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Role of surface tension and ellipticity in laminar film condensation on a horizontal elliptical tube

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Abstract—An analytical study is made into the process of heat transfer with the vapor condensation on a horizontal elliptical tube under the simultaneous effects of the forces of surface tension and gravity on the condensate film. Analytical expressions for both local condensate film thickness and heat transfer coefficient around the elliptical periphery have been derived under the effects of gravity and surface tension for various values of ellipticity, respectively. The dimensionless mean heat transfer coefficient $-\overline{Nu}$ for any ellipticity e and various Bond numbers Bo has been obtained; however, it is almost unaffected by surface tension force due to surface curvature changing. For special objects (vertical plate $e = 1$, circular tube $e = 0$), the results are identical to some classical Nusselt-type solutions.

1. INTRODUCTION

A NUMBER of condensing systems, such as flat plates [1, 2], circular cylinders [3, 4], spheres [5], and non-circular cylinders [6] have been extensively studied regarding the heat transfer process of film-wise condensation. This is true, for example, in space applications, in certain heat pipe configurations, and in chemical engineering processes. According to Semenov *et al.* [7], they found that if the contour of the cross-section of a non-circular horizontal tube on such condensation takes place is so elongated in the gravitational direction that its curvature decreases continuously from the upper generatrix to the lower generatrix of the tube, the condensation heat transfer is enhanced.

Cheng and Tao [8] in 1987, and Wang *et al.* [9] in 1988 have confirmed theoretically and experimentally that an elliptical tube did possess some advantages over a cylindrical one. However, they studied laminar film condensation on a horizontal elliptical tube on the basis of Nusselt theory under the effect of gravity force alone, and they miscalculated the mean condensation coefficient $\bar{h}$ for an ellipse by using

$$\bar{h} = \frac{1}{\pi}\int_0^{\pi} h \, \mathrm{d}\theta$$

instead of taking an averaged value over the entire perimeter. It is to be noted that, for an elliptical tube, the radius of surface curvature is not a constant and cannot be omitted in evaluating $\bar{h}$. In addition to the effect of gravity force, there exists the effect of surface tension forces due to the non-uniform curvature of an elliptical surface on the film flow. Hence, we take into further account the effect of surface tension forces on the condensate film flow outside a horizontal elliptical tube. Besides, from the mathematical point of view, a circular tube is one kind of elliptical tube with zero ellipticity; a flat plate is another kind of elliptical tube with ellipticity equal to one. Our major aim is expected to extend the horizontal elliptical tube in the engineering applications and also see the effect of surface curvature upon the heat transfer rate and hydrodynamics characteristics.

2. ANALYSIS

Consider a horizontal elliptical tube, with major axis '$2a$' in the direction of gravity and minor axis '$2b$', situated in a quiescent pure vapor which is at its saturation temperature T_{sat}. The wall temperature T_w is uniform and below the saturation temperature. Thus, condensation occurs on the wall and a continuous film of the liquid runs downward over the tube under the actions of the component of gravity, which is parallel to the tangent of the tube wall, and of the surface tension forces.

The physical model under consideration is shown in Fig. 1 where the curvilinear coordinates (x, y) are aligned along the elliptical wall surface and its normal. Their corresponding velocities u and v are accordingly assigned. For a laminar, steady-state condensate film with constant fluid properties, the boundary layer equations governed by the basic conservation principles: mass, momentum, and energy are

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

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = \mu\frac{\partial^2 u}{\partial y^2} + (\rho - \rho_v)g\sin\phi - \frac{\partial P}{\partial x} \tag{2}$$

$$\rho C_p\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = k\frac{\partial^2 T}{\partial y^2} \tag{3}$$

where $\phi = \phi(x)$ is the angle between the horizontal direction and the tangent to the tube wall at the pos-

