

A General Correlation for Heat Transfer During Dispersed-Flow Film Boiling in Tubes

M. MOHAMMED SHAH

Riyadh, Saudi Arabia

M. ALTAMUSH SIDDIQUI

Department of Mechanical Engineering, Aligarh Muslim University, Aligarh, U.P., India

A general correlation for heat transfer during film boiling in tubes is presented. It is based on the two-step model. It has been verified with data for nine fluids flowing up in tubes. The fluids include water, cryogenics, refrigerants, and chemicals. The range of data includes pressures from 1 to 215 bar, reduced pressures from 0.0046 to 0.97, mass velocities from 4 to 5,176 kg/m² s, tube diameters from 1.1 to 24.3 mm, and qualities from 0.1 to 2.4. The 546 data points are predicted with a mean deviation of 15.2%. Deviation is defined as the difference between the measured and predicted heat transfer coefficients divided by the measured heat transfer coefficient, the heat transfer coefficients being based on the saturation temperature. Three other well-known correlations are also compared to the same data and found to have much larger deviations. The correlation is also compared with a limited amount of data from horizontal tubes; the results are encouraging.

A wide variety of heat exchangers involve film boiling during normal or upset conditions. Examples are cryogenic coolers, refrigerant evaporators, and nuclear reactors after a postulated loss-of-coolant accident. Hence reliable prediction of film boiling heat transfer is of much practical importance. Consequently, a large number of experimental studies have been done and numerous predictive techniques have been proposed for heat transfer estimation. These have been reviewed by, among others, Andreani and Yadigaroglu [1], Chen and Costigan [2], Shah [3], and Groeneveld and Snoek [4].

In most of the experimental studies, fluid temperature was not measured. Therefore the presence of nonequilibrium has been inferred from the behavior of measured wall temperatures. However, there have been several studies in recent years which have directly shown the presence of nonequilibrium.

Most of the early predictive techniques were modifications of single-phase heat transfer correlations and took no account of the physical phenomena such as nonequilibrium. One of the most successful correlations of this type is that of Groeneveld [5]. Such correlations have only a limited range of validity and are applicable to only a single fluid.

Researchers at the Massachusetts Institute of Technology and the U.K. Atomic Energy Authority independently developed the two-step model, according to

Address correspondence to M. Mohammed Shah, whose present address is MBC-38, P.O. Box 3354, Riyadh 11211, Saudi Arabia. E-mail: mmshahpe@hotmail.com

which heat is first transferred from the wall to the vapor and then from the vapor to the liquid droplets suspended in the vapor stream. Dry or wet interactions between the liquid droplets and the wall are also considered possible. Radiation heat transfer is also considered in some models. Many mechanistic analysis techniques have been developed based on this model. Examples are those of Bennet et al. [6], Forslund and Rohsenow [7], and Yoder and Rohsenow [8]. A recent mechanistic analysis is the three-dimensional model of Andreani and Yadigaroglu [9]. While this rational approach is highly desirable, the solution of the model equations still involves the use of empirical relations and factors for calculating parameters such as droplet size and distribution, interfacial heat transfer, effect of droplets on vapor velocity profile, etc. Calculations are very laborious and take a lot of time even on high-speed computers.

A number of correlations have been developed which take into account the nonequilibrium and are much easier to use than the mechanistic analyses. The most verified among such correlations appear to be those of Groeneveld and Delorme [10], Saha [11], and Hein and Kohler [12]. The last mentioned was applicable to water only but has been generalized by Katsounis [13] to make it applicable to other fluids; in this article this generalized form is called the Kohler correlation. These correlations are in mathematical form and hence suitable for computerized calculations, as indeed are most of the other correlations.

A well-verified correlation was earlier presented by one of the present authors [14]. It is based on the two-step model and is consistent with the physical phenomena. It was verified with data for water and several other fluids over a wide range of parameters. Yoder and Rohsenow [8] also presented a graphical predictive technique which was verified with data for three fluids. While the graphical correlations are very convenient for hand calculations, their use is limited because most calculations these days are done on computers.

Review articles cited earlier indicate that the empirical and nonequilibrium correlations available at present for computerized calculations are not fully satisfactory. The mechanistic models have had limited verification and require a large amount of computer time. Therefore, there is still need for a simple and accurate predictive technique for computerized calculations. This research was undertaken to fulfill this need. The starting point for this work was the Shah correlation [14]. Presented here is a general correlation for heat transfer during film boiling which has been verified with data for water and eight other fluids flowing upward in vertical tubes, over a wide range of parameters. The correlation can be used easily for hand calculations as well as for computerized calculations. Results of comparison of the new corre-

lation as well as three other leading correlations with data are presented. A limited amount of data for horizontal tubes is also analyzed, and applicability of the new correlation to horizontal tubes is discussed.

It is to be noted that in this article, the terms CHF occurrence, dryout, burnout, and boiling crisis have been used interchangeably; no particular mechanism is implied.

PHYSICAL MODEL FOR THE CORRELATION

This correlation is intended for dispersed-flow conditions. It was decided to limit the equilibrium vapor qualities to 10% or greater to exclude other flow patterns. The same limit has also been used by Hammouda et al. [15]. The physical model assumed for developing the correlation is as follows.

1. Flow consists entirely of liquid droplets dispersed in the vapor stream.
2. At the dryout point, there is complete thermodynamic equilibrium. This assumption is common to almost all analyses, including those of Bennet et al. [6] and Yoder and Rohsenow [8]. This assumption appears to be very reasonable, as there is equilibrium just upstream of the dryout point.
3. The turbulence structure and the temperature distribution of vapor flow is not affected by the presence of droplets. Thus the wall-to-vapor heat transfer can be calculated by single-phase correlations. This assumption is common to most predictive techniques, though differing views have been expressed by some researchers, e.g., Varone and Rohsenow [16].
4. At low to moderate reduced pressures, the effects of liquid droplet-wall interactions are negligible. Essentially all heat is removed from the wall by vapor which in turn heats the liquid droplets.
5. At high reduced pressures, the effects of liquid droplet-wall interactions become significant, as shown by Bailey et al. [17]. Heat is then removed from the wall by droplet-wall interactions as well as through convection by vapor.
6. Heat transfer by radiation is negligible. This assumption is common to most predictive techniques, though some researchers have expressed differing views—for example, Chen [18].

DEVELOPMENT OF THE CORRELATION

Developing the correlation involved the selection of physically meaningful dimensionless groups/factors on which nonequilibrium may depend and then finding the relation between them. This effort is briefly described in the following.

Several studies have showed that the burnout quality x_c affects nonequilibrium. A notable example is the work of Hynek et al. [19]. They carried out tests in which all parameters were the same except the critical quality. The wall temperatures were found to be very different in such runs, indicating that for the same equilibrium qualities, actual vapor qualities were different.

Several experimental studies show that nonequilibrium is affected by heat flux. Hence it will be reasonable to expect the nonequilibrium to depend on the nondimensional heat flux. The boiling number, Bo, is defined as

$$\text{Bo} = \frac{q_w}{Gi_{fg}} \quad (1)$$

where q_w is the total wall heat flux, G is the mass velocity, and i_{fg} is the latent heat. Bo may be interpreted as the dimensionless heat flux. It may also be mentioned that the mechanistic analysis of Yoder and Rohsenow [8] shows that Bo affects the magnitude of nonequilibrium.

Examination of data from most experimental studies indicates that nonequilibrium increases with decreasing mass velocity. A study which shows it particularly well is that of Forslund and Rohsenow [7] with nitrogen. Thus it appears that as the mass velocity decreases, heat transfer between vapor and droplets deteriorates more than the heat transfer between vapor and tube wall. A dimensionless group containing G is the Froude number, Fr_L :

$$\text{Fr}_L = \frac{G^2}{\rho_L^2 g D} \quad (2)$$

where ρ_L is the liquid density, g is the acceleration due to gravity, and D is the tube diameter. The Froude number is the ratio of inertia forces to gravitational forces. Soo [20] has shown through an analysis of the governing differential equations that the Froude number is a parameter for the similarity of dilute suspensions of solids in gases flowing through pipes. Varone and Rohsenow [16] have shown that the behavior of liquid droplet suspensions during film boiling in tubes is similar to that of gas–solid suspensions in tubes. Thus the Froude number can be expected to be a similarity parameter for dispersed-flow film boiling in tubes. It has been used in many two-phase heat transfer correlations, for example, by Shah [21] for critical heat flux in vertical tubes [21], for heat transfer in tubes prior to dryout [22]. Griffith and Wallis [23] used the Froude number in their correlation for flow patterns in vertical tubes. A Weber number based on the tube diameter was also tried in the early stages, but it did not give satisfactory correlation.

Thus x_c , Bo, and Fr were identified as possible correlating parameters. Data at moderate pressures for various fluids and from many studies were analyzed in terms of these parameters and a graphical correlation was developed. When it was compared to high-pressure data, it was found to overpredict wall temperatures, i.e., underpredict heat transfer coefficients. Bailey et al. [17] had reported similar results with the analytical solution of Bennet et al. [6] and attributed it to droplet–wall interactions. Hewitt [24] also expressed the same opinion. A factor to account for this effect was developed and incorporated into the correlation.

The correlation was developed using data analyzed by Shah [14]. The other data were used for verification. Thus none of the data for helium, R-12, R-22, and R-134a were used in developing the correlation. Also not used in developing the correlation were the low-pressure water data of Annunziato et al. [25], Evans et al. [26], Chen [18], and Gottula et al. [27].

THE CORRELATION

This correlation is described by the following equations.

