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Surface Tension Effects in Turbulent Film Boiling on a Horizontal Elliptical Tube

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The present theoretical study investigates turbulent film boiling on an isothermal elliptical tube under free convection. The effects of surface tension, eccentricity, and radiation are included in the analysis. Temperature, velocity, and heat transfer characteristics are discussed. The temperature and velocity of the vapor film under the eddy diffusivity present the nonlinear distribution. In addition, under higher eccentricity values, the eccentricity and the surface tension have only little influence on the mean Nusselt number. Furthermore, for film boiling with higher Rayleigh values, the elliptical tube can get better heat transfer efficiencies than can a circular tube. However, when Rayleigh values are low, the eccentricity of an elliptical tube seldom influences the heat transfer.

Nomenclature

a, b	=	semimajor and semiminor axes of the ellipse
Bo	=	Bond number, $(\rho_l - \rho_v)ga^2/\sigma_{Bo}$
C_p	=	specific heat capacity, $\mathrm{J/kg \cdot K}$
D_e	=	equivalent circular diameter of the elliptical tube, Eq. (9)
D_e^+	=	shear parameter, $D_e u^*/\nu_s$
e	=	eccentricity of the ellipse, $\sqrt{1-(b/a)^2}$
Gr_{D_e}	=	modified Grashof number
g	=	acceleration due to gravity, $\mathrm{m/s^2}$
h	=	heat transfer coefficient, $\mathrm{W/(m^2K)}$
h_{fg}	=	latent heat, J/kg
k	=	thermal conductivity, $\mathrm{W/m \cdot K}$
k^+	=	$k(T)/k(T_s)$, Eq. (20)
NR	=	radiation parameter, $\varepsilon\sigma T_s^3(D_e/2)/k_s$
Nu_{D_e}	=	local Nusselt number, $h(D_e/2)/k_s$
Nu_m	=	mean Nusselt number
Pr	=	Prandtl number, $C_p\mu/k$
Ra	=	modified Rayleigh number, $Gr_{D_e}Pr[0.5+1/S(T_r-1)]$
S	=	heat capacity parameter, C_pT_s/h_{fg}
T	=	temperature, K
T_r	=	temperature ratio, T_w/T_s
T^+	=	dimensionless temperature, $(T-T_s)/(T_w-T_s)$
u	=	vapor velocity in the x direction, m/s
u^*	=	shear velocity, $\sqrt{\tau_w/\rho}$
u^+	=	dimensionless velocity, u/u^*
$\dot{V}$	=	local acceleration, $\mathrm{m/s^2}$
v	=	velocity normal to the direction of flow, m/s
x	=	peripheral coordinate, m
y	=	coordinate measured distance normal to the elliptical surface, m
y^+	=	dimensionless distance, yu^*/ν_s
δ	=	vapor-film thickness, m
δ^+	=	dimensionless film thickness, $\delta u^*/\nu_s$
ε	=	emissivity
ε_h	=	eddy diffusivity for energy
ε_m	=	eddy diffusivity for momentum
θ	=	angle measured from the bottom of the tube
μ	=	absolute viscosity, kg/ms
μ^+	=	μ/μ_s, Eq. (18)
ν	=	kinematic viscosity, $\mathrm{m^2/s}$
ρ	=	density, $\mathrm{kg/m^3}$
σ	=	Stefan–Boltzmann constant, $\mathrm{W/m^2K^4}$
σ_{Bo}	=	surface-tension coefficient
τ	=	shear stress, $\mathrm{N/m^2}$
ϕ	=	angle between the tangent to the tube surface and normal to the direction of gravity

Subscripts

l	=	liquid
s	=	vapor at the saturation temperature
v	=	vapor
w	=	tube wall
x	=	x direction
δ	=	vapor–liquid interface

Introduction

FILM boiling plays an important role in heat transfer engineering. Some significant efforts have been directed toward research into related fields. Such research was conducted by the pioneering investigator Bromley [1], who assumed that the temperature distribution within the vapor is linear, neglected the inertia force of the vapor film, and obtained a certain degree of results in the case of film boiling. Pomerantz [2] investigated the influence of acceleration on saturated pool film boiling from the outside of a horizontal tube by experiment. In 1966, Nishikwa and Ito [3] analyzed the two-phase boundary-layer treatment of free-convection film boiling. The theoretical study has been made of film boiling from an isothermal vertical plate and a horizontal cylinder without considering radiative effects. Jordan [4] investigated the laminar film boiling and transition