NOMENCLATURE

a	semi-major axis of ellipse
b	semi-minor axis of ellipse
Bo	Bond number, $(\rho-\rho_v)ga^2/\sigma$
C_p	specific heat of condensate at constant pressure
D_e	equivalent diameter of elliptical tube, defined in equation (21)
e	ellipticity of ellipse
F	dimensionless function, defined in equation (20)
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 condensation corrected for condensate subcooling
Ja	Jakob number, $C_p(T_{sat}-T_w)/h'_{fg}$
k	thermal conductivity of condensate
l	length of flat plate
$\dot{m}$	condensate mass flow rate per unit length of elliptical tube
Nu	local Nusselt number, hD_e/k
$\overline{Nu}$	mean Nusselt number, $\bar{h}D_e/k$ for elliptical tube
$\overline{Nu_l}$	mean Nusselt number, $\bar{h}l/k$ for flat plate
P	static pressure of condensate
Pr	Prandtl number
r	radial distance from centroid of ellipse to the tube wall
R	radius of elliptical surface curvature
Ra	Rayleigh number, $(\rho-\rho_v)\rho g\ Pr\ D_e^3/\mu^2$
s	dimensionless streamwise length, defined in equation (30)
S_f	dimensionless integral function, defined in equation (27)
T_{sat}	saturation temperature of vapor
T_w	wall temperature
u	velocity component in x direction
v	velocity component in y direction
x	coordinate measuring distance along circumference from top of tube
y	coordinate normal to the elliptical surface.

Greek symbols

δ	thickness of condensate film
δ^*	dimensionless thickness of condensate film, defined in equation (22)
θ	angle measured from top of tube
μ	absolute viscosity of condensate
ρ	density of condensate
ρ_v	density of vapor
σ	surface tension coefficient in the film
ϕ	angle between the tangent to tube surface and the normal to direction of gravity
ϕ_c	critical angle making $\sin\phi+Bo(\phi)=0$
φ	inclined angle of flat plate with direction of gravity.

Subscripts

l	flat plate
sat	saturation
v	vapor
w	tube wall.

Superscripts

*	indicates dimensionless
‾	indicates average.

ition (r,θ). Here, θ is the angle measured from the tube upper generatrix; r is the radial distance from the centroid of the ellipse and can be expressed as

$$r = a[(1-e^2)/(1-e^2\cos^2\theta)]^{0.5} \tag{4}$$

where $e = \sqrt{(a^2-b^2)}/a$ is the ellipticity.

Owing to the very thin film thickness, compared with the radius of surface curvature, one may approximately express the pressure gradient as

$$-\frac{\partial P}{\partial x} = \frac{\sigma}{R^2}\frac{\partial R}{\partial x} \tag{5}$$

where R, the radius of curvature of the ellipse at the position (r,θ) can be derived as

$$R = \frac{a}{\sqrt{(1-e^2)}}\left[\frac{1+e^2(e^2-2)\cos^2\theta}{1-e^2\cos^2\theta}\right]^{3/2}. \tag{6}$$

In free convection, inertia and convective terms are neglected, as is usual in Nusselt-type condensation theory. The momentum and energy equations reduce to

$$\mu\frac{\partial^2 u}{\partial y^2} = -(\rho-\rho_v)g\sin\phi - \frac{\sigma}{R^2}\frac{\partial R}{\partial x} \tag{7}$$

and

$$k\frac{\partial^2 T}{\partial y^2} = 0. \tag{8}$$

It is further assumed that at the interface no vapor shear is considered to exert upon the condensate. Thus, the boundary conditions are

$$\frac{\partial u}{\partial y} = 0; \quad T = T_{sat} \quad \text{at } y = \delta \tag{9}$$

