*Int. J. Heat Mass Transfer.* Vol. 34, No. 2, pp. 473–481, 1991
Printed in Great Britain

0017–9310/91 \$3.00+0.00


# Dryout stability and inception at low flow rates

R. B. DUFFEY and E. D. HUGHES

Energy and Systems Technology, EG&G Idaho, Inc., P.O. Box 1625, Idaho Falls, ID 80415-2403, U.S.A.

(*Received* 26 *January* 1990 *and in final form* 26 *April* 1990)

**Abstract**—A theoretical model for the heat flux required to maintain a stable hot patch on a heated surface cooled by a falling liquid film is developed. The theoretical model is based on an existing analytical solution of the two-dimensional heat equation with boundary conditions supplied by heat transfer coefficient correlations appropriate to the fluid conditions at the hot dry patch and at the wet region from the patch. Using physically reasonable correlation forms for the dry patch and wet region heat transfer coefficients, the heat flux is expressed as a function of the liquid film Reynolds number. Existing experimental data were compiled, collected, and used to obtain the correlating lines. The data cover the one-dimensional (low Biot number) region, and indicate that laminar and turbulent liquid film conditions occur. Corresponding correlating equations are obtained from the analytical results. For laminar flow, the dryout heat flux is $q_D = 0.27Re^{0.5}$, and for turbulent flow $q_D = 0.017Re^{0.9}$, with an uncertainty of the order of $\pm 50\%$.

## 1. INTRODUCTION

Consider pouring water down a heated tube or annulus to keep it cool and wetted. At some flow rate, the wall will become dry and if heating continues, the heated-surface temperature will rise. The prediction of the dryout heat flux for downflow is presently based entirely on experimental data. At low flow rates, the occurrence of dryout is characterized by initial fluctuations in wall temperature, followed by a rapid increase. Shires *et al.* [1], Fujita and Ueda [2], Gogonin *et al.* [3], and Steimke [4] have determined thermal conditions by imposing a steady downflow of subcooled water, and raising the annulus power until dryout was observed. At present, no theory is available for predicting the dryout occurrence, although Lussie [5] has reviewed available film flow breakdown and dryout data and proposed a purely empirical fit to the then available data. Berta *et al.* [6], Shires *et al.* [1], and Steimke [4] noted that changes in the (boiling) flow regimes, and the presence of countercurrent and churn-turbulent flows, can be associated with the dryout occurrence. Differences in dryout power for steel and aluminum surfaces also have been proposed.

This paper presents a new treatment of dryout for this downflow case based on Kovalev's [7, 8] classic thermal stability theory. In this analysis, the heat transfer is coupled to conduction in the wall to predict the heat flux that is needed to create a stationary but metastable dryout patch [9, 10]. This approach is consistent with the observation of Shires *et al.* [1] that dry patches occurred at high subcoolings in downward flow. That is, dry patches were formed well before the bulk liquid reached the saturation state. This has been subsequently observed and confirmed by Gogonin *et al.* [3] and Fujita and Ueda [2]. This analysis differs from the theory of film breakdown or dry patch formation based on purely hydrodynamic force considerations (e.g. Zuber and Staub [11] and Fujita and Ueda [2]). Film breakdown is likely to be a necessary, but not sufficient condition for surface dryout, and indeed the heat flux at dryout has been shown by Fujita and Ueda [2] to be systematically greater. It also differs from the more recent analyses of the rewetting rate, which do not consider the steady-state dryout patch solution.

Our earlier attempts to utilize a purely hydrodynamic theory based on two-phase drift-flux modelling were neither fruitful nor gave the observed trends. The present use of the stability theory to correlate the data is quite new. It leads to predictions of the effect of surface conductivity, provides a simple method for interpreting the available data, and gives confidence in understanding the physics of the dryout phenomena. The important point is that local dryout can occur before the total flow reaches the saturation or boiling point, as observed in the experiments.

## 2. THEORY

### 2.1. *Model and assumptions*

We follow the analysis of Porthouse [10], which is based on the pioneering studies of Kovalev [7] and Semeria and Martinet [9]. The approach is to solve the two-dimensional thermal conduction equation in the heated surface, subject to heat flux boundary conditions.

Consider the formation of a hypothetical region of film boiling, or dry patch, on an initially wetted wall; the patch will grow if the heat flux is sufficiently large and collapse if not. The onset of the dry patch could be due to purely hydrodynamic or flow-regime effects causing local deficiencies of water (e.g. countercurrent flooding or film flow deficiency), but the exact initiating mechanism does not yet need to be defined. The patch is classically metastable, and the dryout condition is hence taken to be that flux that gives a (constant-sized) patch of film boiling.

**NOMENCLATURE**

| | | | |
|---|---|---|---|
| $A_f$ | channel flow area | $T_s$ | sink temperature |
| $A_w$ | wall area for heat transfer | $T_{sat}$ | coolant saturation temperature |
| $Bi$ | Biot number | $Nu$ | Nusselt number |
| $C_p$ | specific heat of coolant | $q$ | heat flux. |
| $D$ | diameter | | |
| $G$ | mass flux | Greek symbols | |
| $H$ | heat transfer coefficient for wetted region | $\Gamma$ | film flow rate |
| $h$ | heat transfer coefficient for dry region | $\mu$ | kinematic viscosity. |
| $j_l$ | coolant flow velocity | | |
| $k$ | thermal conductivity | Subscripts | |
| $P$ | perimeter | e | equivalent heated |
| $q_D$ | dryout heat flux | h | heated |
| $Q_D$ | dryout power | i | inner rod |
| $Re$ | Reynolds number | o | outer tube |
| $T$ | temperature | w | wetted |
| $T_c$ | coolant temperature | y | equivalent hydraulic |
| $T_o$ | effective interface temperature | $\varepsilon$ | metal surface thickness. |

The following assumptions are made:

(a) The surface is homogeneous, and the curvature small over the scales of interest, so we may treat the problem as a two-dimensional plane.

