Available online at www.sciencedirect.com

SCIENCE DIRECT®

International Journal of Heat and Mass Transfer 49 (2006) 805–809

International Journal of HEAT and MASS TRANSFER

www.elsevier.com/locate/ijhmt

Technical Note

Interface temperature and heat transfer in forced convection laminar film condensation of binary mixtures

K. Stephan

Institute of Technical Thermodynamics and Thermal Process Engineering, University of Stuttgart, D-70550 Stuttgart, Germany

Received 27 May 2005; received in revised form 27 September 2005
Available online 1 December 2005

Abstract

Based on numerical solutions of the balance equations accurate empirical correlations for filmwise condensation in forced convection flow over a horizontal flat plate are presented. The correlations are designed to be also correct for the limiting case of zero and infinite condensation rate. They are applied to determine the interface temperature between liquid and vapour phase and hence the condensation rate and heat flux. The results are in excellent agreement with those from numerical solutions of the balance equations. It turns out that the usually adopted film theory for the vapour-phase mass transfer overestimates the size of a heat exchanger.

Keywords: Mixture condensation; Filmwise condensation; Interface temperature; Binary mixtures

1. Introduction

Film condensation of mixed vapours, particularly of binary mixtures, has been the subject of a great number of publications in the last two or three decades. Fundamental studies were first made by Sparrow and Lin [1], Minkowycz and Sparrow [2] and Sparrow et al. [3]. They explored analytically the condensation of gases containing non-condensables in forced convection flow along a horizontal flat plate. Following their studies a very large number of publications have been devoted to condensation of gas mixtures in laminar or turbulent flow along horizontal, vertical or inclined plates, inside or outside tubes and in tube bundles. An overview can be found in a paper of Lucas [4] or in the text-books and hand-books, e.g. in [5,6]. In many of these studies numerical solutions of the governing equations for energy and mass have been presented for a wide variety of circumstances.

The driving force for heat transfer in the liquid film is the difference between interface temperature T_i and wall temperature T_w. As the wall temperature T_w often is known or can be expressed with the aid of the heat transfer rate between wall and cooling liquid, the main remaining task is the determination of the interface temperature T_i. It cannot be found easily from a solution of the balance equations, because this requires several tedious iteration procedures. To avoid them heat exchanger design handbooks [5–7] recommend the film theory as an approximate method, though it is known that mass fluxes predicted with the film theory for convective flow with suction, as is the case for condensing vapour, are too low and approach exact values only for vanishing suction [8].

The present work provides correlations for the problem of condensing binary mixtures to determine the interface temperature in forced convective flow along a horizontal flat plate. The correlations were derived from exact numerical solutions of the balance equations. The results obtained with these correlations are in excellent agreement with those from the numerical solutions, but avoid the many iteration procedures of numerical solutions of the balance equations. An example demonstrates the application. As can be shown the usually adopted film theory underestimates the condensation rate and thus leads to oversized condensers.

E-mail address: Stephan@itt.uni-stuttgart.de

0017-9310/$ - see front matter © 2005 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2005.10.007

Nomenclature

c_{pL}	specific heat of condensate, J kg^{-1} K^{-1}
c_F	friction factor, $c_F = \tau_\delta/\lfloor(\rho_G u_\infty)^2/2\rfloor$
D	binary diffusion coefficient, m^2 s^{-1}
F_0	suction parameter, Eqs. (2) and (5)
Δh_v	condensation enthalpy, J kg^{-1}
$\dot{m}$	interfacial mass flux, kg m^{-2} s^{-1}
T	temperature, K
t	temperature, °C
u_∞	free-stream velocity, m s^{-1}
x	liquid mass fraction
x	co-ordinate in flow direction, m
$\tilde{x}$	liquid molar fraction
y	gas mass fraction
$\tilde{y}$	gas molar fraction
Y	mass fraction ratio, $Y = (y_i - y_\infty)/(y_i - x_i)$
z	co-ordinate perpendicular to wall, m

Greek symbols

α	heat transfer coefficient, W m^{-2} K^{-1}
β	mass transfer coefficient, m s^{-1}
δ_L	liquid film thickness, m
η_δ	dimensionless film thickness, Eq. (3)
λ	thermal conductivity, W m^{-1} K^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}