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boiling, and the region which separates had also been discussed. Kalinin et al. [5] presented the pool film boiling, including the reports of laminar flow of vapor inside a film with a stable interface and unstable interface. Sakurai et al. [6] presented the pool film boiling on a horizontal cylinder with theoretical solutions. The analytical heat transfer model is based on laminar boundary theory, including radiation effects. Orozco et al. [7] then focused on the flow boiling of a sphere and a horizontal cylinder with a porous medium. The theoretical model relies on the Brinkman-extended flow models to describe the flow inside the vapor layer occupying the neighborhood near the heated surface. For the forced convection film boiling, Chang and Witte [8] investigated the nature of vapor-film crossflow over a cylindrical heater. The experimental data are obtained from R-11 boiling from a 6.35-mm-diam heater, revealing the details of wake formulation and behavior near the minimum heat-flux condition. Furthermore, for a pipe of the noncircular section (e.g., a horizontal elliptical tube), Yang and Hsu [9] presented the simple analysis of free-convection film boiling on an isothermal elliptical tube with surface tension. The solutions are accurate for elliptical tubes with intermediate diameters.

Laminar film boiling has been widely discussed in published literature, and there is also some development in the research of turbulent film boiling. For example, Sarma et al. [10] presented turbulent film boiling under a uniform heat-flux condition on a horizontal cylinder. In the research, the assumption of equal shear condition both at the wall and the vapor–liquid interface is reasonable. About the turbulent film boiling on a horizontal isothermal circular cylinder, Sarma et al. [11] presented some theoretical results. The analysis compared the theoretical results with previous experimental results and found that their results were in a good agreement with the experimental data.

Even though there is some research in the field of laminar and turbulent film boiling, there are relatively few recent investigations of the issues relating to turbulent film boiling on horizontal elliptical tubes. Therefore, the aim of this present study is to investigate turbulent film boiling on an isothermal elliptical tube in terms of the local vapor-film thickness and heat transfer characteristics and to compare the present results with the results obtained from previously published theoretical and experimental models.

Description of the Physical Model

Consider an elliptical tube immersed in a quiescent pure liquid at the saturation temperature T_s. The isothermal wall temperature T_w is assumed to be high enough for turbulent film boiling to occur on the surface of the elliptical tube, and then a continuous film of vapor runs upward over the tube. The physical model and the coordinate system adopted in the present study are shown in Fig. 1, in which the coordinates use a two-dimensional orthogonal curvilinear coordinate system. The governing equations for the turbulent vapor film are described as follows:

Continuity equation:

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

Momentum equation:

$$\rho_v \left\{ u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} \right\} = g(\rho_l - \rho_v)[\sin\phi + Bo(\phi)] + \frac{\partial}{\partial y}\left\{ \mu\left(1 + \frac{\varepsilon_m}{\nu}\right)\frac{\partial u}{\partial y} \right\} \tag{2}$$

where $Bo(\phi)$, the surface-tension effect, is as follows [9]:

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

Boundary equations:

$$\mu\left(1 + \frac{\varepsilon_m}{\nu}\right)\frac{\partial u}{\partial y} = \tau_w \qquad u = v = 0 \quad \text{at } y = 0;$$
$$\mu\left(1 + \frac{\varepsilon_m}{\nu}\right)\frac{\partial u}{\partial y} = \tau_\delta \quad \text{at } y = \delta \quad \text{for } 0 \le \phi \le \pi \tag{4}$$

Energy equation:

$$\rho_v C_p \left\{ u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} \right\} = \frac{\partial}{\partial y}\left\{ k\left(1 + \frac{\varepsilon_h}{\alpha}Pr\right)\frac{\partial T}{\partial y} \right\} \tag{5}$$

Boundary equations:

$$T = T_w \quad \text{at } y = 0; \qquad T = T_s \quad \text{at } y = \delta \quad \text{for } 0 \le \phi \le \pi \tag{6}$$

Thermal energy balance equation:

$$\frac{\mathrm{d}}{\mathrm{d}x}\int_0^\delta \rho_v u\,\mathrm{d}y = -\frac{k_w}{h_{fg}}\left.\frac{\mathrm{d}T}{\mathrm{d}y}\right|_{y=0} + \frac{\varepsilon\sigma(T_w^4 - T_s^4)}{h_{fg}} \tag{7}$$