and

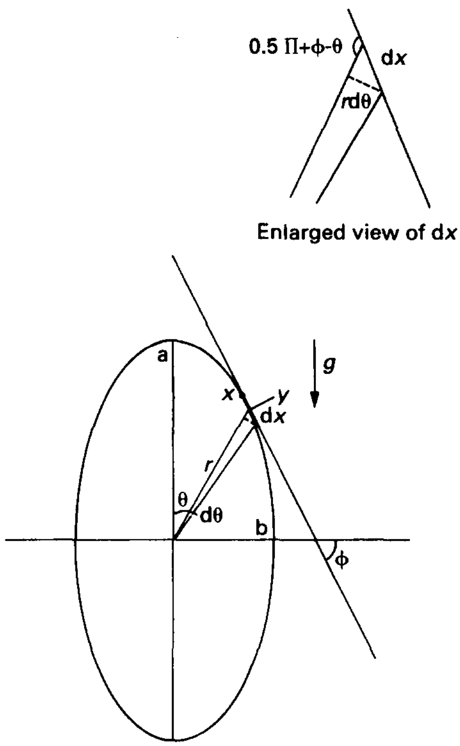

FIG. 1. Schematic and coordinate system for the condensate film flow on the elliptical surface.

$$u = 0, \quad T = T_w \quad \text{at } y = 0. \tag{10}$$

Consequently, the momentum and energy equations can be solved as follows:

$$u = \frac{(\rho-\rho_v)g\delta^2}{\mu}[\sin\phi + Bo(x)]\left[\frac{y}{\delta} - \frac{1}{2}\left(\frac{y}{\delta}\right)^2\right] \tag{11}$$

$$T = (T_{sat} - T_w)\frac{y}{\delta} + T_w \tag{12}$$

where

$$Bo(x) = \frac{1}{Bo}(a/R)^2\frac{\partial R}{\partial x}; \quad Bo = (\rho-\rho_v)ga^2/\sigma$$

the Bond number. With the help of equation (12), the heat flux at the liquid–vapor interface is related to the rate of condensation by

$$h'_{fg}\frac{d\dot{m}}{dx} = k\frac{dT}{dy} = k\frac{T_{sat}-T_w}{\delta} \tag{13}$$

where $\dot{m}$ is the rate of the condensate mass flow over an elliptical perimeter per unit tube length, and $h'_{fg} = h_{fg} + 0.68C_p(T_{sat} - T_w)$, latent heat of condensation corrected for condensate subcooling by Rohsenow [2]. Utilizing equation (11), one obtains the local rate of the condensate mass flow per unit tube length as follows:

$$\dot{m} = \frac{(\rho-\rho_v)\rho g\delta^3}{\mu}[\sin\phi + Bo(x)]. \tag{14}$$

In order to derive the local film thickness δ at the circumferential arc length x (or angle θ) in terms of ϕ, one can substitute equation (14) into equation (13) and obtain

$$\rho\frac{(\rho-\rho_v)g}{3\mu}h'_{fg}\frac{d}{dx}\{\delta^3[\sin\phi + Bo(x)]\} = \frac{k}{\delta}(T_{sat} - T_w). \tag{15}$$

It is more convenient at this point to express dx in polar coordinates. With reference to Fig. 1, the differential elliptical arc length may be written as

$$dx = \frac{r\,d\theta}{\cos(\phi-\theta)}. \tag{16}$$

By using the geometric relationship of an ellipse, it may be shown that the tangent at any point is given as

$$\tan\phi = \tan\theta/(1-e^2). \tag{17}$$

Furthermore, with the help of equations (4) and (17) in equation (16) and in the pressure gradient term, one may obtain the following expressions in terms of e and ϕ:

$$dx = a[(1-e^2)/\sqrt{((1-e^2\sin^2\phi)^3)}]\,d\phi \tag{18}$$

and

$$Bo(x) = Bo(\phi) = \frac{3e^2}{2Bo}\left(\frac{1-e^2\sin^2\phi}{1-e^2}\right)^2\sin(2\phi). \tag{19}$$

Substituting equations (18) and (19) into equation (15), and introducing the transformation of the variable from x to ϕ, one can obtain the local film thickness at θ as follows:

$$\delta = \left[\frac{2ak\mu(T_{sat}-T_w)}{h'_{fg}(\rho-\rho_v)\rho g}\right]^{1/4}F(\phi) \tag{20}$$

where

$$F(\phi) = [\sin\phi + Bo(\phi)]^{-1/3}\left\{2(1-e^2)\int_0^{\phi}[\sin\phi + Bo(\phi)]^{1/3}(1-e^2\sin^2\phi)^{3/2}\,d\phi\right\}^{1/4}.$$

