Dryout Heat Flux and Size of Entrained Drops in a Flow Boiling System

By Tatsuhiro UEDA** and Kyungkun KIM***

Boiling heat transfer experiments are performed on a Freon 113 upward flow in a uniformly heated tube, and the liquid film flow rate and the size of drops entrained in the vapor core are measured at the exit end of the heated tube. The exit film flow rate at the critical heat flux condition is near to zero in a range of low heat fluxes, but it shows a trend to increase with increasing heat flux over a certain value. An empirical equation for the arithmetical mean diameter of the entrained drops is proposed, and it has been shown that the size distribution of drops can be suitably described by a gamma distribution.

1. Introduction

The critical heat flux in a high quality region, so-called the dryout heat flux, is an important design factor of practical heat exchangers in which evaporation is occurring. The flow state in uniformly heated tubes is developed to an annular-dispersed type as the quality increases, and the liquid film flow rate decreases along the tube length owing to its evaporation and drop interchange between the vapor core and the liquid film. When the supplied heat flux is increased, the liquid film flow rate at the exit end decreases progressively, and then a point is reached where a sharp rise in wall temperature takes place at the exit end. The heat flux just before the sharp rise in wall temperature is therefore considered a critical value in the flow boiling system.

In this study, the process to attain the dryout heat flux condition was investigated for forced flow boiling in a uniformly heated tube in which R 113 was vaporized from a subcooled liquid at the inlet to a relatively high quality at the exit. Measurements were made of:
(a) heat transfer coefficient along the heating length and the dryout heat flux,
(b) variation of the liquid film flow rate at the exit end of test tube with the heat flux added, and
(c) size distribution of the drops entrained in the vapor core.

Hewitt et al. (1) examined the relationship between the heat input and the liquid film flow rate with climbing films on a heated surface, and demonstrated that an excursion in wall temperature at exit end of the heated section occurred when the exit film flow rate decreased smoothly to zero. Fujita and Ueda (2) showed in their study on falling liquid films with nucleate boiling that a sharp rise in wall temperature due to film breakdown took place under the condition of the exit film flow rate being reduced to some value near zero, i.e. the film flow rate per unit periphery \( T_f = 0.01 \sim 0.02 \) kg/ms. These observations were obtained with relatively low heat fluxes. In case of high heat fluxes where intensive nucleate boiling occurs under the liquid film, however, the critical condition may be reached under the condition of more remaining liquid film flow rate than in the case mentioned above. The measurement (b) was undertaken to examine the exit film flow rate at the critical condition in a range of high heat fluxes.

The size of drops entrained in the vapor core has an important effect on the mass transfer of drops onto the liquid film and also on the heat transfer process in the post-dryout region. However, experimental data are limited on the size of drops in flow boiling systems. In this study, size distribution of the drops in and near dryout region was measured and compared with the data obtained for adiabatic annular flow systems.

Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>D</td>
<td>tube diameter</td>
<td>m</td>
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<tr>
<td>d</td>
<td>drop diameter</td>
<td>( \mu ) m</td>
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<tr>
<td>( \bar{d} )</td>
<td>mean diameter of drops</td>
<td>(D/m^2) ( \bar{d} )</td>
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<td>( \sigma_0 )</td>
<td>standard deviation of drop size</td>
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Subscripts

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** Professor, Faculty of Engineering, University of Tokyo, Bunkyo-ku, Tokyo.
*** Assistant Professor, Korea Merchant Marine College, Busan, Korea.
2. Apparatus and procedure

A schematic diagram of the apparatus is shown in Fig.1. The test tube is a stainless-steel SUS 304 tube of 10.0 mm I.D., 12 mm O.D., and its heating section is 2.45 m in length. The test liquid R 113 supplied at a subcooled condition was forced to flow upwards in the test tube, and heated to a high quality of annular-dispersed flow type. At the exit end of the test tube, the liquid film flowing on the wall surface was removed through a gap of 15 mm provided between the exit of the test tube and the inlet of the upper tube of 10 mm I.D. This film separator was of the same geometry as one used by Ueda, Tanaka and Koizumi.13) The vapor phase at the exit of the test tube, flowed into the upper tube and the drops entrained in it were separated at a cyclone separator.