According to Yang and Hsu [9], the differential arc length for ellipse can be expressed with the following equation:

$$\mathrm{d}x = I(\phi)\frac{D_e}{2}\,\mathrm{d}\phi \tag{8}$$

where

$$D_e = \frac{2a}{\pi}(1 - e^2)\int_0^\pi (1 - e^2\sin^2\phi)^{-3/2}\,\mathrm{d}\phi \tag{9}$$

$$I(\phi) = \pi(1 - e^2\sin^2\phi)^{-3/2} \Big/ \int_0^\pi (1 - e^2\sin^2\phi)^{-3/2}\,\mathrm{d}\phi \tag{10}$$

where D_e is an equivalent diameter based on the equal outside surface area, which is compared with the circular tubes.

Substitute dx into the thermal energy balance equation, and Eq. (7) can be modified as follows:

$$\frac{\mathrm{d}}{I(\phi)\,\mathrm{d}\phi}\int_0^\delta \rho_v u\,\mathrm{d}y = -\frac{k_w}{h_{fg}}\frac{D_e}{2}\left.\frac{\mathrm{d}T}{\mathrm{d}y}\right|_{y=0} + \frac{\varepsilon\sigma D_e(T_w^4 - T_s^4)}{2h_{fg}} \tag{11}$$

Boiling under the free-convection turbulence vapor, the viscosity component and the buoyancy effect are assumed to be more significant than the inertia force. Momentum equation (2) can be expressed as

$$\tau_w - \tau_\delta = g\delta(\rho_l - \rho_v)[\sin\phi + Bo(\phi)] \tag{12}$$

Because $\mu_v \ll \mu_l$, it is tentatively assumed that the wall shear is the same order to the interfacial shear [10]. In addition, the velocity gradients at the liquid–vapor interface and tube wall have opposite signs, leading to $\tau_w = -\tau_\delta$. The force balance equation (12) yields the following equation:

$$\tau_w = 0.5g\delta(\rho_l - \rho_v)[\sin\phi + Bo(\phi)] \tag{13}$$

The following dimensionless parameters and equations are defined:

$$u^* = \sqrt{\tau_w/\rho} \qquad u^+ = \frac{u}{u^*} \qquad T^+ = \frac{T - T_s}{T_w - T_s}$$
$$y^+ = \frac{yu^*}{\nu_s} \qquad D_e^+ = \frac{D_e u^*}{\nu_s} \qquad \delta^+ = \frac{\delta u^*}{\nu_s} \tag{14}$$

The force balance equation (13) yields the following dimensionless equation:

$$D_e^{+^3} = 0.5Gr_{D_e}\delta^+[\sin\phi + Bo(\phi)] \tag{15}$$

where Gr_{D_e} is the modified Grashof number:

$$\frac{g(D_e/2)^3}{\nu_s^2} \quad \frac{\rho_l - \rho_v}{\rho_v}$$

It is assumed that the turbulent conduction term across the vapor layer is more significant than the convective term, and hence the convective term can be neglected. Energy equation (5) yields the following dimensionless energy equation:

$$\frac{\mathrm{d}}{\mathrm{d}y^+}\left[\left(1 + \frac{\varepsilon_m}{\nu_s}\frac{Pr}{\mu^+(T^+)(1+T^+(T_r-1))}\right)\frac{\mathrm{d}T^+}{\mathrm{d}y^+}\right] = 0 \tag{16}$$

The dimensionless boundary conditions of Eq. (16) are

$$\mathrm{T}^+ = 1 \quad \text{at } y^+ = 0; \qquad T^+ = 0 \quad \text{at } y^+ = \delta^+ \tag{17}$$

where the absolute viscosity equation μ^+ in dimensionless energy equation (16) is expressed as the nitrogen at the saturation temperature corresponding to an operation pressure of 1 atm [10]; that is,

$$\mu^+ = \frac{\mu}{\mu_s} = 2.656 - 3.804\left(\frac{T}{T_s}\right) + 2.148\left(\frac{T}{T_s}\right)^2 \tag{18}$$

