JOURNAL OF THERMOPHYSICS AND HEAT TRANSFER
Vol. 19, No. 1, January–March 2005

Turbulent Film Condensation on an Isothermal Sphere

Hai-Ping Hu*
Tainan Woman's College of Arts and Technology, Tainan, 710 Taiwan, Republic of China
and
Cha'o-Kuang Chen†
National Cheng Kung University, Tainan, 701 Taiwan, Republic of China

An investigation is presented of turbulent film condensation on a sphere. It begins by considering the case of vapor flow past an isothermal sphere. The high tangential velocity of the vapor flow at the boundary layer is determined from potential flow theory. The Colburn analogy is used to define the local liquid–vapor interfacial shear that occurs for high-velocity vapor flow across the spherical surface. The results generated for the local dimensionless film thickness and heat transfer characteristics are discussed. Finally, the results developed in the current study are compared with those generated by previous theoretical and experimental results for laminar flow. It is found that the correlation between the two sets of data is quite satisfactory for low-vapor velocity. However, at the higher vapor velocities, the rate of increase of the coefficient with velocity is greater than that associated with laminar theory. It other words, for high-vapor velocity, the onset of turbulence in the condensate causes discrepancies between the two sets of data. Note that the results of the present study may be used to estimate the heat transfer coefficient more accurately than the result of previous laminar studies.

Nomenclature

Cp	=	specific heat of condensate at constant pressure
F	=	condensation parameter, $2/(SFr)$
Fr	=	Froude number, u_∞^2/gR
f	=	average friction coefficient
Gr	=	Grashof number, gR^3/ν_l^2
g	=	acceleration due to gravity, m/s^2
h	=	condensing heat transfer coefficient at angle θ
h_{fg}	=	latent heat of condensate, J/kg
Ja	=	Jacob number, $C_p(T_s - T_w)/h_{fg}$
j	=	heat transfer factor
k	=	thermal conductivity, w(mk)$^{-1}$
Nu	=	Nusselt number
Nu_m	=	mean Nusselt number
P	=	system pressure parameter
P	=	static pressure of condensate
Pr	=	Prandtl number
R	=	outer radius of the sphere
Re	=	Reynolds number
Re^+	=	Reynolds shear number
Re^*	=	wall shear parameter, $R^+/Gr^{1/3}$
S	=	sub cooling parameter, $C_p(T_s - T_w)/(h_{fg}Pr)$
T	=	temperature
T^+	=	dimensionless temperature
u	=	velocity component in x direction
u_e	=	tangential vapor velocity at the edge of the boundary layer
u^*	=	shear velocity
u_∞	=	vapor velocity of the freestream
u^+	=	dimensionless velocity
x	=	coordinate measured distance along circumference
y	=	coordinate measured distance normal to the wall surface
y^+	=	dimensionless distance
α	=	coefficient of thermal diffusivity
δ	=	condensate film thickness
δ^+	=	dimensionless film thickness
θ	=	angle measured from top of sphere
μ	=	absolute viscosity
ν	=	kinematic viscosity
ρ	=	density
τ	=	shear stress
τ_δ	=	interfacial vapor shear stress
ϕ	=	interfacial shear parameter, $2^{0.8}\pi C(\rho_v/\rho_l)(\nu_l/\nu_v)^{-0.2}Gr^{1.4/6}$

Subscripts

l	=	liquid
s	=	saturation
v	=	vapor
w	=	tube wall
$-$	=	vapor–liquid interface

Superscript

z	=	first guessed value of thickness

Introduction

FILM Condensation plays an important role in heat exchanger design, chemical engineering applications, electronic element cooling, and design of power plant cycles. Consequently, significant effort has been directed toward research into related fields. One such effort was conducted by the pioneeing investigator, Nusselt,[1] who investigated laminar film condensation on a quiescent vapor. In 1996, Shekriladze and Gomelauri[2] analyzed film condensation on horizontal tubes under low-velocity vapor flow conditions. Since then, much literature relating to research of laminar film condensation on horizontal tubes has been published, for example, Refs. 3–6. Yang and Hsu[7] analyzed the problem of combined free- and forced-convection on a horizontal elliptical tube and paid particular attention to the effects of vapor shear and pressure gradient.

Studies of the heat transfer process often consider the case of a spherical body. Typical studies include those conducted by Yang[8] and Karimi.[9] Both researchers considered the case of an isothermal wall with natural convection condensation. The case of forced-convection film condensation on a sphere using constant properties

Received 7 June 2002; revision received 18 February 2004; accepted for publication 18 February 2004.

*(**Job title**), Department of Information Management, 529 Chung Cheng Road Yung Kang City; th0003@ms.twcat.edu.tw.