The test section was heated uniformly by passing an alternate current through it, and the axial wall temperature distribution was measured with forty eight C-A thermo-couples fitted to the outer surface along the tube length. The saturation temperature distribution along the heating length was calculated with the assumption of a linear decrease in pressure along the test section. Pressure taps were provided at inlet and outlet of the test section. The inlet pressure of the test section was kept at a value P_in = 3.24 ata throughout this experiment. The experimental ranges covered are

- Mass velocity \( \dot{G} = 485 \sim 1155 \text{ kg/m}^2\text{s} \)
- Heat flux \( q = (2.5 \sim 11) \times 10^3 \text{ W/m}^2 \)
- Inlet quality \( x_{in} = 0.178, 0.247 \) and \( 0.332 \).

3. Heat transfer to boiling two-phase flow

3.1 Variation of heat transfer coefficient along tube length

The heat transfer to a boiling two-phase flow shows nucleate boiling characteristics in subcooled and low-quality regions, and characteristics of forced convection of liquid film (forced convection evaporation) in high-quality region where the heat transfer coefficient increases with increasing quality. However, it has been noted that when the quality is increased furthermore, a dryout condition is reached and the heat transfer coefficient is suddenly reduced from a very high value to a low value. In this experiment, the inner surface temperature \( T_w \) of the test section was calculated from the temperature measured at the outer surface, and the local heat transfer coefficient was determined by

\[
\frac{h_{TP}}{q} = \frac{T_w - T_b}{T_w - T_b} \quad (1)
\]

where, \( T_b \) is the bulk mean temperature of the fluid and was assumed to be the saturation temperature \( T_{sat} \) in the saturated region.

Figures 2 and 3 show some values of the local heat transfer coefficient plotted against the fluid quality. Fig.2 shows the results at a mass velocity \( \dot{G} = 754 \text{ kg/m}^2\text{s} \), representing an effect of the heat flux. Fig.3 shows variations of the heat transfer coefficient obtained at heat fluxes a little higher than the dryout condition at the exit end of the heating section. These results indicate that the heat transfer coefficient in the low-quality region depends most on the heat flux and increases with increasing heat flux, and that both the transition point to forced convection evaporation and the
dryout point move to lower qualities as the mass velocity is increased.

3.2 Correlation for heat transfer coefficient in saturated region

The heat transfer coefficient in the region of forced convection evaporation has been usually expressed in terms of the Lockhart-Martinelli flow parameter

$$\frac{1}{X_{tt}} = \left( \frac{x}{1-x} \right)^{0.9} \left( \frac{\nu_g}{\nu_1} \right)^{0.1} \left( \frac{P_1}{\rho_1} \right)^{0.5}$$

(2)

Dengler and Addoms(4) correlated the heat transfer coefficient $h_T$ in a form of $h_T = f(1/X_{tt})$, where $h_{10}$ is the heat transfer coefficient for the single-phase liquid flow of total mass velocity $G_T$ expressed as

$$h_{10} = 0.023 \frac{k_1}{D} \left( \frac{D G_T}{\nu_1} \right)^{0.8} Pr_1^{0.4}$$

(3)

Guerrieri and Talty(5), Schrock and Grossman (6), Bennett(7) and Collier(8) used a form of $h_T/h_{10} = f(1/X_{tt})$, where $h_{10}$ is the heat transfer coefficient for the single-phase liquid flow of the liquid component $G_T(1-x)$.

As an extension of the above method, Schrock and Grossman (9), Wright(10), Pujol and Stingem(11) have proposed a method to correlate $h_T/h_{10}$ or $h_T/h_{fg}$ for both regions of nucleate boiling and forced convection evaporation with $1/X_{tt}$ and the boiling number defined by

$$Bo = q/(G_T h_{fg})$$

(4)

Figure 4 shows results obtained at an inlet quality $x_{in} = 0.178$, where $h_T/h_{10}$ is plotted against the flow parameter $1/X_{tt}$. In a range of low values of $1/X_{tt}$ where the nucleate boiling is dominant, $h_T/h_{10}$ is approximately constant, but it increases gradually with increasing quality, and then converges to a region of convective characteristics where the heat transfer coefficient is insensitive to the heat flux.