It is to be noted that the above relation applies to the angles from $\phi = 0$ to $\phi_c(<\pi)$. The critical angle, ϕ_c, is the root of $\sin\phi + Bo(\phi) = 0$. For $\phi \geqslant \phi_c$, since the condensate film layer is dripping off the tube, $F(\phi)$ and δ are considered as infinity. In order to compare with circular tubes, based on the same condensing area, or the same perimeter per unit length of tube, one may express the film thickness in terms of equivalent diameter D_e

$$D_e = 2\frac{a}{\pi}\int_0^{\pi}[(1-e^2)/\sqrt{((1-e^2\sin^2\phi)^3)}]\,d\phi \tag{21}$$

and obtain the local dimensionless film thickness

$$\delta^* = \delta\left[\frac{D_c k\mu(T_{sat}-T_w)}{h'_{fg}(\rho-\rho_v)\rho g}\right]^{-1/4} = F(\phi)\left\{\frac{1}{\pi}\int_0^{\pi}[(1-e^2)\Big/\sqrt{((1-e^2\sin^2\phi)^3)}]\,d\phi\right\}^{-1/4}. \quad (22)$$

Similar to equation (20), the above relation also applies to the angles from $\phi = 0$ to ϕ_c. After the critical angle ($\phi_c \leqslant \phi \leqslant \pi$), the condensate is dripping off the tube. Hence, during the performing of calculation of equation (22), after this singularity $\phi = \phi_c$, $F(\phi)$ and δ should be considered as infinity because the assumption of equation (5) does not apply for the infinite value of δ. Hence, the local heat transfer coefficient at a particular angle $\phi < \phi_c$ may be expressed as

$$h = \left[\frac{(\rho-\rho_v)\rho g h'_{fg} k^3}{2a\mu(T_{sat}-T_w)}\right]^{1/4} \Big/ F(\phi) \quad (23)$$

and

$$h = 0 \quad \text{for } \phi \geqslant \phi_c.$$

Consequently, the local dimensionless heat transfer coefficient may be obtained as

$$Nu = \frac{hDe}{k} = [Ra/Ja]^{1/4}/\delta^* \quad (24)$$

where

$$Ra = (\rho-\rho_v)\rho g\, Pr\, D_e^3/\mu^2 \quad \text{and}$$

$$Ja = C_p(T_{sat}-T_w)/h'_{fg}.$$

Next, in the procedure to obtain the expression of the mean heat transfer coefficient, firstly, insertion of equation (14) into (15) gives

$$\dot{m}^{1/3}\,d\dot{m} = \frac{k(T_{sat}-T_w)}{h'_{fg}}\left[\frac{\rho(\rho-\rho_v)g}{3\mu}\right]^{1/3} a(1-e^2) \times \frac{[\sin\phi + Bo(\phi)]^{1/3}}{(1-e^2\sin^2\phi)^{3/2}}\,d\phi.$$

Integration of the above equation from $\phi = 0$ to $\phi = \pi$ gives the condensate production from one side as

$$\dot{m} = \frac{1}{3}\left[\frac{64k^3a^3(T_{sat}-T_w)^3\rho(\rho-\rho_v)g}{\mu(h'_{fg})^3}\right]^{1/4} \times \left\{(1-e^2)\int_0^{\phi_c}\frac{[\sin\phi + Bo(\phi)]^{1/3}}{(1-e^2\sin^2\phi)^{3/2}}\,d\phi\right\}^{3/4}. \quad (25)$$

