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Pressure gradient and variable wall temperature effects during filmwise condensation from downward flowing vapors onto a horizontal tube

C.-H. Hsu, S.-A. Yang*

Department of Mold and Die Engineering, National Kaohsiung Institute of Technology, Kaohsiung 807, Taiwan

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Abstract

A simple model is developed to study the process of combined forced and natural convection film condensation from downward flowing vapors onto a horizontal tube with variable wall temperature, including effects of the pressure gradient and vapor shear stress. In the present work, the result of mean heat transfer shows that, for forced-convection film condensation, as the wall temperature variation amplitude, A, increases, the value of $\overline{Nu}\,Re^{-1/2}$, with inclusion of the pressure gradient effect, goes down appreciably. However, the value of $\overline{Nu}\,Re^{-1/2}$ when ignoring the pressure gradient effect will increase with A at a smaller pace. As for natural-convection film condensation, the mean heat transfer coefficients remain almost uniform for varying A, which are in good agreement with the previous work.

Key words: Film condensation; Pressure gradient; Variable wall temperature

Nomenclature

A the wall temperature variation amplitude
C_P specific heat of condensate at constant pressure
D diameter of circular tube
F dimensionless parameter, $(Ra/Ja)/Re^2$
g acceleration due to gravity
h condensing heat transfer coefficient at angle ϕ
$\bar{h}$ mean value of condensing heat transfer coefficient
h_{fg} latent heat of condensate
Ja Jakob number, $C_P\Delta T/h'_{fg}$
k thermal conductivity of condensate
m'' condensate mass flux (per unit area)
Nu local Nusselt number hD/k
$\overline{Nu}$ mean Nusselt number $\bar{h}D/k$
p static pressure of condensate
P the dimensionless pressure gradient parameter $(\rho_v/\rho)Pr/Ja$
Pr Prandtl number
r radius of tube
Ra Rayleigh number, $\rho(\rho-\rho_v)g\,Pr\,D^3/\mu^2$
Re two-phase mean Reynolds number $\rho DU_\infty/\mu$
T_{sat} saturation temperature of vapor
T_w wall temperature
u_e the tangential vapor velocity at the edge of the boundary layer
U_∞ the vapor velocity of the main free stream
u velocity component in x-direction
x coordinate measuring distance along circumference from top of tube
y coordinate normal to the elliptical surface.

Greek symbols

δ local thickness of condensate film
δ^* dimensionless thickness of condensate film
μ absolute viscosity of condensate
ρ density of condensate
ρ_v density of vapor
τ_δ interface vapor shear
ϕ the angle between the tangent to tube surface and the normal to direction of gravity.

Subscripts

c critical condition
sat saturation

* Corresponding author.

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v vapor
w tube wall.

Superscripts
* dimensionless
– averaged
(Note: when not subscripted, a property is taken to be that of the liquid phase.)

1. Introduction

Filmwise condensation heat transfer is widely applied in power plant systems, air-conditioning equipment and chemical industrial process equipment. Since the pioneering investigator Nusselt [1], the problem of vapor condensation on horizontal cylinders has received considerable attention; a number of workers [2, 3] modified the simple Nusselt theory. One of the important factors related to the problem and greatly influencing the mechanism of heat transfer is the velocity of the oncoming vapor. When the vapor surrounding a horizontal tube is moving at a high velocity, the problem becomes a type of forced convection film condensation and the analysis must consider two complicated factors: (1) the interfacial vapor shear forces, and (2) the effect of condensate or vapor separation must be accurately treated.

Shekriladze and Gomelauri [4] realized that the primary contribution to the surface shear is due to the change in momentum across the interface. Hence, they equated shear stress at interface to change in momentum flux of the condensing vapor so that they might eliminate the momentum equation in the vapor phase. They first proposed that the mean heat transfer coefficient decreases by 35% if the separation point occurs at 82° and agrees with the experimental data. Denny and Mills [5] applied the asymptotic shear model of Shekriladze and Gomelauri to condensation on a circular cylinder, ignoring the pressure gradient, and obtained results very close to the boundary-layer solutions (within 1–2%). Fujii et al. [6] studied the equation of motion in the vapor boundary layer. However, they approximated the velocity profile in that layer by a quadratic formula, and still ignored in the condensate film inertia and pressure gradient, energy convection and liquid subcooling. By employing Shekriladze and Gomelauri's model, Rose [7] took further account of the pressure gradient effect upon the forced-convection film condensation on a horizontal circular tube with vertical vapor downflow using potential flow theory. Rose proposed a good empirical expression for the mean Nusselt number and found that the inclusion of pressure gradient led to a small decrease in the mean heat-transfer coefficient. Gaddis [8] treated the full two-phase boundary-layer equations through series expansion to find that the separation angle of condensate film is around 0.62π rad agreed with Honda and Fujii's [9] value and also confirmed the relation, $\overline{Nu} \propto \sqrt{Re}$. As for the experimental works and effects of the vapor boundary-layer separation, they may be seen in a review paper by Rose [10]. It indicated that measurements for steam at high velocity give lower heat-transfer coefficients, while those data for R-113 give heat-transfer coefficients higher than the theoretical values, compared with Fujii et al. [11].