Dimensionless numbers

E	dimensionless driving temperature difference, $E = c_{pL}(T_i - T_w)/(Pr_L \Delta h_v)$
Nu	Nusselt number, $Nu = \alpha x/\lambda$
Re	Reynolds number, $Re = u_\infty x/\nu$
R	property parameter, $R = (\rho_L/\rho_G)(\nu_L/\nu_G)^{1/2}$
Pr	Prandtl number, $Pr = \nu\rho c_p/\lambda$
Sc	Schmidt number, $Sc = \nu/D$
Sh	Sherwood number, $Sh = \beta x/D$

Subscripts

c	coolant
i	interface
G	gas
L	liquid
o	vanishing suction
∞	in free stream
ref	reference
x	with x as length in dimensionless quantity
w	wall

2. The interface temperature

We consider the laminar condensation of a binary vapour mixture, as shown in Fig. 1. The vapour is assumed to be saturated with a free-stream equilibrium composition y_∞ of the low boiling component. Its free-stream temperature is T_∞ and the free-stream velocity u_∞. At the interface between vapour and liquid film the two phases have equilibrium composition y_i in the vapour and x_i in the liquid phase. The temperature T_i then follows from an energy balance at the interface [9]

$$\dot{q} = \frac{\lambda_L}{\delta_L}(T_i - T_w) = \alpha_G(T_\infty - T_i) + \rho_G\beta_G Y \Delta h_v \tag{1}$$

with $Y = (y_i - y_\infty)/(y_i - x_i)$ when both components are condensable.

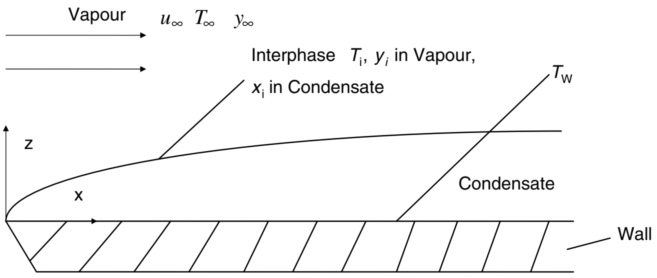

Fig. 1. Physical model.

We have $0 \leqslant Y \leqslant 1$ with $Y = 0$ when no condensation occurs and $Y = 1$, namely $x_i = y_\infty$, for total condensation. If the low boiling component is incondensable, for example in a steam–air mixture, we have $x_i = 0$, and Y reduces to $Y = (y_i - y_\infty)/y_i$. The condensation rate in Eq. (1) is $-\dot{m} = \rho_G\beta_G Y$.

The negative sign herein indicates that the mass flow is directed in the negative direction of the co-ordinate z in Fig. 1.

In Eq. (1) the film thickness δ_L, the heat transfer coefficient α_G on the vapour side of the interface and the corresponding mass transfer coefficient β_G are all complicated functions of the interface temperature T_i, because these coefficients depend on the driving temperature difference $T_w - T_i$, and the thermophysical properties depend on T_w and T_i, as well.

At present δ_L, α_G and β_G must be determined from numerical solutions of the balance equations for mass and energy. These are non-linear differential equations and can only be solved numerically. In order to solve them an iterative procedure is imperative: If the wall temperature T_w and the streamline temperature T_∞ are given we need to estimate the interface temperature T_i, and in addition the so-called suction parameter defined as

$$F_0 = \frac{-\dot{m}}{\rho_G u_\infty} 2Re_{Gx}^{1/2}. \tag{2}$$

Both, F_0 and T_i, should fit together to fulfill Eq. (1). Otherwise the balance equations must be solved repeatedly until both parameters fit together. This laborious procedure can be avoided with the aid of approximate correlations for δ_L, α_G and β_G of sufficient high accuracy as presented in the following.

