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Experimental study on the dryout heat flux of falling liquid film

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Abstract

A combined analytical and experimental investigation was carried out for the critical heat flux leading to dryout of falling liquid films. A modified model for predicting the dryout heat flux was correlated with the recent experimental results, with an average relative deviation of 12.7%, which is significantly less than that of previously reported ones. This improvement is believed to be the result of including the capillarity-induced interfacial evaporation [B.X. Wang, J.T. Zhang, X.F. Peng, Effects of interfacial evaporation on flow and heat transfer of thin falling liquid film, Science in China Ser. E, 1999, submitted] and the modification of streamwise thermocapillarity. The interfacial heat flux is showed to be important for heat transfer and for predicting dryout heat flux of falling liquid films. The new model indicates that, the bulk outlet temperature of falling film is required for accurate prediction of the critical heat flux leading to dryout. The results, compared with the current experimental results, demonstrate that the new model provides better predicted results in general. © 2000 Elsevier Science Ltd. All rights reserved.

Keywords: Falling liquid film; Film dryout; Dryout wall heat flux

1. Introduction

The prediction of dryout heat flux is most crucial in determining the application of falling liquid film. When dryout occurs, the surface being cooled is no longer in intimate contact with the liquid film. As a result, the heat transfer ability will decrease dramatically and the corresponding wall temperature will rise rapidly, even to melt the heat transfer surface. Therefore, the prediction of the dryout heat flux draws extensive concerns in the past decades [1,2]. There have been a number of semi-empirical and experimental studies conducted on this problem, the predicting formulas given by these studies mainly related the critical or dryout well heat flux with the mass flow rate (Reynolds number). In most of these investigations, the causes for film dryout were attributed to the streamwise or lateral thermocapillary effect [2–5]. Owing to the considerable complexion, although there has been encouraging agreement between the experimental and predicted results, there is still a great deal for further study. For example, the interfacial heat transfer was neglected by the cited studies [2–5].

In our previous papers, the thermal non-equilibrium effect on stability of wavy falling liquid films was analyzed [6], the capillarity-induced interfacial evaporation of falling films with/without wall heating were studied analytically and experimentally [7,8]. The experimental

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Nomenclature

C_p	specific heat
g	acceleration of gravity
h_0	mean substrate thickness
J	mass flux
k	thermal conductivity
L	latent heat of liquid
M	molecular weight
q_w	wall heat flux
q_w^s	coefficient in Eq. (17)
Re	Reynolds number, $4\Gamma/\mu$
R_g	universal gas constant
r_e	effective radius of wave trough
T	temperature
U_H	hydrodynamic speed
U_{TC}	thermocapillary speed

Greek symbols

Γ	liquid mass flow rate per unit perimeter
α	accommodation coefficient
μ	dynamic viscosity
ν	kinetic viscosity, μ/ρ
ρ	density
σ	surface tension

Subscript

l	liquid
v	vapor
x	streamwise
y	lateral

results [7] indicated that the interfacial evaporation exists even for the case without wall heating, and the heat flux due to the capillarity-induced interfacial evaporation is to be comparable with the total input of wall heat flux [8]. The proposed semi-empirical correlation in this paper required about the information of the outlet temperature of falling films, which was omitted in all the previously published experimental investigations except the experimental results of Bohn and Davis [5]. But the uncertainty lies on the location of the measuring spot that is unknown. Our experiments demonstrated that the outlet temperature varies a lot with the location of temperature probes for the non-uniform lateral distribution of film temperature due to thermocapillarity. Hence, the proposed semi-empirical correlation will be correlated with our own experimental results, and then, compared with the predicted results of some existed predicting methods [2–5].

2. Semi-empirical correlation of dryout heat flux

There existed several viewpoints on the dryout mechanism of falling films. The approach of Fujita and Ueda [3] was based on force balance within liquid films, such that the streamwise thermocapillary effect is crucial to the mechanism for dryout of falling liquid films, and hence, the dryout condition was really based on the force balance between gravity, thermocapillarity and viscosity in the streamwise direction. Simon and Hsu [2] noted early in 1970 that, lateral film redistribution, caused by thermocapillarity at the free interface, became pronounced as heat flux was increased, which distorts the films flow until the thinnest region reached the condition to sustain a dry patch. Hoke and Chen [4], Bohn and Davis [5] developed their equation for predicting dryout heat flux, by analyzing force balance in the streamwise and lateral direction simultaneously. Our experimental results support the viewpoint of these cited literature [3–5].