All the above works relate to natural convection and/or a forced flowing condensation of an isothermal circular tube. Although, for laminar film condensation with uniform properties and negligible vapor velocity, the assumptions of the simple Nusselt theory have been found in later and more complete studies to be generally valid and the tube wall temperature has been observed to vary around the tube by amounts comparable with the mean temperature difference across the condensate film. The theory seems to be somewhat in accordance with experiment. Fujii et al. [6] showed that the wall temperature may often vary significantly over the circumferential length of the tube, even if the coolant temperature inside the tube is constant. Michael et al. [12] also found that the difference between vapor saturated temperature and the local wall non-uniform temperature usually varies with '$1-A\cos\phi$' profile for forced convection film condensation problem, while at high vapor velocities the measured wall temperature profile is more uniform than predicted. For both natural convection and forced convection film condensation on a circular tube with a variable wall temperature (a cosine distribution), Memory and Rose [13] found that the local condensate film thickness and heat flux depend markedly on the amplitude of the surface temperature variation. However, the mean gravity-dominated condensation heat transfer coefficient is virtually unaffected by a surface temperature variation. As for the forced-convection film condensation, ignoring the pressure gradient case investigated by Memory et al. [14], the mean condensation heat transfer increases as the wall temperature variation amplitude goes up. However, their higher mean condensing heat-transfer coefficients seem to be in contradiction with Honda and Fujii's [9] lower coefficients through a conjugate approach. They showed that reasons for this apparent anomaly are advanced. In fact, the present study, with further inclusion of pressure gradient effect, can modify and try to explain the significant discrepancy between them.

Our major aim is to present a generalization of the model of Memory et al. [14] by further inclusion of the pressure gradient effect. The extended model will be applicable to filmwise condensation from flowing pure vapors onto horizontal tubes, including taking account of the pressure gradient and vapor shear effects in a general fashion, and being amendable to any physically relevant initial condition.

2. Analysis

Consider a horizontal circular tube immersed in a downward flowing pure vapor which is at its saturation temperature T_{sat} and moves at uniform velocity U_∞. The wall temperature T_w may be non-uniform and below the saturation temperature. Thus, condensation occurs on the wall and a continuous film of liquid condensate runs downward over the tube under the combined action of gravity, pressure gradient forces and interfacial vapor shear.

The physical model under consideration is shown in Fig. 1. The conservation of mass, momentum and energy for the steady laminar layer flow are described by the following equations:

$$m'' = \rho \frac{\mathrm{d}}{\mathrm{d}x}\int_0^\delta u\,\mathrm{d}y \tag{1}$$

$$\mu\frac{\partial^2 u}{\partial y^2} + (\rho-\rho_v)g\sin\phi - \frac{\mathrm{d}p}{r\,\mathrm{d}\phi} = 0. \tag{2}$$

$$h_{fg}m'' = k\frac{\partial T}{\partial y} = k\frac{\Delta T}{\delta} \tag{3}$$

where $h'_{fg} = h_{fg} + 3C_P\Delta T/8$ is the latent heat of condensation corrected for condensate subcooling by Rohsenow [15].