†(**Job title**), Department of Mechanical Engineering; ckchen@mail.ncku.edu.tw.

analysis has also been investigated, e.g., by Hu and Jacobi[10] and by Jacobi.[11] These studies developed a generalized heat transfer model based on the vapor shear approximation model proposed by Shekriladze and Gomelauri.[12]

From the preceding paragraphs, it will be clear that there has been significant discussion of laminar film condensation within the published literature. However, it is also worthwhile developing an understanding of turbulent flow condensation because, according to Michael et al.,[12] condensate film may be partially turbulent at high vapor velocities. Sarma et al.[13] carried out theoretical research into turbulent film condensation on a horizontal tube and found that their results were in good agreement with experimental data relating to the condensation of steam flowing under a turbulent regime. However, there has been relatively little recent investigation of the issues relating to turbulent film condensation on a spherical body, for example, the relationships between the various heat transfer characteristics of the condensation film. Therefore, the aim of this present study is to investigate turbulent film condensation on a sphere by employing the model of Sarma et al. This paper also persents a comparison between the current results and those presented previously by Hu and Jacobi[10] and Hsu and Yang.[14]

Analysis

This paper considers a sphere immersed in the downward flow pure vapor that is at its saturation temperature T_s and that moves at a uniform velocity u_∞. The uniform wall surface temperature is T_w. Condensation occurs on the wall of the sphere, and a continuous film of the liquid runs downward over its surface under the combined effects of wall resistance, gravitational force, and the shear force caused by the external flow of the pure vapor. The physical model and the coordinate system adopted are shown in Fig. 1.

It is known that condensation may be partially turbulent at high vapor flow velocities. If turbulent flow if considered to occur near the wall of the sphere, then the momentum and energy equations are given as follows:

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = -\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x} + \frac{\partial}{\partial y}\left[(\nu + \varepsilon_m)\frac{\partial u}{\partial y}\right] + g_x \tag{1}$$

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \frac{\partial}{\partial y}\left[(\alpha + \varepsilon_H)\frac{\partial T}{\partial y}\right] \tag{2}$$

These equations are subject to three major assumptions. First, if turbulent flow is considered to occur near the wall of the sphere, then it may be assumed that the thickness of the liquid film is much smaller than the radius of the sphere. Second, regarding the heat transfer component of the turbulent boundary-layer equation, because the turbulent layer rubs against the wall and carries heat away from it, it is possible to make the assumption that it is sufficiently close to the wall that the left-hand components of Eqs. (1) and (2) may be neglected. Finally, the influence of turbulent conduction across the condensate layer is much more significant that the convective component. These assumptions allow the momentum and energy equation to be expressed in the following simplified form:

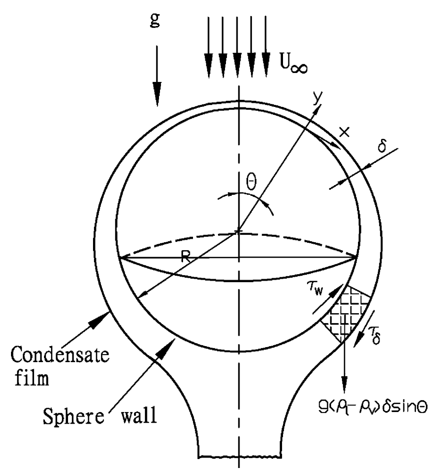

Fig. 1 Physical model and coordinate system.

$$-\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x} + \frac{\partial}{\partial y}\left[(\nu + \varepsilon_m)\frac{\partial u}{\partial y}\right] + g_x = 0 \tag{3}$$

$$\frac{\partial}{\partial y}\left[(\alpha + \varepsilon_h)\frac{\partial T}{\partial y}\right] = 0 \tag{4}$$

Additionally, a heat balance is achieved within the vapors at the interface, which is equal to the sum of all of the heat conducted across the film, that is,

$$\frac{\rho_l}{\sin\theta}\frac{\mathrm{d}}{\mathrm{d}\theta}\int_0^\delta \sin\theta u\,\mathrm{d}y = R\frac{K_l}{h_{fg}}\frac{\mathrm{d}T}{\mathrm{d}y}_{y=0} \tag{5}$$

The boundary conditions are 1) at $y = 0$,

$$T = T_w \tag{6a}$$

and 2) at $y =$ symbol of condensate film thickness,

$$T = T_s \tag{6b}$$

It is assumed that the condensate film is in a turbulent region in all regions other than at the upper stagnation point and that boundary-layer separation of the vapor may be neglected. The momentum equation, then, can be expressed as the following forced balance equation:

$$\tau_w - \tau_\delta - g\delta(\rho_l - \rho_v)\sin\theta = 0 \tag{7}$$

It can be known from the former equation, that the force balance for an element of the condensate film (Fig. 1) is governed by the wall stress, the inertia force, and the liquid–vapor interface shear stress.

