Int. J. Heat Mass Transfer. Vol. 24, pp. 165–170
Pergamon Press Ltd. 1981. Printed in Great Britain

0017-9310/81/0101-0165 $02.00/0

IMPACT OF WALL TEMPERATURE VARIATION ON THE FILM CONDENSATION OF BINARY GAS–VAPOUR MIXTURES

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(*Received* 27 *December* 1979 *and in revised form* 17 *April* 1980)

Abstract—Laminar film condensation of binary vapour mixtures in forced flow along a horizontal or down a vertical flat plate is presented. The complete system of partial differential equations is numerically integrated using a finite-difference method of Hermitian type. Firstly, heat transfer results are presented. Additionally, interface characteristics such as tangential and normal velocity components are reported, which sheds some light on their axial distribution as well as on the liquid-side Reynolds number. Secondly, the effect of a linearly varied wall temperature, compared to the isothermal case, is investigated in terms of heat transfer to the wall and interface velocities.

NOMENCLATURE

c_p,	heat capacity;
D_{12},	binary diffusion coefficient;
g,	gravitational acceleration;
h_f,	latent heat of vaporization;
k,	thermal conductivity;
$\dot{m}$,	mass flux;
p,	total pressure;
Pr,	Prandtl number, $Pr = \mu c_p/k$;
$\dot{q}$,	heat flux;
R,	universal gas constant;
Re_x,	Reynolds number, $Re = \rho u_\infty x/\mu_\infty$ $Re_\delta = u_\delta \cdot \delta\rho_{L\delta}/\mu_{L\delta}$;
Sc,	Schmidt number, $Sc = \rho D_{12}/\mu$;
T,	temperature;
u, v,	axial, normal velocity component;
v^*,	interface mass flux, $v^* = (\rho v_\delta/\rho_\Delta u_\Delta)\, Re_x^{1/2}$ at the vapour-side.
x, y,	axial, normal coordinate;
X,	molar concentration;
Y,	mass concentration.

Greek symbols

α,	weighting factor, reference temperature;
δ,	condensate layer thickness;
Δ,	gas layer thickness;
μ,	dynamic viscosity;
ρ,	density;
τ,	shear stress.

Subscripts

Δ,	at the edge of the vapour boundary-layer;
δ,	at the gas–liquid interface;
L,	in the liquid phase;
w,	at the wall;
x,	at position x;
1,	water steam;
2,	methanol, air;
Nu,	Nusselt-type analysis, $\dot{q}_{Nu} = [k_L^3(T_\Delta - T_w)^3\rho^2 g h_f/4x\mu]^{1/4}$;
ref,	reference quantity, $\dot{q}_{\text{ref}} = k_{L\Delta}(T_\Delta - T_w)/\delta$;
E, A,	E = entrance, A = exit.

Superscripts

κ,	type of mixture, $\kappa = 0$, steam–air; $\kappa = 1$, steam–methanol;
$*$,	with respect to v^*.

INTRODUCTION

THE OBJECTIVE of this investigation dealing with laminar film condensation of binary gas mixtures undergoing forced flow along vertical and horizontal flat walls are twofold. Firstly, to provide additional information on the gas–liquid interphase in terms of tangential and normal velocity, u_δ and v_δ, respectively, apart from heat transfer characteristics. Secondly, to shed some light on the impact of a non-isothermal wall on the overall heat transfer process.

