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APPROXIMATE EQUATIONS FOR FORCED-CONVECTION CONDENSATION IN THE PRESENCE OF A NON-CONDENSING GAS ON A FLAT PLATE AND HORIZONTAL TUBE

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Abstract – For condensation from a vapour–gas mixture flowing parallel to a plane horizontal condensing surface and normal to a horizontal tube, approximate theoretically-based equations are obtained relating the mass flux of vapour to the condensing surface (condensation rate) to the free-stream and condensate surface conditions. These may be used with suitable equations, giving the heat flux (or condensation rate) in terms of the temperature drop across the condensate film, to calculate the heat flux for given free-stream velocity, composition and temperature and condenser surface temperature. The equations are designed to be correct for the limiting cases of zero and infinite condensation rate. For the flat-plate case the present result agrees very closely with earlier numerical solutions covering a wide range of condensation rates and for various values of Schmidt number. For the horizontal-cylinder case the present result is in good agreement with experimental data for steam–air mixtures covering wide ranges of velocity, composition, condensation rate and pressure.

NOMENCLATURE

D,	diffusion coefficient;
d,	diameter of tube;
g,	specific gravitational force;
g_m,	mean mass-transfer coefficient $m_D/(W_{v\infty} - W_{v0})$, [see also equation (20)];
g_{mx},	local mass-transfer coefficient $m_{xD}/(W_{v\infty} - W_{v0}) = \frac{\rho D}{(W_{v\infty} - W_{v0})}\left(\frac{\partial W_v}{\partial y}\right)_0$ [see also equation (6)];
h_{fg},	specific enthalpy of evaporation;
k_c,	thermal conductivity of condensate;
M_g,	molar mass of non-condensing gas;
M_v,	molar mass of vapour;
m_x,	local (inward) surface vapour mass flux, condensation rate;
m,	mean (inward) surface vapour mass flux, condensation rate;
m_{xD},	diffusive component of local (inward) surface vapour mass flux;
m_D,	diffusive component of mean (inward) surface vapour mass flux;
Nu,	mean Nusselt number;
P,	pressure of vapour–gas mixture;
$P_{g\infty}$,	free-stream partial pressure of air;
$P_{sat}(T)$,	saturation pressure at temperature T;
P_{s0},	partial pressure of steam adjacent to condensate surface;
$P_{s\infty}$,	free-stream partial pressure of steam;
Pr,	Prandtl number;
Q,	mean heat flux;
R_g,	specific ideal-gas constant of air;
R_s,	specific ideal-gas constant of steam;
Re,	mean Reynolds number, $u_\infty \rho d/\mu$;
Re_x,	local Reynolds number, $u_\infty \rho x/\mu$;
r,	$(\rho_c\mu_c/\rho\mu)^{1/2}$;
Sc,	Schmidt number;
Sh,	mean Sherwood number, $g_m d/\rho D$;
Sh_x,	local Sherwood number, $g_{mx}x/\rho D = x\left(\frac{\partial \phi}{\partial y}\right)_0$;
Sh_1,	$g_m d/\rho_\infty D$;
Sh_2,	$g_m d/\rho_0 D$;
T,	thermodynamic temperature;
T_0,	temperature at vapour–condensate interface;
T_w,	temperature at plate or tube surface;
T_∞,	free-stream temperature;
u_∞,	free-stream velocity;
v_0,	y-direction velocity at $y = 0$;
v_s,	mean outward surface radial velocity;
W,	mass fraction of non-condensing gas;
W_0,	mass fraction of non-condensing gas at vapour–condensate interface;
W_∞,	free-stream mass fraction of non-condensing gas;
W_{v0},	mass fraction of vapour at vapour–condensate interface;
$W_{v\infty}$,	free-stream mass fraction of vapour;
x,	distance measured along surface;
y,	distance measured normal to x-direction;
z,	$ShRe^{-1/2}$;
z_x,	$Sh_xRe_x^{-1/2}$;
z_1,	$Sh_1Re^{-1/2}$;
z_2,	$Sh_2Re^{-1/2}$

Greek symbols

β_x, $-\frac{v_0}{u_\infty} Re_x^{1/2}$, [see also equation (7)];
β, $-\frac{v_s}{u_\infty} Re^{1/2}$, [see also equation (22)];
ΔP_s, $P_{s\infty} - P_{s0}$;
ΔT, $T_0 - T_w$;
ε_g, $P_{g\infty}/P$;
μ, viscosity, viscosity of vapour–gas mixture;
μ_c, viscosity of condensate;
π, $\Delta P_s/P$;
ρ, density, density of vapour–gas mixture;
ρ_c, density of condensate;
ρ_0, density of vapour–gas mixture at vapour–condensate interface;
ρ_∞, free-stream density of vapour–gas mixture;
ϕ, $(W - W_0)/(W_\infty - W_0)$;
ω, W_∞/W_0.