(b) Two-dimensional conduction occurs within the surface, with an average surface heat transfer coefficient, $H$, existing on the wetted region, and a much lower value, $h$, on the dry, or film-boiling region.

(c) The interface between these wet and dry regions is characterized by some characteristic temperature, $T_o$ (a Leidenfrost, interface, minimum film boiling, or rewetting temperature).

(d) Liquid deficient or dry regions occur due to hydrodynamic causes, but the dryout condition is for a transiently stable patch.

(e) The patch is modelled as semi-infinite, which is justified since the thermal boundary layer at the edges (of order $\sim(k\varepsilon/h)^{1/2}$) is of small extent compared to the patch dimensions.

We note here that although these assumptions idealize the problem they do enable a tractable solution. Mathematically, the Dirac delta function in heat transfer leads to steep Von Neuman conditions at the interface; but the heat transfer coefficients are averaged with respect to temperature, and do reflect the orders of magnitude observed. The use of a constant interface temperature also simplifies the problem, but is also consistent with the overall shape of the Nukiyama boiling curve.

### 2.2. *Solutions for the two-dimensional case*

The assumptions listed above enable an analytical solution to be obtained to the Fourier heat diffusion equation for a stationary patch. The steady-state heat equation

$$\nabla^2 T = 0 \tag{1}$$

which in two-dimensions is (see Fig. 1)

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} = 0 \tag{2}$$

subject to the boundary conditions of constant temperature far from the interface

$$x \to -\infty, \quad y = \varepsilon, \quad T \to T_s + \frac{q}{H}$$

$$x \to +\infty, \quad y = \varepsilon, \quad T \to T_s + \frac{q}{h} \tag{3a}$$

where the interface is given by

$$x = 0, \quad y = \varepsilon, \quad T = T_o. \tag{3b}$$

The heat flux boundary conditions are

$$y = 0, \quad k\left.\frac{\partial T}{\partial y}\right|_o = -q$$

$$x < 0, \quad y = \varepsilon, \quad k\left.\frac{\partial T}{\partial y}\right|_\varepsilon = -H(T_\varepsilon - T_s)$$

$$x > 0, \quad y = \varepsilon, \quad k\left.\frac{\partial T}{\partial y}\right|_\varepsilon = -h(T_\varepsilon - T_s). \tag{3c}$$

In equation (3), $T_s$ is the sink temperature and $T_o$ the temperature at the edge of the dry path.

Solving equation (2) by the method of separation of variables and eliminating constants of integration using the boundary conditions of equation (3) enables the explicit solution for the temperature profiles to be obtained according to refs. [10, 12]. These have the algebraic form of the summation of an infinite series.