Furthermore, the thermal energy balance equation (11) can be rewritten in dimensionless form as follows:

$$\frac{\mathrm{d}}{I(\phi)\,\mathrm{d}\phi}\int_0^{\delta^+}\frac{u^+}{1+T^+(T_r-1)}\,\mathrm{d}y^+ = \frac{S}{Pr}\left\{-T_r^{2/3}\frac{D_e^+}{2}(T_r-1)k^+\frac{\partial T^+}{\partial y^+}\bigg|_{y^+=0} + T_r NR(T_r^4 - 1)\right\} \tag{19}$$

where the absolute conductivity equation k^+ in thermal energy balance equation (19) is expressed as the nitrogen at the saturation temperature corresponding to a system pressure of 1 atm [10]:

$$k^+ = \frac{k}{k_s} = 236.1 - 811.5\left(\frac{T}{T_s}\right) + 1045.4\left(\frac{T}{T_s}\right)^2 - 596.2\left(\frac{T}{T_s}\right)^3 + 127.2\left(\frac{T}{T_s}\right)^4 \tag{20}$$

The local vapor-film shear stress can be expressed as

$$\tau = \mu\left(1 + \frac{\varepsilon_m}{\nu}\right)\frac{\partial u}{\partial y} \tag{21}$$

Further, the dimensionless thermal energy balance equation (19) requires the velocity profile u^+ in the boiling film. The integration terms of dimensionless velocity can be attained by modifying the local vapor-film shear stress equation (21):

$$\frac{\mathrm{d}u^+}{\mathrm{d}y^+} = \frac{\tau}{\tau_w}\Bigg/\left((T/T_w)T_r\mu(T^+) \times \left[1 + \frac{\varepsilon_m}{\nu_s}\frac{1}{\mu^+(T^+)(1+T^+(T_r-1))}\right]\right) \tag{22}$$

The shear stress profile of Eq. (22) is expressed as

$$\frac{\tau}{\tau_w} = 1 - 2\frac{y^+}{\delta^+} \tag{23}$$

The boundary condition is

$$u^+ = 0 \qquad \tau/\tau_w = 1 \quad \text{at } y^+ = 0 \tag{24}$$

The eddy diffusivity distribution presented by Kato et al. [12] is expressed as

$$\frac{\varepsilon_m}{\nu_s} = 0.4y^+[1 - \exp(-0.0017y^{+2})] \tag{25}$$

Considering the boiling heat transfer, the heat transfer coefficient is given by

$$h(T_w - T_s) = -k_w\frac{\partial T}{\partial y}\bigg|_{y=0} + \sigma\varepsilon(T_w^4 - T_s^4) \tag{26}$$

Obviously, the local Nusselt number can be expressed as

$$Nu_{D_e} = -k_w^+ D_e^+\frac{\mathrm{d}T^+}{\mathrm{d}y^+}\bigg|_{y^+=0} + 2NR\left(\frac{T_r^4 - 1}{T_r - 1}\right) \tag{27}$$

The mean Nusselt number for the entire surface of the tube can be written as

$$Nu_m = \frac{1}{\pi}\int_0^\pi Nu_{D_e}\,\mathrm{d}\phi \tag{28}$$

According to the governing equations, the mean Nusselt number can obtained as a function of the following parameters:

$$Nu_m = f(Gr, T_r, NR, S, e, 1/Bo) \tag{29}$$

Numerical Method

The dimensionless governing equations (15–20) and (22), subject to the relevant boundary conditions given, can be used to estimate δ^+, D_e^+, and the Nusselt number [Eq. (27)] for the film vapor by means of the following procedures:

1) Suitable dimensionless parameters such as T_r, S, NR, e, $1/Bo$, and Gr_{D_e} are specified.

2) The boundary conditions of velocity and temperature are as follows:

$$\begin{aligned} &T^+ = 1 \qquad u^+ = 0 \qquad \mu^+ = 2.656 - 3.804T_r + 2.148T_r^2 \\ &k^+ = 236.1 - 811.5T_r + 1045.4T_r^2 - 596.2T_r^3 + 127.2T_r^4 \\ &\quad \text{at } y^+ = 0; \\ &T^+ = 0 \quad \text{at } y^+ = \delta^+ \end{aligned} \tag{30}$$

3) At the bottom of the tube, $\phi = 0$, $i = 0$, the dimensionless film thickness δ^+ is zero. At the next node (i.e., $i = i + 1$), the value of ϕ is given by $\phi_{i+1} = \phi_i + \Delta\phi$, where $\Delta\phi = (\pi/360)$.

4) Guess an initial value of δ^+; substitute Eqs. (18) and (25) into Eqs. (16) and (17) and then get the value of $(\mathrm{d}T^+/\mathrm{d}y^+)|_{y^+=0}$.