INTRODUCTION

With the aid of computers, 'exact' numerical solutions may be obtained for many problems in fluid mechanics and heat transfer. Variations in thermophysical properties with temperature, pressure and composition may be included. In practice however, approximate results, giving relatively simple relations between the surface transfer parameters, are often adequate owing to non-ideal geometry or imprecisely known property values.

The present work provides, for the problem of condensation in the presence of a non-condensing gas, approximate equations for calculating the transfer-rate of vapour, to the condensate surface. The results are based on the uniform-property boundary-layer equations. For the flat plate case, numerical solutions have earlier been obtained for a wide variety of circumstances. The present result is in excellent agreement with the numerical solutions. For the case of the horizontal tube, experimental data are available only for the steam–air case. The present result agrees well with these data.

HORIZONTAL PLATE

The problem considered is illustrated in Fig. 1. The momentum, energy and diffusion equations for the vapour–gas mixture and the momentum and energy equations for the condensate film have to be solved simultaneously subject to boundary conditions of specified uniform u_∞, T_∞, W_∞ and T_w, together with the interface conditions of continuity of velocity, shear stress and temperature, conservation of energy and mass of the condensing constituent and the condition that the interface mass flux of the non-condensing gas is zero.

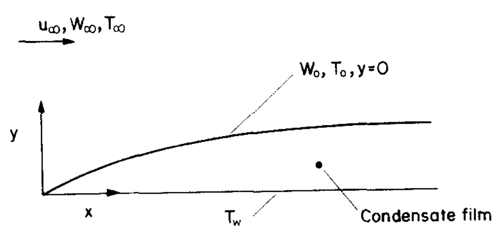

Fig. 1. Condensation on a horizontal plane surface.

Koh [1] and recently Fujii, Uehara, Mihara and Kato [2] have obtained numerical solutions for a variety of circumstances. For the usual case where $\rho_c\mu_c/\rho\mu \gg 1$ Sparrow, Minkowycz and Saddy [3] have indicated that it is permissible to take the interface velocity to be zero when considering the vapour–gas boundary layer. They further simplified the problem by neglecting the contribution to the heat flux in the condensate arising from the temperature gradient in the vapour–gas mixture, so that the energy equation for the vapour–gas mixture is no longer required. As will be seen, the present approximate result for the simplified case considered by Sparrow *et al.* [3] is in excellent agreement also with the solutions of Koh [1] and of Fujii *et al.* [2] confirming the validity of the simplifying approximations.

The analysis of Sparrow *et al.* [3] indicates that the thickening of the condensate film leads to a decrease in condensation rate in the direction of flow such that $v_0 \propto x^{-1/2}$. Furthermore, for thermodynamic equilibrium at the liquid–vapour interface, the interface temperature and composition of the vapour–gas mixture are independent of x. Under these conditions the diffusion problem for the vapour–gas boundary layer becomes identical to the case of heat transfer for flow over an isothermal plate with surface suction and $v_0 \propto x^{-1/2}$, i.e. the governing equations and boundary conditions for the two problems, when appropriately non-dimensionalized, are the same.

An approximate equation (equation (3) of [4]) for the heat-transfer problem, which relates the local surface heat- and mass-transfer parameters, has been given. This equation is correct for zero and infinite suction and agrees very closely with 'exact' numerical solutions for intermediate values of the suction parameter and for various values of Prandtl number. The corresponding equation for the present case is obtained by replacing the local Nusselt number of the earlier problem by a local Sherwood number defined:

$$Sh_x = x\left(\frac{\partial \phi}{\partial y}\right)_0. \quad (1)$$

In addition, the Schmidt number replaces the Prandtl number so that we have for the present case:

$$Sh_x Re_x^{-1/2} = \zeta\{1 + a\beta_x^b Sc^c\}^{-1} + \beta_x Sc \quad (2)$$

where

$$\zeta = Sc^{1/2}(27.8 + 75.9\, Sc^{0.306} + 657\, Sc)^{-1/6} \quad (3)$$

$$\beta_x = -(v_0/u_\infty)Re_x^{1/2} \tag{4}$$
$$a = 0.941$$
$$b = 1.14$$
$$c = 0.93.$$