Forming an energy balance such that the energy

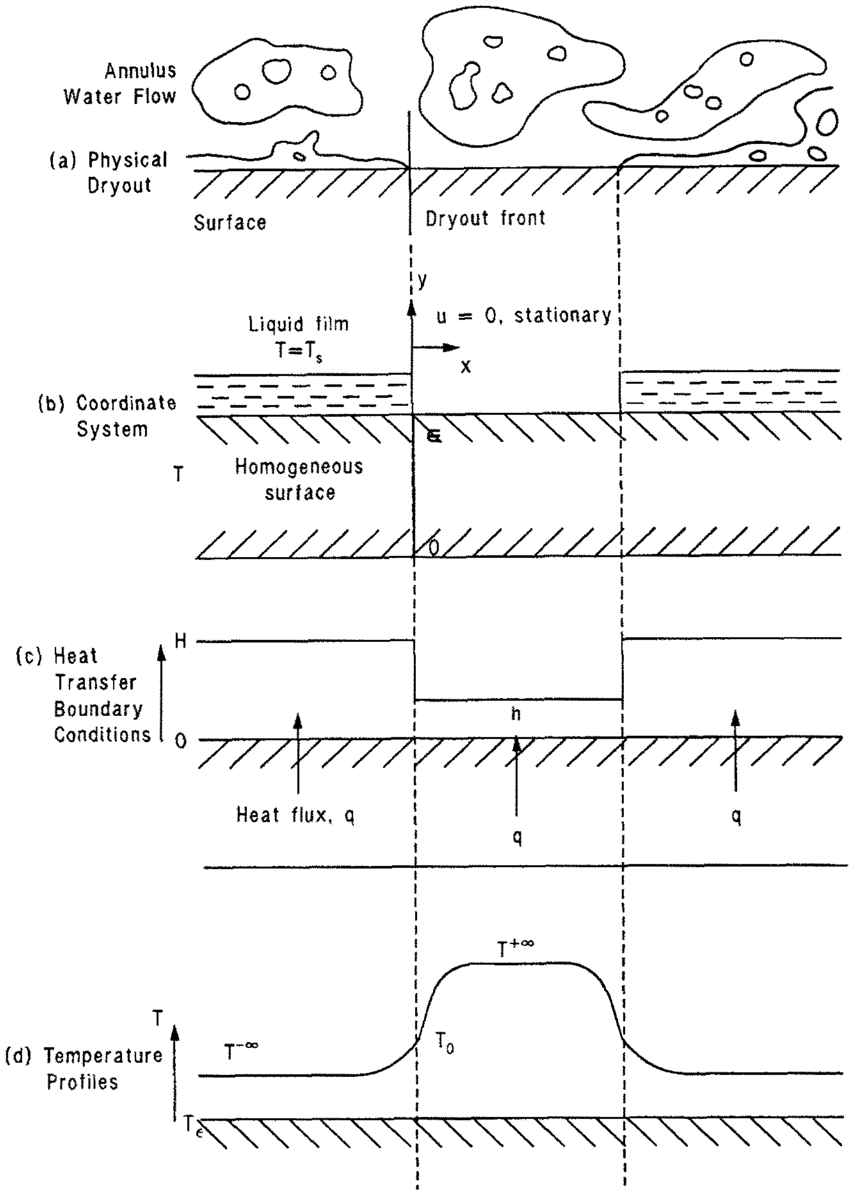


FIG. 1. Theoretical model definition for analysis of dryout.

removed in the wet and dry regions is equal (i.e. the front is stationary), and drawing from refs. [9, 10] it can be shown that

$$\frac{H}{\theta}\left(T_o-T_s-\frac{q_D}{H}\right)=\frac{h}{\phi}\left(\frac{q_D}{h}-T_o+T_s\right) \tag{4}$$

where

$$\theta \tan \theta\varepsilon = \frac{H}{k}$$

$$\phi \tan \phi\varepsilon = \frac{h}{k} \tag{5}$$

and $q_D$ is the *dryout* heat flux. Rearranging equation (4) we have

$$q_D = \left\{\frac{h\theta+H\phi}{\theta+\phi}\right\}(T_o-T_s) \tag{6}$$

which is the general result for dryout.

### 2.3. *The dryout heat flux equations*

To make this elegant solution useful for correlating purposes we proceed to simplify. The limiting cases for equation (6) are when the conduction is essentially one-dimensional (i.e. no $y$ gradients for a high-conductivity, low Biot number material like copper or aluminum), or highly two-dimensional (high Biot number). It has been shown [12] that for the former case, if and only if

$$h, H \ll \frac{\pi^2 k}{4\varepsilon}, \text{ then } \theta = \left(\frac{H}{k\varepsilon}\right)^{1/2}, \phi = (h/k\varepsilon)^{1/2} \quad (7)$$

and for the latter case, if and only if

$$h, H \gg \frac{\pi^2 k}{4\varepsilon}, \text{ then } \theta = \phi \simeq \frac{\pi}{2\varepsilon}. \quad (8)$$

Thus, substituting equations (7) and (8) into equation (6) we find the dryout heat fluxes, $q_{D1}$ and $q_{D2}$ for the one- and two-dimensional limits, respectively, are

$$q_{D1} = (Hh)^{1/2}(T_o - T_s) \quad (9)$$

and

$$q_{D2} = \left(\frac{H+h}{2}\right)(T_o - T_s). \quad (10)$$

We also note that for $h \ll H$, then

$$\frac{q_{D2}}{q_{D1}} = \frac{1}{2}\left(\frac{H}{h}\right)^{1/2}. \quad (11)$$

Note these solutions predict a different variation of dryout flux with heat transfer coefficient, varying from a square root to linear dependency for high and low conductivity surfaces, respectively, with a continuous transition region between them. Also the result is independent of the thermal inertia when the patch is stable, which is to be expected for the steady-state temperature distribution at that condition. Obviously, an unstable patch cools (and rewets) or heats (and grows), and the thermal inertia determines the heating or cooling rate.

The values of $h$ and $H$ at low flow depend on *flow* parameters, such as film flow rate and subcooling as observed in quenching studies [12, 13]. In fact, we argue here a direct extension of this dependency to the present case of downflow in an annulus so we can make use of the extensive previous information, and regard the present case as analogous, that is, dryout of thin liquid films on a surface can be treated theoretically in a similar manner as quenching and rewetting.

## 3. COMPARISONS WITH DATA FOR STEEL AND ALUMINUM SURFACES

### 3.1. *Form of the heat transfer coefficients*

We need to examine the range of possible flow variations in the heat transfer coefficients.

Of course, we do not know the precise variations and values to adopt, so we proceed utilizing intuitive reasoning and physical postulates to reduce the (limitless) range of possibilities, and then test the veracity of these arguments against data in the traditional manner. If we choose similar dependencies as adopted by Porthouse [10], derived from extensive upflow and downflow (falling film) studies on quenching and post-dryout heat transfer for a subcooling, $\Delta T_s$, we would have

$$H = H_s + \Phi G \Delta T_s$$
$$= \sigma G^m + \Phi G \Delta T_s$$

where $G$ is the coolant mass flux, and $H_s$ an effective heat transfer coefficient to saturated liquid with weak flow dependency (to order one-fifth power or less, $m \sim 0.2$), and $\sigma$ and $\Phi$ are constants. More generally, the film flow heat transfer coefficient, $H$, varies with the Reynolds number as $(4\Gamma/\mu)^m$, where $\Gamma$ is the film flow rate per unit *heated* perimeter, so that

$$H \propto Re^m. \quad (12)$$

Specifically, for laminar flow, the classic Nusselt draining film result of Chun and Seban [14] is

$$H \propto Re^{-1/3}$$

and for wavy films

$$H \propto Re^{-0.22}.$$

For turbulent flow, the empirically observed variation for evaporating films is $H \propto Re^{-0.4}$.