5) Substitute $(\mathrm{d}T^+/\mathrm{d}y^+)|_{y^+=0}$ and Eqs. (20) and (22–24) into Eq. (19) to get the value of D_e^+. Then substitute D_e^+ into Eq. (15).

6) The criterion for the accuracy of δ^+ is assessed by Eq. (15), and it can be expressed as the following unequal equation:

$$D_e^{+^3} - 0.5Gr_{D_e}\delta^+[\sin\phi + Bo(\phi)] \leq 10^{-6}$$

If the calculation is a convergence, process the film thickness of the next angular position. If the calculation is not a convergence, guess a new thickness and repeat processes 4–6.

7) The preceding process is repeated at the next node position (i.e., $\phi_{i+1} = \phi_i + \Delta\phi$) and then subsequently at all nodes within the range $0 \leq \phi \leq \pi$.

8) The local Nusselt number [Eq. (27)] and mean Nusselt number [Eq. (28)] are then calculated.

9) The flowchart for calculating the vapor thickness is shown in Fig. 2.

Results and Discussion

Figure 3 presents the distribution of dimensionless velocity for entire angular positions, 0 deg $\leq \phi \leq$ 180 deg. The figure shows that the velocity is zero on the wall with no slip condition. Under the effects of the turbulent model, along the film thickness, the temperature distribution shows the nonlinear profile. And the velocity increases to the maximum value at about $y/\delta = 0.5$. Beyond the maximum value, the velocity gradually decreases to the vapor–liquid interface. Furthermore, the dimensionless velocity increases as the angular positions increase. The physical reason for the former

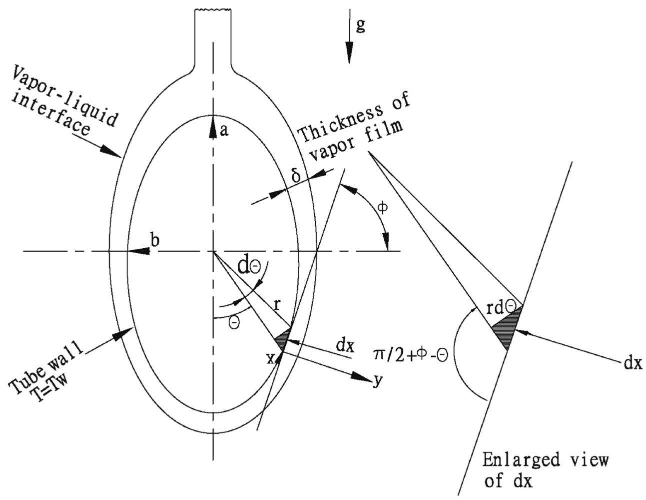

Fig. 1 Physical model and coordinate system.

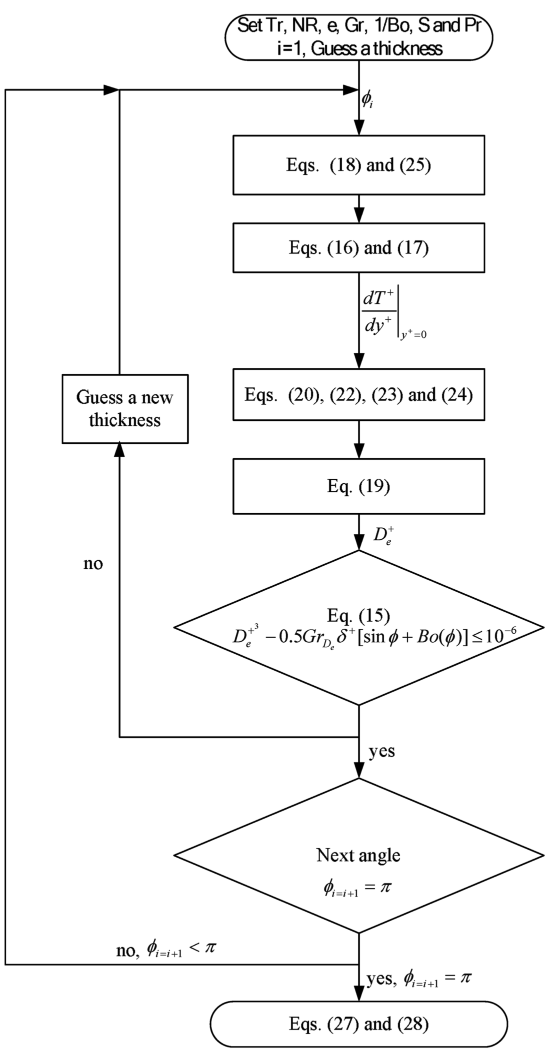

Fig. 2 Flowchart for calculating the vapor thickness and Nusselt number.