For high vapor velocity, the semi-empirical equation that describes heat transfer in the flow parallel to a moderately curved surface may also be used to describe the heat transfer in the flow on a spherical surface. Jakob[15] proposed that this situation may be described for any fluid by the following expression:

$$Nu_v = C Re_v{}^{0.8} Pr^{\frac{1}{3}} \tag{8}$$

where C is a constant depending on flow configuration.

Colburn[16] suggested that the heat transfer factor could be represented as follows:

$$j = St Pr^{\frac{2}{3}} \tag{9}$$

This is in good agreement with $f/2$, and clearly in turbulent flow the two expressions may be compared, that is,

$$f/2 = St Pr^{\frac{2}{3}} \tag{10}$$

The average friction coefficient in the streamwise direction may then be written as

$$f = \frac{1}{2}\int_0^\pi \sin\theta\,\mathrm{d}\theta \tag{11}$$

With

$$f_\theta = 2C Re_v^{-0.2} \tag{12}$$

As stated in Ref. 17, the turbulent boundary layer exerts a friction force on the liquid–vapor boundary. The shear stress is estimated by considering the external flow of vapors across the sphere's surface when there is no condensation on that surface. The local friction coefficient is defined as

$$f_\theta = \tau_\delta \Big/ \tfrac{1}{2}\rho_v u_v^2 \tag{13}$$

According to potential flow theory, for a uniform vapor flow of velocity u_∞ passing a spherical surface, the vapor velocity at the edge of the boundary is given by

$$u_e = \tfrac{3}{2} u_\infty \sin\theta \tag{14}$$

Combining Eqs. (12–14) allows the local shear stress to be expressed as

$$\tau_\delta = \tfrac{9}{4}\rho_v u_\infty^2 C Re_v^{-0.2} \sin^2\theta \tag{15}$$

where the value of constant C is 0.034 and $Re_v = u_\infty D/\nu_v$.

Incorporating the interfacial vapor shear stress τ_δ given by Eq. (15) into the elemental force balance enables Eq. (7) to be rewritten in the following form:

$$\tau_w - \tfrac{9}{4}\rho_v u_\infty^2 C Re_v^{-0.2}\sin^2\theta - g\delta(\rho_l - \rho_v)\sin\theta = 0 \tag{16}$$

The following dimensionless variables and equations are now defined.

Friction velocity:

$$u^* = (\tau_w/\rho)^{\frac{1}{2}} \tag{17a}$$

Dimensionless velocity:

$$u^+ = u/u^* \tag{17b}$$

Dimensionless temperature:

$$T^+ = (T - T_w)/(T_s - T_w) \tag{17c}$$

Dimensionless distance:

$$y^+ = yu^*\big/\nu_l \tag{17d}$$

Dimensionless thickness:

$$\delta^+ = \delta u^*\big/\nu_l \tag{17e}$$

Shear parameter:

$$Re^* = Re^+\big/Gr^{\frac{1}{3}} \tag{17f}$$

Shear Reynolds:

$$Re^+ = Ru^*\big/\nu_l \tag{17g}$$

Froude number:

$$Fr = u_\infty^2\big/gR \tag{17h}$$

Modified Grashof number:

$$Gr = gR^3\big/\nu_l^2[(\rho_l - \rho_v)/\rho_l] \tag{17i}$$

The eddy diffusivity for momentum ε_m is assumed to be equal to that for energy ε_h. Substitution of these variables into the energy equation yields the following dimensionless energy equation:

$$\frac{d}{dy^+}\left[\left(1 + \frac{\varepsilon_m}{\nu_l}Pr\right)\frac{dT^+}{dy^+}\right] = 0 \tag{18}$$

The dimensionless boundary conditions of Eq. (18) are as follows:
1) At $y^+ = 0$,

$$T^+ = 0 \tag{19a}$$

and 2) at $y^+ = \_^+$,

$$T^+ = 1 \tag{19b}$$

Substitution of Eq. (17) into Eq. (5) gives

$$\frac{1}{\sin\theta}\frac{d}{d\theta}\int_0^{\delta+}\sin\theta u^+\,dy^+ = \frac{Ja}{Pr}Gr^{\frac{1}{3}}Re^*\frac{dT^+}{dy^+}_{y^+=0} \tag{20}$$

The ratio Ja/Pr is defined as the subcooling parameter S, and the Jacob number is

$$Ja = Cp(T_s - T_w)\big/h_{fg}$$

Substitution of Eq. (17) into Eq. (16) yields

$$Re^{*3} = (9/8\pi)Re^{*2}\,2^{0.8}\,\pi\cdot C(\rho_v/\rho_l)(\nu_l/\nu_v)^{-0.2}Gr^{1.4/6}Fr^{0.9}\sin^2\theta + \delta^+\sin\theta \tag{21}$$

where $2^{0.8}\pi C(\rho_v/\rho_l)(\nu_l/\nu_v)^{-0.2}Gr^{1.4/6}$ is defined as the interfacial shear parameter ϕ.