The heat transfer coefficient in the dry patch region, $h$, corresponds to post-dryout convective film boiling where the variation with Reynolds number corresponds to studies by White and Duffey [15] and Sun *et al.* [16] and is observed to be

$$h \propto Re^n \quad (13)$$

where $n$ lies in the range 0.6–1.4 for quenching from below or above, respectively.

Substituting equations (12) and (13) into equation (9), we obtain, straightforwardly

$$q_{D1} = \text{const.} \times Re^{(m+n)/2}(T_o - T_s)$$
$$= a_1\, Re^{(m+n)/2}(T_o - T_s), \text{ say} \quad (14a)$$

and, substituting equations (12) and (13) into equation (10)

$$q_{D2} = a_2[Re^m + a_3\, Re^n](T_o - T_s). \quad (14b)$$

Typical values for $(T_o - T_s)$ lie in the range 50–300°C, depending on the conditions and definition. As shall be seen, we do not need a precise value, and hence can sidestep the usual controversial arguments over its physical meaning also. We may also write equations (14a) and (14b) in non-dimensional form

$$Nu_{\lambda+1} = a_{\lambda+1}[Re^{(m-(\lambda-1)n)/(2-\lambda)} + a_{\lambda+2}\lambda\, Re^n] \quad (15)$$

where $\lambda = 0$, 1 for the one- and two-dimensional limits, respectively.

We conclude, therefore, that there are no first principle means of determining $(m-(\lambda-1)n)/(2-\lambda)$ other than by plotting the data according to the theory (i.e. $q_D$ vs $Re$). For the one-dimensional ($Bi < 1$) case, we *expect* the flow dependency to be such that $0.15 < (m+n)/2 < 0.65$ for laminar subcooled film flows, and $0.5 < (m+n)/2 < 0.9$ for turbulent films, based on the above summary of available information.

Table 1. Relevant parameters for the experimental data

| Derived dimensions | Shires *et al.* [1] (1964) Heated rod in annulus | Gogonin *et al.* [3] (1979) Heated rod in annulus | Steimke [4] (1989) Heated shroud in annulus | Berta *et al.* [6] (1989) Heated rod in annulus | Fujita and Ueda [2] (1978) Flow outside rod |
|---|---|---|---|---|---|
| Heated perimeter, $P_h$ (m) | 0.0299 | 0.088 | 0.245 | 0.0499 | 0.0503 |
| Wetted perimeter, $P_w$ (m) | 0.1297 | 0.339 | 0.466 | 0.1297 | 0.0503 |
| Equivalent diameter, $D_e$ (m) | 0.0095 | 0.052 | 0.0075 | 0.0095 | 0.0160 |
| Length, $L$ (m) | 3.66 | 1.0 | 3.96 | 3.66 | 0.6, 1.0 |
| Flow area, $A_f$ (m) | $3.088\times10^{-4}$ | $4.41\times10^{-3}$ | $8.74\times10^{-4}$ | $3.088\times10^{-4}$ | N/A |

### 3.2. *Comparison of theory and experiment: local heat flux*

There are five data sets available for comparison to the model which cover very wide ranges of geometric and operating parameters and are summarized in Table 1, *and all are for* $Bi \gtrsim 1$ ($\lambda = 0$). These data have been reported by Shires *et al.* [1], Fujita and Ueda [2], Gogonin *et al.* [3], Steimke [4], and Berta *et al.* [6]. The latter have been summarized in a plot by Muhlbaier.† These workers poured known flow rates of water down a uniformly electrically-heated annulus, with stainless steel or aluminum inside walls. The observed power level for dryout was converted to heat flux by dividing by surface area (which is strictly not the *local* flux at dryout).

The data of Shires *et al.* [1] were obtained for the case of water flowing down a heated rod of 15.875 mm (0.625 in.) diameter, contained inside an unheated shroud of 2.54 mm (2.00 in.) diameter. The heated length of the rod was 3.667 m (12 ft.), and the experiments were all conducted at atmospheric pressure. The water flow rate varied from about $1.8\times10^{-3}$ kg s$^{-1}$ (0.004 lbm s$^{-1}$) to about $1.8\times10^{-2}$ kg s$^{-1}$ (0.04 lbm s$^{-1}$), with inlet water temperature ranging from 294 to 366 K. Shires *et al.* [1] also conducted other tests in rod bundles that are not considered here.

The data of Fujita and Ueda [2] were obtained for water flow down the outside of 16 mm stainless steel tubes 0.60 and 1.00 m long. The water flow rate varied from about $4.5\times10^{-3}$ to about 2 kg s$^{-1}$, with water inlet temperatures from 295 to 352 K.