3. Correlations for heat and mass transfer

The film thickness δ_L in Eq. (1) was first calculated by Emmons and Leigh [10]. They numerically solved the balance equations and tabulated the results, which are indeed very accurate. We extended their tables to a greater number of values F_0 solving again numerically the balance equations. The results could be correlated by

$$Nu_{Lx}Re_{Lx}^{-1/2} = \frac{1}{\eta_\delta} = \left(\frac{0.332 - 0.08(RE)^{0.443}}{1.745(RE)^{2/3} + 0.992(RE)^{1.66}} + 0.25\right)^{1/2} \tag{3}$$

with $Nu_{Lx} = \frac{\alpha_L x}{\lambda_L} = \frac{x}{\delta_L}$, $R = \frac{\rho_L}{\rho_G}\left(\frac{\nu_L}{\nu_G}\right)^{1/2}$, $E = \frac{c_{pL}(T_i - T_w)}{Pr_L\,\Delta h_v}$, and $\eta_\delta = \frac{\delta_L}{x}Re_{Lx}^{1/2}$.

The dimensionless parameters R, E, η_δ herein were first introduced by Minkowycz and Sparrow [2].

The maximum error in Nu_{Lx} is 0.27% over the entire range $0 \leqslant RE \leqslant \infty$. As can be seen from Eq. (3) the film thickness in forced convection condensation increases with length according to $\delta_L \sim x^{1/2}$.

The conduction term $\alpha_G(T_G - T_i)$ in Eq. (1) is often much smaller than the condensation term $\rho_G\beta Y\,\Delta h_v = -\dot{m}\,\Delta h_v$, except for small condensation rates $Y \to 0$. Most often this term therefore can be neglected. In the following we do so. How to proceed when heat conduction may not be neglected is explained in the appendix of the paper.

A correlation for mass transfer was dealt with by Rose [11]. It can be written in the form

$$Sh_{Gx}Re_{Gx}^{-1/2} = \varphi_1(Sc)\frac{1}{1 + 0.427F_0^{1.14}Sc^{0.93}} + \frac{F_0 Sc}{2}. \tag{4}$$

Herein the suction parameter $F_0(RE)$ is given by

$$F_0 = 2RE/\eta_\delta \tag{5}$$

with $\eta_\delta(RE)$ from Eq. (3).

Eq. (4) is valid in the entire range of the suction parameter $0 \leqslant F_0 \leqslant \infty$, and also valid for $0 \leqslant Sc \leqslant \infty$.

The function $\varphi_1(Sc)$ was approximated by

$$\varphi_1(Sc) = \frac{Sc^{1/2}}{\sqrt{\pi}(1 + 1.973Sc^{0.272} + 21.29Sc)^{1/6}} \tag{6}$$

also valid over $0 \leqslant Sc \leqslant \infty$.

Eq. (6) is of higher accuracy than previous correlations given by Rose [12] and by Baehr and Stephan [13], though these correlation agreed already well with numerical results obtained from the balance equations. The maximum error in Sh_{Gx}, Eq. (4), is over the entire range of RE and Sc below 3%, and below 1% in the interesting range $0.3 \leqslant Sc \leqslant 1$ for gases within $0.01 \leqslant RE \leqslant 10$.

As follows from this, Eq. (4) is of the form

$$Sh_{Gx}Re_{Gx}^{-1/2} = f(Sc, RE). \tag{7}$$

The mass transfer coefficients β_G is a function of Sc and RE, and thus contains also the unknown interface temperature T_i.

For vanishing suction $F_0 \to 0$, respectively $RE \to 0$, Eq. (4) reduces to

$$Sh_{Gx}Re_{Gx}^{-1/2} = \varphi_1(Sc) \tag{8}$$

with $Sh_{Gx} = \beta_{G0}x/D$ and $\varphi_1(Sc)$ from Eq. (6).

From the definition of the suction parameter, Eq. (2), we obtain the condensation rate $\dot{m}$ after elimination of Re_{Gx} with the aid of Eq. (8).

$$\frac{-\dot{m}}{\rho_G\beta_{G0}} = \frac{1}{2}\frac{Sc}{\varphi_1(Sc)}F_0 \tag{9}$$

with $F_0(RE)$ from Eq. (5).