Furthermore, Eq. (20) requires the knowledge of the velocity profile in the liquid film, and it is obtained from the following expression[13]:

$$\left(1 + \frac{\varepsilon_m}{v}\right)\frac{du^+}{dy^+} = 1 \tag{22}$$

The boundary condition of Eq. (22) is at $y^+ = 0$

$$u^+ = 0 \tag{23}$$

The eddy diffusivity distribution applied in the present study was presented by Kato et al.[18] and is given by the following equation:

$$\varepsilon_m/\nu_l = 0.4y^+[1 - \exp(-0.0017y^{+2})] \tag{24}$$

Substituting the condition into Eq. (18) yields the following equation:

$$\frac{dT^+}{dy^+} = 1\Bigg/\left[\left(\int_0^{\delta^+}\frac{dy^+}{1 + 0.4y^+\left[1 - \exp\left(-0.0017y^{+^2}\right)\right]Pr}\right)\Bigg/\left(1 + \left[0.4y^+\left(1 - \exp\left(-0.0017y^{+^2}\right)\right]Pr\right)\right] \tag{25}$$

Substitution of Eq. (22) into Eq. (20) gives

$$Re^* = \left\{\int_0^{\delta^+}\sin\theta\int_0^{\delta^+}\sin\theta\,\frac{dy^+}{1 + 0.4y^+\left[1 - \exp\left(-0.0017y^{+^2}\right)\right]}dy^+\right\}\Bigg/SGr\frac{dT^+}{dy^+}\bigg|_{y^+=0}\sin\theta \tag{26}$$

From Eq. (21) the local dimensionless thickness is

$$\delta^+ = \frac{Re^{*3} - Re^*\phi Fr^{0.9}\sin^2\theta}{\sin\theta} \tag{27}$$

When the condensation heat transfer is considered, interpretation of the results of the model is straightforward if the following heat transfer coefficient is adopted:

$$K_l\frac{\partial T}{\partial y}\bigg|_{y=0} = h(T_s - T_w) \tag{28}$$

In a dimensionless form, Eq. (28) can be expressed as a local Nusselt number, that is,

$$Nu = Re^+\frac{dT^+}{dy^+}\bigg|_{y^+=0} \tag{29}$$

Obviously, the local Nusselt number can also be expressed as follows:

$$\frac{Nu}{Re_l^{\frac{1}{2}}} = \frac{Gr^{\frac{1}{2}}}{Fr^{\frac{1}{4}}}Re^*\frac{dT^+}{dy^+}\bigg|_{y^+=0} \tag{30}$$

where $Re_l = u_\infty R/\nu_l$ and $Nu = hR/k_l$. Integration of this expression over the complete circumference of the sphere provides an overall value of the mean Nusselt number, that is,

$$\frac{Nu_m}{Re^{\frac{1}{2}}} = \frac{1}{2}\int_0^\pi \frac{Nu}{Re^{\frac{1}{2}}} \sin\theta \, \mathrm{d}\theta \tag{31}$$

The equations just developed will now be applied in the following section of this paper, which relates to the numerical method.

Numerical Method

The dimensionless governing equations (25–27) can be used to estimate the values of δ^+ and Re^*. The estimation process can be described as follows.

1) Suitable dimensionless parameters, such as S, Pr, Gr, and ϕ, are specified.

2) Boundary conditions corresponding to the system conditions are input; that is, at $i = 0$, the dimensionless film thickness $^{+}_{-}$ is zero. At the next node, that is, $i = i + 1$, the value of θ is given by $\theta_{i+1} = \theta_i + \Delta\theta$, where $\Delta_\theta = (\pi/300)$. In this way, the angle is within the range $0 < \theta \leq \pi$.

3) In Eq. (25), the dimensionless film thickness is assumed to be $\delta^+_{i=1}$ at the first node, that is, $i = 1$. The temperature gradient

$$\left.\frac{\mathrm{d}T^+}{\mathrm{d}y^+}\right|_{y^+=0}$$

can be calculated and then substituted into Eq. (26).

