Turbulent Film Heat-transfer Coefficients during Condensation in Tubes

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Equations for the local film heat-transfer coefficient with a turbulent film are obtained based on an approximate boundary layer treatment and compared with experiment and other well-known predictive methods.

NOTATION

D	Tube diameter
G	Mass flux of mixture
k_L	Thermal conductivity of liquid
Nu	Nusselt number, $\alpha D/k_L$
Nu_{LO}	Nusselt number if mixture flows as liquid, $\alpha_{LO} D/k_L$
Pr	Prandtl number
Dp_f	Two-phase pressure gradient due to friction
Dp_{LO}	Pressure gradient due to friction if mixture flows as liquid
Re_L	Liquid Reynolds number, $\{(1-x)GD/\mu_L\}$
Re_{LO}	Reynolds number if mixture flows as liquid
R_ϕ	Group defined by equation (12)
V^+	Friction velocity, $\{(\tau_w/\rho_L)^{1/2}\}$
x	Mass dryness fraction
α	Heat-transfer coefficient
α_{LO}	Heat-transfer coefficient if mixture flows as water
δ	Thickness of condensate film
δ^+	Dimensionless thickness of condensate film, $(\delta\rho_L/V^+/\mu_L)$
λ_{LO}	Friction factor if mixture flows as liquid
μ_L	Absolute viscosity of liquid
ρ_G	Density of vapour
ρ_L	Density of liquid
τ_w	Shear stress at wall
ϕ^2_{LO}	Two-phase friction multiplier, (Dp_f/Dp_{LO})

1 INTRODUCTION

A well-known method (**1**, **2**) of predicting the heat-transfer coefficient across a turbulent film in condensation is to use the equation

$$\frac{\alpha}{\alpha_{LO}} = \phi_{LO} \tag{1}$$

which relates the ratio of the film coefficient to the single-phase coefficient if the mixture flows as liquid to the square root of the two-phase multiplier for friction.

Starting from boundary layer theory, a simple modification to this method which improves its performance at high dryness fractions is obtained in this note.

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2 BASIC EQUATIONS

Boundary layer theory (**3**, **4**) leads to the following equations for the heat-transfer coefficients for a turbulent film

$\delta^+ < 30$

$$\frac{\alpha\mu_L}{k_L\rho_L V^+} = \left[5 + \frac{5}{Pr}\ln\left\{1 + Pr\left(\frac{30}{\delta^+} - 1\right)\right\}\right]^{-1} \tag{2}$$

$\delta^+ > 30$

$$\frac{\alpha\mu_L}{k_L\rho_L V^+} = \left\{5 + \frac{5}{Pr}\ln(1 + 5Pr) + \frac{2{\cdot}5}{Pr}\ln\frac{\delta^+}{30}\right\}^{-1} \tag{3}$$

This assumes a uniform film round the wall, and neglects the influence of gravity on the shear distribution. Kutateladze (**4**), Bae *et al.* (**5**), and Rohsenow, Webber, and Ling (**6**) have developed equations including gravitational effects.

The dimensionless thickness, δ^+, is given with sufficient accuracy by

$$\delta^+ = (Re_L/2)^{1/2} \quad \text{for } Re_L < 1000 \tag{4}$$

$$\delta^+ = 0{\cdot}0504 Re_L^{7/8} \quad \text{for } Re_L > 1000 \tag{5}$$

Equations (2)–(5) can be expressed, for convenience, as

$$\frac{\alpha\mu_L}{k_L\rho_L V^+} = F(Pr, Re_L) \tag{6}$$

The Nusselt number, where the total mixture flows as liquid, as used by Ananiev *et al.* (**1**), is

$$Nu_{LO} = 0{\cdot}021 Re_{LO}^{0{\cdot}8} Pr^{0{\cdot}43} \tag{7}$$

where
$$Re_{LO} = \frac{GD}{\mu_L} \tag{8}$$

The wall shear stress can be expressed as

$$\begin{aligned}\tau_w &= -\frac{D}{4}Dp_f \\ &= -\frac{D}{4}Dp_{LO}\phi^2_{LO} \\ &= \frac{\lambda_{LO}}{8}\left(\frac{\mu_L}{D}\right)^2\frac{Re^2_{LO}}{\rho_L}\phi^2_{LO}\end{aligned} \tag{9}$$

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Table 1

R_ϕ as function Re_L and Pr

Reynolds No	Prandtl No 0·5	0·75	1·0	1·5	2·0	5·0	10·0	50·0
100	2·23	1·90	1·69	1·45	1·30	0·93	0·73	0·40
200	1·82	1·59	1·44	1·28	1·17	0·90	0·73	0·42
300	1·64	1·46	1·34	1·21	1·13	0·90	0·74	0·44
400	1·54	1·38	1·29	1·17	1·10	0·90	0·75	0·45
500	1·46	1·33	1·25	1·15	1·08	0·90	0·76	0·46
750	1·35	1·25	1·19	1·11	1·06	0·90	0·77	0·47
1 000	1·28	1·20	1·15	1·09	1·05	0·91	0·79	0·48
1 500	1·14	1·10	1·07	1·03	1·01	0·91	0·80	0·50
2 000	1·10	1·07	1·05	1·03	1·01	0·92	0·81	0·51
5 000	1·00	1·01	1·02	1·02	1·02	0·96	0·87	0·56
10 000	0·94	0·98	1·00	1·02	1·03	1·00	0·91	0·60
15 000	0·92	0·96	0·99	1·02	1·03	1·02	0·94	0·62
20 000	0·90	0·95	0·99	1·02	1·04	1·03	0·96	0·64

where, using a Blasius equation,

$$\lambda_{LO} = \frac{0{\cdot}186}{Re_{LO}^{0{\cdot}2}} \tag{10}$$

Equation (6) can therefore be expressed as

$$Nu = 0{\cdot}151 Re_{LO}^{0{\cdot}9} \phi_{LO} F(Pr, Re_L) \tag{11}$$

and from eqs. (7) and (11)

$$\frac{\alpha}{\alpha_{LO}} = \frac{R_\phi \phi_{LO}}{(1-x)^{0{\cdot}1}} \tag{12}$$

where
$$R_\phi = \frac{7{\cdot}2 Re_L^{0{\cdot}1} F(Pr, Re_L)}{Pr^{0{\cdot}43}} \tag{13}$$

Table 1 presents the term R_ϕ for a range of Reynolds and Prandtl numbers.