For the sake of brevity a review of prior investigations on this physical problem is omitted, and the reader is referred to publications by Mynkowicz and Sparrow [1], Denny and Mills [2], and Denny and South [3]. The physical model is such that a steam–gas or steam–steam mixture condenses onto a horizontally or vertically positioned flat surface. (The case of binary condensation for flow around a horizontally positioned circular cylinder in crosswise forced laminar flow is presented elsewhere [4].) The complete system of partial differential equations describing the gas and the liquid phase in terms of momentum, energy, and species concentration conservation is solved numerically using a finite difference method [5, 6]. Variable thermophysical properties are accounted for despite the fact that the results presented encourage to employ the temperature reference model [4] as advocated by Sparrow *et al.* [7], Mynkowicz and Sparrow [1], and Denny and Mills [2]. Restrictions arise from omitting surface tensions at the interphase. The interphase is considered to be ripple-free since for all cases calculated the liquid phase

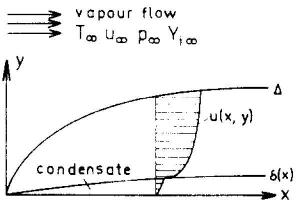

FIG. 1. Schematic representation of the model and the coordinates.

Reynolds number $Re_\delta = u_\delta \delta \rho_L/\mu_L$ is well below the critical Reynolds number for falling films. Figure 1 illustrates the model and the coordinates.

ANALYSIS

Using the boundary layer approach the conservation of mass, momentum, energy, and species concentration for the liquid phase of a steam–steam system gives, respectively:

$$\frac{\partial}{\partial x}(\rho_L u) + \frac{\partial}{\partial y}(\rho_L v) = 0 \tag{1}$$

$$g(\rho_L - \rho_\Delta) + \frac{\partial}{\partial y}\left(\mu_L \frac{\partial u}{\partial y}\right) = 0 \tag{2}$$

$$\frac{\partial}{\partial y}\left(k_L \frac{\partial T}{\partial y}\right) = 0 \tag{3}$$

$$\frac{\partial}{\partial y}\left(\rho_L D_{12,L} \frac{\partial \bar{Y}_1}{\partial y}\right) = 0. \tag{4}$$

Neglecting the inertia and convective terms in equations (2)–(4) has been done for two reasons. Prior investigations, for instance [2], substantiated these assumptions. Additionally, for reasons of comparison this seems to be necessary [1,8]. The influence of various liquid phase momentum models on the overall condensation process, i.e. linear velocity distribution, complete boundary layer equation, or u_δ = const, was presented elsewhere [9]. The results of this investigation [9] agree with the findings by Denny and Mills [2] and Denny and Jusionis [8].

For the gas phase (this jargon is used as an abbreviation for the two systems considered, i.e. condensable, and condensable–noncondensable mixtures) the governing equations are as follows:

$$\frac{\partial}{\partial x}(\rho u) + \frac{\partial}{\partial y}(\rho v) = 0 \tag{5}$$

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = g(\rho - \rho_\Delta) + \frac{\partial}{\partial y}\left(\mu \frac{\partial u}{\partial y}\right) \tag{6}$$

$$c_p \rho\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + (c_{p1} - c_{p2})\rho D_{12}\frac{\partial Y_1}{\partial y}\frac{\partial T}{\partial y} \tag{7}$$

$$\rho\left(u\frac{\partial Y_1}{\partial x} + v\frac{\partial Y_1}{\partial y}\right) = \frac{\partial}{\partial y}\left(\rho D_{12}\frac{\partial Y_1}{\partial y}\right). \tag{8}$$

For the sake of simplicity, the index v for vapour phase has been dropped. Equations (5)–(8) represent the equation of continuity, momentum, energy, and species concentration of component 1. Closure of the component system is given by equation (5) instead of using an additional species equation. Secondary effects such as viscous dissipation or diffusional conduction are omitted [7].