4) The shear Reynolds parameter at the first node $(Re^*_{i=1})$ can be calculated. Substitution of the shear Reynolds parameter Re^*, into Eq. (27) allows the calculation of a new value of the dimensionless film thickness, that is, $\delta^{+^{z+1}}_{i=1}$. If comparison of $\delta^{+^{z}}_{i=1}$ and $\delta^{+^{z=1}}_{i=1}$ reveals no similarity between the two values, a new, corrected value of $\delta^{+^{z}}_{i=1}$ will be assumed, and computation continues until the convergence criterion is attained, that is,

$$\left|\frac{\delta_i^{+^{z+1}} - \delta_i^{+^{z}}}{\delta_i^{+^{z+1}}}\right| \leq 1 \times 10^{-6}$$

5) This process is repeated at the next node position, that is, $\theta_{i+1} = \theta_i + \Delta\theta$, and then subsequently at all nodes within the range $0 < \theta \leq \pi$.

6) The local Nusselt number and the mean Nusselt number are then calculated.

Results and Discussion

Figure 2 presents the variation of the dimensionless film thickness of the condensate around the circumference of the sphere for different values of the subcooling parameter S. Because the shear velocity u^* is zero at the upper stagnation point, that is, $\theta = 0$, the value of the dimensionless film thickness δ^+ is also zero. Figure 2 shows that the dimensionless film thickness increases continuously from its minimum value at the upper stagnation point, that is, $\theta = 0$, as the value of θ increases and that it reaches its maximum value at the lower stagnation point, that is, $\theta = \pi$. Furthermore, note that an increase in either the value of the subcooling parameter S, or in the Froude number, leads to an increase in the local condensate film thickness. Because the dimensionless parameter F is defined as $F = 2/(SFr)$, an increase in either the S or the Froude number Fr value will result in a decrease in the value of F. Conversely, when F increases, the dimensionless film thickness decreases. This result is similar to that presented by Yang[8] in his research on laminar film condensation. Therefore, it may be stated that, in both turbulent film condensation and laminar film condensation, an increase in F will lead to a decrease in δ^+.

Figure 3 presents the variation of the local Nusselt number of the condensate around circumference of the sphere. Figure 3 shows three different subcooling parameter values and two Froude numbers. The plotted data are derived from Eq. (30). Figure 3 shows that, for all values of S, the value of the local Nusselt number at the

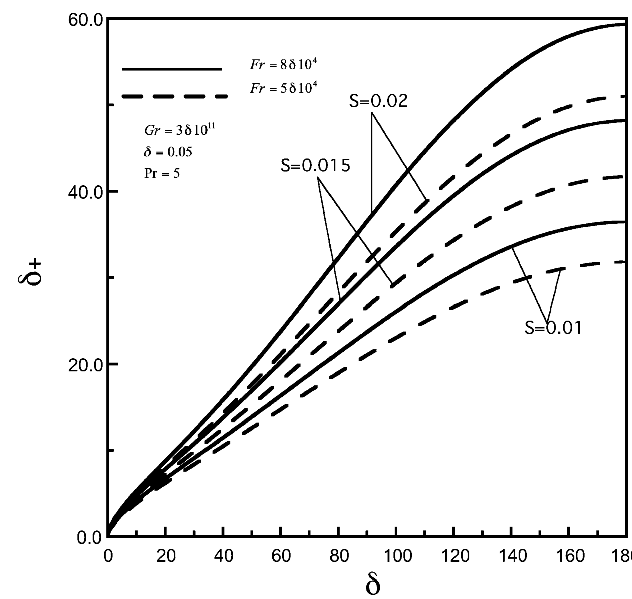

Fig. 2 Variation of dimensionless local film thickness around periphery of sphere.

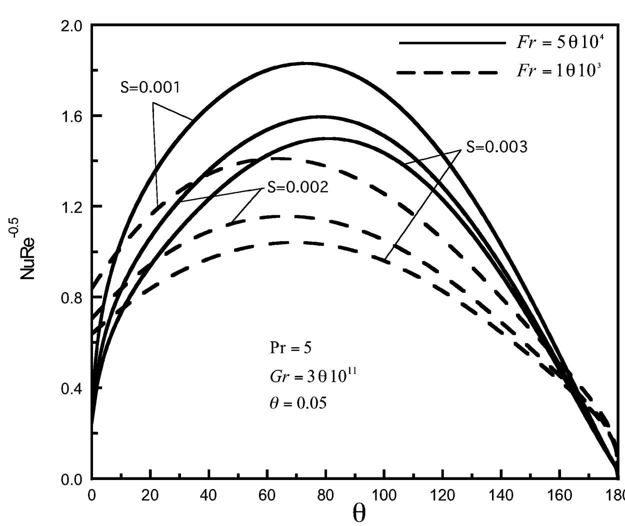

Fig. 3 Variation of local Nusselt number around periphery of sphere.

upper stagnation point, that is, $\theta = 0$, is less in the case of the higher Froude number. However, for the same value of S, the maximum local Nusselt number is higher for the higher Froude number case. Note that the rate of increase in the local Nusselt number is higher in the case of higher Froude numbers. Additionally, it can be noted that as the Froude number increases, the location of the maximum local Nusselt number moves closer to $\theta = \pi/2$. Finally, Figure 3 clearly indicates that the local Nusselt number decreases as the value of the subcooling parameter increases.