Thus the condensation rate depends on parameter RE and the Schmidt number Sc. The temperature difference $T_i - T_w$ in parameter

$$RE = \frac{\rho_L}{\rho_G}\sqrt{\frac{\nu_L}{\nu_G}}\frac{c_{pL}(T_i - T_w)}{Pr_L\,\Delta h_v}$$

is only a few Kelvin. An error in the interface temperature therefore greatly affects parameter RE. On the other hand the suction parameter $F_0(RE)$ in the limit of $RE \to \infty$ becomes $F_0 = RE$ and tends to $F_0 = 0.87(RE)^{2/3}$ in the limit $RE \to 0$. As a consequence the condensation rate is also very sensitive in errors in the interface temperature.

The correlations for heat and mass transfer given here are of sufficient high accuracy and yield very low errors in the interface temperature, as the following example will demonstrate.

The friction factor $c_F = \tau_\delta/((\rho_G u_\infty^2)/2)$ can also be described as a function of parameter RE

$$\frac{c_F}{2}Re_{Gx}^{1/2} = \frac{0.332}{1 + 0.427F_0^{1.14}} + \frac{F_0}{2} \tag{10}$$

with $F_0(RE)$ from Eq. (5).

4. The calculation procedure

The procedure to determine the interface temperature from Eq. (1) is as follows:

For a given wall temperature T_w and gas composition y_∞ we first estimate the interface temperature T_i and then calculate the thermophysical properties of the liquid and gas phase at a reference state. For the liquid Poots and Miles [14] and Denny and Mills [15] recommended as reference state

$$T_{ref,L} = T_w + \tfrac{1}{3}(T_i - T_w), \quad x_{ref,L} = (T_{ref}, p),$$

and for the gas phase Lucas [4] recommended $T_{ref,G} = \frac{1}{2}(T_i + T_\infty)$ and the species concentration $y_{ref,G}$ of the saturated vapour at $T_{ref,G}$.

With these reference states the thermophysical properties can be determined and from them the quantities δ_L and β_G follow from Eqs. (3) and (4). With the compositions

y_i, x_i at the interface determined from the phase equilibrium of the mixture, the value Y in Eq. (1) is known as well.

If the estimated interface temperature T_i was correct, then Eq. (1) is fulfilled. Otherwise iterations become necessary.

If instead of the wall temperature T_w the coolant temperature T_c is given, we have

$$\frac{\lambda_L}{\delta_L}(T_i - T_w) = k(T_w - T_c),$$

where k stands for the overall heat transfer coefficient between coolant and wall. The wall temperature in Eq. (1) then is

$$T_w = \frac{(\lambda_L/\delta_L)T_i + kT_c}{(\lambda_L/\delta_L) + k}. \tag{11}$$

In the case of a constant wall temperature T_w the interface temperature is also constant and independent from the free-stream velocity u_∞ and the length co-ordinate, because they cancel in Eq. (1). Different from this follows from Eqs. (1) and (11) that the interface temperature T_i becomes a function of free-stream velocity and length co-ordinate when the coolant temperature T_c is prescribed, except for the limiting case $\lambda_L/\delta_L \ll k$.

5. An example

As an example we determine the interface temperature of a steam–air mixture with a mole fraction $\tilde{y}_\infty = 0.05$ of air, corresponding to a mass fraction of $y_\infty = 0.078$, when flowing along a horizontal plate of constant wall temperature $T_w = 331.15$ K (60 °C), a free-stream temperature of $T_\infty = 373.15$ K (100 °C) and pressure 0.1 MPa.

As the interface temperature is between the dew point temperature 371.33 K (98.18 °C) and the wall temperature 331.15 K, a reasonable estimate seems to be $T_i = 353.15$ K (80 °C), which is almost the arithmetic mean between dew point and wall temperature.

With the thermophysical properties of the liquid state at

$$T_{\mathrm{ref,L}} = T_w + \tfrac{1}{3}(T_i - T_w) = 337.82\ \mathrm{K}\ (64.67\ °\mathrm{C})$$

and those of the gas state at $T_{\mathrm{ref,G}} = \frac{1}{2}(T_i + T_\infty) = 363.15$ K (90 °C) and $\tilde{y}_{\mathrm{ref,G}} = 0.298$ of the saturated vapour at 363.15 K, we find $RE = 0.198(T_i - T_w) = 0.198(T_i - 331.15)$.