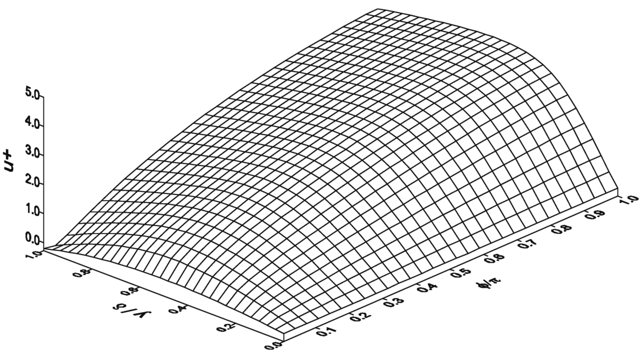

Fig. 3 Local velocity distribution in the vapor film ($Pr = 1$, $Gr = 10^9$, $S = 0.46$, $T_r = 1.6$, $NR = 1.4$, $e = 0.5$, and $1/Bo = 0.05$).

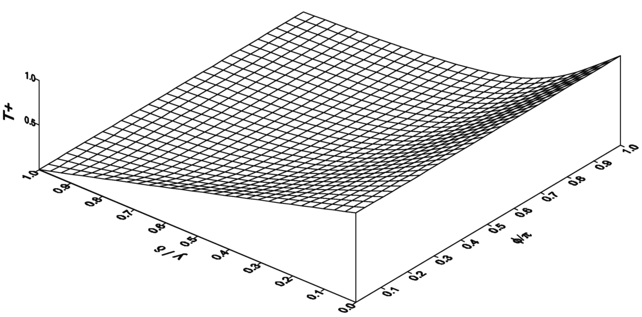

Fig. 4 Local temperature distribution in the vapor film ($Pr = 1$, $Gr = 10^9$, $S = 0.46$, $T_r = 1.6$, $NR = 1.4$, $e = 0.5$, and $1/Bo = 0.05$).

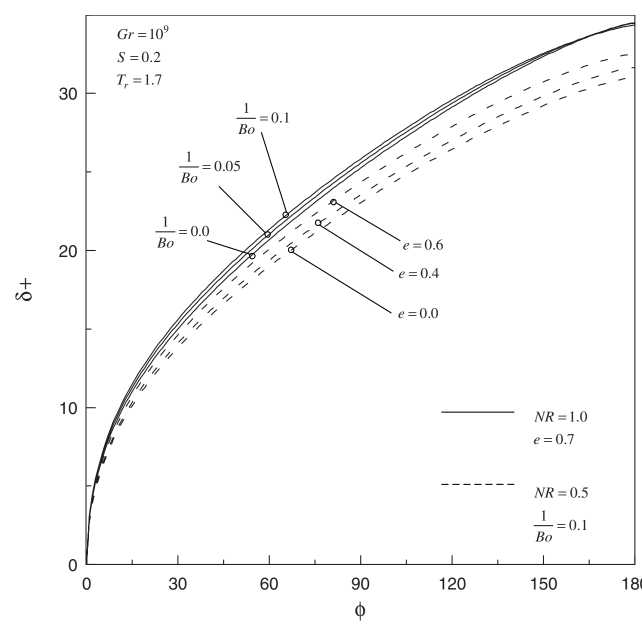

Fig. 5 Dimensionless film thickness on the elliptical tube.

phenomenon is film boiling under free convection: the wall and interface have large shear stress, and it will lead to a lower velocity.

Figure 4 presents the distribution of the temperature profile for all angular positions. In the figure, under the effects of eddy diffusivity ($\varepsilon_m/\nu_s = 0$ at $y^+ = 0$), the bottom of the elliptical tube shows the linear temperature distribution. Then the nonlinear distribution of the profile becomes increasingly significant when the effects of eddy diffusivity become strong.