The total heat flux q_w removed from the wall is expressed by

$$q_w = q_c + q_{dc} = F_{dc} h_g (T_w - T_g) \quad (3)$$

where q_c is the heat flux removed from the wall by convection to vapor, q_{dc} is the heat flux removed by wall–droplet interaction, h_g is the wall-to-vapor convective heat transfer coefficient, T_w is the wall temperature, T_g is the actual bulk vapor temperature, and F_{dc} is a factor to account for the enhancement due to wall–droplet interactions.

T_g is calculated by the following basic heat balance equation:

$$\frac{x_E - x_A}{x_A} i_{fg} = \int_{T_{SAT}}^{T_g} C_{pg} dT = i_g - i_{gSAT} \quad (4)$$

where x_E is the equilibrium quality, x_A is the actual quality, T_{SAT} is the saturation temperature, C_{pg} is the specific heat of vapor at constant pressure, i_g is the enthalpy of vapor at the actual temperature, and i_{gSAT} is the enthalpy of saturated vapor.

At the dryout point x_A and x_E are assumed equal. Downstream of the dryout point, the following relations apply:

$$\text{For } \text{Bo} \geq 5 \times 10^{-4}, \quad x_A = f_n(x_c, x_E, \text{Fr}_L) \quad (5)$$

At $\text{Bo} = 0$ (i.e., zero heat flux), there could obviously be no nonequilibrium. For $\text{Bo} \times 10^4 < 5$, $(x_E - x_A)$ is

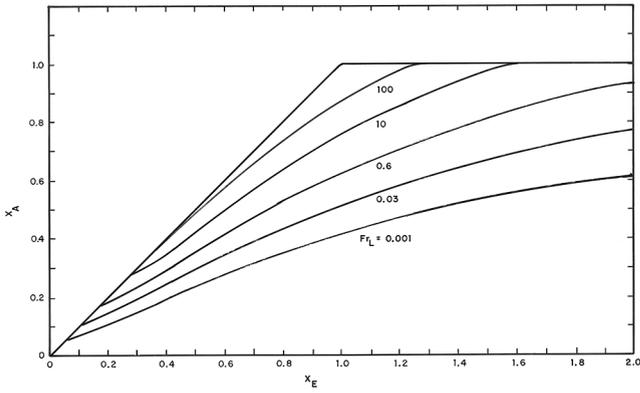


Figure 1 The present correlation for film boiling in tubes in graphical form.

calculated by linear interpolation using the following equation:

$$(x_E - x_A) = (x_E - x_A)_o \frac{Bo}{5 \times 10^{-4}} \quad (6)$$

where $(x_E - x_A)_o$ is the value of $(x_E - x_A)$ at $Bo \geq 5 \times 10^{-4}$ given by Eq. (5).

The functional relation of Eq. (5) is expressed graphically in Figure 1. It is expressed by the following equations.

For $x_E \geq 0.4$,

$$x_A = (A_1 + A_2 x_E + A_3 x_E^2 + A_4 x_E^3) Fr_L^n \quad (7)$$

The value of x_A given by Eq. (7) is corrected by the following relations:

$$\text{If } x_A > x_E, \quad \text{then } x_A = x_E \quad (8)$$

$$\text{If } x_A > 1, \quad \text{then } x_A = 1 \quad (9)$$

For $x_E < 0.4$, the correlation is represented by straight lines joining x_A at $x_E = 0.4$ given by Eq. (7) and intersecting the equilibrium line ($x_A = x_E$) at

$$x_{A,INT} = x_{E,INT} = 0.19 Fr_L^{0.16} \quad (10)$$

In Eq. (7),

$$A_1 = -0.0347 \quad A_2 = 0.9335$$

$$A_3 = -0.2875 \quad A_4 = 0.035$$

$$n = 0.064$$

The curves shown in Figure 1 have been calculated from the above equations.

PROCEDURE FOR CALCULATION WITH THE PRESENT CORRELATION

Calculation of x_A

The procedure for graphical calculations is described in Appendix 1. Calculations using the equations are explained now.

Knowing x_E , x_c , Fr_L , and Bo , the calculations proceed as follows.

1. For the given Fr_L , determine the intersection between the correlating curve and the equilibrium line. For $Fr_L \geq 100$, this is done by simultaneous solution of Eq. (7) and the equation $x_A = x_E$. For $Fr_L < 100$, intersection point is given by Eq. (10).
2. If $x_c \leq x_{E,INT}$, $x_A = x_E$ for $x_E \leq x_{E,INT}$. For $x_E > x_{E,INT}$, x_A is obtained by Eqs. (7)–(10).
3. If $x_c > x_{E,INT}$, determine the point where the tangent from the point $x_E = x_A = x_c$ touches the curve given by Eq. (7). This point of tangency is at the intersection of Eqs. (7) and (11).

$$x_A = x_c + (x_E - x_c)$$

$$\times (A_2 + 2A_3 x_E + 3A_4 x_E^2) Fr_L^n \quad (11)$$

Simultaneous solution of Eqs. (7) and (11) provides the tangent point $(x_{E,TAN}, x_{A,TAN})$. For $x_E > x_{E,TAN}$, obtain x_A from Eq. (7). For $x_E < x_{E,TAN}$, x_A is obtained from the line joining $(x_{E,INT}, x_{A,INT})$ with $(x_{E,TAN}, x_{A,TAN})$. Thus,

$$x_A = \frac{(x_{A,TAN} - x_c)}{(x_{E,TAN} - x_c)} x_E + x_c \frac{(x_{E,TAN} - x_{A,TAN})}{(x_{E,TAN} - x_c)} \quad (12)$$

4. If $Bo < 5 \times 10^{-4}$, apply Eq. (6) to correct the $(x_E - x_A)_o$ calculated by the above procedure.

The above equations can be easily programmed for computerized calculations. In Appendix 2, calculations for the case of nonuniform heat flux are explained and the calculation of x_c is discussed.

Calculation of F_{dc}

If $p_r > 0.8$ and $L/D > 30$, then

$$F_{dc} = 2.64 p_r - 1.11 \quad (13)$$

Otherwise, $F_{dc} = 1$. Note that data for $L/D < 30$ have not been considered in developing the correlation, for reasons discussed later.

Calculation of Single-Phase Heat Transfer Coefficient

To calculate h_g in Eq. (3), a single-phase heat transfer correlation is required. Numerous such correlations are available, many of which have been listed by Bhatti and Shah [28]. The following four correlations were tried.

1. The Dittus-Boelter correlation,

$$Nu = 0.023 Re^{0.8} Pr_g^{0.4} \quad (14)$$

where Nu is the Nusselt number ($h_g D / k_g$), k_g is the vapor thermal conductivity, Re is the Reynolds number, and Pr is the Prandtl number.

2. The correlation of Hallader and Bannerjee [29],

$$Nu = 0.00834 Re^{0.8774} Pr^{0.6112} \quad (15)$$

3. The Gnielinski correlation [30],

$$Nu = 0.012(Re^{0.87} - 280) Pr^{0.4} \quad (16)$$

4. The correlation of Pethukov et al. [31],

$$Nu = \frac{(f/2) Re Pr}{C + 12.7(f/2)^{0.5}(Pr^{2/3} - 1)} \quad (17)$$

where

$$C = 1.07 + \frac{900}{Re} - \frac{0.63}{1 + 10 Pr}$$

and f is the Fanning friction factor for flow of fluids in tubes.

For use in the present film boiling correlation, Re in the above formulas is defined as

$$Re = \frac{GDx_A}{\mu_g \alpha} \quad (18)$$

where μ_g is the vapor viscosity, and α is the void fraction. Calculations were done with all the above four correlations using properties at bulk fluid temperature as well as at film temperature. The film temperature is defined as the arithmetic average of the wall and

bulk fluid temperatures. It was found that for water, film temperature properties gave better agreement, especially at higher pressures. For fluids other than water, bulk temperature properties generally gave better agreement. The deviations of these equations with the various data sets varied, but considering all data together, the mean deviations of all four correlations were about the same. After detailed study of the results, the following recommendations are made.

1. For $Re < 10^4$, use the Dittus-Boelter equation.
2. For $Re > 10^4$, use the Hallader equation.
3. For water, use the film temperature properties. For other fluids, use the bulk temperature properties.

Calculation of Void Fraction

With small liquid droplets suspended in vapor stream, it is reasonable to expect that there will be little slip between the phases. Koizumi et al. [32] report that the slip velocity in their R-113 data was 1 m/s or less. Groenevald and Delorme [10] have quoted other experimental studies showing low slip.

The void fraction was therefore calculated by the homogeneous model, which gives

$$\alpha = \frac{x_A \rho_L}{(1 - x_A) \rho_g + \rho_L x_A} \quad (19)$$

where ρ_g is the vapor density.

COMPARISON WITH EXPERIMENTAL DATA

Property Data

Properties for water/steam were taken from the *ASME Steam Tables*, [33], using the computer program provided with it.

Properties for R-12, R-22, R-113, R-134a, and nitrogen were taken from the AECL code "PROPER." Comparison with the data in the *ASHRAE Handbook* [34], to the extent possible, showed satisfactory agreement. The properties in this book are mostly limited to saturated vapors.

Properties of propane and methane were taken from Vargaftik [35].

For helium, property data from two sources were examined. These are McCarty [36] and Beaton and Hewitt [37]. These two sources differ widely, the latter giving much lower heat transfer. Those by McCarty appear to be more commonly used and gave better agreement with the heat transfer data analyzed here. Therefore the property data of McCarty were used in the final analysis.

Experimental Data Analyzed

This research was primarily intended for upflow in vertical tubes. Attempts were made to collect data for this configuration, covering as wide a range as possible for this configuration. The data for vertical upflow in tubes that have been analyzed are listed in Tables 1 and 2. The range covered by these data is summarized in Table 3. It is seen that it is very wide, including nine fluids over an extreme range of mass flux, pressure, and other parameters. Hydrogen data analyzed by Shah [14] were reevaluated, with essentially the same results. To deal adequately with this subject, a detailed study considering hydrogen data from many sources is needed. Hydrogen data are therefore not discussed here.