Figure 5 shows a series of data comparing some correlations presented so far. Schrock and Wright have adopted an exponent of Prandtl number in Eq.(3) as $1/3$, so that a modification was made on these correlations for this comparison. There are some discrepancies among these correlations. As is pointed out by Chen(12) and Sato et al.(13), the value of $h_T/h_{10}$ seems to be affected by the heat flux and the mass velocity other than Bo and $X_{tt}$. However, the correlations of the above form are simple and useful to estimate roughly the heat transfer coefficient in saturated region. Therefore, simple correlations were derived for the present result dividing the boiling two-phase flow into three regions, namely, the nucleate boiling region, the transition region and the forced convection evaporation region. Empirical correlations thus derived are as follows:

Nucleate boiling region:

$$h_T/h_{10} = 1.20 (Bo \times 10^4)^{-0.5}$$

(5)

Transition region:

$$h_T/h_{10} = 0.98 [Bo \times 10^4]^{-1.5} \left( 1/X_{tt} \right)^{2/3}$$

(6)

Forced convection region:

$$h_T/h_{10} = 4.20 (1/X_{tt})^{0.57}$$

(7)

Solid lines in Fig.4 for $Bo \times 10^4 = 10, 30, 90, 34$ and 80 and those in Fig.5 represent the values calculated from Eqs.(5)-(7). These equations correlate the present data within ± 20%.

3.3 Dryout heat flux

Figure 6 shows the dryout heat flux — the heat flux just before a sharp rise in wall temperature — obtained in this experiment. The dryout heat flux increases with increasing mass velocity, and the quality at the exit end of heating section decreases.
4. Liquid film flow rate at the exit end of heating section

The flow rates of liquid film and entrained drops at the exit end of heating section were obtained by measuring the flow rate of liquid removed at the film separator. The liquid film flow rate at the exit end of heating section was calculated from the removed flow rate \( M_F \) measured at the film separator by the following equation,

\[
M_F = M_F' - M_{dp} + M_{fin} \tag{8}
\]

where \( M_{dp} \) is the drop deposition rate in an unheated tube extending from the exit end of heating section to the film separating gap. A part of the liquid film was splashed at the gap separator and carried to the upper tube. \( M_{fin} \) in the above equation denotes this carry-over flow rate. Since the purpose of this measurement is to examine the exit film of low flow rates near the dryout condition, the drop entrainment from the thin film on the unheated tube surface has been neglected in Eq.(8). \( M_{dp} \) and \( M_{fin} \) depend upon the flow conditions and the dimensions of apparatus. They were calculated, as described in Appendix 2, by applying empirical equations of Leda et al., derived for a film separator of the same dimensions as the present apparatus.

The entrained drop flow rate at the exit end of heating section was calculated by

\[
M_d = M_T (1 - x_0) - M_F \tag{9}
\]

where, \( M_T \) is the total flow rate and \( x_0 \) is the quality at the exit end of heating section. The drop flow rate was also derived from the mass balance

\[
M_d = M_T' - M_{fin} + M_G
\]

where, \( M_T' \) is the liquid flow rate measured at the cyclone separator and \( M_G \) denotes the evaporation rate of drops due to a pressure loss between the end of heating section and the cyclone separator. The values of \( M_d \) derived from this mass balance were a little lower (~10% at the maximum) than those of Eq.(9). This difference seems to result from the amount of drops flowing out of the cyclone separator together with vapor.

A ratio of the entrained drop flow rate obtained by Eq.(9) to the total flow rate is plotted against the equilibrium quality at the exit end of heating section in Fig.7. In the same way as the result of Langner et al., for R 12, the exit drop flow rate takes a peak value at some exit conditions.

Fig.5 Comparison of correlations for heat transfer coefficient

Fig.6 Heat flux and exit quality at dryout condition

Fig.7 Variation of entrained drop flow rate ratio
quality and then decreases slowly as the exit quality is increased. Each solid plot on the line of \((N_d + N_f)/N_F\) represents the exit quality at the dryout heat flux.