In the present problem, we have a second equation relating the quantities in equation (2), given by the condition that the interface is impermeable to the non-condensing gas. This gives

$$m_x = -\rho v_0 = -\frac{\rho D}{W_0}\left(\frac{\partial W}{\partial y}\right)_0 \tag{5}$$

so that the mass-transfer coefficient

$$g_{mx} = \frac{m_x(1-W_{v0})}{(W_{v\infty}-W_{v0})} = \frac{m_x}{1-\omega} \tag{6}$$

where

$$\omega = W_\infty/W_0$$

and

$$\beta_x = (m_x/\rho u_\infty)Re_x^{1/2} \tag{7}$$

$$Sh_x Re_x^{-1/2} = \beta_x Sc/(1-\omega). \tag{8}$$

Equations (2) and (8) may thus be solved simultaneously to give, for specified free-stream conditions, the relationship between the vapour mass flux to the condensate surface and the composition at the interface. Eliminating either z_x $(=Sh_x Re_x^{-1/2})$ or β_x we obtain

$$\omega = \{1+\beta_x Sc(1+0.941\,\beta_x^{1.14}Sc^{0.93})/\zeta\}^{-1} \tag{9}$$

or

$$z_x + 0.941\,Sc^{-0.21}(1-\omega)^{1.14}z_x^{2.14} - \zeta/\omega = 0. \tag{10}$$

If ω is required for a given value of β_x this may be obtained directly from equation (9); otherwise equation (9) or (10) must be solved by iteration. To obtain somewhat less accurate results, which may be used as starting values in the iterative process, the following may be used:

$$\beta_x = \frac{\{1+4.57\,Sc^{-0.04}\zeta(1-\omega)/\omega\}^{1/2}-1}{2.28\,Sc^{0.96}} \tag{11}$$

$$z_x = \frac{\{1+4.57\,Sc^{-0.04}\zeta(1-\omega)/\omega\}^{1/2}-1}{2.28\,Sc^{-0.04}(1-\omega)}. \tag{12}$$

The above were obtained by redetermining the constants a and c in the heat-transfer problem [4] with b forced to unity. This gave [for use in equation (2)] $a = 1.142$, $b = 1$, $c = 0.96$.

In Table 1, the results given by equation (9) are compared with the numerical solutions given by Sparrow *et al.* [3]. As may be seen the agreement is very good.

As indicated earlier, only when the simplifying assumptions used by Sparrow *et al.* [3] are adopted, does the condensation problem become strictly identical to the previously considered [4] problem of heat transfer with surface suction. Koh [1] and Fujii *et al.* [2] did not make these assumptions. When the condition of velocity continuity at the vapour–liquid interface is used rather than taking the interface velocity to be zero when treating the vapour–gas boundary layer, the results are found to depend on the parameter $r = (\rho_c\mu_c/\rho\mu)^{1/2}$. Koh [1] obtained numerical solutions for $r = 500$, 100 and 10 for $Sc = 0.5$ and 1.0. Fujii *et al.* [2] used $r = 1000$, 500 and 100 and $Sc = 0.2$, 0.5, 1.0 and 1.5. Examination of these data reveals that only for the smallest value of r, i.e. 10, do the results depend significantly on this parameter suggesting that, for the usual case where the vapour is not close to its critical state, the approximation used by Sparrow *et al.* [3] is valid.

In Fig. 2, where the results of Koh [1] and Fujii *et al.* [2] are compared with equation (10), the data of Koh [1] for $r = 10$ have been omitted. The good agreement between the exact numerical solutions and equation (10) confirms the validity of the present result and of the simplifications made by Sparrow *et al.* [3].

Table 1. Condensation on a flat plate. Comparison of numerical solutions of Sparrow *et al.* [3] for $Sc = 0.55$ with values given by equation (9)

	ω	
β_x	Sparrow	Equation (9)
0.025	0.951	0.951
0.05	0.905	0.905
0.075	0.863	0.863
0.1	0.823	0.824
0.125	0.787	0.788
0.15	0.752	0.753
0.175	0.720	0.721
0.2	0.690	0.691
0.225	0.662	0.663
0.25	0.635	0.637
0.3	0.587	0.588
0.35	0.543	0.545
0.5	0.438	0.439
0.75	0.319	0.318
1.0	0.241	0.240
1.5	0.149	0.149
2.0	0.100	0.100
2.5	0.0717	0.0713
3.0	0.0532	0.0532
5.0	0.0216	0.0217

HORIZONTAL CYLINDER

Before discussing the condensation problem, we first consider the related case of heat transfer during flow normal to a cylinder with surface suction.