Noting that the above relation gives only half of the condensate mass flow from the tube, one finds that an energy balance within the condensate film over an entire elliptical perimeter per unit tube length yields

$$2\dot{m}h'_{fg} = \bar{h}(\pi D_e)(T_{sat}-T_w). \quad (26)$$

Secondly, inserting equation (25) into (26), one may obtain the mean heat transfer coefficient as

$$\bar{h} = \left(\frac{128}{81\pi}\right)^{1/4}\frac{k}{D_c}[Ra/Ja]^{1/4}S_f(e) \quad (27)$$

where

$$S_f(e) = \left\{\int_0^{\phi_c}\frac{[\sin\phi + Bo(\phi)]^{1/3}}{(1-e^2\sin^2\phi)^{3/2}}\,d\phi \Big/ \int_0^{\pi}(1-e^2\sin^2\phi)^{-3/2}\,d\phi\right\}^{3/4}.$$

At last, the overall Nusselt number can then be presented as follows:

$$\overline{Nu} = \frac{\bar{h}D_e}{k} = \left(\frac{128}{81\pi}\right)^{1/4}[Ra/Ja]^{1/4}S_f(e). \quad (28)$$

In the limit case $e = 1$, it is noted that an elliptical tube becomes a vertical plate of both sides occuring condensation. Hence, its equivalent diameter becomes $D_e = 2(l/\pi)$. Here, l is the length of the vertical plate. Therefore, one should use l instead of D_e for Ra and $\overline{Nu}$ in equation (28) and may obtain the same form as Nusselt's solution, i.e.

$$\overline{Nu}_l = \frac{\bar{h}l}{k} = 0.943[Ra/Ja]^{1/4}. \quad (29)$$

3. RESULTS AND DISCUSSION

Equation (22) has been evaluated numerically for different values of ellipticity and reciprocality of Bo at any particular angular position ϕ, and its corresponding dimensionless streamwise length s.

$$s = 2x/\pi D_c = \int_0^{\phi}(1-e^2\sin^2\phi)^{-3/2}\,d\phi \Big/ \int_0^{\pi}(1-e^2\sin^2\phi)^{-3/2}\,d\phi. \quad (30)$$

The above results are shown in Figs. 2 and 3(a)–(c).

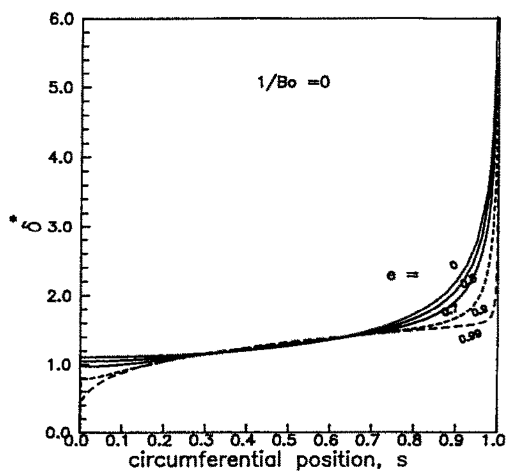

FIG. 2. Dependence of dimensionless local film thickness on ellipticity around periphery of ellipse.