The boundary conditions for both subsystems (liquid phase $0 \leqslant y \leqslant \delta$, gas phase $\delta \leqslant y \leqslant \Delta$) are

$$y = 0 \quad u = 0, \quad v = 0;$$

$$T = T_w(x); \quad \bar{Y}_1 = \bar{Y}_{1w}(T_w, p_\Delta) \tag{9}$$

$$y \to \Delta \quad u = u_\Delta;$$

$$T = T_\Delta; \quad Y_1 = Y_{1\Delta}(T_\Delta, p_\Delta). \tag{10}$$

In order to ensure continuity of velocity and temperature as well as continuity of momentum, energy, and mass transfer at the liquid–gas interphase $y = \delta$ additional equations are required:

$$u = u_\delta; \quad T = T_\delta(x);$$

$$\bar{Y}_1 = \bar{Y}_{1\delta}(p_\Delta, T_\delta); \quad Y_1 = Y_{1\delta}(p_\Delta, T_\delta) \tag{11}$$

$$\mu_L \frac{\partial u}{\partial y} = \mu \frac{\partial u}{\partial y} \tag{12}$$

$$y = \delta$$

$$k_L \frac{\partial T}{\partial y} = k\frac{\partial T}{\partial y} + \dot{m}_\delta h_f \tag{13}$$

$$-\int_0^\delta \frac{\partial}{\partial x}(\rho_L u)\,\mathrm{d}y = \dot{m}_\delta$$

$$= \frac{1}{\bar{Y}^\kappa_{1\delta} - Y_{1\delta}}\rho D_{12}\frac{\partial Y_1}{\partial y} \tag{14}$$

The gas phase exhibits ideal gas behaviour, $p = \rho RT$. The interphase is at saturation state for the condensing vapour and shows thermodynamic equilibrium. Equation (14) represents the restriction that the interphase is impenetrable for noncondensables (superscript $\kappa = 0$). The superscript κ takes the value zero for the mixture water–air, and the value one for water–methanol. For water–air mixtures, equation (4) cancels since the liquid phase consists of condensed pure water steam, only ($\bar{Y}_{1w} = 1$). Consequently, all thermophysical properties in equations (2)–(4) are pure component properties. Otherwise, the thermodynamic and transport properties are locally variable mixture properties which depend on the local temperature and concentration for a given pressure level. The definition of the mass concentrations Y_i, $i = 1, 2$, reads $Y_i \equiv \rho_i/\rho$, $\rho = \rho_1 + \rho_2$.

Equations (1)–(14) describe the physical model whereby equations (1)–(10) serve as a skeleton in order to extract the unknowns u_δ, T_δ and δ from equations

(12)–(14). Thus, the whole problem may be viewed as a mutually depending roots problem in terms of the interphase conditions since the three properties u_δ, T_δ and δ are interdependent and, hence, cannot be solved separately.

THERMOPHYSICAL PROPERTIES

The data of all pure thermophysical properties are compiled from [10] and [11] and fitted by least-squares analysis to polynominals of the order (5), solely for computational reasons. The mixture rules for the condensate ($\kappa = 1$) are simply additive rules weighted by the local mass concentration. For the gas phase mixture, rules were taken from Reid and Sherwood [12] following their recommendations (for details the reader is referred to [4]). The saturation data were cast into polynominals of varying order. Particular difficulties have to be attributed to the curve-fitting of the saturation data of methanol, particularly for concentrations of less than $Y_{2\Delta} = 0.1$ ($p_\Delta = 1.01325$ bar) since the present results, extensively checked against the findings of an investigation by Denny and Jusionis [8] which employed a condensation model very similar to the one used at present by the author, deviated by up to 10%.

NUMERICAL PROCEDURE

Equations (1)–(14) turned dimensionless by introducing a normalized temperature and concentration and, additionally, a stream function, are transformed with respect to their coordinates x and y in order to overcome cumbersome numerical start-off problems. This set of equations is numerically integrated by means of the finite difference method of Hermitian type [5, 6] which generates solutions in terms of the function and its first derivative, simultaneously, a feature which is of importance to the interphase conditions (12)–(14). A variable grid scheme places 2/3 of 31 nodal points in y-direction (even 11 points seem to be sufficient) into the ultimate vicinity of the interphase.