For a specified distribution of surface radial velocity, dimensional analysis suggests:

$$Nu = \xi_1(Re, \beta, Pr) \tag{13}$$

where Nu and β are mean values. By analogy with the flat plate case, and for an appropriately specified

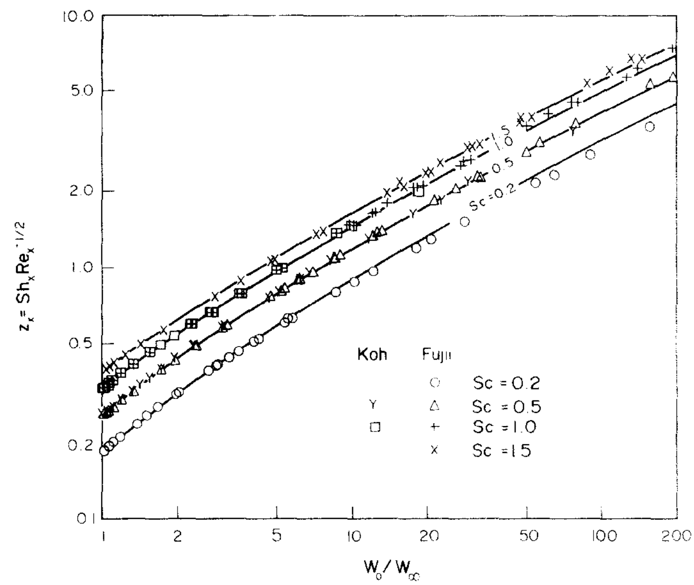

FIG. 2. Condensation on a horizontal plane surface. Comparison of numerical results of Koh [1] and Fujii *et al.* [2] with equation (10) (represented by the lines). *Note*: The accuracy of some of the data given in [2] has been improved [12]. Figure 2 incorporates the revised values.

distribution of surface radial velocity, we might expect equation (13) to reduce to:

$$NuRe^{-1/2} = \xi_2(\beta, Pr). \tag{14}$$

Equation (14) has the correct form for the limiting case of infinite suction and is supported by experimental data over a wide range of Reynolds number for zero suction.

For the zero-suction case, boundary-layer separation precludes straightforward analysis. However, experimental data (see for instance McAdams [5]) may be represented over a wide range of Reynolds number, with sufficient accuracy for the present purposes by:

$$NuRe^{-1/2} = 0.57\,Pr^{1/3} \tag{15}$$

$$(10 < Re < 10^4).$$

As in the case of the flat plate, the boundary layer thins with increasing suction and the separation point moves towards the rear of cylinder [6]. For the infinite suction limit, as the boundary-layer thickness approaches zero and separation is completely suppressed, an energy balance yields:

$$NuRe^{-1/2} = \beta Pr. \tag{16}$$

In general therefore we propose:

$$NuRe^{-1/2} = 0.57\,Pr^{1/3}\xi_3(\beta, Pr) + \beta Pr \tag{17}$$

where

$$\xi_3(0, Pr) = 1$$

$$\xi_3(\infty, Pr) = 0$$

and suggest, on the basis of [4]:

$$\xi_3(\beta, Pr) = (1 + a\beta^b Pr^c)^{-1} \tag{18}$$

where a, b and c are positive constants near to unity.

Turning now to the condensation problem, for the flat plate case where the condensate motion results solely from the vapour flow, it was seen that the velocity of the condensate surface could be neglected when considering the vapour–gas boundary layer. In a case when the condensing surface is not horizontal, gravity also plays a role in the flow of the condensate film. However, for condensation on horizontal tubes, the gravity-induced condensate velocity is low, even at high condensation rates, owing to the small effective height of the condensing surface. Thus, in this case, neglect of the condensate velocity when considering the vapour–gas boundary layer will generally be satisfactory.

As in the case of the flat plate, the diffusion equation for the vapour–gas boundary layer is identical in form to the energy equation in the corresponding heat-transfer problem discussed above so that, if it is assumed (as is true for the normal surface velocity in the flat plate problem) that the distribution of surface radial velocity is the same in both cases, we have, for the vapour–gas boundary layer, by reference to equation (17):

$$ShRe^{-1/2} = 0.57\,Sc^{1/3}\xi_3(\beta, Sc) + \beta Sc \tag{19}$$

The condition that the condensate surface is impermeable to the non-condensing gas gives:

$$g_m = \frac{m(1 - W_{v0})}{W_{v\infty} - W_{v0}} = \frac{m}{1 - \omega} \tag{20}$$

$$Sh = \frac{g_m d}{\rho D} = \frac{md}{\rho D(1 - \omega)} \tag{21}$$

$$\beta = (m/\rho u_\infty)Re^{1/2} \quad (22)$$

$$ShRe^{-1/2} = \beta Sc/(1-\omega). \quad (23)$$

Thus, with an expression for $\xi_3(\beta, Sc)$, equations (19) and (23) may be solved simultaneously to give the relationship between ω and either of the parameters containing the condensation rate, i.e. $ShRe^{-1/2}$ and β.