Figure 8 shows the variation of the exit film flow rate \(N_f\) obtained from the measurement for inlet qualities \(x_{in} = -0.178\) and \(-0.332\). The exit film flow rate decreases steadily with increasing heat flux. The solid symbols plotted on the abscissa represent the heat fluxes at the dryout condition. In a range of low heat fluxes, the dryout condition occurs, as in the results of Hewitt (15) and Bennetts (16) for water, and those of Staniforth and Stevens (17) for \(R\) 12, when the exit film flow rate decreases to some value near zero. By analyzing experimental data on \(R\) 113 high-quality flow in a uniformly heated tube with heat fluxes less than about \(8 \times 10^4\) \(\text{W/m}^2\), Ueda et al. (3) have shown that the position where a sharp rise in wall temperature takes place coincides well with the position where the film flow rate is predicted to be zero. However, in a range of heat fluxes higher than about \(9 \times 10^4\) \(\text{W/m}^2\), the dryout condition is reached at a state where some extent of liquid film — \(N_f = 5 \sim 15\) \(\text{kg/h} / (T_g = 0.04 \sim 0.13\) \(\text{kg/m}^2\text{s}) — is remaining at the exit end of heating section, and the exit film flow rate at the dryout heat flux shows a trend to increase with increasing heat flux. A possible explanation for this trend may be that, in case of high heat fluxes, the dry area caused by local film breakdown or film separation is easily stabilized, because it reaches rapidly such a high temperature that the liquid film can no longer rewet.

As is seen in Fig. 6, the exit qualities at a dryout heat flux \(q_o = 9 \times 10^6\) \(\text{W/m}^2\) are in a range of 30 \(\sim 55\%\).

5. Size of drops entrained in dryout region

The drops entrained in the vapor flow were sampled at the exit end of test tube under conditions of \(q_T = 220 \sim 890\) \(\text{kg/m}^2\text{s}\) and \(x_{in} = -0.178\). The heat input in this experiment was adjusted to be near the dryout heat flux for each mass velocity \(q_T\). A sampling device, which was composed of a rotating shutter and a glass plate mounted in it, was installed in the upper tube of the film separator. The glass plate coated with magnesium oxide was exposed to the mist flow for a short time. Impinging drops impressed craters on the coating, then the diameter of the craters was measured by taking microscopic photographs. For determining the actual drop diameter from the crater diameter, a correlation equation (3) derived from the data of May's experiment (18) was used.

5.1 Mean diameters of entrained drops

The measured arithmetical mean diameter of drops \(d_1 = \text{Edm/N}\) ranged from 23 to 52 \(\mu\)m and showed a trend to decrease with increasing superficial vapor velocity \(U_g = q_T \times x_{in} / \rho_g\) at the exit of test tube. The present data are compared with the data obtained by Ueda et al. (3) for \(R\) 113 adiabatic annular flow in Fig. 9. It is interesting to note that the mean diameter of entrained drops in the dryout region of an evaporating tube is approximately equal to that in a steady adiabatic two-phase flow. Fig. 9 shows a correlation of the present data along with the previous data for air-water systems by Ueda (19). From this result, an empirical equation for the mean diameter was derived as follows:

\[
\frac{d_1}{D} = 6.1 \times 10^{-3} \left(\frac{\sigma}{\mu_g \rho_g}\right)^{1.25} x \left(\frac{\rho_g}{\rho_1}\right)^{0.50} \left(\frac{\rho_1}{\rho_1}\right)^{0.50}
\]

The vapor velocity in an evaporating tube increases along the heating section, then, there is a problem of selecting an appropriate value. For correlating the present data, however, the superficial vapor velocity at the exit end was used as \(U_g\) in the above equation. The values of the mean diameter \(d_m = (Edm/E)\) of the present data were about 10% higher than those predicted by the following empirical equation (19) proposed for an adiabatic two-phase flow:

\[
\frac{d_m}{D} = 5.8 \times 10^{-3} \left(\frac{\sigma}{\mu_g \rho_g}\right)^{1.25} x \left(\frac{\rho_g}{\rho_1}\right)^{0.34} \left(\frac{\rho_1}{\rho_1}\right)^{0.34}
\]
5.2 Distribution of entrained drop sizes