Applying the Bernoulli equation to the pressure gradient term in equation (2) along the interface and assuming the condensate film thickness to be neglected when compared with the radius of the tube, one may rewrite the momentum equation

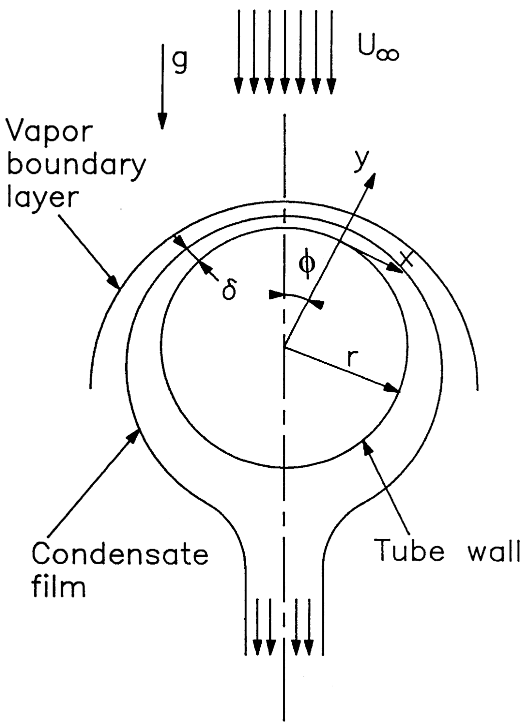

Fig. 1. Physical model and coordinate system.

$$\mu\frac{\partial^2 u}{\partial y^2} = -(\rho-\rho_v)g\sin\phi - \rho_v u_e\frac{\mathrm{d}u_e}{r\,\mathrm{d}\phi} \tag{4}$$

subjected to the following boundary conditions:

$$\frac{\partial u}{\partial y} = \tau_\delta/\mu, \quad \text{at } y=\delta \tag{5}$$

and

$$u = 0, \quad \text{at } y = 0. \tag{6}$$

Next, the interfacial boundary condition, i.e. the vapor shear, is to be modeled. A good approximation for high condensation rates, is given by Shekriladze and Gomelauri's [4] model as:

$$\tau_\delta = m''u_e. \tag{7}$$

According to potential flow theory, for a uniform flow with velocity U_∞ past a circular tube, one may derive the vapor velocity at the edge of boundary-layer as:

$$u_e = U_\infty 2\sin\phi \tag{8}$$

and then obtain the pressure gradient and interface vapor shear by using equation (8):

$$\rho_v u_e\frac{\mathrm{d}u_e}{r\,\mathrm{d}\phi} = \rho_v U_\infty^2 4\sin 2\phi/D \tag{9}$$

and

$$\tau_\delta = m''U_\infty 2\sin\phi. \tag{10}$$

Substituting equations (8) and (9) into equation (4) and its boundary conditions, one can solve the momentum equation as follows:

$$u = m''U_\infty 2y\sin\phi/\mu + [(\rho-\rho_v)g\sin\phi + \rho_v U_\infty^2 4\sin 2\phi/D]\left(y\delta - \frac{1}{2}y^2\right)/\mu. \tag{11}$$

The energy equation (3), is a balance between the latent heat released at the interface through condensation and heat flux conducted through the condensate film to the tube wall surface. However, the wall temperature distribution should be specified or fitted by measured data, then one can calculate the mean wall temperature as:

$$\bar{T}_w = \frac{1}{\pi}\int_0^\pi T_w(\phi)\,\mathrm{d}\phi \tag{12}$$

and express the temperature difference across the film by adopting the following form from Memory et al. [14]:

$$\Delta T = (T_{sat} - \bar{T}_w)(1 - A\cos\phi) = \overline{\Delta T}(1 - A\cos\phi) \tag{13}$$

where, A is a constant ($0 \leqslant A \leqslant 1$) and denotes the wall temperature variation amplitude.

Inserting equation (3) into equations (1) and (11) to eliminate m'', and integrating the updating equation (1) by using equation (11) and introducing the dimensionless parameters yields:

$$2\delta^*\frac{\mathrm{d}}{\mathrm{d}\phi}\left[\delta^*(1-A\cos\phi)\sin\phi+\frac{1}{3}\delta^{*3}F\sin\phi+\frac{4}{3}\delta^{*3}P\sin 2\phi\right]=1-A\cos\phi \quad (14)$$

with the boundary condition:

$$\mathrm{d}\delta^*/\mathrm{d}\phi=0 \quad \text{at } \phi=0 \quad (15)$$

where:

$$\delta^*=(\delta/D)\sqrt{Re} \quad (16)$$

$$F=(Ra/Ja)/Re^2;\quad P=(\rho_v/\rho)Pr/Ja. \quad (17)$$

The first term inside the derivative in equation (14) results from the interfacial shear stress while the term involving P is the effect of pressure gradient due to potential flow. When both of these terms are omitted, equation (14) reduces to the pure natural-convection film condensation, i.e. the Nusselt-type condensation problem.