One way to proceed at this point would be to adopt equation (18) for ξ_3 and, by comparing the result obtained with experimental data, obtain values of the constants a, b and c so as to achieve the best fit. This, ideally, would require accurate measurements for a wide range of vapour velocity, composition and pressure and for different vapour–gas combinations. Satisfactory data for steam–air mixtures only are at present available. In view of this and of the fact that good agreement with the available data was found when tentatively using the convenient values $a = b = c = 1$, further refinement of the values is postponed.

With the above values equations (18) (with Sc replacing Pr) (19) and (23) give the following equivalent results:

$$\omega = \{1 + 1.75\beta Sc^{2/3}(1+\beta Sc)\}^{-1} \quad (24a)$$

or

$$\omega = \{1 + z^{-1} - [1 + 2z^{-1} + z^{-2}(1 - 2.28\,Sc^{1/3})]^{1/2}\}/2 \quad (24b)$$

or

$$z = \{[1 + 2.28\,Sc^{1/3} \times (\omega^{-1} - 1)]^{1/2} - 1\}/2(1-\omega) \quad (24c)$$

or

$$\beta = \{[1 + 2.28\,Sc^{1/3} \times (\omega^{-1} - 1)]^{1/2} - 1\}/2\,Sc \quad (24d)$$

where

$$z = ShRe^{-1/2}. \quad (25)$$

A correlation based on measurements for steam–air mixtures for a wide range of Reynolds number, air concentration and pressure has been given by Berman [7]. This may be written:

$$\frac{mdR_sT_\infty}{\Delta P_s D} = 0.47\,Re^{1/2}\pi_g^{-1/3}\varepsilon_g^{-0.6} \quad (26)$$

where

$$\Delta P_s = P_{s\infty} - P_{s0}$$

$$\pi_g = \Delta P_s/P$$

$$\varepsilon_g = P_{g\infty}/P.$$

Using the Gibbs–Dalton ideal-gas mixture relations, and taking $R_g/R_s = 0.622$, equation (26) gives:

$$z_1 = Sh_1Re^{-1/2} = \frac{0.455}{(W_0 - W_\infty)^{1/3}} \times \frac{W_0}{W_\infty^{0.6}}\,\frac{(1 - 0.378\,W_\infty)^{0.933}}{(1 - 0.378\,W_0)^{2/3}} \quad (27)$$

where the density in z_1 and Sh_1 is ρ_∞. When the temperature dependence of the density is neglected in comparison with density variations arising from composition differences we obtain:

$$z_2 = Sh_2Re^{-1/2} = \frac{0.455}{(W_0 - W_\infty)^{1/3}} \times \frac{W_0}{W_\infty^{0.6}}\,\frac{(1 - 0.378\,W_0)^{1/3}}{(1 - 0.378\,W_\infty)^{0.067}} \quad (28)$$

where the density in z_2 and Sh_2 is ρ_0.

As may be seen from equations (27) and (28), z, as given by the correlation of Berman [7], depends separately on W_0 and W_∞ and will also differ according to the choice of density. Equation (24c) on the other hand, indicates that z depends only on the ratio W_∞/W_0 (for a given Sc) and stems from a uniform-density analysis.

In Table 2 values of z given by equation (24c) with $Sc = 0.55$ (a mean value for steam–air mixtures) are compared with z_1 and z_2 given by equations (27) and (28).

It may be seen that the values of z given by the Berman correlation are not greatly affected by the choice of density and that the dependance of z on W_∞ for a given value of the ratio W_0/W_∞ is relatively weak. Moreover, the values are in quite good general agreement with those given by equation (24c). When considered in the light of the scatter of the data used by Berman, the comparisons given in Table 2 are very satisfactory.

Further measurements have recently been reported by Mills, Tan and Chung [8]. Mills *et al.* measured the heat-transfer rate to a horizontal water-cooled condenser tube in a downward flowing steam–air mixture. Since the temperature and vapour composition at the condensate surface are related by the thermodynamic equilibrium condition, equation (24) (which relates the condensation rate, the remote vapour composition and velocity and the composition, and hence temperature, at the condensate surface) may be used in conjunction with an equation relating the heat-transfer rate (and hence the condensation rate) to the temperature difference across the condensate film, to determine the heat-transfer rate for specified bulk vapour conditions and condenser tube-wall temperature.

