intensity relationship of equation (5) is without a direct theoretical foundation, it was decided that an equally valid approach would be the application of a correction depending on (D_o/D_i) to the unaltered round tube prediction. We therefore defined the factor ϕ such that

$$(q''_{CHF})_{annulus} = (q''_{CHF})_{round\,tube}\,\phi. \quad (6)$$

Based on the results of the previous analysis

$$\phi = 1.46 - 0.27(D_o/D_i). \quad (7)$$

Application to this correction factor to the CHF data in annuli eliminated the (D_o/D_i) effect and resulted in $\mu(R) = 1.01$ and $\sigma(R) = 0.10$. Figure 1 compares the predicted and measured results for a random sampling of one fourth of the data points. Note that the data considered includes tests in which only the outer wall was heated and tests in which only the inner wall was heated.

It is instructive to compare the (D_o/D_i) corrections applied here to the correction to round tube heat transfer required by the recommended equation for single phase heat transfer in annuli [10]. Since the recommended design equation for annuli, due originally to Monrad and Pelton [11], is given by

$$Nu = 0.02 Pr^{1/3} Re^{0.8} (D_o/D_i)^{0.53} \quad (8)$$

the correction factor is found to be

$$(h_c)_{tube}/(h_c)_{annuli} = 1.15(D_o/D_i)^{-0.53} \quad (9)$$

when we divide the Dittus–Boelter equation by equation (8). Figure 2 compares the ratio of convective heat transfer coefficients derived from equation (9) to the corrections of equations (5) and (7). The strong similarity of these corrections to the ratio from equation (9) is evident.

The substantial underprediction seen by Wittekind [5] at high void fractions in horizontal lines was not seen with the vertical line data considered here. At void fractions of approximately 0.77, predicted values were only about 8% below observations. There was, however, a tendency for predicted CHFs to increase as the void fraction and qualities

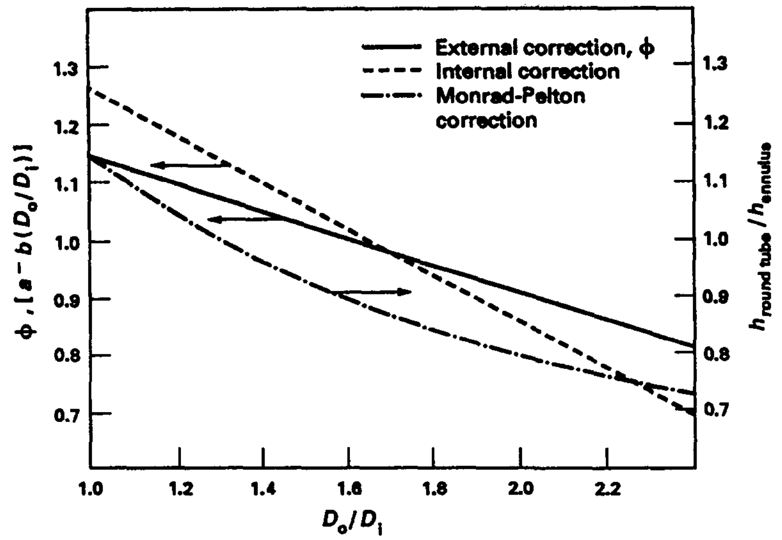

FIG. 2. Comparison of $[(h_c)_{round\ tube}/(h_c)_{annulus}]$ to corrections made to CHF predictions for annuli.

decreased. At exit qualities of -0.05, predictions were about 10% above observations. This variation of $\mu(R)$ with quality might indicate that there is a small enthalpy gradient at subcooled conditions and low qualities. The change in behavior at higher qualities might be due to both an increased eddy diffusivity in the presence of voids as well as some differences between the void profile in round tubes and annuli. Lim [12] developed a model for computing the fluid quality adjacent to the bubbly layer by using these concepts. While he was able to eliminate the variation with exit quality, he was not able to significantly reduce $\sigma(R)$. It appears that the magnitudes of the effects examined by Lim are small and satisfactory CHF predictions can be obtained without their consideration.

CONCLUSION

The revised Weisman and Pei CHF model can be applied to annuli with only one heated wall [1, 3]. Accurate predictions require: (a) the use of an average heat flux in computing the quality of the fluid adjacent to the bubbly layer and (b) a correction which accounts for the effect of the diameter ratio on the turbulent intensity. The proposed diameter ratio correction is consistent with the revision required to convert single phase heat transfer predictions for round tubes to predictions for annuli.

Since Wittekind [5] found that his empirical coefficient in the equation for i_b applied whether one or both walls of the annulus was heated, it is believed that the modification of Weisman and Pei's approach suggested herein also applies to annuli when both walls are heated.

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