Applying the boundary condition, equation (15) into equation (14), one may obtain the expression for the condensate film thickness at $\phi=0$ as:

$$1/(2\delta_0^*)-\delta_0^*-\frac{1}{3}(F+8P)\delta_0^{*3}=0. \quad (18)$$

Note that the above nonlinear equation can be solved for the condensate film thickness at $\phi=0$, δ_0^* by using the Newton–Raphson iteration method.

Before proceeding to obtain the solution of equation (14), and thence to calculate the heat transfer rate for the horizontal tube, it is to be noted that the condensate film flow may separate at the following condition:

$$\partial u/\partial y|_{y=0}\leqslant 0. \quad (19)$$

Thus, this condition may also be obtained by equation (14) in the following relationship:

$$1-A\cos\phi+F(1+8P\cos\phi)\delta^{*2}=0. \quad (20)$$

If $\phi=\phi_c$ satisfies the above equation, ϕ_c is called the critical angle, i.e.

$$\mathrm{d}\delta^*/\mathrm{d}\phi\to\infty \quad \text{as } \phi\to\phi_c. \quad (21)$$

Although δ^* is unknown, it may be solved by means of a fourth-order Runge–Kutta integration by taking step size $\Delta\phi=0.005^\circ$ and then substituting into equation (20) by bisection method to determine the position or value of ϕ_c. The algorithm is very unstable and sensitive to the calculated δ^* at ϕ close to ϕ_c, so we are required to check if the condensate film thickness will abruptly become extra thick, i.e.

$$\delta^*\to\infty \quad \text{as } \phi\to\phi_c. \quad (22)$$

Obviously, when $P=0$ or $F\geqslant 8P$ it satisfies, $\phi_c=\pi$. In this case the condensate film will separate or drip off at the bottom of the tube. Otherwise, $F<8P$, the critical angle lies in, $\pi/2<\phi_c\leqslant\pi$, and the condensate film will drip off before reaching the bottom of the tube. Since for the latter case, solutions will not be possible beyond ϕ_c, one may ignore the contribution to heat transfer due to an extra large film thickness.

The local heat flux q is given by:

$$q=k\Delta T/\delta=k\Delta\bar{T}(1-A\cos\phi)/\delta \quad (23)$$

which may be non-dimensionalized to give:

$$q^*=qD/(\sqrt{Re}\,k\,\Delta\bar{T})=(1-A\cos\phi)/\delta^*. \quad (24)$$

The mean heat flux for the circular tube is given by:

$$\bar{q}=\frac{2}{\pi D}\int_0^{\pi D/2}q\,\mathrm{d}x=\int_0^{\pi}q\,\mathrm{d}\phi/\pi. \quad (25)$$

As in the Nusselt theory, the dimensionless local heat transfer coefficient can be shown to be:

$$Nu=\frac{D}{\delta}=\sqrt{Re}/\delta^*. \quad (26)$$

We are also interest in an expression for the overall mean heat transfer coefficient. Integrating equation (26) over a whole tube, but neglecting the contribution to the heat transfer beyond ϕ_c based on the whole surface area gives:

$$\overline{Nu}=\bar{q}D/(k\Delta\bar{T}) \quad (27)$$

$$\overline{Nu}Re^{-1/2}=\int_0^{\phi_c}(1/\delta^*)(1-A\cos\phi)\,\mathrm{d}\phi/\pi. \quad (28)$$

It is to be noted that, at low vapor velocity, equation (28) blends with the Nusselt type solution. Furthermore, there are two asymptotic cases explained as follows.

Firstly, for the case of $F\gg 1$, i.e. for very slow vapor flow, the gravity force is much larger than vapor shear force and the pressure gradient due to potential flow. Hence, the problem reduces to the natural-convection film condensation, i.e. Nusselt-type condensation. After omitting the first term and final term in equation (14), one has:

$$\frac{2}{3}\delta^*\frac{\mathrm{d}}{\mathrm{d}\phi}(\delta^{*3}F\sin\phi)=1-A\cos\phi. \quad (29)$$

By separation of variables, one may obtain:

$$\delta^*=F^{-1/4}(\sin\phi)^{-1/3}\left[2\int_0^{\phi}(1-A\cos\phi)(\sin\phi)^{1/3}\,\mathrm{d}\phi\right]^{1/4}. \quad (30)$$

Thus, one can obtain the local Nusselt number from equation (26). Next, one may also obtain the overall mean Nusselt number from equation (28) as follows:

$$\overline{Nu}(Ja/Ra)^{1/4}=\int_0^{\phi_c}\left\{(1-A\cos\phi)(\sin\phi)^{1/3}\,\mathrm{d}\phi\left[2\int_0^{\phi}(1-A\cos\phi)(\sin\phi)^{1/3}\,\mathrm{d}\phi\right]^{-1/4}\right\}\mathrm{d}\phi/\pi. \quad (31)$$

It is to be noted that, for $A=0$, an isothermal wall surface, $\overline{Nu}(Ja/Ra)^{1/4}=0.728$.