Based on the criterion of Bohn and Davis [5], a new semi-empirical correlation was proposed. The main points of this approach are as follows.

The analytical model for a liquid film flowing down a vertical heated plate is shown in Fig. 1. Referring to the analysis of Bohn and Davis[5], and assuming a laminar-flow balance, yields

$$\mu_l \frac{\partial^2 u}{\partial z^2} = \rho_l g \tag{1}$$

where u is the streamwise velocity, g is the gravitational acceleration, ρ_l and μ_l is the mass density and viscosity for the liquid film, respectively.

From Eq. (1), a scale of the hydrodynamic speed, U_H, can be expressed as

$$U_H = \frac{\rho_l g h_0^2}{\mu_l} \tag{2}$$

with h_0 being an estimate of the mean substrate thickness. Notify the kinetic viscosity, ν_l, as $\nu_l = \mu_l/\rho_l$, then

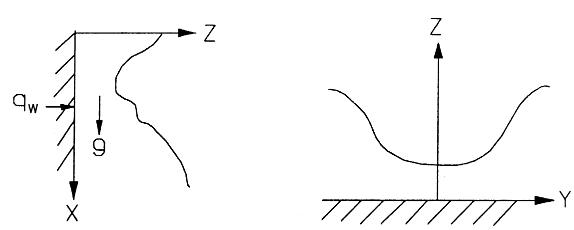

Fig. 1. Nomenclature of the breakdown model.

the Reynolds number of the substrate film flow, Re_s, can be written as

$$Re_s = \frac{U_H h_0}{\nu_l} \tag{3a}$$

where h_0 is the mean substrate film thickness calculated by [9]

$$h_0 = \left(\frac{\nu_l^2}{g} Re_s\right)^{1/3} \tag{4}$$

In the cross-stream direction, the thermocapillarity creates surface stresses to drive fluid from troughs into crests [10–13]. Approximately, the cross-stream force balance at the interface would be [5]

$$\mu_l \frac{\partial u}{\partial z} \cong \frac{d\sigma}{dy} = -\left|\frac{d\sigma}{dT}\right| \frac{\partial T}{\partial y} \tag{5}$$

where σ is the surface tension and $|\frac{d\sigma}{dT}|$ is the surface tension gradient due to temperature variation. The estimated thermocapillary speed, U_{TC}, is given as

$$U_{TC} = \left|\frac{d\sigma}{dT}\right| \frac{\Delta T}{\mu_l} \frac{h_0}{l_y} \tag{6}$$

where ΔT is an estimate of the temperature differences induced by heating, and l_y is a lateral length scale. The integral heat balance could be represented approximately by

$$\int_0^{h_0} \rho_l C_{p,1} \left(U_H \frac{\partial T}{\partial x} + U_{TC} \frac{\partial T}{\partial y}\right) dh = q_w - JL \tag{7}$$

where q_w is the wall heat flux, L is the latent heat, and J is the interfacial mass flux of evaporation.

For thin film, it is reasonable to assume $\frac{\partial T}{\partial x}$ and $\frac{\partial T}{\partial y}$ being constant, respectively, Hence, Eq. (7) can be reduced to

$$U_H \rho_l C_{p,1} \frac{\partial T}{\partial x} + U_{TC} \rho_l C_{p,1} \frac{\partial T}{\partial y} = \frac{1}{h_0}(q_w - JL) \tag{8}$$

The interfacial mass flux of evaporation J was given as [7,8]

$$J = \left(\frac{2\alpha}{2-\alpha}\right)\left(\frac{M}{2\pi R_g}\right)^{1/2} T_{iV}^{-1/2} \frac{2\sigma}{r_e} \tag{9}$$

where α is the accommodation coefficient; M is the molecular mass of the working fluid; R_g is the universal gas constant; and T_{iV} is the interfacial temperature in the interface region; σ is the surface tension; and r_e is the effective radium of wave troughs.