The measured drop size distributions well fitted a gamma distribution like the previous data for R 113 adiabatic annular flow \(^{(3)}\). Denoting the drop diameter normalized by the arithmetical mean diameter as \(t = d / \bar{d}_1\), the gamma distribution and its standard deviation are expressed as follows:

\[
\frac{\Delta n}{N} = f(t) \Delta t
\]

\[
f(t) = \frac{\Gamma(m)}{\Gamma(m) \Gamma(\lambda)} t^{m-1} \exp(-\lambda t)
\]

and

\[
o_o = \sqrt{m/\lambda} = 1/\sqrt{\lambda}, \quad (m = \lambda)
\]

where, \(\Delta n\) is the number of drops in a range of \((d - \Delta d/2) \sim (d + \Delta d/2)\), \(N\) is the total number of the counted drops \((N \approx 600\) in this measurement\), \(\Delta t = \Delta d / \bar{d}_1\), and \(\Gamma(m)\) is the gamma function.

The standard deviation of the measured size distribution can be calculated from its definition,

\[
o_o = \Gamma(1-t) \frac{\Delta n}{N} \]

The curves in Fig.10 show the drop size distributions obtained by applying the standard deviations thus derived in Eqs. (12) and (13). In this calculation, \(\Delta d\) was taken as 10 \(\mu m\). Whereas the value of \(o_o\) ranged from 0.33 to 0.55 for the R 113 adiabatic annular flow, the present data showed a little higher values of \(o_o = 0.40 \sim 0.63\), the average being 0.50. Therefore, substituting \(o_o = 0.50\) into Eqs. (12) and (13), the averaged drop size distribution of the present data was expressed as

\[
\frac{\Delta n}{N} = 42.7 \left( \frac{d}{\bar{d}_1} \right)^{3.0}
\]

Beside the arithmetical mean diameter \(\bar{d}_1\), several kinds of mean values of the drop sizes have been used. When the gamma distribution is assumed for the drop sizes, these mean values can be expressed as simple functions of \(\bar{d}_1\) and \(\lambda\). For example,

\[
\bar{d}_2 = \left[ \frac{\Sigma d^2 \Delta n}{N} \right]^{1/2} = \bar{d}_1 \left[ \frac{\Gamma(3/2)}{\Gamma(1/2)} \right]^{1/2}
\]

and

\[
\bar{d}_3 = \left[ \frac{\Sigma d^3 \Delta n}{N} \right]^{1/3} = \bar{d}_1 \left[ \frac{\Gamma(5/2)}{\Gamma(1/2)} \right]^{1/2}
\]

\[
d_m = \left[ \frac{\Sigma d^3 \Delta d}{\Sigma d^2 \Delta d} \right]^{1/2}
\]

The averaged values of \(\bar{d}_2, \bar{d}_3, d_m\) and

![Fig.10 Distribution of drop sizes](image-url)

**Table 1.** Experimental data on size of entrained drops \((x_{in} = 0.178)\)

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<th>No.</th>
<th>(C_T) (kg/m^2.s)</th>
<th>(q/10^8) (W/m^2)</th>
<th>(x_a)</th>
<th>(u_a) (m/s)</th>
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<td>7.50</td>
<td>0.606</td>
<td>25.18</td>
<td>42.4</td>
<td>74.8</td>
<td>0.69</td>
<td>104.4</td>
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<td>754</td>
<td>8.09</td>
<td>0.661</td>
<td>27.53</td>
<td>33.4</td>
<td>49.9</td>
<td>0.54</td>
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<td>0.670</td>
<td>27.87</td>
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<td>887</td>
<td>8.80</td>
<td>0.610</td>
<td>31.03</td>
<td>27.4</td>
<td>36.6</td>
<td>0.46</td>
<td>44.7</td>
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$d_{2}$ derived from the measured drop size distribution were 1.12 $d_{1}$, 1.25 $d_{1}$, 1.41 $d_{1}$ and 1.57 $d_{1}$, respectively. On the other hand, these values are predicted from Eq. (15) to be 1.12 $d_{1}$, 1.23 $d_{1}$, 1.37 $d_{1}$ and 1.50 $d_{1}$, respectively. The data obtained in this measurement are summarized in Table 1.