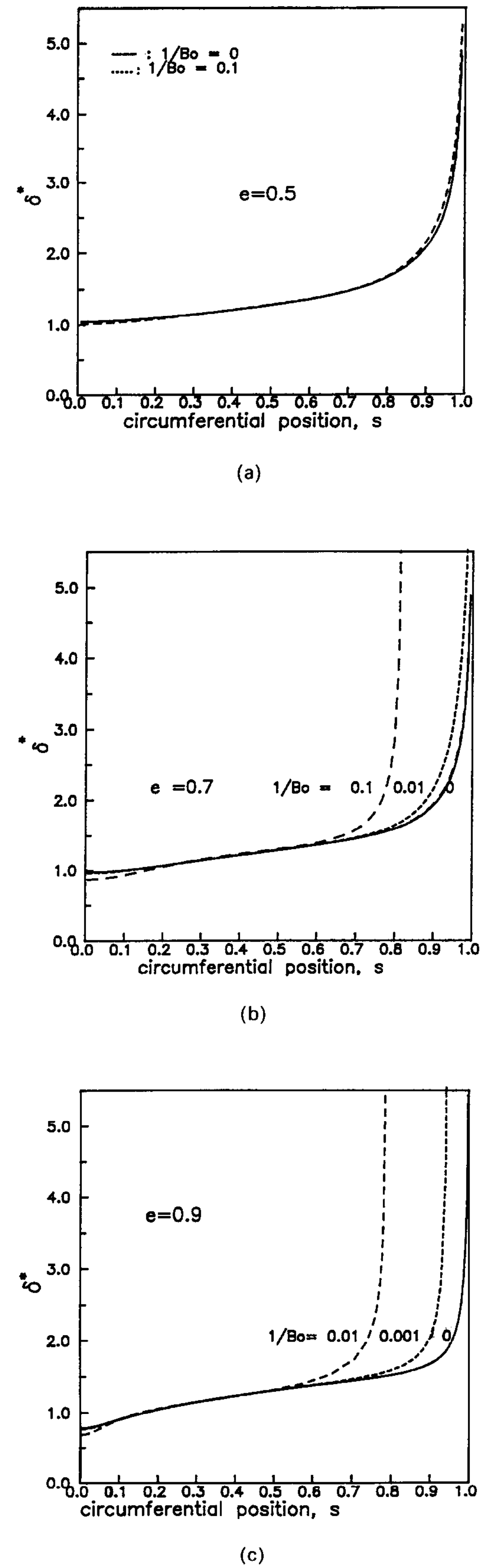

FIG. 3. Effect of surface tension on local film thickness around periphery of ellipse.

When $e = 0$ (circular tube), the dependence of dimensionless film thickness on dimensionless streamwise length (or angle) coincides with Nusselt's solution. It is noted that, in this case, the surface tension term $Bo(\phi)$ automatically vanishes. As e approaches 1, δ^* at the top ($s = 0$) becomes close to 0, which is the same as in the vertical plate. Figures 3(a)–(c) show that in the upper half of the tube, the positive effect of surface tension forces, owing to the decreasing surface curvature, pulls the condensate film down and, thus, makes the condensate film thinner. However, in the lower half of the tube, the increasing surface curvature makes the film thicker than the one without considering effect of surface tension, i.e. the negative pressure gradient ($-\partial P/\partial x < 0$) due to the reverse effect of surface tension $Bo(\phi) < 0$, tends to retard the condensate film flow down and thus accumulate the condensate mass. Once the condensate mass gravity forces outweigh its surface tension forces, the condensate will drip off the tube surface. Thus, for the cases $1/Bo \neq 0$, condensate drips off the tube at $\phi = \phi_c$ more ahead than at $\phi = \pi$ in the case $1/Bo = 0$.

Next, the dependence of local heat transfer coefficient on ellipticity and the reciprocal of Bo around the periphery of the ellipse is shown in Figs. 4(a)–(c) and 5. It may be seen that the local heat transfer coefficients increase with increasing ellipticity near both the top and bottom of the tube significantly for $1/Bo = 0$. Besides, the local Nusselt number also increases as $1/Bo$ increases in the upper half of the tube because there exists an additional effective action of surface tension forces $Bo(\phi) > 0$ due to the increasing radius of surface curvature. In the lower half of the tube, however, the pressure gradient term ($-\partial P/\partial x$) is negative since $Bo(\phi)$ is negative due to the decreasing radius of the surface curvature. This negative pressure gradient caused by the reversal surface tension effect will retard the condensate film flow down and accumulate the mass, and subsequently, the film will become thicker than the case $1/Bo = 0$. Consequently, Nu decreases with the increase of $1/Bo$.