The data of Gogonin *et al.* [3] were also obtained for the case of a 28 mm diameter heated rod inside an unheated 80 mm diameter shroud. Test section heated lengths of 0.50, 1.0, and 2.2 m were used (for compactness, we have only used the data from the 0.50 and 1.0 m heated lengths), and the experiments were conducted at atmospheric pressure. Gogonin *et al.* [3] reported the liquid flow rate as a volumetric flow per unit heated perimeter, and these values were converted to the liquid velocity via the known flow area by

$$J_l = \frac{G_l P_h}{A_f}$$

† Private communication with D. R. Muhlbaier, Westinghouse Savannah River Company (1989).

where $G_l$ is the irrigation flow reported by Gogonin *et al.* [3]. These data are for very low $J_l$ and $\Gamma$ (mass flow rate per unit *heated* perimeter) values. The water inlet temperature for the tests was 298.15 K (25°C). Gogonin *et al.* [3] stated that dry patches formed after the water had been heated by only 5 K above the inlet temperature, and thus the resulting dryout power was very low.

The data of Berta *et al.* [6] are for the case of a 38.1 mm heated rod inside a 50.8 mm diameter shroud, with a heated length of 3.81 m. The tests were conducted at about 0.090 MPa (13 psia), and the water flow rate varied from about 0.15 to 0.5 kg s$^{-1}$. The inlet temperature ranged from 314 to 317 K.

The data reported by Steimke [4] are for the case of flow down an annulus in which the outer tube was heated, with an internal diameter of 7 mm and an external diameter of 7.79 mm, with a water flow from 0.24 to 0.72 kg s$^{-1}$ and inlet at a temperature of 292 K.

Other data for downward flow at low coolant flow rates have been reported by Mishima and his coworkers [17–19]. Most of these data are for the case of net bulk boiling of the coolant and will not be considered further in this paper.

### 3.3. *Correlation of high flow data*

Consider first the data for the higher heat fluxes and flow velocities in Fig. 2 plotted as a function of superficial flow velocities (which is proportional to *Re*). The data show considerable scatter, but we proceed to strike a line through them based on the theoretical prediction and form of equation (14a). A typical correlation is

$$q_{Dl} = Kj_l^{(m+n)/2} \tag{16}$$

$$\simeq 200 j_l^{0.5} \tag{17}$$

where $q_{Dl}$ is in kW m$^{-2}$, and $j_l$ is in m s$^{-1}$, for the range $0.005 < j_l < 1.0$ m s$^{-1}$. The agreement shown in Fig. 2 is striking; for comparison, the purely empirical correlation of Lussie [5] is based on film flow rate and would lie below the data. Therefore, we may conclude that film flow breakdown by itself may be a necessary but not a *sufficient* condition for dryout.

The aluminum data of Berta *et al.* [6] also seem to agree well. A lower bound to all these data, which

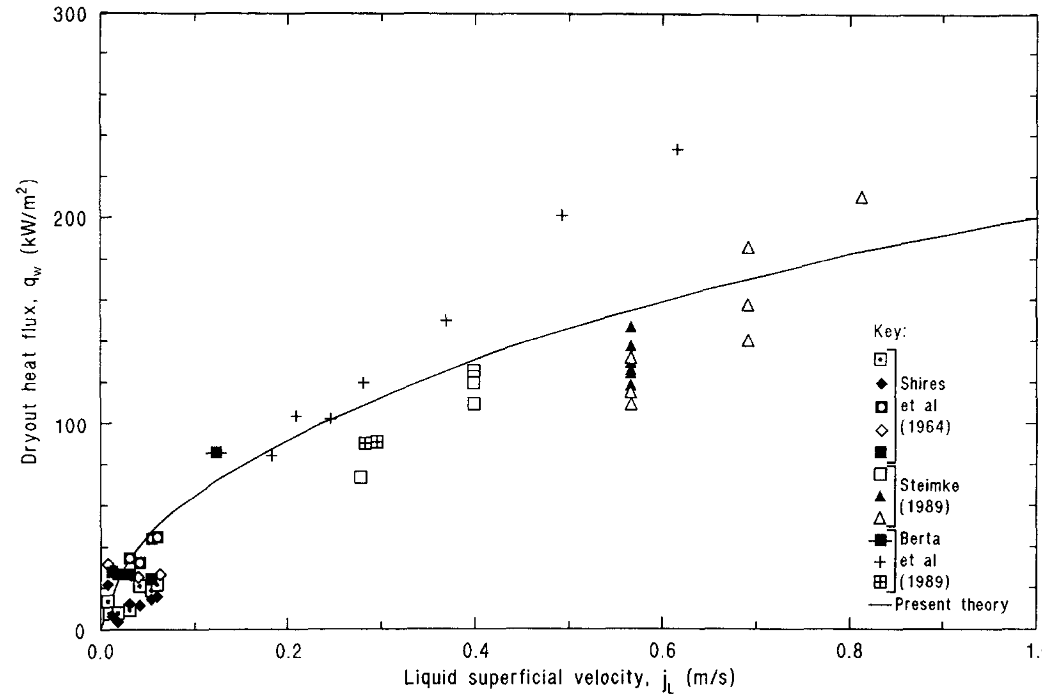


Fig. 2. Comparison of theory to data.

may be of interest and use for design purposes, is given by

$$q_{DI}|_{min} = 105j_l^{0.5}. \quad (18)$$

Overall, we regard the agreement as surprisingly satisfactory. We expect and predict high Biot number surfaces (e.g. silica), to possess a different dependence in flow rate.