Secondly, for the $F \ll 1$ cases, the other asymptotic cases: forced-convection dominated film condensation, their critical angles $\phi_c < \pi$ always when $F < 8P$. For pure forced-convection film condensation, one may put $F = 0$ in equation (17) and then obtain the mean heat transfer coefficients calculations using equations (14)–(28). By the way, if one put $P = 0$, the present work will reduce to be the same results with Memory et al. [14].

3. Results and discussion

In this section, numerical results for mixed-convection film condensation of downward flowing vapor on a horizontal tube are presented in two parts. In the first part, results of the condensate film thickness and its corresponding critical angles are obtained and discussed for a wide range of *F*, and a practical range of *A* and *P*. In practical condensation, we choose the cases of $P = 1$ and 3, because *P* is usually less than 1.0 for steam, and less than 4.0 for the refrigerants. Then, the second part will indicate and discuss the performance of heat transfer rates or Nusselt number for the same range of *F*, *A*, and *P* as the first part.

3.1. Characteristics of flow dynamics: condensate film thickness δ^*; critical angle ϕ_c

Firstly, for $F = 0$ and $P = 0$ case, i.e. for pure forced-convection film condensation, the results of numerical solutions from equation (14) are shown in Fig. 2a and just coincide with those of Memory et al. [14]. It is seen that for isothermal tube wall (Shekriladze–Gomelauri case, $A = 0$), the film thickness increases continuously with ϕ. For larger values of *A* (stronger temperature variation around the tube) the film thickness at first goes down to a minimum before increasing. It is noted that when $A = 1$, i.e. $\Delta T = 0$ at $\phi = 0$, its δ^* is minimum but not zero, which may be obtained directly from equation (14). Secondly, for $F = 10$, $P = 1$ and $P = 3$ two cases shown in Fig. 2b and c, i.e. for natural convection dominated film condensation, δ^* increases directly with ϕ. In fact, these results blend with Nusselt's solution. Besides, at the extreme value $A = 1$, when $\Delta T = 0$, at $\phi = 0$, it is seen that δ^* is zero and also agrees with Memory and Rose [13].

Furthermore, it also indicates that as compared to Fig. 2b and c, the condensate film for $P = 3$ case separates more ahead or at smaller ϕ_c than that for $P = 1$ case. This difference shows that the pressure gradient plays a significant effect upon the condensate film flow and cannot be ignored like Memory et al. [14]. If one ignores the effect of pressure gradient, the condensate film flow remain intact over the entire tube. However, according to the later works from Memory and Rose [16] and Yang and Hsu [17], the phenomenon of condensate film flow