3 THE TWO-PHASE MULTIPLIER

The boundary layer theory used above suggests the use of annular flow theory to evaluate the two-phase multiplier in eq. (12). As shown by Wallis (**7**) annular flow theory gives

$$\phi_{LO}^2 = \frac{(1-x)^{2-n}}{\alpha_L^2} \tag{14}$$

where α_L is the liquid fraction. It is well known, however, that the multiplier is given to equal accuracy using

$$\phi_{LO} = \left\{1 + x\left(\frac{\rho_L}{\rho_G} - 1\right)\right\}^{1/2} \tag{15}$$

This is obtained from homogeneous theory. Equations (14) and (15) have in fact been combined (**8**) to give an equation for the phase velocity ratio which is in good agreement (**9**) with data. As eq. (15) is more readily used than eq. (14), and as it is the equation used by Ananiev (**1**) and Boyko (**2**), it is used in the subsequent analysis.

From eqs. (12) and (15)

$$\frac{\alpha}{\alpha_{LO}} = \frac{R_\phi}{(1-x)^{0{\cdot}1}}\left\{1 + x\left(\frac{\rho_L}{\rho_G} - 1\right)\right\}^{1/2} \tag{16}$$

From Table 1 it can be seen that, for a considerable range of conditions, R_ϕ approximates to unity; for example, in the range $1500 < Re_L < 20000$ and $0{\cdot}75 < Pr < 5{\cdot}0$ assuming R_ϕ to be unity introduces an error no greater than 10 per cent. Hence approximately

$$\frac{\alpha}{\alpha_{LO}} = \frac{1}{(1-x)^{0{\cdot}1}}\left\{1 + x\left(\frac{\rho_L}{\rho_G} - 1\right)\right\}^{1/2} \tag{17}$$

Equations (1) and (15), of course, give

$$\frac{\alpha}{\alpha_{LO}} = \left\{1 + x\left(\frac{\rho_L}{\rho_G} - 1\right)\right\}^{1/2} \tag{18}$$

Figure 1 compares data obtained by Kosky and Staub

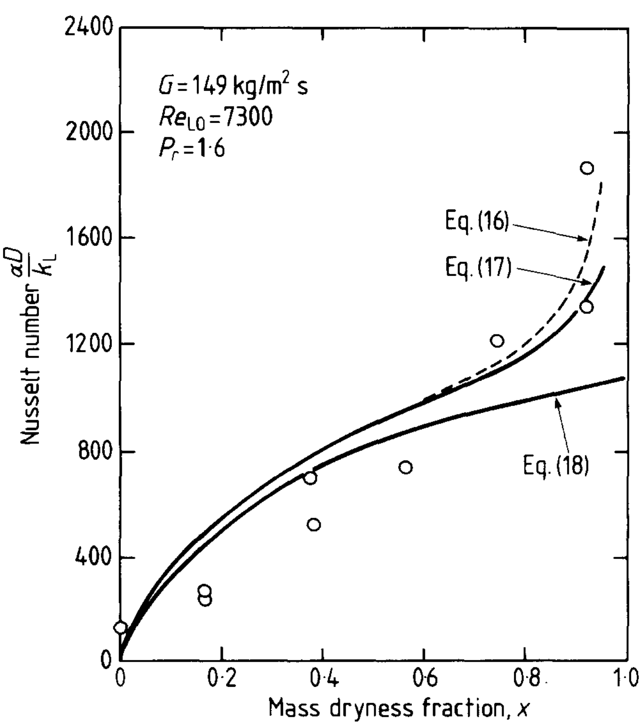

Fig. 1. Comparison of equations for local heat-transfer coefficients: steam at pressure of 1·45 bar condensing in 12·6 mm bore tube

(**3**) for steam being condensed in a pipe with curves corresponding to eqs. (16), (17), and (18).

The trends of the data are correctly predicted by both eqs. (16) and (17). All three equations tend to overpredict at lower mass dryness fractions. There is, however, ample evidence (**1**, **2**) that this overprediction is not generally true in the case of eq. (18).

References (**1**) and (**2**) show that eq. (17) gives good agreement with experiment; however, examination of the comparison with experiment in those papers suggests a further improvement would be obtained using eq. (16).

Equation (7) has been used for the single phase coefficient to facilitate comparison with references (**1**) and (**2**). If the Dittus–Boelter equation were used (**10**) then

$$R_{\phi,\,\mathrm{DB}} = \frac{0{\cdot}875 R_\phi}{Pr^{0\cdot17}} \tag{19}$$

The choice of equation does not influence the comparison in Fig. 1 except where x is practically zero.

4 CONCLUSIONS

The analysis has served the following purposes.

(1) It provides the theoretical basis of eq. (1).

(2) It indicates that at high mass dryness fraction the approximate eq. (17) should be more accurate than eq. (18).

(3) The tabulated values of R_ϕ indicate when the approximate equations may be used.

(4) Equation (12) and Table 1 provide a convenient method of using boundary layer theory. There is some evidence that this procedure may overpredict the film coefficient.

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