Data for $L/D < 30$ have not been included in Tables 1–3, L being the distance from the CHF point. The phenomena in this region are very complex. These

include entrance effects, developing boundary layers, splashing by liquid droplets, rapidly changing temperatures, etc. A separate study is needed to deal adequately with this region. Some researchers, for example, Nishikawa et al. [38] and Saha [11], have applied entrance effect factors for single-phase heat transfer in analyzing the film boiling data in the region close to the burnout point and found adequate agreement. This approach was also tried in the present work and the results are discussed later.

To ensure exclusion of the data for transition boiling, data for wall temperature below the Leidenfrost temperature have been excluded. The following equation of Spiegler et al. [39] was used to predict the Leidenfrost temperature, T_{LEID} :

$$T_{LEID} = (0.13 p_r + 0.84)T_c \quad (20)$$

Table 1 Water data analyzed and results of their comparison with the present correlation

| Source | D (mm) | G (kg/m ² s) | q_w (kW/m ²) | Bo ×10 ⁴ | Re ×10 ⁻³ | Fr_L | x_E | p_r | Percent deviation of correlation | | | |
|-----------------------------|-------------|------------------------------|-------------------------------|------------------------|-------------------------|---------|-------|--------|----------------------------------|------|--------|-------|
| | | | | | | | | | Number of data points | | | |
| | | | | | | | | | New | G-D | Kohler | Saha |
| Annunziato et al. [25] | 12.4 | 3.7 | 2.4 | 2.6 | 1.3 | 0.00012 | 0.65 | 0.0046 | 22.0 | 51.1 | NC | NC |
| Chen [18] | 12.0 | 25 | 20 | 2.4 | 1.5 | 0.0057 | 0.20 | 0.005 | 22.0 | 36.4 | 15.6 | 182.3 |
| | | 417 | 100 | 6.4 | 30.0 | 2.2 | 0.34 | 0.16 | 16 | 16 | 16 | 6 |
| Nijhawan et al. [40, 41] | 14.1 | 30 | 22 | 2.4 | 3.3 | 0.0074 | 0.25 | 0.012 | 15.6 | 21.7 | 37.0 | 15.1 |
| | | 42 | 29 | 4.4 | 5.3 | 0.0150 | 0.40 | | 9 | 9 | 9 | 5 |
| Evans et al. [26] | 15.4 | 15 | 26 | 8.1 | 5.0 | 0.0017 | 0.74 | 0.017 | 19.2 | 11.3 | 39.0 | NA |
| | | | | | | | | | | | | |
| Gottula et al. [27] | 15.4 | 17 | 16 | 4.3 | 4.1 | 0.0023 | 0.42 | 0.018 | 13.2 | 18.0 | 34.3 | NA |
| | | | 18 | 4.9 | 5.8 | | | | | | | |
| Ling et al. [46] | 14.9 | 500 | 510 | 6.2 | 177 | 2.51 | 0.70 | 0.13 | 9.4 | 35.9 | 15.9 | 20.9 |
| | | | 560 | 8.5 | 687 | | | | | | | |
| Janssen & Kervinnen [58] | 12.52 | 1,016 | 838 | 5.5 | 358 | 15.3 | 0.71 | 0.32 | 2.8 | 28.1 | 30.7 | 7.6 |
| | | | | | | | | | | | | |
| Era et al. [48] | 5.97 | 1,100 | 540 | 2.7 | 251 | 37.7 | 0.59 | 0.32 | 11.4 | 22.1 | 18.0 | NA |
| | | 6.00 | 2,200 | 900 | 3.3 | | | | | | | |
| Bennet et al. [6] | 12.6 | 379 | 511 | 2.3 | 129 | 2.1 | 0.27 | 0.32 | 7.9 | 26.0 | 36.1 | 14.4 |
| | | 5,176 | 1,833 | 8.9 | 1,078 | | | | | | | |
| Kastner et al. [43] | 12.5 | 510 | 300 | 3.6 | 252 | 3 | 0.45 | 0.23 | 21.3 | 47.7 | 40.1 | 18.4 |
| | | 1,001 | 497 | 4.6 | 475 | | | | | | | |
| Kohler [44] | 12.5 | 1,000 | 488 | 2.9 | 264 | 13.7 | 0.52 | 0.23 | 16.5 | 60.9 | 6.2 | 30.4 |
| | | 1,017 | 500 | 5.0 | 434 | | | | | | | |
| Bailey et al. [17] | 12.8 | 2,000 | 242 | 1.5 | 543 | 108.0 | 0.40 | 0.82 | 15.0 | 11.1 | 19.2 | 33.1 |
| | | | 614 | 3.9 | 737 | | | | | | | |
| Herkenrath et al. [49] | 10.0 | 1,530 | 720 | 6.4 | 308 | 84.4 | 0.37 | 0.84 | 13.4 | 5.3 | 5.6 | 27.3 |
| | | | | | | | | | | | | |
| Bishop et al. [50] | 2.54 | 1,355 | 661 | 6.8 | 100 | 201 | 0.13 | 0.88 | 8.6 | 31.2 | 23.8 | 44.4 |
| | | 5.10 | 3,374 | 1,921 | 21.0 | | | | | | | |
| Swenson et al. [52] | 10.4 | 948 | 309 | 6.4 | 196 | 40 | 0.18 | 0.94 | 5.8 | 22.4 | 33.2 | NC |
| | | 1,355 | 565 | 8.2 | 469 | | | | | | | |
| Schmidt [51] | 8.0 | 700 | 470 | 18.3 | 131 | 34 | 0.35 | 0.97 | 29.6 | 18.4 | NA | NA |
| | | | | | | | | | | | | |
| All data | 2.54 | 3.7 | 16 | 2.4 | 1.3 | 0.00012 | 0.13 | 0.0046 | 13.5 | 31.0 | 27.8 | 40.2 |
| | | | | | | | | | | | | |

Table 2 Data analyzed for fluids other than water and results of their comparison with the present correlation

| Source | Fluid | D (mm) | G (kg/m ² s) | q_w (kW/m ²) | Bo ×10 ⁴ | Re ×10 ⁻³ | Fr _L | x_E | p_r | Percent deviation of correlation | | | |
|-------------------------|----------|-------------|------------------------------|-------------------------------|------------------------|-------------------------|-----------------|-------|-------|----------------------------------|------|--------|------|
| | | | | | | | | | | Number of data points analyzed | | | |
| | | | | | | | | | | New | G-D | Kohler | Saha |
| Ogata & Sato [53] | Helium | 1.09 | 82 | 0.63 | 3.8 | 38 | 41.7 | 0.5 | 0.49 | 9.3 | 48.1 | 16.1 | 17.2 |
| | | | 92 | 1.4 | 7.6 | 58 | 52.0 | 1.0 | 8 | 8 | 8 | 8 | |
| Glickstein [54] | Propane | 8.9 | 251 | 397 | 23.2 | 46 | 3.0 | 0.19 | 0.24 | 7.7 | 62.3 | 4.7 | 22.5 |
| | | | 652 | 561 | 59.0 | 207 | 20.3 | 1.00 | 14 | 14 | 14 | 4 | |
| Glickstein [54] | Methane | 8.9 | 209 | 214 | 24.7 | 66 | 3.8 | 0.50 | 0.22 | 1.2 | 68.1 | 5.1 | 20.9 |
| | | | | | 78 | | 0.75 | 2 | 2 | 2 | 2 | | |
| Hynek et al. [19] | Nitrogen | 10.2 | 42 | 23 | 26.2 | 10 | 0.027 | 1.00 | 0.05 | 9.6 | 53.6 | 34.4 | 29.3 |
| | | | 44 | | 27.5 | 19 | 0.031 | 2.23 | 11 | 11 | 11 | 7 | |
| Forslund & Rohsenow [7] | Nitrogen | 8.2 | 95 | 214 | 6.3 | 7 | 0.17 | 0.14 | 0.05 | 16.5 | 58.8 | NA | NA |
| | | | 257 | | 34.1 | 90 | 1.29 | 1.92 | 79 | 79 | | | |
| Laverty & Rohsenow [55] | Nitrogen | 8.1 | 95 | 12 | 6.3 | 10 | 0.18 | 0.15 | 0.05 | 24.9 | 38.9 | NA | NA |
| | | | 286 | 54 | 29.0 | 76 | 1.61 | 1.8 | 25 | 14 | | | |
| Cumio [56] | R-12 | 4.75 | 600 | 87 | 8.7 | 86 | 7.2 | 0.35 | 0.33 | 24.8 | 43.6 | 57.1 | 22.4 |
| | | | 1,230 | 110 | 15.8 | 285 | 32.0 | 2.00 | 0.69 | 40 | 40 | 40 | 12 |
| Schnittger [45] | R-12 | 12.5 | 326 | 41 | 3.1 | 248 | 0.55 | 0.35 | 0.23 | 18.1 | 24.0 | 23.2 | 15.8 |
| | | | 24.3 | 1,255 | 62 | 12.3 | 715 | 13.2 | 2.85 | 0.68 | 52 | 52 | 52 |
| Hammouda et al. [15] | R-12 | 4.4 | 530 | 39 | 5.2 | 16 | 4.0 | 0.10 | 0.20 | 11.2 | 25.0 | NA | NA |
| | | | 950 | 51 | 8.6 | 66 | 12.9 | 0.50 | 12 | 12 | | | |
| Groeneveld [57] | R-12 | 7.8 | 665 | 123 | 8.6 | 182 | 3.9 | 0.40 | 0.27 | 16.9 | 19.7 | 17.0 | 23.6 |
| | | | 2,713 | 290 | 14.8 | 957 | 6.6 | 0.95 | 11 | 11 | 11 | 11 | |
| Nishikawa [37] (1983) | R-22 | 13.0 | 400 | 20 | 4.7 | 136 | 1.9 | 0.60 | 0.68 | 9.1 | 42.3 | 36.4 | 23.8 |
| | | | | | 244 | | 1.01 | 11 | 11 | 11 | 11 | | |
| Nishikawa [47] | R-22 | 9.0 | 400 | 20 | 3.7 | 126 | 2.7 | 0.70 | 0.68 | 11.3 | 24.2 | 36.0 | 16.0 |
| | | | 1,000 | 40 | 6.7 | 470 | 16.8 | 1.42 | 0.84 | 31 | 31 | 31 | 31 |
| Koizumi [32] | R-113 | 10.0 | 542 | 21 | 1.8 | 322 | 1.5 | 0.55 | 0.08 | 12.2 | 44.8 | 4.4 | 17.4 |
| | | | 1,154 | 72 | 7.1 | 573 | 6.7 | 1.10 | 10 | 10 | 10 | 10 | |
| Hammouda et al. [15] | R-134a | 4.4 | 950 | 64 | 4.0 | 33 | 15.2 | 0.10 | 0.20 | 20.9 | 17.4 | NA | NA |
| | | | | | 4.2 | 75 | 16.5 | 0.25 | 0.28 | 8 | 8 | | |
| All data | | 1.09 | 42 | 0.6 | 3.8 | 7 | 0.027 | 0.10 | 0.05 | 16.6 | 39.7 | 28.8 | 18.9 |
| | | | 24.3 | 2,034 | 561 | 59 | 957 | 52.0 | 2.35 | 0.84 | 314 | 314 | 190 |