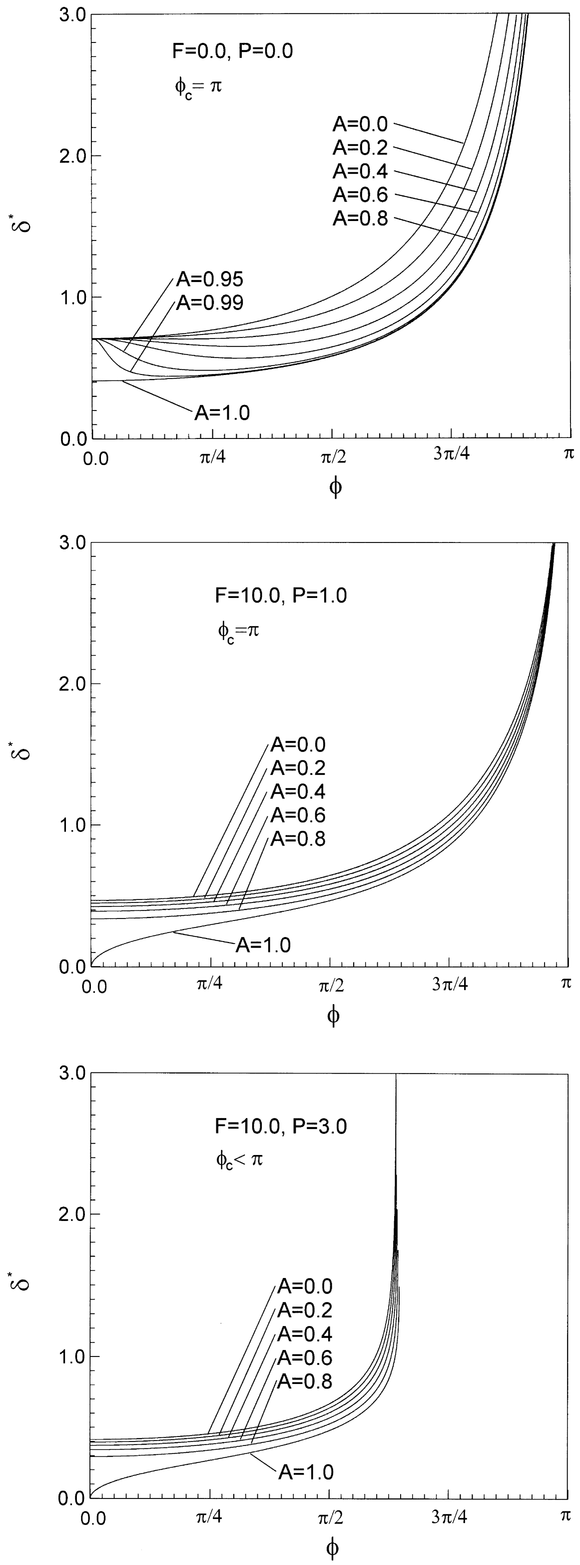

Fig. 2. (a) (b) (c). Dependence of dimensionless local film thickness on the wall temperature variation amplitude.

separation occurs because the pressure gradient effect exists in a forced flow. This effect may explain the contradictions between Memory et al. [14] and Honda and Fujii [9].

3.2. *Performance of heat transfer: local heat flux* q*, *mean heat transfer* $\overline{\text{Nu}}\ \text{Re}^{-1/2}$

3.2.1. *Profile of the local heat flux*

In Fig. 3, the results from equation (24) show that, for the isothermal wall, the local heat flux decreases continuously around the tube. As A increases, the local heat flux first rises where the effect of the increasing ΔT outweighs that of the increasing film thickness. Subsequently, the local heat flux reaches a maximum at a location on the rear half of the tube before decreasing to zero as the film thickness becomes infinite. All three cases have the similar trend, the higher F or lower vapor velocity is, the higher the local heat flux is.

3.2.2. *Effect of* F *(vapor flow velocity) and* P *(pressure gradient)*

The following sections regarding the mean heat transfer coefficient or mean Nusselt number are obtained numerically from equation (28). Numerical integrations with a step size twice as large produced a change in the predicted mean Nusselt number of less than 0.1%. Firstly, for no pressure gradient effect ($P = 0$), the mean Nusselt result is just the same form as that of Memory et al. [14] and illustrated in Fig. 4a. In addition, for an isothermal wall case ($A = 0$), the present result reduces to the Shekriladze–Gomelauri model. However, Memory et al. [14] used the Shekriladze–Gomelauri shear stress approximation without considering the pressure gradient

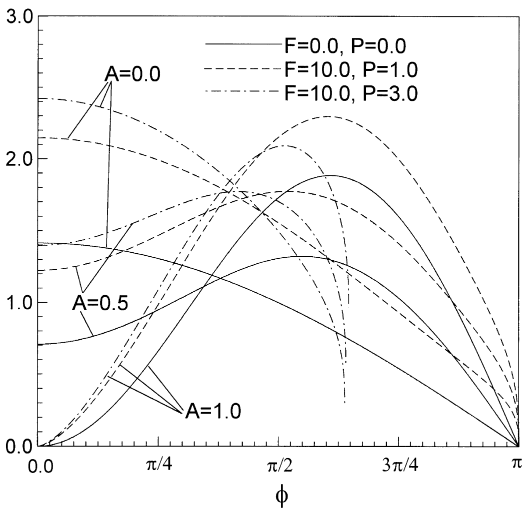

Fig. 3. Dependence of dimensionless local heat flux on the wall temperature variation amplitude A.