RESULTS AND DISCUSSION

A variation of the free stream velocity u_Δ of a water steam–methanol mixture exhibits a marked impact on the heat transfer to an isothermal vertical wall and on the mass transfer across the interphase (Fig. 2). In qualitative terms, a ten-fold increase in velocity u_Δ results in a two-fold increase in heat flux to the wall (full lines). This is due to an enhanced mass transfer to the interphase and due to vapour drag [8]. This is in general agreement with the $x^{-1/2}$- and $u_\Delta^{1/2}$-dependence of $\dot{q}/\dot{q}_{Nu}$ as pointed out by Denny *et al.* [14]. The non-similar character of the results gets more pronounced the larger the free stream velocity. Deviations of the present results (full line) from those by Denny and Jusionis [8] (circle) may be attributed to a variety of influencing factors: (i) saturation data approximation, (ii) heat flux reference $\dot{q}_{Nu}$ instead of $\dot{q}_{Nu,H_2O}$ and (iii) numerical solution procedure and iteration scheme (for comparison, see reference [6]). The heat flux reference $\dot{q}_{Nu}$ is the classical Nusselt heat flux

$$\dot{q}_{Nu} = [k_L^3(T_\Delta - T_w)^3 g\rho_L^2 h_f/4\,\mu_L x]^{1/4}, \qquad (15)$$

where all thermophysical mixture properties except h_f are evaluated at a reference temperature $T_{ref} = T_w + \alpha(T_\Delta - T_w)$ and at a concentration $Y_{i\Delta} = Y_{i\Delta}(T_\Delta)$. h_f is evaluated at T_Δ and $Y_{i\Delta}$. This is done purely for the sake of a trend-wise comparison with the other authors' results, i.e. [13]. Departure of $\dot{q}/\dot{q}_{Nu}$ from $\dot{q}/\dot{q}_{Nu,H_2O}$ [8] is less than 4% for the severest case ($u_\Delta = 30$ m/s).

The variation of the reduced interface velocity u_δ/u_Δ with increasing x/L is little for case 1 (30 m/s) whereas in case 2 (3 m/s) the reduced velocity increases noticeably. These findings are somewhat misleading with respect to the actual quantities since the interface velocity ratio $u_{\delta 1}/u_{\delta 2}$ (subscripts 1 and 2 refer to case 1 and 2) has a value of 1.96 for $u_{\Delta 1}/u_{\Delta 2} = 10$ at $x/L = 1$.

In quantitative terms, these results are reflected by the interface mass transfer v^*. The ratio $v_{\delta 1}/v_{\delta 2}$ amounts to a value of 1.98 for a ten-fold increase in free stream velocity at $x/L = 1$ assuming the thermophysical properties ρ and ρ_Δ to be equal. This enhanced mass transfer in terms of steam for an increase in u_Δ results in an augmentation of heat transfer to the wall [7, 8] as already pointed out.

The finite value of u_δ/u_Δ for $x/L = 0$ appears to be puzzling and physically wrong, yet, it merely is the outcome of the assumption of the continuity of shear stress at the interface, even for $x/L = 0$. Simplification of the film condensation model assuming the interphase velocity u_δ to be constant or negligibly small seems to predict inaccurate results for certain thermal and flow conditions.

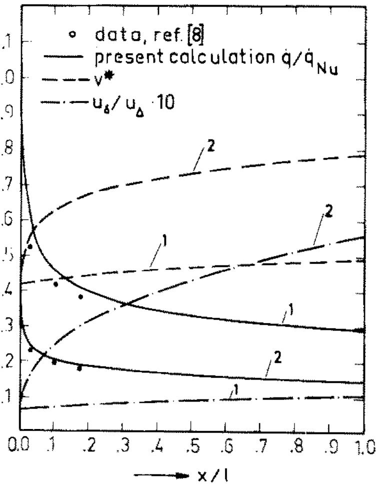

FIG. 2. Effects of u_Δ on methanol–water condensation; reduced wall heat transfer $\dot{q}/\dot{q}_{Nu}$; normalized interphase mass transfer v^*, and interphase velocity u_δ/u_Δ; specifications: vertical flat plate; variable thermophysical properties; $p_\Delta = 1$ atm; $\Delta T = 20$ F; $Y_{2\Delta} = 0.409$; $L = 1$ m; (1) 30 m/s, (2) 3 m/s.