With this we obtain the suction parameter F_0 from Eq. (5) as a function of the interface temperature, and the film thickness δ_L from Eq. (3). Together with the Schmidt number $Sc \cong 0.55$ of the steam–air mixture we obtain the mass transfer coefficient β_G from Eq. (4) as a function of RE. In a first calculation we neglected the conduction term $\alpha_G(T_\infty - T_i)$ in Eq. (1).

Calculation of the interface temperature performed with a personal computer delivers an interface temperature $T_i = 350.96$ K (77.81 °C) in very good agreement with the exact value from a numerical solution of the balance equations $T_i = 350.93$ K (77.78 °C). Both interface temperatures are almost the same as the estimated value of 80 °C. With the exact value $T_i = 350.93$ K of the interface temperature we obtain from Eq. (9) a condensation rate of $-\dot{m}/\rho_G\beta_{G0} = 3.73$. If the conduction term in Eq. (1) is taken into account we obtain a slightly higher interface temperature $T_i = 351.65$ K (78.50 °C) as follows from the equations given in the appendix.

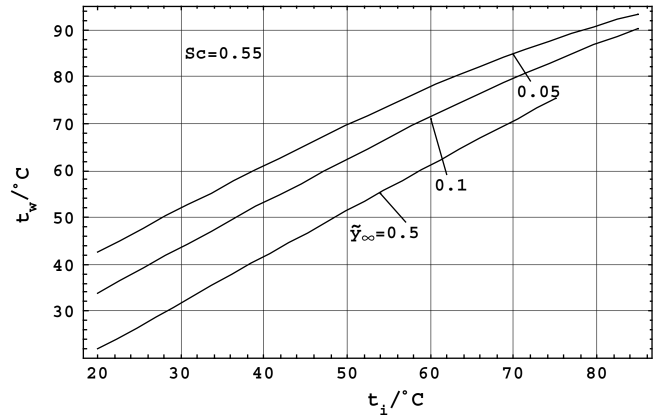

Fig. 2. Variation of interface temperature T_i with wall temperature T_w and mole fraction $\tilde{y}_\infty$ of air.

Fig. 2 shows the interface temperature plotted over the wall temperature for different mole fractions $\tilde{y}_\infty$ of the air, a total pressure of 0.1 MPa and a constant Schmidt number $Sc = 0.55$. The interface temperature increases with wall temperature and decreases with the air content. At an air content of $\tilde{y}_\infty = 0.5$ the interface temperature is only slightly higher than the wall temperature. The heat flux tends to zero.

6. Comparison with results from the film theory

The usually adopted procedure to determine the interface temperature is based on the assumption that the gas-side heat and mass transfer may be calculated with the aid of the film theory. In this theory we can either assume a constant molar reference volume in the gas phase equivalent to the assumption $p, T = \mathrm{const}$ for ideal gases, or use a constant reference density. Because the previous equations were based on the concept of a constant reference density, the following application of the film theory is also based on this assumption. The mass flux towards the interface is then given by

$$\dot{m} = \rho_G\beta_{G0}\ln\frac{y_\infty - x_i}{y_i - x_i}, \tag{12}$$

an equation equivalent to that assuming a constant molar volume, e.g. [16].

The mass transfer coefficient β_{G0} without suction is given by Eq. (8). A similar equation holds for heat transfer without suction

$$Nu_{Gx}Re_{Gx}^{-1/2} = \varphi_2(Pr_G) \tag{13}$$

with $\varphi_2(Pr_G)$ equivalent to Eq. (6)

$$\varphi_2(Pr_{\mathrm{G}}) = \frac{Pr_{\mathrm{G}}^{1/2}}{\sqrt{\pi}(1 + 1.973 Pr_{\mathrm{G}}^{0.272} + 21.29 Pr_{\mathrm{G}})^{1/6}}. \tag{14}$$