As a first approximation, assuming J being constant, and substituting Eq. (6) into Eq. (8), would yield

$$\rho_l C_{p,1} \left|\frac{d\sigma}{dT}\right| \frac{\Delta T_y}{\mu_l} \frac{h_0}{l_y} \frac{\Delta T_y}{l_y} = \frac{1}{h_0}[q_w - JL] - \rho_l C_{p,1} U_H \frac{\Delta T_x}{l_x} \tag{10}$$

Then, $\frac{\Delta T_y}{l_y}$ can be solved as

$$\left(\frac{\Delta T_y}{l_y}\right)^2 = \frac{\mu_l}{\rho_l C_{p,1} \left|\frac{d\sigma}{dT}\right| h_0} \left[\frac{q_w}{h_0} - \frac{L}{h_0}\left(\frac{2\alpha}{2-\alpha}\right)\left(\frac{M}{2\pi R_g T_{iV}}\right)^{1/2} \frac{2\sigma}{r_e} - \rho_l C_{p,1} U_H \frac{\Delta T_x}{l_x}\right] \tag{11}$$

Substituting Eq. (11) into Eq. (6), yields

$$U_{TC} = \left|\frac{d\sigma}{dT}\right| \frac{h_0}{\mu_l} \sqrt{\frac{\mu_l}{\rho_l C_{p,1} \left|\frac{d\sigma}{dT}\right| h_0} \left[\frac{q_w}{h_0} - \frac{L}{h_0}\left(\frac{2\alpha}{2-\alpha}\right)\left(\frac{M}{2\pi R_g T_{iV}}\right)^{1/2} \frac{2\sigma}{r_e} - \rho_l C_{p,1} U_H \frac{\Delta T_x}{l_x}\right]} \tag{12}$$

Following the same way conducted by Bohn and Davis [5], a criterion for dryout will be such that the "thermocapillary" speed should be large enough compared to the "hydrodynamic" speed, i.e.

$$\frac{U_{TC}}{U_H} \geq n \tag{13}$$

where n is a constant to be determined. This criterion indicates that, as long as the lateral film flow large enough to produce sufficient thinning of the substrate between roll waves, the substrate will break to form a stable dry area.

Substituting Eq. (12) into Eq. (13), yields the following expression

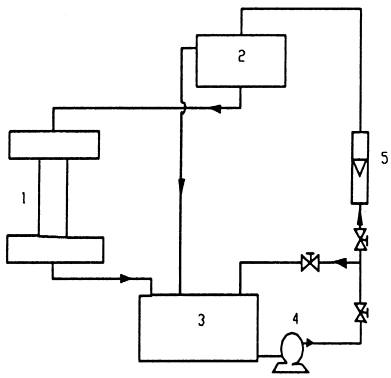

Fig. 2. Sketch of the experimental system. 1 — entrance and test section; 2 — upper tank; 3 — lower tanker; 4 — pump; 5 — flowmeter.

Table 1
The experimental results of present authors

Number	Γ (kg/s m)	q_w (J/s m^2)	T_{in} (°C)	T_{out} (°C)
1	0.307	13240.0	17.9	20.8
2	0.375	17420.0	19.1	22.1
3	0.375	17220.0	18.8	22.1
4	0.375	17630.0	18.8	22.2
5	0.430	17850.0	16.7	18.7
6	0.430	19460.0	16.7	19.1
7	0.450	24860.0	17.9	21.9
8	0.470	23800.0	17.5	23.0
9	0.470	25465.0	17.7	23.7
10	0.470	27200.0	17.8	24.1
11	0.470	13950.0	17.6	20.7
12	0.470	20140.0	17.9	22.5
13	0.470	21480.0	18.1	22.9
14	0.470	20800.0	18.1	23.0
15	0.470	18870.0	17.5	21.8
16	0.474	23500.0	17.6	21.3
17	0.641	21710.0	18.4	20.6

$$\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|\frac{h_0}{U_H\mu_l}\sqrt{\frac{\mu_l}{\rho_l C_{p,l}\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|h_0}\left[\frac{q_w}{h_0}-\frac{L}{h_0}\left(\frac{2\alpha}{2-\alpha}\right)\times\left(\frac{M}{2\pi R_g T_{iV}}\right)^{1/2}\frac{2\sigma}{r_e}-\rho_l C_{p,l}U_H\frac{\Delta T_x}{l_x}\right]}\geq n \tag{14}$$