According to numerical results from equation (28), the mean heat transfer coefficient is nearly unaffected by surface tension forces at small ellipticity e but is somewhat influenced at large e for the whole perimeter ($0 \leqslant \phi \leqslant \pi$). However, since the present model takes no account of contribution to the overall mean heat transfer coefficient after the separation point ($\phi \geqslant \phi_c$) due to δ approaching infinity, this causes a difference between the cases $1/Bo = 0$ and $1/Bo \neq 0$, especially for larger e. For example, for a horizontal elliptical tube with a vertical major axis and $e = 0.9$, its overall mean heat transfer coefficient decreases less than 16% as $1/Bo$ goes from 0 to 0.01. But, if integrated with respect to the same base, before the separation point $0 \leqslant \phi \leqslant \phi_c$ (or $0 \leqslant s \leqslant s_c$), the effect of surface tension force upon the mean heat transfer coefficient will normally be less than the no surface tension case by 2% or so. In other words, if further included, the contribution of the neglected condensate mass after

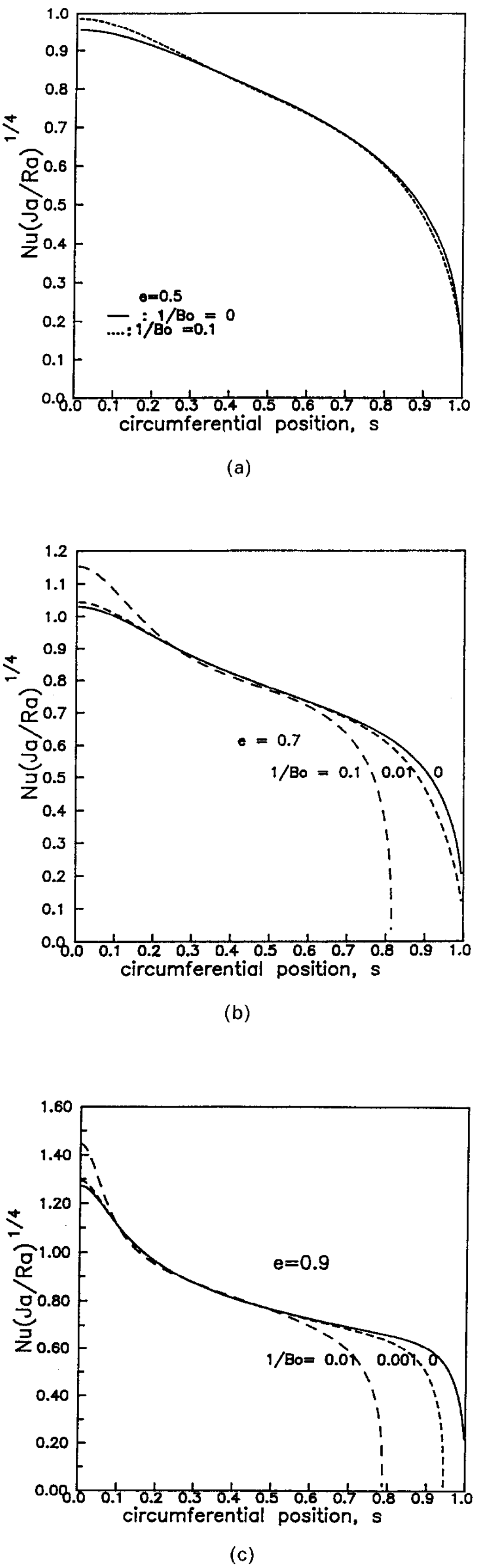

FIG. 4. Dependence of dimensionless local heat transfer on surface tension around periphery of ellipse.

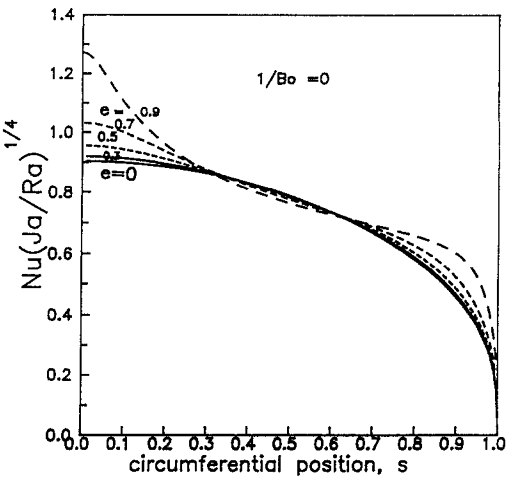

FIG. 5. Dependence of dimensionless local heat transfer on ellipticity around periphery of ellipse.