The heat transfer coefficient α_{G} according to the film theory is given by

$$\alpha_{\mathrm{G}} = \alpha_{\mathrm{G0}}\varsigma \quad \text{with } \varsigma = \frac{\phi}{e^{\phi} - 1} \quad \text{and} \quad \phi = \frac{\dot{m} c_{p\mathrm{G}}}{\alpha_{\mathrm{G0}}}. \tag{15}$$

With Eqs. (8), (12) and (13) the factor ϕ can be written

$$\phi = \frac{\varphi_1(Sc)}{Sc} \frac{Pr_{\mathrm{G}}}{\varphi_2(Pr_{\mathrm{G}})} \ln \frac{y_\infty - x_{\mathrm{i}}}{y_{\mathrm{i}} - x_{\mathrm{i}}}. \tag{16}$$

With this, and with Eq. (3) and $\rho_{\mathrm{G}}\beta_{\mathrm{G}}Y = -\dot{m}$, Eq. (1) converts into an equation containing the unknown interface temperature T_{i}:

$$\frac{2RE}{\eta_\delta} = F_0 = 2\left[\frac{\varphi_1(Sc)}{Sc} \ln \frac{y_{\mathrm{i}} - x_{\mathrm{i}}}{y_\infty - x_{\mathrm{i}}} + \frac{\varphi_2(Pr_{\mathrm{G}})}{Pr_{\mathrm{G}}} \varsigma \frac{c_{p\mathrm{G}}(T_\infty - T_{\mathrm{i}})}{\Delta h_{\mathrm{v}}}\right] \tag{17}$$

with η_δ from Eq. (3).

If we neglect the heat conduction towards the interface represented by the last term in Eq. (17) we obtain for the previous example of condensation of a steam–air mixture an interface temperature $T_{\mathrm{i}} = 343.6$ K (70.5 °C) instead of 350.96 K (77.8 °C). The condensation rate is $-\dot{m}/\rho_{\mathrm{G}}\beta_{\mathrm{G0}} = 2.29$ instead of the exact value $-\dot{m}/\rho_{\mathrm{G}}\beta_{\mathrm{G0}} = 3.73$ in the previous example. If the conductive heat flux is taken into account the results are slightly different $T_{\mathrm{i}} = 343.8$ K (70.7 °C) and the condensation rate becomes $-\dot{m}/\rho_{\mathrm{G}}\beta_{\mathrm{G0}} = 2.30$. Thus, the condensation rate when determined with the film theory is too low. In this example it is about 38% lower than the exact value. As a consequence a condenser would be oversized when designed with the film theory.

Appendix. Interface temperature when heat conduction from gas to interface may not be neglected

The heat transfer in the gas phase is comparable to that along a flat plate with suction, neglecting the influence of the mass diffusion in the balance equation of the temperature field. It may be written in analogy to Eq. (4) as

$$Nu_{\mathrm{G}x} Re_{\mathrm{G}x}^{-1/2} = \varphi_2(Pr_{\mathrm{G}}) \frac{1}{1 + 0.427 F_0^{1.14} Pr_{\mathrm{G}}^{0.93}} + \frac{F_0 Pr_{\mathrm{G}}}{2}. \tag{A.1}$$

In Eq. (1) we introduce the definitions

$$\eta_\delta = \frac{\delta_{\mathrm{L}}}{x} Re_{\mathrm{L}x}^{1/2}, \tag{A.2}$$

$$-\dot{m} = \rho_{\mathrm{G}}\beta_{\mathrm{G}} Y = \tfrac{1}{2}\rho_{\mathrm{G}} u_\infty Re_{\mathrm{G}x}^{-1/2} F_0. \tag{A.3}$$

With this we eliminate δ_{L} and $\rho_{\mathrm{G}}\beta_{\mathrm{G}}Y$ in Eq. (1) and after some rearrangement obtain

$$\frac{1}{\eta_\delta} RE = \left\{\varphi_2(Pr_{\mathrm{G}}) \frac{1}{1 + 0.427 F_0^{1.14} Pr_{\mathrm{G}}^{0.93}} + \frac{F_0 Pr_{\mathrm{G}}}{2}\right\} E_{\mathrm{G}} + \frac{F_0}{2} \tag{A.4}$$

with RE as before and $E_{\mathrm{G}} = \frac{c_{p\mathrm{G}}(T_\infty - T_{\mathrm{i}})}{Pr_{\mathrm{G}}\,\Delta h_{\mathrm{v}}}$, $\eta_\delta(RE)$ from Eq. (3) and $\varphi_2(Pr)$ from Eq. (14). As we can see from Eq. (A.4) heat conduction may be neglected, when the first term on the right-hand side is much smaller than the last term. Then Eq. (A.4) becomes identical with Eq. (5).