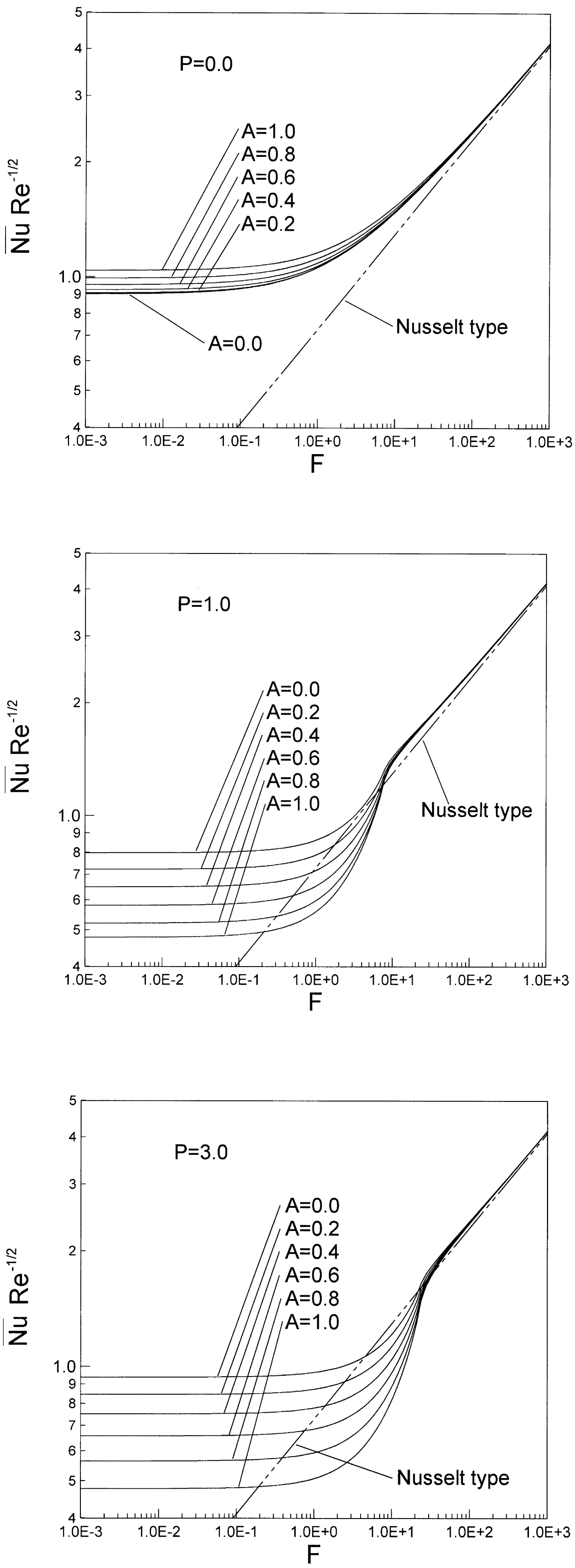

Fig. 4. (a) (b) (c). Dependence of mean Nusselt number on F for combined forced and natural convection condensation.

effects, so they overestimate the mean vapor-side heat-transfer coefficient. Hence, the present results with further accounting for the pressure gradients (including $P = 1$ and $P = 3$), as shown in Fig. 4b and c, make the mean heat transfer coefficients lower and reasonable, above all for lower F. Besides, there exists a transition zone near the Nusselt type's line around from $F = 1$ to $F = 10$.

3.2.3. Effect of A (*wall temperature variation amplitude*)

Figure 5 shows that in general, the mean Nusselt number increases insignificantly with A for $P = 0$ case no matter what the values of F are. But, as P increases, the mean Nusselt numbers decrease more appreciably with increasing A for both natural convection dominated ($F \geqslant 10$) condensation and forced convection dominated ($F < 1$) condensation. This is the reason that if one ignores the pressure gradient i.e. $P = 0$, the mean Nusselt number will be overpredicted.

4. Concluding remarks

(1) The present model with further inclusion of pressure gradient effect may compensate the discrepancy between the works from Memory et al. [14] and Honda and Fujii [9]. The present result appears to be adequate and can apply to the combined natural convection and forced convection film condensation.
(2) When $P = 0$, the mean heat transfer coefficient is increasing insignificantly with A, whereas as P is included and increases, the mean heat transfer coefficient decreases appreciably with A.