Aszopardi et al. (20) measured the size of entrained drops in an air-water annular flow by applying a laser scattering technique, and expressed the cumulative mass fraction distribution by the Rosin-Rammler equation,

$$R = \exp \left[ -\left( \frac{d}{d_{c}} \right)^{n_{0}} \right] \quad (20)$$

where $R$ is the cumulative mass fraction of drops with diameters greater than $d$ mm, and $d_{c}$ and $n_{0}$ are the characteristic diameter and exponent, respectively. In case of entrained drops of the gamma distribution given in this paper, the cumulative mass fraction can be calculated by the following equation,

$$R = 1 - \int_{0}^{t} f(t) t^{3} dt / \int_{0}^{\infty} f(t) t^{3} dt \quad (21)$$

Fig. 11 shows comparison of the cumulative mass fraction derived from the measured drop size distribution with that calculated from Eq. (21). Dotted lines in this figure show the Rosin-Rammler equation with values of $d_{c}$ and $n_{0}$ so determined as to fit Eq. (21) at $(1 - R) = 0.2$ and 0.8. Both equations are in good agreement. The characteristic values of the Rosin-Rammler equation thus derived for the present data were $d_{c} = 40 \sim 115$ mm and $n_{0} = 2.4 \sim 3.5$, as shown in Table 1. Meanwhile, the characteristic exponents derived by Aszopardi et al. were $n_{0} = 2.0 \sim 2.5$, and those obtained by Karabelas (21) for liquid-liquid dispersions were in a range of $n_{0} = 2.3 \sim 3.3$.

6. Conclusions

As a result of experimental investigation on the heat transfer coefficient and the dryout condition of R-113 upward flow boiling in a uniformly heated tube, the following conclusions were obtained.

(1) Dividing the saturated boiling process into three regions, i.e. the nucleate boiling region, the transition region and the forced convection evaporation region, empirical correlations for the heat transfer coefficient in these regions are derived as the equations (5), (6) and (7), respectively.

(2) As the heat flux is increased, the liquid film flow rate at the exit end of the heating section decreases steadily and reaches the dryout condition. In a range of heat fluxes less than $9 \times 10^{6}$ W/m$^2$, the dryout condition occurs when the exit film flow rate decreases near to zero. However, in a range of higher heat fluxes, the dryout condition is reached at a state where some extent of liquid film remains at the exit end of heating section, and the liquid film flow rate at the dryout condition shows a trend to increase with increasing heat flux.

(3) The arithmetical mean diameter of entrained drops measured under the heat fluxes near the dryout condition is expressed as Eq. (10), the same as in the results with an adiabatic two-phase flow. The droop size distribution of the present data is described by the gamma distribution of a standard deviation $\sigma_{0} = 0.50$ as Eq. (15).

Fig. 11 Cumulative mass fraction distribution of entrained drops

Fig. 12 Viscous sublayer thickness of liquid film in forced convection evaporation region

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Appendix 1. Heat transfer characteristics in forced convection evaporation region

In the forced convection evaporation region, the dominant mechanism of heat transfer is the convection through a thin liquid film on the tube wall, but there may be some nucleate boiling under the film when the heat flux is increased. Assuming that the liquid film consists of a viscous sublayer and a turbulent layer, and that heat transfer resistances through the turbulent layer and the liquid-vapor interface are negligible, the thickness of the viscous sublayer is given by

$$\delta = k_{1} (T_{w} - T_{sat}) / q_{w} = k_{1} / h_{TP}$$

For investigating heat transfer characteristics in the forced convection evaporation region, the nondimensional thickness of
the viscous sublayer

\[
\delta^* = \frac{\delta}{\nu_1 \rho_1^{1/2} D_p} = \frac{\delta}{\nu_1} \left( \frac{D_p \delta_p}{4 \rho_1} \right)^{1/2}
\]

was calculated. The frictional pressure loss gradient \((\partial p/\partial \delta_p)_{D_p}\) in the above equation was predicted by applying the relationship between \(\delta_{1tt} = (\partial p/\partial \delta_p)_{D_p} (\partial p/\partial \delta_p)_{D_p}^{1/2}\) and \(x_{tt}\) proposed by Lockhart and Martinelli (22).