An appropriate equation for the condensate film has been obtained by Fujii, Uehara and Kurato [9]:

$$\frac{Qd}{\Delta T k_c} = \left\{0.656\left(\frac{u_\infty \rho_c d}{\mu_c}\right)^2\left(1 + \frac{1}{r}\cdot\frac{\mu_c h_{fg}}{k_c \Delta T}\right)^{4/3} + \frac{0.276\,\rho_c^2 d^3 h_{fg} g}{\mu_c k_c \Delta T}\right\}^{1/4} \quad (29)$$

Table 2. Condensation from steam–air mixture on a horizontal tube. Comparison of the Berman [7] correlation [see equations (27) and (28)] with the present result [equation (24c)]

	W_0/W_∞	1.5	3	5	10	15	25	35	50	70	100
W_∞	z	0.59	0.88	1.19	1.79	2.26	3.01	3.62	4.40	5.27	6.38
0.001	z_1	0.54	0.68	0.91	1.38	1.79	2.50	3.13	3.97	4.99	6.36
	z_2	0.54	0.68	0.90	1.38	1.78	2.48	3.09	3.90	4.86	6.13
0.002	z_1	0.57	0.72	0.95	1.45	1.88	2.64	3.30	4.21	5.32	6.84
	z_2	0.57	0.72	0.95	1.44	1.86	2.59	3.22	4.06	5.04	6.33
0.005	z_1	0.60	0.76	1.01	1.55	2.02	2.86	3.61	4.66	5.99	7.93
	z_2	0.60	0.76	1.00	1.53	1.97	2.73	3.38	4.23	5.20	6.44
0.01	z_1	0.63	0.80	1.06	1.65	2.16	3.09	3.96	5.24	6.99	
	z_2	0.63	0.79	1.05	1.59	2.04	2.81	3.45	4.27	5.16	
0.02	z_1	0.66	0.84	1.12	1.76	2.35	3.47	4.62			
	z_2	0.66	0.83	1.09	1.64	2.10	2.83	3.42			
0.05	z_1	0.71	0.91	1.23	2.02	2.85					
	z_2	0.70	0.87	1.14	1.67	2.08					
0.1	z_1	0.74	0.97	1.36							
	z_2	0.73	0.90	1.15							
0.2	z_1	0.78	1.07								
	z_2	0.75	0.90								
0.5	z_1	0.84									
	z_2	0.75									

Taking the vapour as an ideal-gas mixture, the interface equilibrium condition gives:

$$W_0 = \frac{P - P_{sat}(T_0)}{P - \{1 - (M_v/M_g)\}P_{sat}(T_0)}. \tag{30}$$

For the vapour conditions and tube-wall temperatures used by Mills *et al.* [8] equations (24), (29) and (30) were solved simultaneously (using a suitable iterative technique and taking $Q = mh_{fg}$) to determine the corresponding heat fluxes. These are compared with the observed values in Table 3. The properties of the condensate film were evaluated at $T_w + (T_0 - T_w)/3$ as suggested by Sparrow *et al.* [3]. The specific enthalpy of phase change was evaluated at T_0. For the vapour–gas mixture the density and viscosity were taken as the arithmetic means of their values at, and remote from, the vapour–condensate interface, the densities being evaluated on the basis of ideal-gas mixtures and the viscosities by the method of Wilke [10]. The diffusion coefficient was obtained from:

$$\frac{D}{\text{m}^2/\text{s}} = \frac{7.65 \times 10^{-5}}{P/\text{Pa}} \cdot \left(\frac{T}{\text{K}}\right)^{11/6} \tag{31}$$

as recommended by Fujii, Kato and Mihara [11] and taken at $(T_\infty + T_0)/2$.

* It may be noted that, for the small values of vapour velocity used by Mills *et al.* [8], the results obtained differed only slightly from those found when setting $u_\infty = 0$ in equation (29) so that the result of Fujii *et al.* [9], for the condensate film, reduces to the simple Nusselt equation.

As may be seen from the calculated temperature differences in the vapour phase and across the condensate film*, the measurements of Mills *et al.* [8] extend from those cases where the dominant resistance is that of the condensate film (lowest gas concentrations and highest vapour velocities) to those where the gas-phase resistance is the controlling factor. The fact that the agreement between the observed and calculated values is excellent over the whole range lends strong support to equation (24).