Figure 4 presents the relationship between the mean Nusselt number of the sphere and the subcooling parameter for different Froude numbers. For any particular Froude value, it can be seen that the mean Nusselt number increases as the value of the subcooling parameter S decreases. However, for a given value of S, the mean Nusselt number is dependent on the Froude number and can be seen to increase as the Froude value increases.

Figure 5 shows the influence of the interfacial shear parameter ϕ on the mean Nusselt value for three different Froude number values. The system pressure parameter is given by $P = \rho_v Pr/\rho_l Ja$, thus, when ρ_v/ρ_l is higher, $(P \cdot S)$ is also higher (where $S = Ja/Pr$). Figure 5 shows that, for a fixed value of S, the system pressure increases as the value of ϕ increases. As a result, the increase in system pressure leads to an increase in the mean Nusselt number.

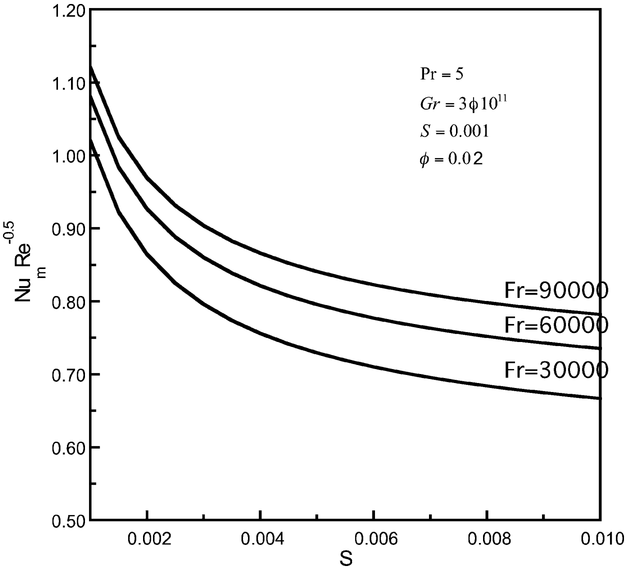

Fig. 4 Effect of subcooling parameter on mean Nusselt number.

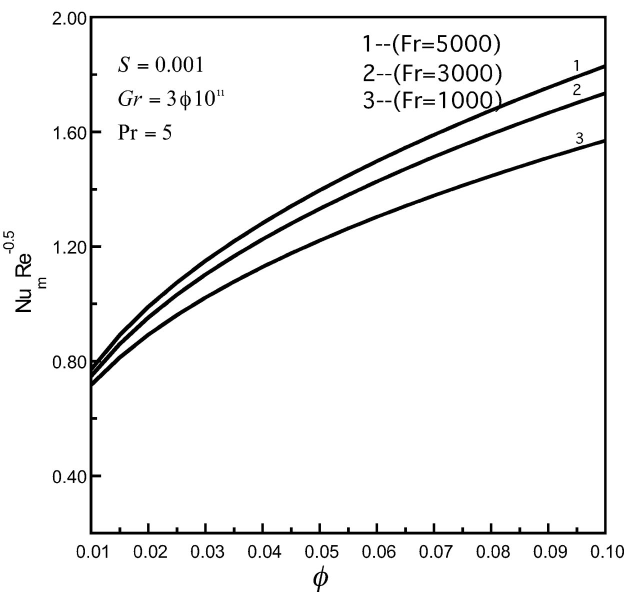

Fig. 5 Dependence of mean Nusselt number on ϕ effect of *Fr*.

Figure 6 shows the influence of the system pressure on the mean Nusselt number for five different values of the subcooling parameter S within the range 0.001–0.005. It can be seen that the mean Nusselt number decreases as the value of S increases. Additionally, note that an increase in system pressure results in an increase in the mean Nusselt number.