### 3.4. *General correlation of available world data*

In order to determine the variations over a wider flow range, Fig. 3 was prepared and covers two orders of magnitude in heat flux and over three in film flow Reynolds number, $4\Gamma/\mu$. The latter was chosen because superficial velocity is not meaningful in large annuli at low flow rates because all the water is on the wall and the theory predicts a dependency of Reynolds number. A transition in the flow dependency can be observed at $Re \sim 10^3$, implying a change in film flow heat transfer from laminar to turbulent, in accord with the observations of Fujita and Ueda [2]. An overall best fit line for turbulent flows ($Re > 1000$) is given by

$$q_D = 0.017Re^{0.9} \quad (19)$$

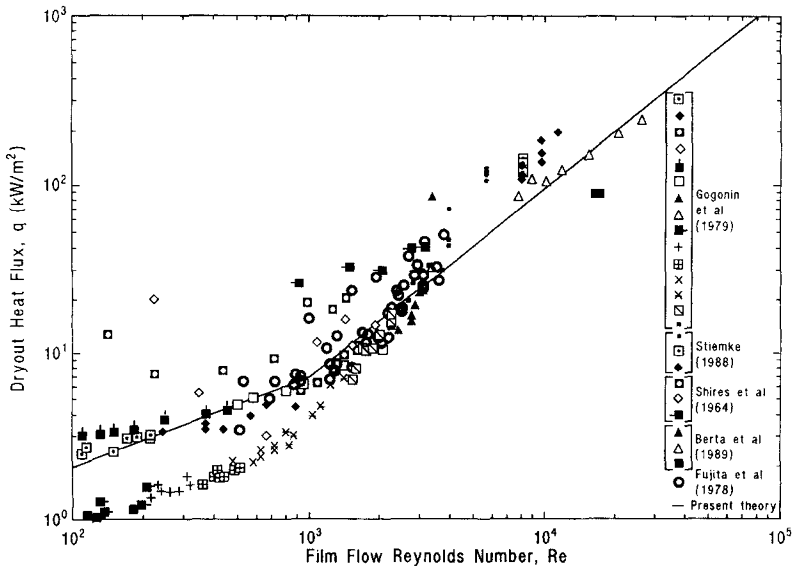


Fig. 3. Correlation of theory to data.

over the range $7 < q_w < 10^3$ kW m$^{-2}$ and $10^3 < Re < 10^5$, which is an extremely simple result. For the one-dimensional data given, this value of 0.9 for $(m+n)/2$ falls at the upper end at the expected range 0.5–0.9 for turbulent films ($Re > 10^3$). For laminar flows, the flow dependency varies as $O[Re^{1/2}]$, which falls within the expected range of 0.15–0.65 (i.e. for $Re < 1000$), and the dryout heat flux is

$$q_D = 0.27 Re^{1/2}. \tag{20}$$

Although a range of exponents is possible within the data scatter, equations (19) and (20) give the transition between the laminar and turbulent fits.

### 3.5. *The saturation ratio*

An alternative presentation of the data is to determine the ratio, $R^*$, of the observed annulus dryout power, $Q_D$, to the power required to raise the inlet flow to boiling point (saturation) (i.e. $Q_s \simeq GA_f C_p \Delta T_s$). There is then a simple 'design' parameter given by

$$R^* = \frac{Q_D}{Q_s} \tag{21}$$

called here the 'saturation ratio'.

Now, assuming there is a relation between the local flux and the total power at dryout then

$$Q_D \propto \int_A q_D \, dA. \tag{22}$$

If some proportionality, $\psi$, holds to account for all quantities and variations in the flow distribution, power peaking, geometry and deviation of the local dryout flux from the annulus average, then

$$Q_D = \psi q_D. \tag{23}$$

From equations (9), (19), and (21)

$$R^* = \frac{\psi (Hh)^{1/2} (T_o - T_s)}{GA_f C_p \Delta T_s} \tag{24}$$

or from equation (16)

$$R^* = \psi \frac{K j_1^{(m+n)/2-1}}{\rho_1 A_f \Delta T_s}. \tag{25}$$

Now as observed by Gogonin *et al.* [3] and Shires *et al.* [1] dry patches *form* before the total flow is saturated ($R^* < 1$). Therefore, this type of comparison is not useful in relation to determining local dryout.

Two observations should be made. Firstly, we expect and predict a secondary variation due to subcooling effects to be still present in these data plots (not yet accounted for). Secondly, we also predict that at very high flows ($j_1$ increasing), the $R^*$ curve will increase again due to the second (subcooling) term in the expression for $H$. The data presumably do not cover this range at this time, and are saturated liquid near the dryout point. Data covering a wider range of flows are needed to further test the correlation.

## 4. DISCUSSION

### 4.1. *Further thoughts*

The present work can be regarded as a significant new application, but of course not complete at this time. It is, however, largely consistent with all the available observations and data trends. Further theoretical refinement and extension of the data base are both desirable; determination of *local* heat fluxes at dryout patches is also clearly needed. In terms of a saturation ratio, the data do show that this may be less than unity, which confirms the hypothesis of local dryout as preceding bulk boiling for some of the data.

### 4.2. *Heat fluxes*

The question of whether the combined heat flux *magnitudes* implied by the theory and data correlation equation (17) are at all reasonable requires some consideration.