where p_r is the reduced pressure and T_c is the critical temperature. (Note that both T_c and T_{LEID} are to be in K or R.) The Leidenfrost temperature predicted by this

formula is a little lower than the critical temperature of the fluid. It is realized that this equation may not be accurate, but a rough approximation is adequate for our purpose. Groeneveld and Delorme [10] had deleted data for wall temperature below 400°C in evaluating their correlation against water data for the same reason; the critical temperature of water is 374°C.

Where the publications in hand gave a large amount of data, representative data were picked out at reasonable intervals of parameters including the extreme values of parameters. This was to avoid excessive effort which would not have given any new information. Thus the data points analyzed are representative of many more data points.

Other Correlations Which Have Been Evaluated

So that the present correlation may be viewed in perspective, it was decided to compare the same database with the best available nonequilibrium correlations. From the literature survey, it appeared that the among

Table 3 Complete range of all vertical tube data analyzed

| Fluid | Water, helium, nitrogen, methane, propane, R-12, R-22, R-113, R-134a |
|---------------------------|--|
| Tube ID, mm | 1.09–24.3 |
| Pressure, bar | 1.02–215 |
| p_r | 0.0046–0.97 |
| q_w , KW/m ² | 0.6–1,921 |
| G , kg/m ² s | 3.7–5,176 |
| T_w , K | 5–1,065 |
| Fr _L | 0.00012–1,600 |
| Bo × 10 ⁴ | 1.5–59 |
| Re × 10 ⁻³ | 1.3–1,078 |
| x_E | 0.13–2.4 |
| x_c | –1.4 to +0.96 |
| No. of data points | 546 |
| Mean deviation | 15.2 |

the best are the correlations of Saha [11], Kohler (Hein and Kohler [12], as generalized by Katsounis [13], and Groeneveld-Delorme [10]. Attempt was made to compare the entire database with these correlations together with the present correlation.

The values of x_c for the data analyzed here were the measured values given in these data themselves. General methods for calculating x_c are discussed in Appendix 2.

Results of Data Analysis

Results of comparison of the measured heat transfer coefficients in vertical tubes with the present and the other three correlations are given in Tables 1 and 2. These show that the 546 data points analyzed are predicted by the present correlation with a mean deviation of 15.2%. The deviations of the other three correlations are seen to be considerably higher. In these tables, deviation is defined as follows:

$$\text{Deviation} = \frac{(\text{predicted } h_{TP} - \text{measured } h_{TP})}{\text{measured } h_{TP}} \quad (21)$$

where

$$h_{TP} = \frac{q_w}{T_w - T_{SAT}} \quad (22)$$

where T_{SAT} is the saturation temperature.

The complete range of vertical tube data analyzed is summarized in Table 3. Figures 2–7 show comparison of wall temperatures from some runs with the various correlations. Figures 8 and 9 show comparison of measured vapor temperatures with various correlations.

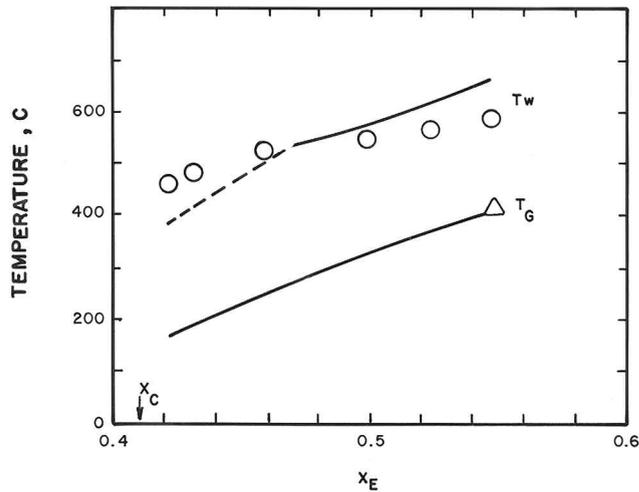


Figure 2 Comparison of the present correlation with the measured wall and vapor temperatures. Fluid water, $D = 15.4$ mm, $p = 3.7$ bar, $G = 17.3$ kg/m² s, $q_w = 18$ kW/m². Dashed line is the prediction for $L/D < 30$. Data of Gottula et al. [27].

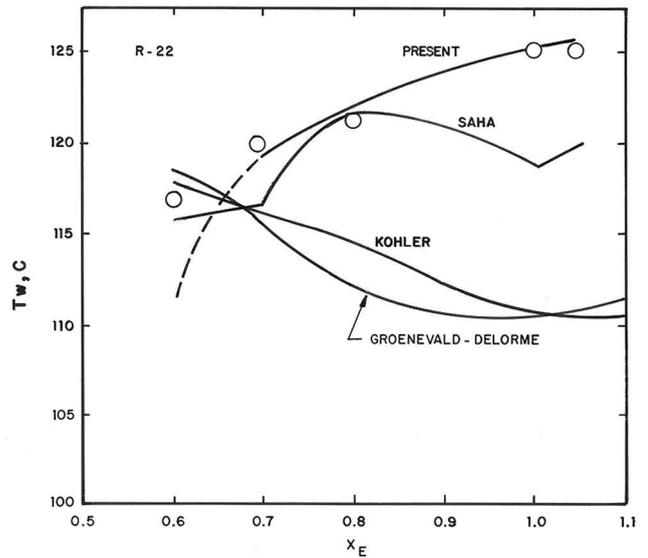


Figure 3 Comparison of the various correlations with the data of Nishikawa et al. [47] for R-22. $D = 9$ mm, $p = 42$ bar, $G = 400$ kg/m² s, $q_w = 20$ kW/m², $x_c = 0.54$. Dashed line is for $L/D < 30$.

DISCUSSION OF RESULTS OF DATA ANALYSIS

Range of Data Analyzed

The data analyzed include nine fluids with widely differing properties: cryogenics, chemicals, refrigerants, and water. The range of dimensional and dimensionless parameters is very wide. Mass velocity of water ranges

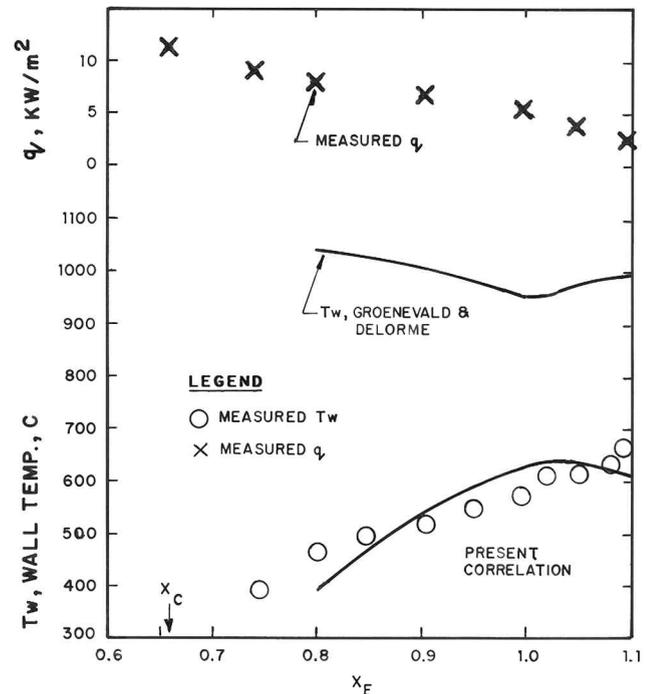


Figure 4 Comparison of the variable heat flux data of Annunziato et al. [25] for water with the present and the Groeneveld-Delorme correlations. $D = 12.4$ mm, $p = 1.01$ bar, $G = 4$ kg/m² s.