Based on the criterion of Eq. (14), the least dryout heat flux of falling films can be obtained as

$$q_w = q_w^s Re_s^{4/3}\left[n^2+\frac{\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|L}{\mu_l U_H^2\rho_l C_{p,l}}\left(\frac{2\alpha}{2-\alpha}\right)\times\left(\frac{M}{2\pi R_g T_{iV}}\right)^{1/2}\frac{2\sigma}{r_e}+\frac{\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|}{\rho_l g h_0}\frac{\Delta T_x}{l_x}\right] \tag{15}$$

where

$$q_w^s = \left[\rho_l C_{p,l}^3\mu_l^5 g^2\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|^{-3}\right]^{1/3} \tag{16}$$

$$Re_s = \frac{U_H h_0 \rho_l}{\mu_l} \tag{3b}$$

On the right-hand side of Eq. (15), the first term is the same that of Bohn and Davis [5], the second and third term represents, respectively, the effects of the capillarity-induced interfacial evaporation and modification of streamwise thermocapillarity. In order to correlate the experimental data with Eq. (15), the Reynolds number Re_s is replaced by $4\Gamma/\mu_l$ with Γ defined as liquid mass flow rate per unit perimeter, and ΔT_x is replaced by the difference of inlet and outlet temperature. By so doing, the third term on the right of Eq. (15) should be modified by an unknown parameter b, for the revised $\Delta T_x/l_x$ is not the real local temperature gradient. Then, Eq. (15) is reduced to

$$q_w = q_w^s Re^{4/3}\left[n^2+\frac{\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|L}{\mu_l U_H^2\rho_l C_{p,l}}\left(\frac{2\alpha}{2-\alpha}\right)\times\left(\frac{M}{2\pi R_g T_{iV}}\right)^{1/2}\frac{2\sigma}{r_e}+b\frac{\left|\frac{\mathrm{d}\sigma}{\mathrm{d}T}\right|}{\rho_l g h_0}\frac{\Delta T_x}{l_x}\right] \tag{17}$$

The effective radius in Eq. (17) can be obtained from the following equation [7]:

$$r_e = L\left(\frac{2\alpha}{2-\alpha}\right)\left(\frac{M}{2\pi R_g}\right)^{1/2}T_{iV}^{-1/2}\times 2\sigma \times 10^{-3}\left(-0.8408\Gamma^2+4.1384\Gamma-0.8236\right)^{-1} \tag{18}$$

Then, the two unknown coefficients n and b in Eq. (17)

could be determined by correlating our experimental data.

3. Experimental system

In order to provide experimental data by which the unknown coefficients n and b in Eq. (17) can be determined, the experimental apparatus shown in Fig. 2 was designed and constructed [14]. This test apparatus is similar to those previously described by the authors [7], and some of the results were also cited here in Table 1. The entrance sector is a 90 mm long and 18 mm O.D. plexiglass cylinder. The test section is a 300 mm long, 0.5 mm thick and 18 mm O.D. stainless cylinder. The surface finish and measurement uncertainty in the outer diameter of these cylinders are 1.6 μm and 0.05 mm, respectively. Falling liquid films are formed from a 0.5 mm width circular orifice. A 100 mm inside diameter, 150 mm depth water container is provided at the upper end of the entrance section. The upper water tank is used to keep water level stable. There were four pairs of T-type thermal couples used to measure the inlet temperature, the temperature difference between inlet and outlet and the temperature of two spots on the vertical wall. The uncertainty of these thermocouple records is within $\pm 0.1°C$. The difference of inlet and outlet temperature was measured by putting the reference and sensor end of the same thermocouple in the inlet and outlet temperature measuring spot, respectively. So, the electrical potential difference of these two ends gives the temperature difference of the inlet and outlet temperature exactly. This method ensures that the uncertainty of the inlet and outlet temperature difference will be $\pm 0.1°C$. The outlet temperature is the average one, the outlet temperature would be actually non-uniform around the exit of the tested column due to the effect of lateral thermocapillarity. The inlet temperature measuring spot was set in the top container of the test section. In this way, the average bulk temperature of the inlet and outlet fluids can be measured accurately.