Figure 7 provides a comparison between the present results for the mean Nusselt number for a sphere and those generated in two previous studies. At low-to-moderate velocity, that is, a higher value of F, the current results are in good agreement with the laminar condensate flow results published by Hsu and Yang[14] and Hu and Jacobi.[10] In the case of high vapor velocity, that is, $F < 0.1$, it is immediately obvious that the rate of increase of the mean Nusselt number in the present research is greater than that determined by Hsu and Yang and Hu and Jacobi. Hence, under high vapor velocity, the mean Nusselt number is significantly influenced by vapor velocity "because of the onset of turbulence in the condensate film" as stated by Lee et al.[19] Michael et al.[12] also mentioned in their 1989 study that "laminar condensate flow theories are accurate only at low vapor velocities." It can be seen that the results of the present paper support these comments. Figure 7 also shows that, for high vapor velocity, the mean Nusselt number increases for higher values of ϕ. In contrast, for low vapor velocity, the influence of the ϕ value on the heat transfer coefficient is less significant.

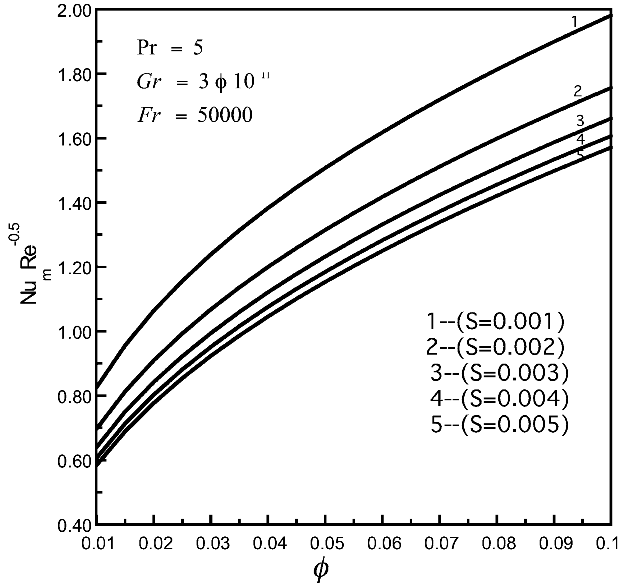

Fig. 6 Dependence of mean Nusselt number on ϕ effect of subcooling parameter.

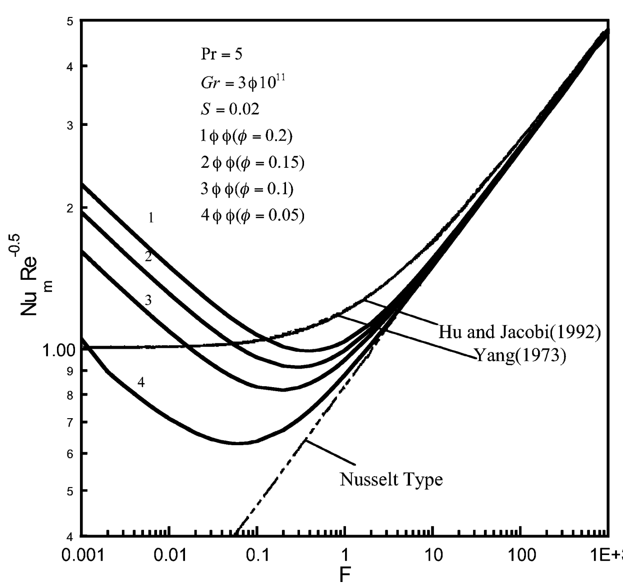

Fig. 7 Dependence of mean Nusselt on F, $F = 2/(SFr)$.

Conclusions

The following conclusions may be drawn from the results of this study:

1) At low-to-moderate velocity flow, that is, higher values of F, the present theoretical results for turbulent flow are in fair agreement with the theoretical results for laminar condensate flow. For high vapor velocity flow, that is, $F < 0.1$, the rate of increase of the mean Nusselt number identified in the present research is greater than previously reported data. When $F < 0.1$, the mean Nusselt number is significantly influenced by the vapor velocity due to the onset of turbulence in the condensate film.

2) It has been found that, for higher Froude numbers, the rate of increase in the local Nusselt number is higher and that the maximum value of the local Nusselt number occurs near to $\theta = \pi/2$. Furthermore, the maximum value of the local Nusselt number occurs for the higher Froude numbers. Moreover, it has been determined that in the turbulent condensate film under high vapor velocities the variation

of system pressure and subcooling parameter around the circumference may have significant effects on heat transfer coefficient.

3) The present results confirm Jakob's theory that the semi-empirical equation that describes the heat transfer in a flow parallel to a moderately curved surface may also be used to describe the heat transfer in the turbulent region of a flow on a spherical surface. Furthermore, the present results also confirm that Colburn's analogy may be used successfully to estimate the interfacial shear at the interface for high-velocity vapors on a spherical surface.