In the case of small suction rates, $Y \to 0$, when heat conduction is no more negligible we may proceed as follows: We first estimate the interface temperature T_{i} and with this determine the values RE, Pr_{G} and E_{G}. Keeping in mind that η_δ in Eq. (A.4) depends on parameter RE according to Eq. (3), we can then determine the suction parameter F_0 from Eq. (A.4). As follows from Eqs. (A.3) and (4) the suction parameter should fulfill the equation

$$\varphi_1(Sc) \frac{1}{[1 + 0.427 F_0^{1.14} Sc^{0.93}] F_0 Sc/2} + 1 = \frac{1}{Y}. \tag{A.5}$$

If this equation is not fulfilled, the interface temperature must be estimated once again until Eq. (A.5) is fulfilled.

References

[1] E.M. Sparrow, S.H. Lin, Condensation heat transfer in the presence of a noncondensable gas, J. Heat Transfer 86C (1964) 430–436.

[2] W.J. Minkowycz, E.M. Sparrow, Condensation heat transfer in the presence of noncondensables, interfacial resistance, superheating variable properties and diffusion, Int. J. Heat Mass Transfer 9 (1966) 1125–1144.

[3] E.M. Sparrow, W.J. Minkowycz, M. Saddy, Forced convection in the presence of noncondensables and interfacial resistance, Int. J. Heat Mass Transfer 10 (1967) 1829–1845.

[4] K. Lucas, Combined body force convection in laminar film condensation of mixed vapours—integral and finite difference treatment, Int. J. Heat Mass Transfer 19 (1976) 1273–1280.

[5] VDI-Wärmeatlas, 9th ed. section Jba Springer-Verlag, Berlin, 2002.

[6] G.F. Hewitt, G.L. Shires, Y.V. Polezhaev (Eds.), International Encyclopaedia of Heat and Mass Transfer, CRC Press, Boca Raton, 1996.

[7] Heat Exchanger Design Handbook, vol. 2, VDI-Verlag, Düsseldorf, and Hemisphere Publishing Corporation, Washington, 1986.

[8] R.B. Bird, W.E. Stewart, E.N. Lightfoot, Transport Phenomena, second ed., John Wiley & Sons, New York, 2002.

[9] K. Stephan, A. Laesecke, The influence of suction on heat and mass transfer in condensation of mixed vapours, Wärme Stoffübertrag. (Thermo Fluid Dyn.) 13 (1980) 115–123.

[10] H.W. Emmons, D.C. Leigh, Tabulation of the Blasius function with blowing and suction, Fluid Motion Sub-Committee, Aeronaut. Res. Com., Report no. FM 1915, 1953.

[11] J.W. Rose, Approximate equations for forced-convection condensation in the presence of a non-condensing gas on a flat plate and horizontal tube, Int. J. Heat Mass Transfer 23 (1980) 539–546.

[12] J.W. Rose, Boundary-layer flow on a flat plate, Int. J. Heat Mass Transfer 22 (1979) 969.

[13] H.D. Baehr, K. Stephan, Heat and Mass Transfer, Springer-Verlag, Berlin, 1998, p. 322.

[14] G. Poots, R.G. Miles, Effects of variable properties on laminar film condensation of saturated steam on a vertical flat plate, Int. J. Heat Mass Transfer 10 (1967) 1677–1692.

[15] V.E. Denny, A.F. Mills, Nonsimilar solutions for laminar film condensation on a vertical surface, Int. J. Heat Mass Transfer 12 (1969) 965–979.

[16] K. Stephan, Heat Transfer in Condensation and Boiling, Springer-Verlag, Berlin, 1992.