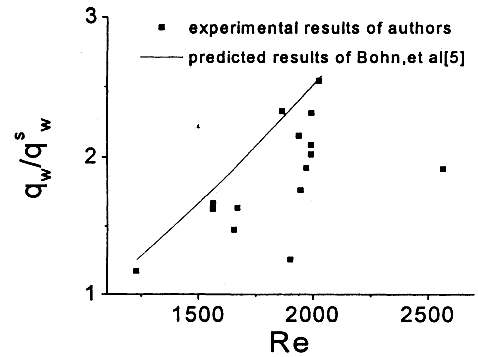

Fig. 4. Correlated results of literature [5].

The test section was heated by electricity directly, and the two electric poles located at the two ends of the test sector.

A HP 34970A Data Acquisition/Switch Unit and HP 34902A 16-channel multiplexer were coupled and used as the data acquisition system. It has $6_{1/2}$-digit multimeter accuracy, stability and noise rejection resulted in a high degree of accuracy. The reading rate is up to 600 readings per second on a single channel. In this experiment, the integrating interval was set to 1PLC (equivalent to 20 ms); the corresponding resolution was $6_{1/2}$; the attached noise error is (range of measurement) $\times$ 0.1%.

In every test run, the experimental data were collected by a 586/300 MHz PC. The time interval was approximately 0.030 s. In every test run, the temperature was obtained by averaging the 100 data of every channel.

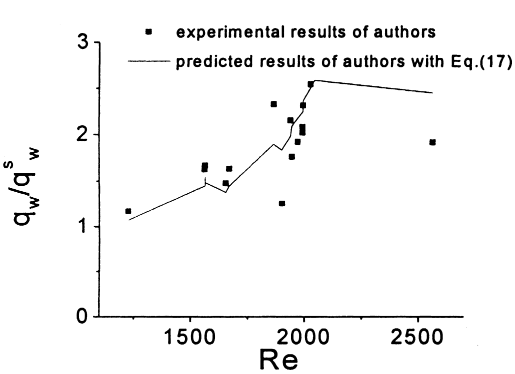

Fig. 3. Experimental and correlated results of authors.

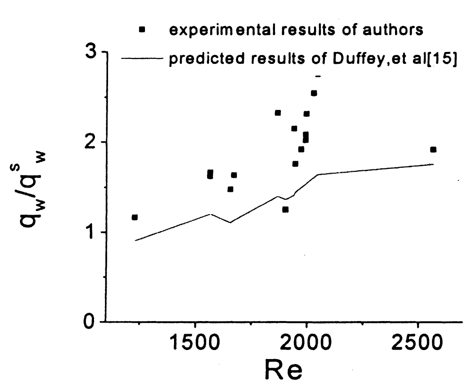

Fig. 5. Correlated results of literature [15].

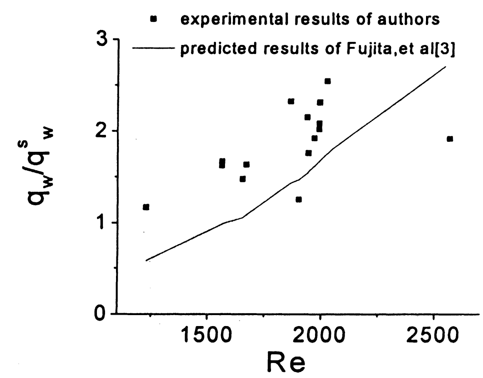

Fig. 6. Correlated results of literature [3].