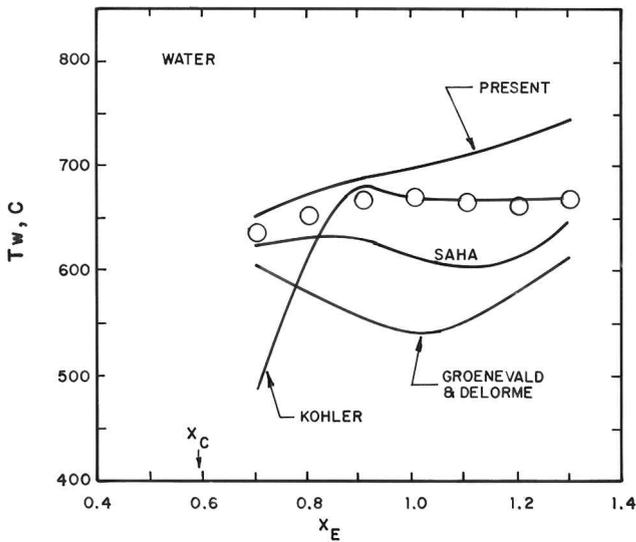


Figure 5 Comparison of the data of Ling et al. [46] for water with various correlations. $D = 14.9$ mm, $p = 119$ bar, $G = 500$ kg/m² s, $q = 510$ kW/m².

from 4 to 5,176 kg/m² s. The reduced pressure varies from 0.0046 to 0.97, an extremely wide range. Data for all fluids over the entire range are reasonably well correlated.

Very low velocities together with low pressures occur during reflooding of nuclear reactors. Such data are adequately predicted by the present correlation as seen in Figures 2 and 4 as well as in Table 1, while the deviations of other correlations are considerably higher.

Mode of Dryout Occurrence

In the data analyzed here, the boiling crisis was produced in several ways. These include increasing heat flux until dryout occurs (e.g., Bennet et al. [6]), quenching a preheated tube (for example, Hynek et al. [19]),

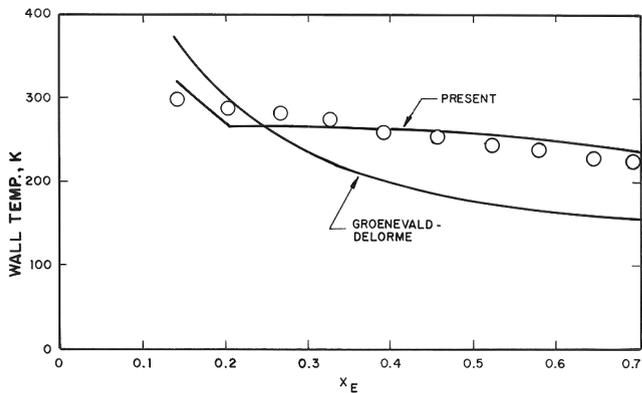


Figure 6 Comparison of the data of Forslund and Rohsenow [7] for nitrogen with present and Groenevald-Delorme correlations. $D = 8.2$ mm, $p = 1.72$ bar, $G = 257$ kg/m² s, $q_w = 31.5$ kW/m², $x_c = 0$.

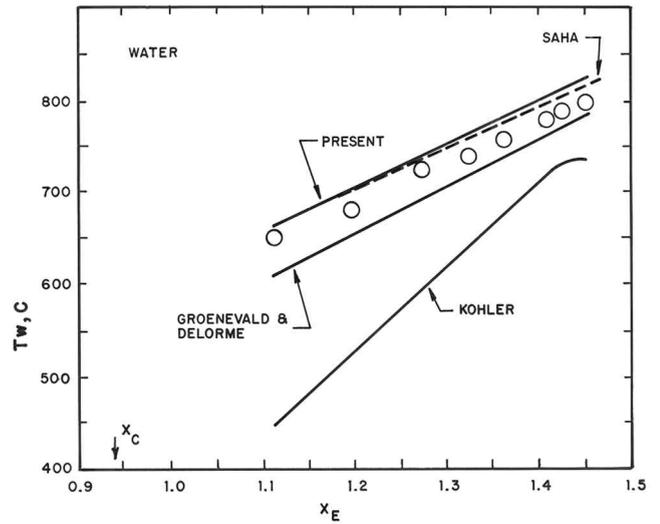


Figure 7 Comparison of the data of Bennet et al. [6] for water with various correlations. $D = 12.6$ mm, $p = 70$ bar, $G = 379$ kg/m² s, $q = 511$ kW/m².

and application of hot patch (for example, Hammouda et al. [15]). While most of the data are from steady-state runs, some quasi-steady-state data for slowly moving quench fronts are also included, for example, Evans et al. [26]. Data from runs of all these types are satisfactorily correlated.

Nonuniform Heat Flux

The only data for nonuniform heat flux included here are those of Annunziato et al. [25]. These also happen to be the data at the lowest mass flux and also at the lowest pressure. The agreement of the present correlation is

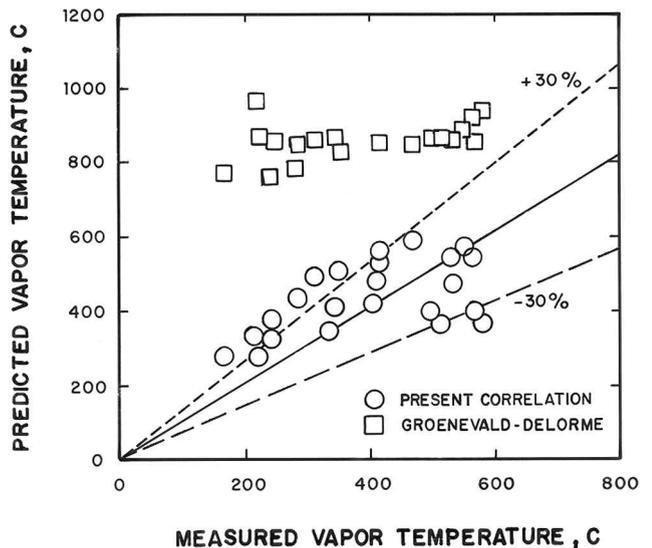


Figure 8 Comparison of vapor temperatures measured by Annunziato et al. [25] with the present and the Groenevald-Delorme correlations.

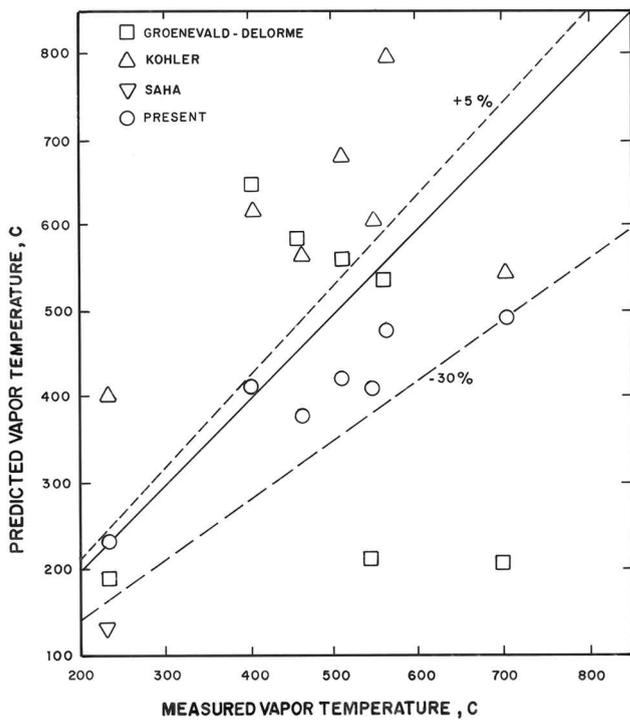


Figure 9 Comparison of Lehigh University measurements of water vapor temperatures with predictions of various correlations.

reasonably good, and is comparable to that of other low-pressure and low-mass-flux data at uniform heat flux. Thus the results with nonuniform heat flux data are encouraging, but more verification is needed to confirm the applicability to nonuniform heat flux.

The Groeneveld-Delorme correlation performed poorly with these data. These data were not compared with the Kohler and Saha correlations.

Vapor Temperature

There have been very few studies in which vapor temperature inside the tube has been measured. Such studies whose data have been analyzed here are those of Evans et al. [26], Gottula et al. [27], Nijhawan et al. [40, 41], and Annunziato et al. [25]; the first three of these are from a laboratory at Lehigh University. All of these data are for water at low pressure and flow rate to simulate reactor reflooding conditions.

Figure 2 compares the present correlation with the data of Gottula et al. [27]. The present correlation is seen to predict the vapor temperature accurately.

Figure 8 shows the comparison of the data of Annunziato et al. [25] with the present and the Groeneveld-Delorme correlations. The present correlation's predictions are in reasonable agreement with the data, while the Groeneveld-Delorme correlation predicts very high temperatures.

Figure 9 shows the comparison of the Lehigh University data with the various correlations. It is seen

that the present correlation predicts these data fairly well. The deviations of the other correlations are much larger.

It should also be noted that accurate measurement of vapor temperature in dispersed-flow film boiling is quite difficult, and some of the reported temperatures may not be accurate.

Thus the present correlation performs reasonably well in the prediction of vapor temperatures, and considerably better than the other correlations. However, the range of data available and analyzed is rather narrow.

As the agreement of the present correlation with measured vapor temperatures is better than those of the other correlations, its comparative performance in predicting heat transfer coefficients would have been even better had the heat transfer coefficients been based on the actual vapor temperatures.

The Region $L/D = 0$ to 30

For $L/D < 30$, we tried the following entrance effect factor given by McAdams [42] for single-phase flow:

$$\frac{h_z}{h} = 1 + \left(\frac{D}{L}\right)^{0.7} \quad (23)$$

where h_z is the heat transfer coefficient at $L/D = z$ and h is the heat transfer coefficient at a large L/D . The 76 data points in this region had a mean deviation of 20.7%. This is quite good. However, the results were mixed: some data had very low deviations, while some had large deviations.