References

[1]Nusselt, W., "Die Oberflachen Kondensation Des Wasserdampers," *Zeitsehrift des Vereines Deutsher Ingenieure*, Vol. 60, No. 4, 1916, pp. 541–546.

[2]Shekriladze, I. G., and Gomelauri, V. I., "Theoretical Study of Laminar Film Condensation of Flowing Vapor," *International Journal of Heat and Mass Transfer*, Vol. 9, No. 6, 1966, pp. 581–591.

[3]Fujii, T., Uehara, H., and Kurata, C., "Laminar Filmwise Condensation of Flowing Vapor on a Horizontal Condenser," *International Journal of Heat and Mass Transfer*, Vol. 15, No. 20, 1972, pp. 235–246.

[4]Fujii, T., Honda, H., and Oda, K., "Condensation of Steam on a Horizontal Tube the Influence of Oncoming Velocity and Thermal Condition at the Tube Wall," *Proceedings of the 18th National Heat Transfer Conference, Condensation Heat Transfer*, ASME, San Diego, CA, 8 Aug. 1979, pp. 35–43.

[5]Honda, H., Zozu, S., Uchima, B., and Fujii, T., "Effect of Vapor Velocity on Film Condensation of R-113 on Horizontal Tubes in Across Flow," *International Journal of Heat and Mass Transfer*, Vol. 29, No. 3, 1986, pp. 429–438.

[6]Sarma, P. K., Subrahmanyam, T., Dharma Rao, V., and Rao, P. V. S. K., "Condensation of Vapor Flowing Normal to the Axis of a Horizontal Tube," *Engineering Foundation Conference, Condensation and Condenser Design*, American Society of Mechanical Engineers, New York, 1993.

[7]Yang, S. A., and Hsu, C. H., "Free- and Forced-Convection Film Condensation from a Horizontal Elliptic Tube with a Vertical Plate and Horizontal Tube as Special Cases," *International journal of Heat and Fluid Flow*, Vol. 18, No. 6, 1997, pp. 567–574.

[8]Yang, J. W., "Laminar Film Condensation on a Sphere," *Journal of Heat Transfer*, Vol. 95C, No. 2, 1973, pp. 174–178.

[9]Karimi, A., "Laminar Film Condensation on Helical Reflux Condensers and Related Configurations," *International Journal of Heat and Mass Transfer*, Vol. 20, No. 11, 1977, pp. 1137–1144.

[10]Hu, H., and Jacobi, A. M., "Vapor Shear and Pressure Gradient Effects During Filmwise Condensation from a Flowing Vapor onto a Sphere," *Experimental Thermal and Fluid Science*, Vol. 5, No. 4, 1992, pp. 548–555.

[11]Jacobi, A. M., "Filmwise, Condensation from a Flowing Vapor onto Isothermal, Axisymmetric Bodies," *Journal of Thermophysics and Heat Transfer*, Vol. 6, No. 2, 1992, pp. 321–325.

[12]Michael, A. G., Rose, J. W., and Daniels, L. C., "Forced Convection Condensation on a Horizontal Tube-Experiments with Vertical Downflow of Steam," *Journal of Heat Transfer*, Vol. 111, No. 3, 1989, pp. 792–797.

[13]Sarma, P. K., Vijayalakshmi, B., and Mayinger, F., "Turbulent Film Condensation on a Horizontal Tube with External Flow of Pure Vapors," *International Journal of Heat and Mass Transfer*, Vol. 41, No. 3, 1998, pp. 537–545.

[14]Hsu, C. H., and Yang, S. A., "Mixed Convection Film Condensation from Downward Flowing Vapors onto a Sphere with Variable Wall Temperature," *Heat and Mass Transfer*, Vol. 18, No. 1–2, 1997, pp. 85–91.

[15]Jakob, M., *Heat Transfer*, Wiley, New York, 1949, pp. 578–580.

[16]Colburn, A. P., "A Method of Correlation Forced Convection Heat Transfer Data and a Comparison with Fluid Friction," *Transactions of the AICE*, Vol. 29, 1933, pp. 174–210.

[17]Bejan, A., *Convection Heat Transfer*, Wiley, New York, 1995, pp. 343–347.

[18]Kato, H., Shiwaki, N. N., and Hirota, M., "On the Turbulent Heat Transfer by Free Convection from a Vertical Plate," *Internal of Heat and Mass Transfer*, Vol. 11, No. 7, 1968, pp. 117–1125.

[19]Lec, W. C., Rahbar, S., and Rose, J. W., "Film Condensation of Refrigerant 113 and Ethanediol on a Horizontal Tube—Effect of Vapor Velocity," *Journal of Heat Transfer*, Vol. 106, No. 3, 1984, pp. 524–530.