the separation point, the difference based on the whole perimeter between the cases $1/Bo = 0$ and $1/Bo \neq 0$ will locate in the range from about 2% to 16% in evaluating the overall mean heat transfer coefficient. In Fig. 6, the mean heat transfer coefficient for an ellipse with its major axis oriented in the direction of gravity is compared with that for an ellipse with its minor axis oriented in the direction of gravity. In the former case, $\overline{Nu}$ increases with increasing e very slowly at small e, and at much greater pace at large e. When $e = 0$, it is identical to Nusselt's solution for a circular tube.

As for the inclined flat plate, except the horizontal case $\varphi = \pi/2$, taking the limit $e = 1$ for the major axis inclined at a specified angle φ with the direction of gravity yields

$$\overline{Nu}_l = \frac{\bar{h}l}{k} = 0.943 \cos \varphi [Ra/Ja]^{1/4}.$$

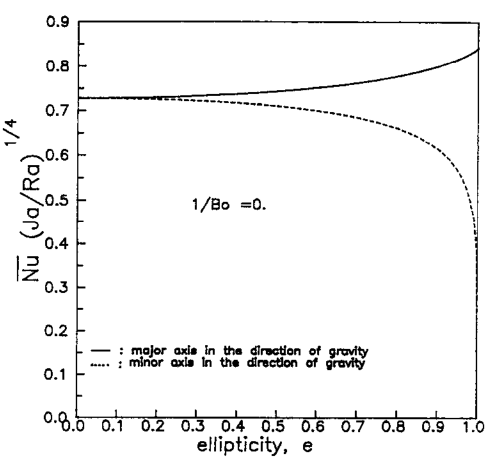

FIG. 6. Dependence of dimensionless mean heat transfer coefficient on ellipticity of ellipse.

For the excluded case of the horizontal flat plate, the condensate film flows by the aid of a pressure gradient due to variations of the film thickness. This effect of film thickness varying is the only action to make the film flow in this case (for more details, please see our paper ref. [10]), but it is neglected, based on the assumption $\delta \ll R$ in the present model. So the present analysis does not include the horizontal flat plate case.

4. CONCLUDING REMARKS

This is the first analytical approach to resolve the laminar film condensation outside a horizontal elliptical tube by introducing the role of ellipticity. It should be noted that the numerical results use Nusselt numbers based on an equal condensing area diameter rather than a hydraulic diameter. The result obtained only applies to the very slow or quiescent vapor condensing outside horizontal elliptical tubes and also very long inclined circular tubes (see ref. [6]), with negligible interfacial vapor shear drag. Because in the range near the angle $\phi = \phi_c$ which the film drops off the tube surface and its film surface is changing from 'convex' to 'concave', neglecting the film thickness compared with the radius of the elliptical surface curvature in calculating the effect of surface tension might cause error, the present theory cannot predict the surface tension correctly for $\phi_c < \phi < \pi$. Thus, for $\phi_c < \phi < \pi$, we do not use $F(\phi)$ to express $\delta(\phi)$ but simply consider $\delta(\phi)$ to be infinite because the film is dripping off the tube. In a similar case for a circular tube, Taghavi [11] confirmed the good significance concerning the effect of neglected film thickness on the mean heat transfer coefficient. Two major conclusions are warranted in the present study.

(1) The condensation heat transfer performance of the horizontal elliptical tube surface with vertical major axis is superior to those of the circular tube surface and horizontal tubes with inclined major axis.

(2) Unlike the Nusselt model, considering the gravity drain alone, the present analysis considers both the gravity and surface tension forces. The results indicate the surface tension has an influence on the local heat transfer rate and hydrodynamics characteristics, but the effect of surface tension force on the mean heat transfer coefficient is nearly insignificant especially for small ellipticity.

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