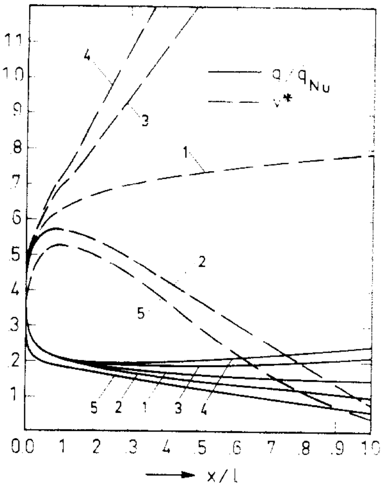

FIG. 3. Effects of variable wall temperature T_w on methanol–water condensation; reduced wall heat transfer $\dot{q}/\dot{q}_{Nu}$; normalized interphase mass transfer v^*; specifications as Fig. 2, case 2. (1) $T_{wA} = T_{wE}$, (2) $T_{wA} - T_{wE} = 10$ K, (3) $T_{wA} - T_{wE} = -10$ K, (4) $T_{wA} - T_{wE} = -15$ K, (5) $Y_{2\Delta} = 0.75$; $T_{wA} - T_{wE} = 10$ K.

The following figures deal with the influence of a linearly varied wall temperature on the heat and mass transfer to the wall and across the interface, respectively: $T_w(x/L) = T_{wE} + (T_{wA} - T_{wE})x/L$.

Figures 3 and 4 concern a fixed steam–methanol mixture for 'standard' conditions, i.e. assigned values of $Y_{2\Delta} = 0.409$ and $\Delta T = T_\Delta - T_{wE} = 20$ F at $x/L = 0$, curves 1–4 (see [8]). The solid lines represent the reduced wall heat flux $\dot{q}/\dot{q}_{Nu}$, the dashed lines the interface mass transfer, v^* (Fig. 3). Lowering the wall temperature, curves 3 and 4, increases the overall thermal driving force $\Delta T = T_\Delta - T_w(x)$, and hence the rate of change of heat compared to the isothermal case, curve 1. This applies likewise, yet in the opposite sense, for an increase in wall temperature, curve 2. The variation of the heat transfer along x/L, however, is non-linear. The variation of curve 2, compared to curve 1 for instance, depends on two opposing influences. The x^{-n}-dependence of $\dot{q}/\dot{q}_{Nu}$, $n > 0$, reduces the heat flux whereas the increasing driving force $T_\Delta - T_w(x)$ tends to push up the heat flux level.

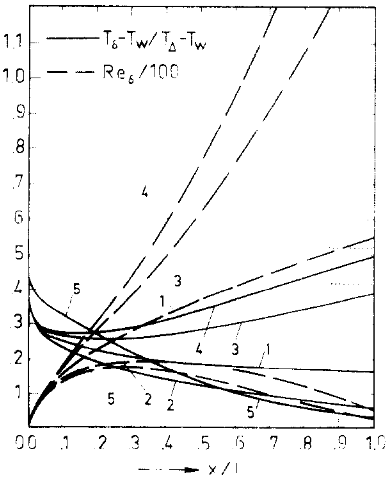

FIG. 4. Impact of variable wall temperature T_w on reduced interphase temperature $T_\delta - T_w/T_\Delta - T_w$ and on interface Reynolds number Re_δ for methanol–water condensation. Specifications see Fig. 3.