Accurate predictions close to the burnout point are very difficult due to the complexity of phenomena as well as the steep rise in temperatures for a small increase in quality. A small error in estimation of burnout quality could result in a large percent error in wall temperature. Hence the results with this simple approach may be considered good.

Performance of Other Correlations

Among the other correlations tested here, only the Groeneveld and Delorme correlation could be compared to all data points. The Kohler and Saha correlations cannot be used when burnout occurs at subcooled conditions. The void fraction equation in the Saha correlation sometimes gave void fractions greater than 1; predictions could not be made for such data points. As a result of these limitations, the number of uniform-heat-flux data points analyzed with the Kohler and Saha

correlations is 331 and 216, respectively, as compared to 530 with the present and the Groeneveld-Delorme correlations.

While the other correlations show good agreement with some of the data sets, their overall performance is not good. For all vertical tube data, the mean deviation of the present correlation is 15.2%, for Groeneveld-Delorme 36.0%, Kohler 28.3%, and Saha 25.6%. While the mean deviation of the Kohler and Saha correlations are lower than that of the Groeneveld-Delorme correlation, these could be applied to many fewer data points.

The major shortcoming of the Saha correlation is its void fraction prediction. As it often gives void fraction greater than 1, thus invalidating the calculations, one never knows whether this correlation can be applied to a particular problem. For the data for which it could be used, its predictions were better than the Groeneveld-Delorme and Kohler correlations. Another limitation is that it cannot be used when the boiling crisis occurs at negative qualities.

It may be mentioned that the computer time required by the Saha correlation is much more than for the other three correlations.

Horizontal Tubes

There have been comparatively few studies on horizontal tubes. Three studies were found in which vapor quality exceeded 10%. Two of these, by Kastner et al. [43] and by Kohler [44], appear to be from the same series of tests. Their range of parameters is listed in Table 4. These studies show that at high mass velocities, the phenomena are the same as in vertical tubes. At comparatively low velocities, the upper surface dries out first and its temperature rises while the bottom of the tube still has nucleate boiling. The bottom of the tube dries out eventually and its temperature also rises.

We applied the present correlation to the data for the top of the tube using the measured x_c for the top of

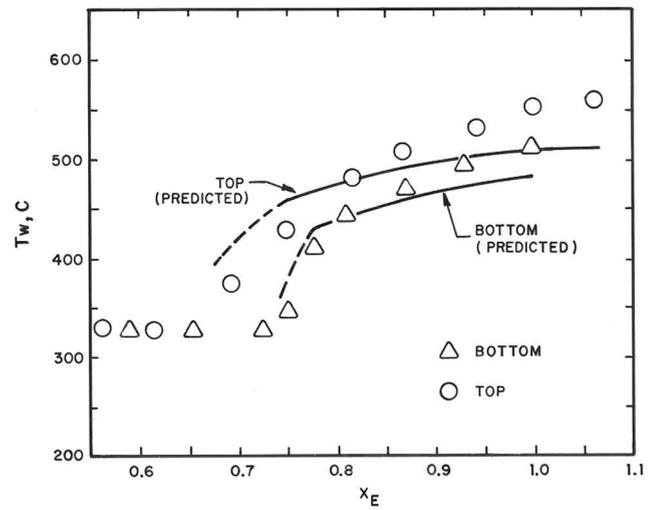


Figure 10 Comparison of the present correlation with the data of Kastner et al. [43] for water in a horizontal tube. $D = 12.5$ mm, $p = 100$ bar, $G = 743$ kg/m² s, $q_w = 391$ kW/m².

the tube. The measured wall temperatures at the side and bottom of the tube were similarly analyzed using the measured x_c at the side and bottom of the tube. The other three correlations were applied in the same way. It should be clarified that this approach has not been recommended by the authors of those correlations. The results are summarized in Table 4. It is seen that the agreement with the data of Schnittger [45] is good, while that with the other two sources (which appear to be really one) is fair. The mean deviation of the present correlation with all data is 25.5%. This is not bad. Figure 10 shows the comparison of data from one of the runs with the present correlation.

The results obtained with this simple approach are encouraging. No definite conclusion can be reached from the limited amount of data analyzed. More study is required using more varied data. Some modifications to the correlation may be needed. A significant factor may be conduction through tube wall.

Table 4 Summary of horizontal tube data analyzed

| Source | Fluid | D (mm) | G (kg/m ² s) | q_w (kW/m ²) | Bo ×10 ⁴ | Re ×10 ⁻³ | Fr _L | x_E | p_r | Percent deviation of correlations | | | |
|---------------------|-------|-------------|------------------------------|-------------------------------|------------------------|-------------------------|-----------------|-------|-------|-----------------------------------|------|--------|------|
| | | | | | | | | | | Number of data points | | | |
| | | | | | | | | | | New | G-D | Kohler | Saha |
| Kastner et al. [43] | Water | 12.5 | 505 | 189 | 2.8 | 244 | 4.4 | 0.70 | 0.45 | 29.9 | 71.0 | 38.3 | NA |
| | | | 743 | 391 | 4.0 | 310 | 9.5 | 1.08 | | | 13 | 13 | 13 |
| Kohler [44] | Water | 12.5 | 521 | 145 | 1.5 | 167 | 0.23 | 0.16 | 0.23 | 28.7 | 50.0 | 28.5 | 20.9 |
| | | | 24.3 | 2,467 | 605 | 6.9 | 1,495 | 80.7 | 1.06 | 0.91 | 40 | 40 | 33 |
| Schnittger [45] | R-12 | 24.3 | 384 | 62 | 4.7 | 290 | 3.9 | 0.38 | 0.45 | 16.7 | 19.4 | 23.7 | 18.1 |
| | | | 1,279 | 63 | 15.5 | 1151 | 5.5 | 0.84 | 0.68 | 27 | 27 | 27 | 27 |
| All data | | 12.5 | 384 | 62 | 1.5 | 167 | 0.23 | 0.38 | 0.23 | 25.5 | 43.6 | 29.6 | 19.1 |
| | | | 24.3 | 2,467 | 605 | 15.5 | 1,495 | 80.7 | 1.08 | 0.91 | 80 | 80 | 73 |

SUMMARY AND CONCLUSION

1. A general correlation, consistent with physical phenomena, is presented for heat transfer during film boiling, which can be conveniently used for computerized calculations as well as for hand calculations.
2. The present correlation has been shown to be in good agreement with a wide range of data for upflow in uniformly heated vertical tubes. The data include reduced pressures, from 0.0046 to 0.97, mass velocities from 4 to 5,176 kg/m² s, and nine fluids (water, helium, nitrogen, propane, methane, R-12, R-22, R-113, and R-134a).
3. A limited amount of data for nonuniformly heated tubes was also satisfactorily correlated.
4. In the region $L/D < 30$, use of a single-phase entrance effect factor gave reasonably good results, even though the phenomena involved are very complex.
5. The same database was also compared to the correlations of Saha [11], Hein and Kohler [12], and Groenevald-Delorme [10]. Their deviations were considerably higher. Further, the Saha and Kohler-Hein correlations are not applicable over the entire range of parameters.
6. The present correlation was also compared to data for horizontal tubes from three sources. The results are encouraging.
7. The correlation is recommended for application to vertical upflow in tubes in the range of data analyzed. This range is wide enough to cover most needs. Use outside this range should be made with caution.
8. More research is needed on horizontal tubes, nonuniformly heated tubes, and the region immediately downstream of the dryout point. Analysis of data for other fluids is also desirable.

NOMENCLATURE

All equations in this article are dimensionless. Hence any consistent system of units may be used.

| | |
|----------|---|
| Bo | boiling number [$= q_w / (G i_{fg})$], dimensionless |
| C_{pg} | specific heat of vapor at constant pressure, J/kg K (Btu/lb F) |
| D | inside diameter of tube, m (ft) |
| f | Fanning friction factor for pipe flow [$= (\Delta p / \Delta L) D \rho_L g_c / G^2$], dimensionless |
| F_{dc} | factor for enhancement resulting from liquid droplet-wall interaction, dimensionless |
| Fr_L | Froude number [$= G^2 / (\rho L^2 g D)$], dimensionless |

| | |
|-----------------------|--|
| g | acceleration due to gravity, m/s ² (ft/s ²) |
| g_c | conversion factor, = 1 for SI units, = 32.2 for British units |
| G | total mass flux, kg/m ² s (lb/ft ² s) |
| h_g | heat transfer coefficient between vapor and tube wall, W/m ² K (Btu/s ft ² °F) |
| i_{fg} | latent heat of vaporization, J/kg (Btu/lb) |
| i_g | enthalpy of vapor at actual temperature, J/kg (Btu/lb) |
| i_{gSAT} | enthalpy of saturated vapor, J/kg (Btu/lb) |
| k_g | thermal conductivity of vapor, W/m K (Btu/s ft °F) |
| L | distance from dryout point, m (ft) |
| Nu | Nusselt number, dimensionless |
| p | absolute pressure, Pa (lb/ft ²) |
| p_c | critical pressure, Pa |
| p_r | $= p / p_c$, dimensionless |
| $\Delta p / \Delta L$ | pressure drop per unit length, Pa/m (lb/ft ² ft) |
| Pr | Prandtl number of vapor, dimensionless |
| q_c | heat flux removed from wall through convection by vapor, W/m ² (Btu/s ft ²) |
| q_{dc} | heat flux removed from wall through liquid droplet cooling W/m ² (Btu/s ft ²) |
| q_w | total heat flux at inside surface of tube, W/m ² (Btu/s ft ²) |
| Re | Reynolds number, dimensionless |
| T_g | actual vapor temperature K (°F) |
| T_{SAT} | temperature of saturated vapor, K (°F) |
| T_w | temperature of wall, K (°F) |
| x_A | actual vapor quality, dimensionless |
| x_c | critical quality, i.e., quality at dryout, dimensionless |
| x_E | vapor quality calculated by assuming thermodynamic equilibrium, dimensionless |
| α | vapor void fraction, dimensionless |
| μ | dynamic viscosity of vapor, N s/m ² (lb/ft s) |
| ρ | density, kg/m ³ (lb/ft ³) |

Subscripts

| | |
|-----|-----------|
| g | of vapor |
| L | of liquid |

Abbreviations

| | |
|-----|--------------------|
| CHF | critical heat flux |
| NA | not applicable |
| NC | not calculated |

REFERENCES

- [1] Andreani, M., and Yadigaroglu, G., Prediction Methods for Dispersed Flow Film Boiling, *Int. J. Multiphase Flow*, vol. 20, suppl., pp. 1-51, 1994.