Fig. 12 shows the nondimensional viscous sublayer thickness thus derived from the measured heat transfer data in the forced convection evaporation region of Fig. 4. The abscissa of this figure represents the nondimensional liquid film thickness

\[
y_1^+ = \frac{y_1}{\nu_1 \rho_1^{1/2}}
\]

which was derived from an assumption of Karman's universal velocity profile being applicable to the liquid film and the following equation for the liquid film flow rate

\[
M_f = \pi D_p y_1^+ \left( \frac{\tau_w}{\rho_1} \right) dy^+ = \pi D_p y_1^+ \left( \frac{\tau_w}{\rho_1} \right)
\]

where, \(u^+ = u \left( \frac{\nu_1}{\rho_1} \right)^{1/2}, y^+ = y \left( \frac{\tau_w}{\rho_1} \right)^{1/2}\)

It has been known that value \(\delta^*\) of the single-phase turbulent flow obtained under the two-layers model assumption varies with the Prandtl number, and \(\delta^* = 11.6\) for \(Pr = 1, \delta^* = 5\) for \(Pr \to \infty\), and \(\delta^* = 7.7\) for the present experimental condition of \(Pr = 4.76\). The values of \(\delta^*\) in Fig. 12 are lower than 7.7 and show a trend to decrease with decreasing \(y_1^+\). Although the values were obtained by rough assumptions, the result of Fig. 12 suggests that some nucleate boiling heat transfer occurs at the same time even in the forced convection evaporation region of the present experiment.

Appendix 2. Drop deposition rate and carry-over flow rate

The drop deposition rate in an unheated tube was calculated by

\[
M_{dp} = K C \rho_d D \delta_n
\]

where, \(K\) is the drop transfer coefficient \(m/s\), \(C\) is the mean drop concentration in the vapor flow \(kg/m^3\), and \(\delta_n\) denotes the length of the unheated tube (0.34 m). The drop concentration is defined as

\[
C = \frac{M_d}{\rho_d \rho_d u_d + \rho_d u_d + \rho_1}
\]

In this calculation, the vapor flow rate was obtained by \(M_g = \rho_v \rho_x D_p \delta_p / (\pi D_p^2 / 4)\) and \(u_d = u_d / 1.1\) respectively. The value of \(K\) was derived by substituting the measured values of \(M_g \rho_d kg/s\) and \(\delta_n \mu m\) at the exit end of heating section into the following empirical equation (3),

\[
K = \frac{v_g}{D \rho_d} d_m^{-1.9} \left( \frac{\rho_g}{\rho_d} \right)^{0.286} \exp\left( -k_2 \frac{M_g}{\pi D^2 / 4} \right)
\]

where, \(k_1 = 2.37 \times 10^6, k_2 = 6.27 \times 10^{-6}\) for \(M_g / (\pi D^2 / 4) \approx 70 \text{ kg/m}^2\text{s}\) and \(k_1 = 1.04 \times 10^6, k_2 = 1.00 \times 10^{-6}\) for \(M_g / (\pi D^2 / 4) > 70 \text{ kg/m}^2\text{s}\).

The carry-over flow rate in the film separator was calculated by the following empirical equations (3) which were determined from the experimental data for R 113 flow in a film separator of the same dimensions as the present apparatus,

\[
M_{fin} = 3.55 \exp \left( 0.0822 M_f(1 - x_f) \right)\]

\[
-10^{-5} \left( \rho_g D / \rho_d \right) \quad \text{kg/h}
\]

for \(M_f(1 - x_f) \leq 42 \text{ kg/h}\)

\[
M_{fin} = 70.8 \exp \left( 0.0109 M_f(1 - x_f) \right)\]

\[
-10^{-5} \left( \rho_g D / \rho_d \right) \quad \text{kg/h}
\]

for \(M_f(1 - x_f) > 42 \text{ kg/h}\)

References


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