These results are analogously displayed for the interface temperature T_δ [Fig. 4 (solid line)], yet more pronounced. The non-linearity is particularly evident for the $T_\delta - T_w/T_\Delta - T_w$ presentation (Fig. 4), assuming curve 1 to be a 'line of symmetry' with respect to the 10° wall temperature increase and decrease of curve 3 and 2, respectively. At station $x/L = 1$, curve 3 reaches a T_δ-level compared to curve 1 which is about twice the difference of curve 2. The difference between the corresponding isothermal (dotted) and non-isothermal values at $x/L = 1$ may be attributed to the upstream accumulation of the second mixture component at the interface which does not allow to reach the prescribed isothermal value.

As for v^* (Fig. 3) and Re_δ (Fig. 4), curves 2–4 tend to follow the curve for the isothermal wall in the vicinity of the leading edge, as expected. The more pronounced the influence of the overall driving force $T_\Delta - T_w(x)$ gets, the more depart the individual curves 2–4 from the isothermal curve 1. The reason for this behaviour is straightforward [7, 8]: The augmentation in rate of heat transfer, curve 3, compared to curve 1, is due to a diminution in wall temperature T_w and this results in an increase in thermal driving force $T_\Delta - T_w$. This increase in $T_\Delta - T_w$, in turn, acts as a suction potential which gives rise to an enhancement in convective interface mass transfer v^*. This mechanism may be compared to a lowering in concentration of a noncondensable (the more volatile component, methanol, may be regarded as a 'noncondensable' gas component) and, hence, in a reduction of the noncondensable build-up at the interface [13].

Figure 5 displays the influence of a linearly varied wall temperature on the heat flux for steam–air under forced flow along a horizontal flat plate. The free stream conditions are: $u_\Delta = 1$ m/s, $Y_{2\Delta} = 0.1$, $T_\Delta = 373.15$ K. The wall temperature variation ranges between 368.15 K and 333.15 K. Additionally, the interface Reynolds number Re_δ is presented. The heat flux $\dot{q}$ to the wall is reduced by $\dot{q}_{ref}$ where $\dot{q}_{ref}$ is a reference flux defined as $\dot{q}_{ref} = k_{L\Delta}(T_\Delta - T_w)/\delta$ with $k_{L\Delta}$ being the heat conductivity of the condensate at temperature T_Δ. Basically, this heat flux ratio $\dot{q}/\dot{q}_{ref}$ may be cut down to $T_\delta - T_w/T_\Delta - T_w$ assuming a linear temperature distribution for the liquid (which appears to be true for not too low free stream velocities) and the ratio of the conductivities to be unity which holds well for the thermal conditions considered.

The linear plotting underlines the overall similarity character of the heat flux results except for the ultimate vicinity of the tip of the plate where solutions are

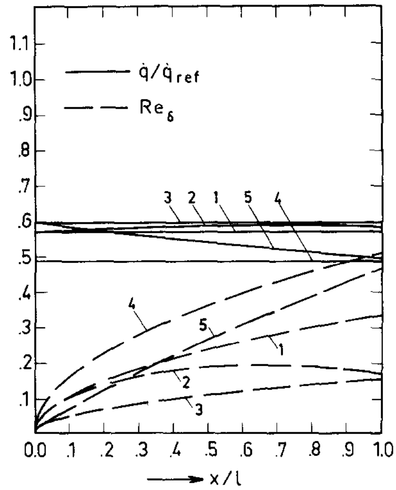

FIG. 5. Impact of variable wall temperature T_w on water steam–air condensation onto a horizontal flat plate; reduced wall heat transfer $\dot{q}/\dot{q}_{\mathrm{ref}}$; Reynolds number Re_δ; specifications: variable thermophysical properties; $u_\Delta = 1$ m/s, $T_\Delta = 373.15$ K, $Y_{2\Delta} = 0.1$, $T_{wE} = 358.15$ K; (1) $T_{wA} = T_{wE}$, (2) $T_{wA} - T_{wE} = 10$ K, (3) $T_{wE} = 368.15$ K; $T_{wA} = T_{wE}$, (4) $T_{wE} = 333.15$ K; $T_{wA} = T_{wE}$, (5) $T_{wE} = 368.15$ K; $T_{wA} - T_{wE} = -35$ K.