- [2] Chen, J. C., and Costigan, G., Review of Post-Dryout Heat Transfer in Dispersed Two-Phase Flow, in G. F. Hewitt et al. (eds.), *Post-Dryout Heat Transfer*, CRC Press, Ann Arbor, MI, 1992.
- [3] Shah, M. M., Forced Convection Boiling, in *Genium Heat Transfer and Fluid Flow Data Books*, Genium, Schenectady, NY, 1992.
- [4] Groeneveld, D. C., and Snoek, C. W., A Comprehensive Examination of Heat Transfer Correlations Suitable for Reactor Safety Analysis, in G. F. Hewitt et al. (eds.), *Multiphase Science and Technology*, vol. 2, Hemisphere, Washington, DC, 1986.
- [5] Groeneveld, D. C., Post-Dryout Heat Transfer at Reactor Operating Conditions, Atomic Energy of Canada Ltd. Rep. AECL-4513, 1973.
- [6] Bennet, A. W., Hewitt, G. F., Kearsley, H. A., and Keays, R. K. F., Heat Transfer to Steam-Water Mixtures Flowing in Uniformly Heated Tubes in Which the Critical Heat Flux Has Been Exceeded, AERE R5373, 1967.
- [7] Forslund, R. P., and Rohsenow, W. M., Dispersed Flow Boiling, *J. Heat Transfer*, vol. 90, no. 11, pp. 399-407, 1968.
- [8] Yoder, G. L., Jr., and Rohsenow, W. M., A Solution for Dispersed Flow Heat Transfer Using Equilibrium Fluid Conditions, *J. Heat Transfer*, vol. 105, pp. 10-17, 1983.
- [9] Andreani, M., and Yadigaroglu, G., A 3-D Eulerian-Lagrangian Model of Dispersed Flow Film Boiling Including a Mechanistic Description of the Droplet Spectrum Evolution—1, the Thermal Hydraulic Model, *Int. J. Heat Mass Transfer*, vol. 40, pp. 1753-1772, 1997.
- [10] Groeneveld, D. C., and Delorme, G. G. J., Prediction of Thermal Non-Equilibrium in the Post-Dryout Regime, *Nuclear Eng. Des.*, vol. 36, pp. 17-26, 1976.
- [11] Saha, P., A Non-Equilibrium Heat Transfer Model for Dispersed Droplet Post-Dryout Regime, *Int. J. Heat Mass Transfer*, vol. 23, pp. 483-493, 1980.
- [12] Hein, D., and Kohler, W., A Simple-to-Use Post Dryout Heat Transfer Model Accounting for Thermal Non-Equilibrium, *First Int. Heat Transfer Workshop on Fundamental Aspects of Post-Dryout Heat Transfer*, Salt Lake City, UT, NUREG/CP-0060, 2-4 April 1984.
- [13] Katsounis, A., Post Dryout Correlations and Models Compared to Experimental Data from Different Fluids, in J. Bougard and N. Afgan (eds.), *Heat and Mass Transfer in Refrigeration and Cryogenics*, Hemisphere, Washington, DC, 1987.
- [14] Shah, M. M., A General Predictive Technique for Heat Transfer during Saturated Film Boiling in Tubes, *Heat Transfer Eng.*, vol. 2, pp. 51-62, 1980.
- [15] Hammouda, N., Groeneveld, D. C., and Cheng, S. C., An Experimental Study of Subcooled Film Boiling of Refrigerants in Vertical Upflow, *Int. J. Heat Mass Transfer*, vol. 39, no. 18, pp. 3799-3812, 1996.
- [16] Varone, A. F., and Rohsenow, W. M., Post Dryout Heat Transfer Prediction, *Nuclear Eng. Des.*, vol. 95, pp. 315-327, 1986.
- [17] Bailey, N. A., Collier, J. G., and Ralph, J. C., Post-Dryout Heat Transfer in Nuclear and Cryogenic Equipment, AERE-R7519, 1973. Quoted in G. F. Hewitt, Heat Transfer in High-Quality Two-Phase Flows, in J. J. Ginoux (ed.), *Two-Phase Flows and Heat Transfer with Application to Nuclear Reactor Design Problems*, p. 130, Hemisphere, Washington, DC, 1978.
- [18] Chen, Y., An Investigation of Dispersed Flow Film Boiling Heat Transfer of Water, in J. F. Kefer et al. (eds.), *Experimental Heat Transfer, Fluid Mechanics and Thermodynamics 1991*, Proc. Second World Conference on Experimental Heat Transfer, Fluid Mechanics, Thermodynamics, Dubrovnik, Croatia, 1991.
- [19] Hynek, S. J., Rohsenow, W. M., and Bergles, A. E., Forced Convection Dispersed Flow Film Boiling, M. I. T. Dept. of Mechanical Engineering Rep. 70586-63, 1969.
- [20] Soo, S. L., Pipe Flow, in G. Hetsroni (ed.), *Handbook of Multiphase Systems*, pp. 3-24 to 3-30, Hemisphere, Washington, DC, 1982.
- [21] Shah, M. M., Improved General Correlation for Critical Heat Flux during Upflow in Uniformly Heated Vertical Tubes, *Int. J. Heat Fluid Flow*, vol. 8, pp. 326-335, 1987.
- [22] Shah, M. M., A New Correlation for Heat Transfer during Boiling Flow through Pipes, *ASHRAE Trans.*, vol. 82, part 2, pp. 66-86, 1976.
- [23] Griffith, P., and Wallis, G. B., Two-Phase Slug Flow, *J. Heat Transfer*, vol. 83, pp. 307-320, 1961.
- [24] Hewitt, G. F., Heat Transfer in High-Quality Two-Phase Flows, in J. G. Ginoux (ed.), *Two-Phase Flow and Heat Transfer*, pp. 111-132, McGraw-Hill, New York, 1978.
- [25] Annuziatio, A., Cumo, M., and Pallazi, G., Post-Dryout Heat Transfer in Uncovered Core Accidents, in M. Merilo (ed.), *Thermal Hydraulics of Nuclear Reactors*, vol. 1, pp. 335-342, American Nuclear Society, 1983. (Additional data sheets provided by the authors.)
- [26] Evans, D., Webb, S. W., and Chen, J. C., Axially Varying Vapor Superheats in Convective Film Boiling, *J. Heat Transfer*, vol. 107, pp. 663-669, 1985.
- [27] Gottula, R. C., Nelson, R. A., Chen, J. C., Neti, S., and Sundaram, R. K., Forced Convective Non-Equilibrium Post-CHF Heat Transfer Experiments in a Vertical Tube, ASME-JSME Thermal Engineering Conf., Honolulu, HI, 1983.
- [28] Bhatti, M. S., and Shah, R. K., Turbulent and Transition Flow Convective Heat Transfer in Ducts, in S. Kakac, R. K. Shah, and W. Aung (eds.), *Handbook of Single-Phase Heat Transfer*, chap. 4, Wiley, New York, 1987.
- [29] Hallader, G., and Bannerjee, S., Heat Transfer to Superheated Steam in Round Tubes, AECL unpublished report, 1969. Quoted in Groeneveld and Delorme [10].
- [30] Gnielinski, V., New Equations for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, *Int. Chem. Eng.*, vol. 16, pp. 359-368, 1976. Quoted in Bhatti and Shah [28].
- [31] Pethukov, B. S., and Kirillov, V. V., The Problems of Heat Exchange in the Turbulent Flow of Liquids in Tubes (in Russian), *Teplotenergetika*, vol. 4, no. 4, pp. 63-68, 1958. Quoted in Bhatti and Shah [28].
- [32] Koizumi, Y., Ueda, T., and Tanaka, H., Post-Dryout Heat Transfer to R-113 Upward Flow in a Vertical Tube, *Int. J. Heat Mass Transfer*, vol. 22, pp. 669-678, 1979.
- [33] Meyers, C. A., McClintock, R. B., Silvestri, C. J., and Spencer, R. C., *ASME Steam Tables*, 6th ed., ASME, New York, 1993.
- [34] *ASHRAE Handbook 1981 Fundamentals*, ASHRAE, Atlanta, GA, 1981.
- [35] Vargaftik, N. B., *Tables on the Thermophysical Properties of Liquids and Gases*, 2d ed., Wiley, New York, 1975.
- [36] McCarty, R. D., *Thermophysical Properties of Helium-4 from 2 to 1500 K with Pressures to 1000 Atmospheres*, NBS-TN-631, National Technical Information Service, Springfield, VA, 1972.
- [37] Beaton, C. F., and Hewitt, G. F., *Physical Property Data for the Chemical and Mechanical Engineer*, Hemisphere, New York, 1989.
- [38] Nishikawa, K., Yoshida, S., Mori, H., and Takamatsu, H., An Experiment on the Heat Transfer Characteristics in the

Post-Burnout Region at High Subcritical Pressures, *ASME-JSME Thermal Engineering Conf.*, vol. 1, pp. 159–164, 1983.