4. Correlation of experimental results with discussion

The experimental results were summarized in Table 1. Using these data and a least squares curve fit technique, the two unknown constants in Eq. (17) were found to be $n = 5.49 \times 10^{-5}$ and $b = 6.55 \times 10^{-2}$. The predicted values with Eq. (17) are compared with the experimental results obtained here, are shown in Fig. 3. As illustrated, the correlation is quite good with the average relative deviation between the experimental and calculated results of 12.7%.

In addition, the existing models reported in the literature [3,5,15,16] are compared with our measurements in Figs. 4–7, and the relative deviations of these comparisons were given in Table 2. The comparisons demonstrated that our proposed correlation has the minimum relative average deviation and the second smallest maximum relative deviation. The improving accuracy of this new model is due to the inclusion of the capillarity-induced interfacial evaporation and modification of streamwise thermocapillarity terms in the correlation. Table 2 exhibited that, the correlation of [3] has a little higher relative deviation, which shows that the lateral thermocapillarity might be the main cause responsible for film dryout.

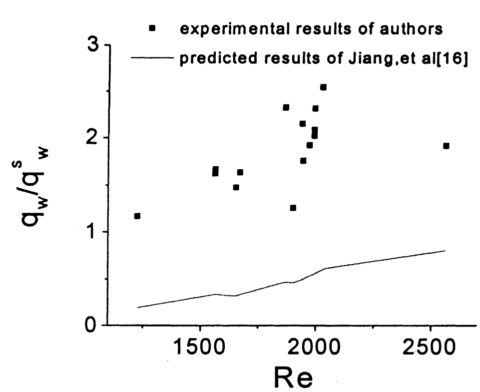

Fig. 7. Correlated results of literature [16].

Table 2
Comparison of the predicting models

Author	Average relative deviation (%)	Maximum relative deviation (%)
Fujita and Ueda	30.5	50.6
Duffey and Hughes	26.5	40.4
Bohn and Davis	22.9	87.3
Jiang et al.	75.3	83.5
Wang et al.	12.7	46.3

As indicated, the values predicted by Bohn and Davis [5] fit the current experimental results well with the average relative deviation of 22.9% which is consistent with that of comparison between the calculated and experimental results of Bohn and Davis [5]. It implies that the correlation of Bohn and Davis [5] might be generally suitable, and the present correlation is really a reasonable modification of Bohn and Davis [5] with a somewhat better prediction technique for the dryout heat flux of falling films, provided if the inlet and outlet temperature are given correctly.

5. Conclusions

The correlation of Bohn and Davis [5] for the prediction of dryout heat flux of falling liquid films has been modified by incorporating the capillarity-induced interfacial evaporation [7] and modification of streamwise thermocapillarity in this paper. The resulting expression was compared with the experimental results of this investigation, and with other correlation available in literature. This comparison demonstrated that the proposed new method has the lowest average relative deviation. It makes clear that the heat flux from interfacial evaporation could not be neglected, especially in the case of low mass flow rate.

The modification of streamwise thermocapillarity has been included in the developed new model, so that the outlet temperature of falling film is needed. This is important for improving the precision for prediction of the dryout wall heat flux. Problem may be existed that the outlet temperature varies with the probe location because of lateral thermocapillarity. This should be verified by further studies.

The mechanism of film dryout employed in [16] is quite different. They argued that the onset of the dry patch could be due to purely hydrodynamic effects, by which the dryout condition should be taken to be the heat flux maintained the dry patch. The drypatch will grow if the heat flux is sufficiently large and collapse if not. The present method is indeed a modification of Bohn and Davis' correlation [5].

It is well known that, film dryout is a complex phenomenon and affected by many known/unknown factors. It was observed that the partial pressure of vapor has possible effect on the dryout heat flux, the lower the partial pressure of vapor is, the higher the dryout heat flux. The reason might be that the lower partial pressure would enlarge the difference of the chemical potential between the liquid and the vapor. Therefore, the enhanced nonequilibrium will drive the liquid to evaporate much more. From Eq. (17), it makes sure that the dryout heat flux grows with increasing interfacial heat flux. It reminds that the second term in the right side of Eq. (17) should involve the effect of partial pressure of vapor, but this is not manifested in this study. Hence, further investigation would be needed.

Acknowledgements

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