markedly non-similar [4, 6, 8]. The general non-similarity of the solutions becomes more obvious for larger temperature differences $T_\Delta - T_w$. In contrast to the situation for the vertically positioned flat plate the linear change in wall temperature along a horizontal flat plate exhibits a linear change in heat flux, curves 2 and 5. The behaviour of Re_δ along the x-axis is somewhat different from that for the vertical plate (Fig. 4). Figure 5 demonstrates a retarded increase in Re_δ whereas in Fig. 3 Re_δ tends to augment strongly, cases 3 and 4.

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INFLUENCE DE LA VARIATION DE LA TEMPERATURE DE PAROI SUR LA CONDENSATION EN FILM DE MELANGES BINAIRES GAZ–VAPEUR

Résumé—On étudie la condensation en film laminaire de mélanges binaires de vapeur en écoulement forcé le long d'une plaque horizontale ou verticale. Le système complet d'équations aux dérivées partielles est intégré numériquement en utilisant une méthode aux différences finies de type hermitien.

On présente tout d'abord des résultats de transfert thermique. Accessoirement, des caractéristiques de l'interface comme les composantes de vitesse tangentielle et normale sont considérées en faisant apparaitre leur distribution axiale ainsi que le nombre de Reynolds du côté liquide. Enfin l'effet d'une température de paroi variant linéairement, en comparaison du cas isotherme, est étudié en fonction du transfert thermique à la paroi et des vitesses à l'interface.

EINFLUß EINER VERÄNDERLICHEN WANDTEMPERATUR AUF DIE FILMKONDENSATION BINÄRER GAS-DAMPFGEMISCHE

Zusammenfassung—Die Filmkondensation binärer Dampfgemische in erzwungener laminarer Strömung entlang einer horizontalen oder vertikalen ebenen Platte wird dargelegt. Das vollständige System partieller Differentialgleichungen, welche in Form der Grenzschichtgleichungen das stark gekoppelte nichtähnliche Zweiphasen-Problem beschreiben, wird mit Hilfe einer Finiten-Differenzen-Methode vom Hermiteschen Typ numerisch integriert. Zunächst werden Wärmeübertragungsergebnisse dargelegt, die mit denen früherer vergleichbarer Untersuchungen recht gut übereinstimmen. Zusätzlich werden Phasengrenzflächengrößen wie Tangential- und Normalgeschwindigkeit dargestellt, was sowohl deren Verhalten in axialer Richtung als auch das der Kondensat-Reynoldszahl näher beleuchtet. Darüberhinaus wird der Einfluß einer sich linear verändernden Wandtemperatur—verglichen mit dem isothermen Fall—im Hinblick auf die Wärmeübertragung zur Wand und bezüglich der Phasengrenzgeschwindigkeiten untersucht.

ВЛИЯНИЕ ИЗМЕНЕНИЯ ТЕМПЕРАТУРЫ СТЕНКИ НА ПЛЕНОЧНУЮ КОНДЕНСАЦИЮ БИНАРНЫХ ПАРО-ГАЗОВЫХ СМЕСЕЙ

Аннотация—Исследуется ламинарная пленочная конденсация бинарных паровых смесей при их вынужденном течении вдоль горизонтальной плоской пластины или при стекании вниз по вертикальной плоской пластине. Полная система дифференциальных уравнений в частных производных интегрируется численно с помощью эрмитовского варианта метода конечных разностей. Представлены результаты по теплообмену. Кроме того, определены такие характеристики границы раздела, как тангенциальная и нормальная компоненты скорости, что позволило определить аксиальное поле этих скоростей, а также число Рейнольдса для жидкости. С помощью исследования переноса тепла к стенке и скорости на границе раздела фаз выявлено влияние линейно изменяющейся температуры стенки на процесс конденсации по сравнению с изотермическим случаем.