[39] Spiegler, P., Hopenfeld, J., Silberberg, M., Bumpus, C. F., and Norman, A., Onset of Stable Film Boiling and Foam Limit, *Int. J. Heat Mass Transfer*, vol. 6, pp. 987–994, 1963. Quoted in Groeneveld and Snoek [4].

[40] Nijhawan, S., Chen, J. C., and Sundaram, R. K., Parametric Effects on Post-Dryout Heat Transfer, ASME Paper 80-WA/HT-50, 1980.

[41] Nijhawan, S., Chen, J. C., Sundaram, N. K., and London, E. J., Measurement of Vapor Superheat in Post-Critical Heat Flux Boiling, *J. Heat Transfer*, vol. 102, pp. 465–470, 1980.

[42] McAdams, W. H., *Heat Transmission*, 3d ed. McGraw-Hill, New York, 1954.

[43] Kastner, W., Kohler, W., Kratzner, W., and Hein, D., Wärmeübergang in Post Dryout-Bereich in Vertikalen und Horizontalen Rohren bei gleichförmiger Beheizung KWU-Abschlussbericht zum Forderungsvorhaben, BMFT ET1409 A, 1981.

[44] Kohler, W., Einfluss des Benetzungszustandes der Heizfläche auf Wärmeübergang und Druckverlust in einem Verdampferrohr, Dr. Ing. Dissertation, Technical University of Munich, Germany, 1984.

[45] Schnittger, R. B., Untersuchungen zum Wärmeübergang bei Vertikalen und Horizontalen Rohrströmungen in Post-Dryout-Bereich, Dissertation, Universität Hannover, Germany, 1982.

[46] Ling, C. H., Becker, K. M., Hedberg, S., and Strand, G., Temperature Distribution for the Post-Burnout Regime in a Round Tube, Royal Institute of Technology, Laboratory of Nuclear Engineering Rep. KTH-NEL-16, Stockholm, 1971. Quoted in Saha [11].

[47] Nishikawa, K., Yoshida, S., Mori, H., and Takamatsu, H., Post-Dryout Heat Transfer to Freon in a Vertical Tube at High Subcritical Pressures, *Int. J. Heat Mass Transfer*, vol. 29, pp. 1245–1251, 1986.

[48] Era, A., Heat Transfer Data in the Liquid Deficient Region for Steam-Water Mixtures at 70 kg/cm² Flowing in Tubular and Annular Conduits, CISE-R-184, 1967. Quoted in Groeneveld and Delmore [10] and Saha [11].

[49] Herkenrath, H., Mork-Morkenstein, P., Jung, U., and Weckerman, F. J., Wärmeübergang an Wasser bei Erzwungener Strömung in Druckbereich von 140 bis 250 Bar, EUR-3658d, 1967. Quoted in Groeneveld and Delorme [10].

[50] Bishop, A. A., Sandberg, R. O., and Tong, L. S., Forced Convective Heat Transfer at High Pressure after the Critical Heat Flux, ASME Paper 65-HT-31, 1965.

[51] Schmidt, K. R., Thermodynamic Investigations of Highly Loaded Boiler Heating Surfaces, AEC-Tr-4033, 1960. Quoted in Groeneveld and Delorme [10].

[52] Swenson, H. S., Carver, J. R., and Szoeko, G., The Effects of Nucleate Boiling versus Film Boiling on Heat Transfer in Power Boiler Tubes, *J. Eng. Power*, pp. 365–371, October 1962.

[53] Ogata, H., and Sato, S., Forced Convection Heat Transfer to Boiling Helium in a Tube, *Cryogenics*, vol. 14, p. 375, 1974.

[54] Glickstein, M. R., Personal communication, 1967. Quoted in Hynek et al. [19].

[55] Lavery, W. F., and Rohsenow, W. M., Film Boiling of Saturated Nitrogen Flowing in a Vertical Tube, *J. Heat Transfer*, vol. 89, no. 2, pp. 90–98, 1967.

[56] Cumo, M., Farello, G. E., and Ferrai, G. The Influence of Curvature in Post Dry-Out Heat Transfer, *Int. J. Heat Mass Transfer*, vol. 15, pp. 2045–2062, 1972.

[57] Groeneveld, D. C., The Thermal Behavior of a Heated Surface at and Beyond Dryout, AECL-4309, 1972.

[58] Janssen, E., and Kervinnen, J. A., Film Boiling and Rewetting, General Electric Rep. NEDO-20975, 1975. Quoted in Saha [11].

[59] Cheng, X., Erbacher, F. J., and Muller, U., Critical Heat Flux in Uniformly Heated Vertical Tubes, *Int. J. Heat Mass Transfer*, vol. 40, no. 12, pp. 2929–2939, 1997.

[60] Kefer, V., Kohler, W., and Kastner, W., Critical Heat Flux (CHF) and Post-CHF Heat Transfer in Horizontal and Inclined Evaporator Tubes, *Int. J. Multiphase Flow*, vol. 15, no. 9, pp. 385–392, 1989.

APPENDIX I: HAND CALCULATIONS WITH THE PRESENT CORRELATION

The steps in calculating x_A by using Figure 1 are as follows:

1. Calculate Fr_L and x_c by a suitable correlation, such as that of Shah (1987).
2. Locate x_c on the equilibrium line AB .
3. If this point is below the intersection of the Fr_L curve and line AB , x_A is read on the equilibrium line AB and then directly along the Fr_L curve.
4. If this point is above the intersection of the Fr_L curve and the equilibrium line, draw a tangent to the Fr_L curve from the x_c point on the the line AB . x_A is then read along the tangent up to the point of contact with the Fr_L curve, and beyond that along the Fr_L curve.

To illustrate these simple steps, the following two cases are shown in Figure 1A:

1. $Fr_L = 1.3$, $x_c = 0.1$
2. $Fr_L = 10$, $x_c = 0.6$

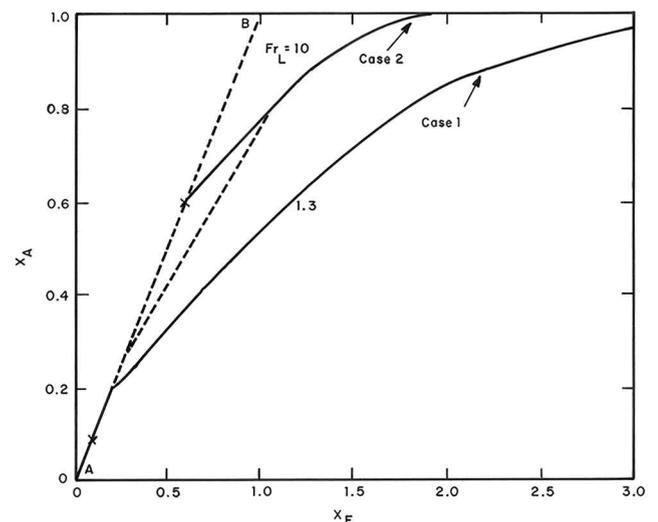


Figure 1A Prediction of x_A for two cases. Predicted x_A is read along the solid lines.

APPENDIX 2: ADDITIONAL INFORMATION FOR HELP IN CALCULATIONS

Calculations When Heat Flux Is a Dependent Variable

If heat flux is an independent variable, as is the case in nuclear reactors and electrode boilers, heat flux and thus Bo are easily calculated. When heat flux is a dependent variable, as is the case when heating is by liquid or condensing vapor, calculations have to be done by trial and error. A heat flux is assumed, the film boiling heat transfer coefficient is calculated, the overall heat transfer coefficient is calculated, and then the heat flux is calculated. If the calculated heat flux differs from the assumed value, a new value of heat flux is assumed and the calculations are repeated. This procedure is continued until convergence is achieved.

Calculation of x_c

The dryout quality can be calculated using any predictive technique for critical heat flux, as the two are related. Numerous such predictive techniques ranging from interpolation formulas to theoretical derivations are available. It is suggested that the reader refer to books and articles dealing with this subject to select the best method for the application. The review article by Groeneveld and Snoek [4] is a good example. Some guidance is provided in the following.

For CHF upflow in uniformly heated vertical tubes, among the most verified is the Shah correlation [21].

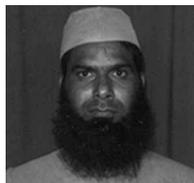
In a recent comparison of several leading predictive techniques, Cheng et al. [59] found it to give the best agreement with data.

Predictive techniques for horizontal tubes are scarce. Kefer et al. [60] have given a simple correlation which agrees with their own data for water. The extent of its agreement with other data is not known.



M. Mohammed Shah obtained degrees in mechanical engineering from Aligarh Muslim University in India and California State University at Los Angeles. He has developed a number of general predictive techniques including those for heat transfer and critical heat flux during boiling and condensation in tubes, heat transfer and critical heat flux during pool boiling of liquid metals, and

heat transfer in fluidized beds. He is also involved in engineering and research in the field of air conditioning and refrigeration. He has made many contributions to engineering handbooks. His previous positions include scientist-in-charge of the Air Conditioning and Refrigeration Department of the Central Mechanical Engineering Research Institute, Durgapur, India, and Principal Engineer with Ebasco Services, Inc., in New York, NY. He is a member of the ASME and a fellow of ASHRAE. He is licensed as a Professional Engineer in California and New York.



M. Altamush Siddiqui is a reader in the Department of Mechanical Engineering, Aligarh Muslim University, India. He obtained his Ph.D. from the same university. Other areas of his research have included boiling in thermosyphons, condensation heat transfer, shell-and-tube heat exchangers, solar thermosyphons and solar refrigeration, advanced cycles for improving the efficiency of

absorption refrigeration systems, and the search for new refrigerants for the absorption cycle. He has published extensively and has guided the research work of many M.S. and doctorate degree candidates.