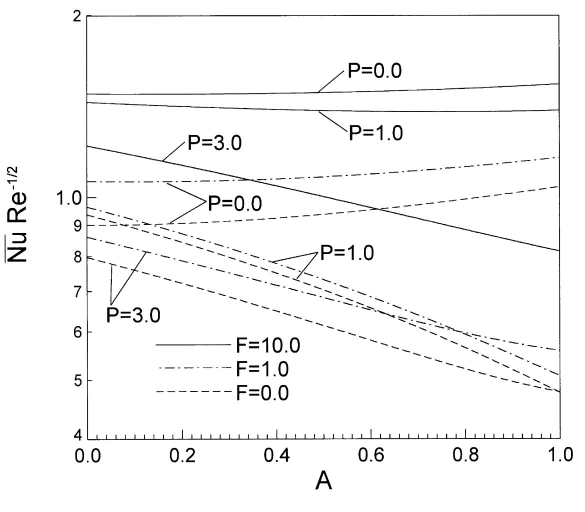

Fig. 5. Dependence of mean Nusselt number on parameter A for combined forced and natural convection condensation.

(3) The mean heat transfer coefficient is also nearly unaffected by the pressure gradient for the lower vapor velocity once its corresponding $\phi_c = \pi$. As for the higher vapor velocity (or higher F), the mean heat transfer coefficient decreases significantly with increasing the pressure gradient effect.
(4) Due to neglecting waviness in the condensate film layer, the present model should be cautiously applied when $Re > 3 \times 10^5$.

References

[1] W. Nusselt, Die oberflachen kondensation des wasserdampes, Zeitschrift des Verreines Deutscher Ingenieure 60 (1916) 541–546.
[2] E.M. Sparrow, J.L. Gregg, Laminar condensation heat transfer on a horizontal cylinder, Trans. ASME, J. Heat Transfer 81 (1959) 291–296.
[3] M.M. Chen, An analytical study of laminar film condensation: part 2—single and multiple horizontal tubes, J. Heat Transfer 83 (c) (1961) 55–60.
[4] I.G. Shekriladze, V.I. Gomelauri, The theoretical study of laminar film condensation of a flowing vapor, Int. J. Heat Mass Transfer 9 (1966) 581–591.
[5] V.E. Denny, A.F. Mills, Laminar film condensation of a flowing vapor on a horizontal cylinder at normal gravity, J. Heat Transfer 91 (c) (1969) 495–501.
[6] T. Fujii, H. Uehara, C. Kruata, Laminar film condensation of flowing vapor on a horizontal cylinder, Int. J. Heat Mass Transfer 15 (1972) 235–246.
[7] J.W. Rose, Effect of pressure gradient in forced convection film condensation on a horizontal tube, Int. J. Heat Mass Transfer 27 (1984) 39–47.
[8] E.S. Gaddis, Solution of the two phaseboundary-layer equations for laminar film condensation of vapor flowing perpendicular to a horizontal cylinder, Int. J. Heat Mass Transfer 22 (1979) 371–382.
[9] H. Honda, T. Fujii, Condensation of flowing vapor on a horizontal tube—numerical analysis as a conjugate heat transfer problem, J. Heat Transfer 106 (1984) 841–848.
[10] J.W. Rose, Fundamentals of condensation heat transfer: laminar film condensation, JSME Int. J. Series II. 31 (3) (1988) 357–375.
[11] T. Fujii, H. Honda, K. Oda, Condensation of steam on a horizontal tube—the influence of oncoming velocity and thermal condition at the wall. Condensation Heat Transfer ASME, Proceedings of the 18th National Heat Transfer Conference, 1979, pp. 35–43.
[12] A.G. Michael, J.W. Rose, L.C. Daniels, Forced convection condensation on a horizontal tube—experiments with vertical downflow of steam, J. Heat Transfer 111 (1989) 792–797.
[13] S.B. Memory, J.W. Rose, Free convection laminar film condensation on a horizontal tube with variable wall temperature, Int. J. Heat Mass Transfer 34 (1991) 2775–2778.
[14] S.B. Memory, W.C. Lee, J.W. Rose, Forced convection film condensation on a horizontal tube—effect of surface

temperature variation, Int. J. Heat Mass Transfer 36 (1993) 1671–1676.

[15] W.M. Rohsenow, Heat transfer and temperature distribution in laminar film condensation, Trans. ASME 78 (1956) 1645–1648.

[16] S.B. Memory, J.W. Rose, Forced convection film condensation on a horizontal tube—influence of boundary-layer separation, J. Heat Transfer 117 (1995) 529–533.

[17] S.-A. Yang, C.-H. Hsu, Superheated mixed convection film condensation in the forward stagnation region of a cylinder or sphere, J. Chinese Soc. Mech. Eng. 18 (1) (1997) 85–93.