CONCLUDING REMARKS

Perhaps the most important practical result of the present work is that for condensation on a horizontal tube [equations (24)]. The fact that this is designed to be correct for high and low condensation rates, together with the excellent agreement with a wide range of steam–air experimental data, suggests that the equation should be generally satisfactory for other vapour–gas combinations. The assumption that the distribution of surface radial velocity is such as to permit a result of the form:

$$ShRe^{-1/2} = F(\beta, Sc), \tag{32}$$

as is the case for the flat plate when $v_0 \propto x^{-1/2}$, is probably more realistic in practice than the widely-used assumption that the surface temperature of a condenser tube is uniform.

Table 3. Condensation on a horizontal tube. Comparison of observations of Mills *et al.* [8] with values calculated using equation (24) for the vapour–gas layer, equation (29) for the condensate film and the interface condition equation (30)

$100\,W_\infty$	u_∞ (m/s)	T_∞ (K)	T_w (K)	$T_\infty - T_0$ (K)	$T_0 - T_w$ (K)	Heat flux (kW/m^2)	
Mills				Present calculation		Mills	Present calculation
0.11	0.689	331.0	322.8	0.36	7.84	95.6	95.1
0.45	0.811	315.8	308.4	1.35	6.05	67.3	71.2
0.75	0.716	317.7	308.6	2.60	6.50	69.0	74.1
1.01	0.674	319.1	309.3	3.44	6.36	68.5	72.2
1.30	0.601	321.0	309.6	4.85	6.55	68.5	72.9
1.62	0.591	321.2	309.5	5.61	6.09	65.7	68.4
1.94	0.552	322.2	309.1	7.01	6.09	66.0	67.6
2.29	0.479	324.9	309.8	8.96	6.14	67.3	67.5
2.62	0.439	326.6	310.1	10.49	6.01	66.1	66.2
2.94	0.415	327.6	309.8	11.93	5.87	65.4	64.8
3.70	0.494	316.1	299.4	12.34	4.36	52.7	49.1
4.13	0.467	316.8	298.6	13.98	4.22	51.6	47.8
5.43	0.372	316.0	294.8	17.90	3.30	41.5	39.0
6.30	0.334	316.4	293.4	20.07	2.93	39.8	35.5
7.10	0.320	318.5	292.9	22.70	2.90	37.8	35.5
7.88	0.299	320.0	295.2	22.22	2.58	34.7	32.8

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REFERENCES

1. J. C. Koh, Laminar film condensation of condensable gases and mixtures on a flat plate, in *Proc. 4th U.S.A. Nat. Cong. Appl. Mech.* **2**, 1327–1336 (1962).
2. T. Fujii, H. Uehara, K. Mihara and Y. Kato, Forced convection condensation in the presence of non-condensables – a theoretical treatment for two-phase laminar boundary layer, Univ. Kyushu Research Institute of Industrial Science Rep. No. 66, 53–80 (1977).
3. E. M. Sparrow, W. J. Minkowycz and M. Saddy, Forced convection condensation in the presence of non-condensables and interfacial resistance, *Int. J. Heat Mass Transfer* **10**, 1829–1845 (1967).
4. J. W. Rose, Boundary-layer flow with transpiration on an isothermal flat plate, *Int. J. Heat Mass Transfer* **22**, 1243–1244 (1979).
5. W. H. McAdams, *Heat Transmission.* McGraw-Hill, New York (1951).
6. D. G. Hurley and B. Thwaites, An experimental investigation of the boundary layer on a porous circular cylinder, Aeronautical Research Council R. and M. No. 1819 (1955).
7. L. D. Berman, Determining the mass transfer coefficient in calculations on condensation of steam containing air, *Teploenergetika* **16**, 68–71 (1969).
8. A. F. Mills, C. Tan and D. K. Chung, Experimental study of condensation from steam–air mixtures flowing over a horizontal tube: overall condensation rates, in *Proc. 5th Int. Heat Transfer Conference (Tokyo)*, Vol. 5, paper CT 1.5, 20–23 (1974).
9. T. Fujii, H. Uehara and C. Kurato, Laminar filmwise condensation of a flowing vapour on a horizontal cylinder, *Int. J. Heat Mass Transfer* **15**, 235–246 (1972).
10. C. R. Wilke, A viscosity equation for gas mixtures, *J. Chem. Phys.* **18**, 517–519 (1950).
11. T. Fujii, Y. Kato and K. Mihara, Expressions of transport and thermodynamic properties of air, steam and water, Univ. Kyushu Research Institute of Industrial Science Rep. No. 66, 81–95 (1977).
12. T. Fujii, Private communication (1978).