We have argued that $h$ is some typical film boiling or post-dryout value, which for low flows is typically less than 1 kW m$^{-2}$ K$^{-1}$. Now the wet side coefficient, $H$, is bounded by either some nucleate boiling value ($\sim 10^2$ kW m$^{-2}$ K$^{-1}$) or a value due to conduction across a non-boiling wetting liquid film. A simple estimate of the latter can be derived for laminar flow from the classic Nusselt draining film thickness, $\delta$, neglecting interface shear, which is given by $(3\mu\Gamma/g\rho_l^2)^{1/3}$, where $\mu$ is the liquid viscosity and $\Gamma$ the film flow rate per unit perimeter. For the film flow rates of present interest, $\Gamma \sim 0.5$ kg s$^{-1}$ m$^{-1}$, which gives a heat transfer coefficient for the film flow $(k/\delta) \sim 20$ kW m$^{-2}$ K$^{-1}$. Utilizing the evaporating film correlation of Chun and Seban [14] results in a turbulent convective coefficient of $\sim 8$ kW m$^{-2}$ K$^{-1}$ as an alternate estimate (McCreery).†

Thus, from these estimates, $(hH)^{1/2}$ is of the order of 3–30 kW m$^{-2}$ K$^{-1}$ and since $(T_o - T_s)$ is $\simeq 100$ K, then $q_{Dl}$ is greater than 300 kW m$^{-2}$, somewhat higher than the average value observed, but not unreasonably so given these order-of-magnitude arguments.

### 4.3. *Effects of heater surface material*

There are two important variations that we predict. Firstly, the physical properties—notably conductivity—will affect the observed variation with flow rate, and these are predictable based on our analysis. Secondly, surface finish and condition—oxidation and roughness—are known to affect the heat fluxes and interface temperature, the former by as much as an order of magnitude based on past quenching studies. The use of 'prototypical' materials and surface condition in experiments are plainly advisable.

## 5. CONCLUSIONS

We have considered the analysis of downflow of water in a heated annulus. Using existing dry patch

† Private communication with G. McCreery, EG&G Idaho, Inc. (1989).

stability theory, we can analytically predict the dryout condition. Based on reasonable arguments for the parametric variations of the wet and dryside heat transfer mechanisms, as observed in past work, all the available downflow data in annuli are correlated by a new theoretically-based equation given by, for turbulent flow, $q_D = 0.017Re^{0.9}$, and for laminar flow, $q_D = 0.27Re^{0.5}$, for $q_D$ from 1 to $10^3$ kW m$^{-2}$, and $Re$ from $10^2$ to $10^5$, with an uncertainty of the order of $\pm 50\%$.

The agreement with the data clearly shows that the hydrodynamics of film flow breakdown is not a sufficient condition for formation of dryout patches, and that *thermal* considerations control. The value of the exponent is consistent with the range expected for film flow heat transfer ranging from laminar to turbulent. The existing data cover the low Biot number range for the surface thermal properties.

New predictions based on the theory are also relevant. The effect of annulus conductivity is estimated as small but significant; and in the extreme of high Biot number, should change the flow dependency to one of near direct proportionality. Surface material and finish are also potentially important variables and will alter the magnitude of the constant in the correlation. Further theoretical developments, plus data for a wider range of flow rates and estimates of local heat fluxes at dryout are recommended.

## REFERENCES

1. G. Shires, A. R. Pickering and P. T. Blacker, Film cooling of vertical fuel rods, United Kingdom Atomic Energy Authority Report AEEW-R 343 (1964).
2. T. Fujita and T. Ueda, Heat transfer to falling liquid films and film breakdown—I, *Int. J. Heat Mass Transfer* **21**, 97–108 (1978).
3. I. I. Gogonin, A. R. Dorokhov and V. N. Bochagov, Stability of "dry patches" in thin, falling liquid films, *Fluid Mech.—Sov. Res.* **8**, 103–109 (1979).
4. J. Steimke, Status of heat transfer experiment (u), Nuclear Reactor Technology and Scientific Computations, DOE Report DPST-88-854 (1989).
5. W. G. Lussie, Local heat flux criterion for the ECS phase of a loss of coolant accident in the Savannah River Plant reactors, DOE Report EG&G-ES&H-8433 (1989).
6. V. T. Berta, K. G. Condie and J. A. Schroeder, Phenomenological scoping studies of downflow in a uniformly heated ribbed vertical annulus, DOE Report ES&H-SRP-3-89 (1989).
7. S. A. Kovalev, An investigation of minimum heat fluxes in pool boiling of water, *Int. J. Heat Mass Transfer* **9**, 780–788 (1966).
8. S. A. Kovalev, On stability of boiling regimes, *Teplofiz. Vysok. Temp.* **2**(5), 1219–1226 (1964).
9. R. Semeria and B. Martinet, Calefaction spots on a heating wall; temperature distribution and resorption. Paper presented at Symposium on Boiling Heat Transfer in Steam-generating Units and Heat Exchangers, I. Mech. Engng, Vol. 180(3C), pp. 192–205 (1965–66).
10. D. T. C. Porthouse, Dryout and rewetting in reactors and heat exchangers by a thermal conduction analysis, Central Electricity Generating Board Report RD/B/N2772 (1973).
11. N. Zuber and F. W. Staub, An analytical investigation of the transient response of the volumetric concentration in a boiling forced-flow system, *Int. J. Heat Mass Transfer* **9**, 897–905 (1966).
12. R. B. Duffey and D. T. C. Porthouse, The physics of rewetting in water reactor emergency core cooling, *Nucl. Engng Des.* **25**, 379–394 (1973).
13. B. D. G. Piggott and D. T. C. Porthouse, A correlation of rewetting data, *Nucl. Engng Des.* **32**, 171–181 (1975).
14. K. R. Chun and R. A. Seban, Heat transfer to evaporating liquid films, *Trans. ASME, J. Heat Transfer* 391–396 (1971).
15. E. P. White and R. B. Duffey, A study of the unsteady flow and heat transfer in the reflooding of water reactor cores, *Ann. Nucl. Energy* **3**, 197–210 (1976).
16. K. H. Sun, G. E. Dix and C. L. Tien, Effect of precursory cooling on falling-film rewetting, *Trans. ASME, J. Heat Transfer* 360–365 (1975).
17. K. Mishima and M. Ishii, Critical heat flux experiments under low flow conditions in a vertical annulus, *Nucl. Engng Des.* **86**, 165–181 (1985).
18. K. Mishima and H. Nishiha, The effect of flow direction and magnitude on chf for low pressure water in thin rectangular channels, Argonne National Laboratory, ANL-82-6 (1982).
19. K. Mishima, H. Nishihara and I. Michiyoshi, Boiling burnout and flow instabilities for water flowing in a round tube under atmospheric pressure, *Int. J. Heat Mass Transfer* **30**, 1169–1182 (1985).

## STABILITE DE L'ASSECHEMENT ET APPARITION AUX FAIBLES DEBITS

**Résumé**—On développe un modèle théorique du flux thermique nécessaire pour maintenir une structure stable sur une surface chaude refoidie par un film tombant liquide. Le modèle théorique est basé sur une solution analytique de l'équation bidimensionnelle de la chaleur avec des conditions aux limites thermiques appropriées aux conditions hydrodynamiques sur la zone chaude sèche et sur la zone mouillée. En utilisant des formes raisonnables physiquement pour les coefficients de transfert thermique, le flux de chaleur est exprimé en fonction du nombre de Reynolds du film liquide. On collecte les données expérimentales existantes; elles couvrent la région monodimensionnelle (faible nombre de Biot) et elles indiquent que se réalisent les conditions de film laminaire et turbulent. On obtient des formules pour les résultats analytiques. Pour l'écoulement laminaire, le flux thermique d'assèchement est $q_D = 0{,}27Re^{0,5}$ et pour l'écoulement turbulent $q_D = 0{,}017Re^{0,9}$, avec une incertitude de l'ordre de $\pm 50\%$.

## STABILITÄT UND EINSETZEN DES "DRYOUT" BEI KLEINEN MASSENSTROMDICHTEN

**Zusammenfassung**—Es wird ein theoretisches Modell zur Berechnung der Wärmestromdichte entwickelt, welche für die stabile Aufrechterhaltung einer Trockenstelle an einer berieselten Heizfläche erforderlich ist. Das theoretische Modell baut auf einer bestehenden analytischen Lösung der zweidimensionalen Wärmeleitgleichung auf. Als Randbedingungen werden Korrelationen für den Wärmeübergang angesetzt, die den Strömungsbedingungen an der heißen Trockenstelle bzw. im benetzten Gebiet in großer Entfernung von der Trockenstelle entsprechen. Bei Verwendung physikalisch sinnvoller Korrelationen für den Wärmeübergang im Bereich der Trockenstelle und im Bereich des benetzten Gebietes läßt sich die Wärmestromdichte als Funktion der Film-Reynolds-Zahl ausdrücken. Für die Korrelation werden vorhandene Versuchsdaten aufbereitet und gesammelt. Diese Daten umfassen den eindimensionalen Bereich (kleine Biot-Zahl) und zeigen, daß sowohl laminare als auch turbulente Filmströmung vorkommt. Aus den analytischen Ergebnissen werden entsprechende Korrelationsgleichungen ermittelt. Die Wärmestromdichte beim Austrocknen beträgt für laminare Strömung $q_D = 0{,}27Re^{0,5}$ und für turbulente Strömung $q_D = 0{,}017Re^{0,9}$ bei einer Unsicherheit der Größenordnung $\pm 50\%$.

## УСТОЙЧИВОСТЬ И НАЧАЛО КРИЗИСА ТЕПЛООБМЕНА ПРИ НИЗКИХ СКОРОСТЯХ ТЕЧЕНИЯ ПЛЕНКИ

**Аннотация**—Разработана теоретическая модель, позволяющая определить тепловой поток, необходимый для сохранения сильно нагретого участка на поверхности, охлаждаемой стекающей жидкой пленкой. Модель основана на имеющемся аналитическом решении двумерного уравнения теплопроводности с граничными условиями, содержащими обобщенные соотношения для коэффициента теплопереноса, которые соответствуют условиям, характерным для рабочей жидкости у сильно нагретого сухого участка и у прилегающей к нему влажной области. С использованием физически оправданных соотношений для коэффициентов теплопереноса на сухом участке и во влажной области тепловой поток представлен в виде функции числа Рейнольдса жидкой пленки. Обработка полученных экспериментальных данных позволила получить обобщающие соотношения. Эти данные охватывают одномерную (с низким числом Био) область и свидетельствуют о наличии условий ламинарного и турбулентного течений жидкой пленки. На основе аналитических результатов получены соответствующие обобщающие уравнения. При ламинарном течении критический тепловой поток составляет $q_D = 0{,}27Re^{0,5}$, а при турбулентном—$q_D = 0{,}017Re^{0,9}$ с погрешностью порядка $\pm 50\%$.