EQUATION APPROCHEES POUR LA CONDENSATION AVEC CONVECTION FORCEE, EN PRESENCE D'UN GAZ INCONDENSABLE, SUR UNE PLAQUE PLANE ET UN TUBE HORIZONTAL

Résumé — Pour la condensation d'un mélange vapeur–gaz en mouvement parallèlement à un plan horizontal et normalement à un tube horizontal, des équations théoriques approchées sont obtenues qui relient le flux massique de vapeur à l'écoulement libre et aux conditions de surface froide. Ceci peur être utilisé avec des équations convenables donnant le flux de chaleur en fonction de la chute de température à travers le film de condensat, pour calculer le flux de chaleur pour une vitesse, une composition et une température au loin données et une température de surface fixée. Les équations sont correctes pour les cas limites de flux de condensat nul et infini. Dans les cas de la plaque plane, le résultat s'accorde bien avec des solutions numériques antérieures qui couvrent un large domaine de flux de condensat et pour différentes valeurs du nombre de Schmidt. Dans le cas du cylindre horizontal, le résultat est en bon accord avec les données expérimentales pour des mélanges vapeur d'eau–air dans un large domaine de vitesse, de composition, de flux de condensat et de pression.

NÄHERUNGSGLEICHUNGEN FÜR DIE KONDENSATION BEI ERZWUNGENER KONVEKTION IN GEGENWART EINES NICHTKONDENSIERENDEN GASES AN EINER EBENEN PLATTE UND AN EINEM WAAGERECHTEN ROHR

Zusammenfassung—Für die Kondensation aus einem Dampf-Gas-Gemisch, welches parallel zu einer ebenen, waagerechten Kondensationsfläche und senkrecht zu einem waagerechten Rohr strömt, wurden theoretisch begründete Näherungsgleichungen aufgestellt, die den Dampfmassenstrom mit der Kondensationsfläche (Kondensationsrate) sowie mit den Bedingungen der freien Strömung und denen der Kondensatoberfläche verknüpfen. Diese Gleichungen können zusammen mit geeigneten Gleichungen angewandt werden, welche den Wärmestrom (oder die Kondensationsrate) in Abhängigkeit vom Temperaturabfall über den Kondensatfilm beschreiben, um den Wärmestrom bei vorgegebener Geschwindigkeit, Zusammensetzung und Temperatur der ungestörten Strömung sowie der Kondensator-Oberflächentemperatur zu berechnen. Die Gleichungen sind so aufgestellt, daß sie die Grenzfälle der verschwindenden und der unendlichen Kondensationsrate richtig beschreiben.

Im Fall der ebenen Platte stimmt das vorliegende Ergebnis sehr gut mit früheren numerischen Lösungen überein, die große Bereiche von Kondensationsraten und verschiedene Werte für die Schmidt-Zahl umfassen.

Im Fall des waagerechten Zylinders stimmt das gegenwärtige Ergebnis gut mit bei Wasserdampf-Luft-Gemischen gewonnenen Meßwerten überein, welche einen großen Bereich der Parameter Geschwindigkeit, Zusammensetzung, Kondensationsrate und Druck umfassen.

ПРИБЛИЖЕННЫЕ УРАВНЕНИЯ ДЛЯ РАСЧЕТА КОНДЕНСАЦИИ ПАРА ПРИ ВЫНУЖДЕННОЙ КОНВЕКЦИИ В ПРИСУТСТВИИ НЕКОНДЕНСИРУЮЩЕГОСЯ ГАЗА НА ПЛОСКОЙ ПЛАСТИНЕ И ГОРИЗОНТАЛЬНОЙ ТРУБЕ

Аннотация — Для расчета конденсации пара из потока паро-газовой смеси, направленного параллельно плоской горизонтальной поверхности и перпендикулярно горизонтальной трубе, получены приближенные уравнения, связывающие плотность потока пара на поверхности конденсации (скорость конденсации) с условиями в свободном потоке и на поверхности конденсата. Эти уравнения могут использоваться совместно с соответствующими уравнениями, связывающими тепловой поток (или скорость конденсации) с перепадом температур поперек пленки конденсата, для расчета плотности теплового потока при заданных скорости, составе и температуре смеси в свободном потоке, а также температуре поверхности конденсатора. Уравнения справедливы для предельных случаев нулевой и бесконечной скорости конденсации. Для случая плоской пластины результаты хорошо согласуются с полученными ранее численными решениями в широком диапазоне скоростей конденсации и при различных значениях числа Шмидта. Для случая горизонтального цилиндра результаты находятся в хорошем соответствии с экспериментальными данными для паровоздушных смесей в широком диапазоне изменения скорости течения и состава смеси, а также скорости конденсации и давления.