Figure 5 presents the variation of the dimensionless vapor-film thickness on an elliptical tube. Because the shear velocity u^* is zero at the bottom of the elliptical tube (i.e., $\phi = 0$), the value of the dimensionless film thickness δ^+ is also zero. Specifically, the film thickness increases continuously from a minimum value at the bottom of the elliptical tube ($\phi = 0$) as the value of ϕ increases. It can be seen that the film thickness reaches its maximum value on the top of the tube at the upper stagnation point ($\phi = \pi$). Furthermore, it is noted that an increase in the values of the eccentricity e and surface-tension effects $1/Bo$ leads to an increase of the vapor-film thickness.

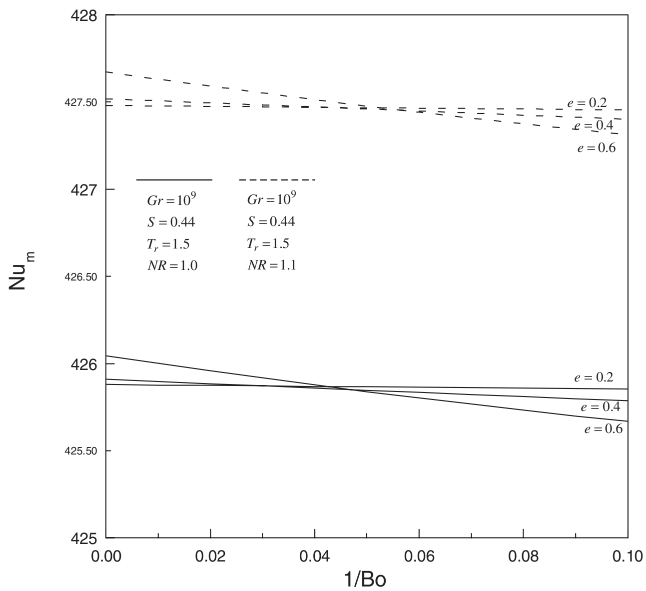

Fig. 6 Effects of surface tension on the mean Nusselt number.

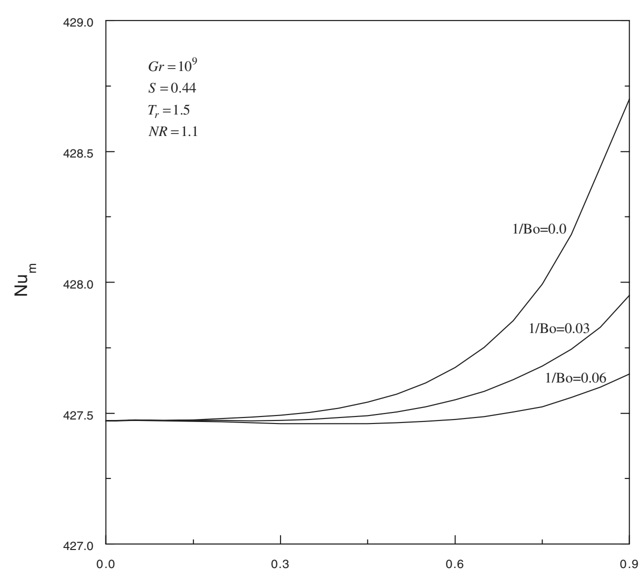

Fig. 7 Effects of eccentricity on the mean Nusselt number.

Under larger values of eccentricity, the vapor velocity will decrease. Thus, a larger film thickness is need for the thermal balance equation.

Figure 6 presents the relationship between the mean Nusselt number and the surface-tension effects for eccentricity $e = 0.2$, 0.4, and 0.6. The figure presents the results of the free-convection film boiling. According to Eq. (3), when eccentricity values are low ($e < 0.3$), the $Bo(\phi)$ values will be small and the surface-tension effects seldom influence the heat transfer. In addition, when the values of eccentricity are larger, the $Bo(\phi)$ values will be larger and the mean Nusselt will be influenced by the surface tension. And when surface-tension effects increase, the mean Nusselt will slightly decrease.

Figure 7 presents the relationship between the mean Nusselt number and the eccentricity for surface-tension effects $1/Bo = 0.0$, 0.03, and 0.06. When the eccentricity values ($e > 0.3$) are larger, the $Bo(\phi)$ will show larger values. And the surface tension has some slight influence on the mean Nusselt number. However, when eccentricity values ($e \leq 0.3$) are smaller, the $Bo(\phi)$ will also be smaller. And the influence of eccentricity and surface-tension effects on the mean Nusselt can almost be neglected.

Figure 8 provides a comparison between the present results for the mean Nusselt number for a horizontal tube and the results generated in a previous study [10] for the following equation:

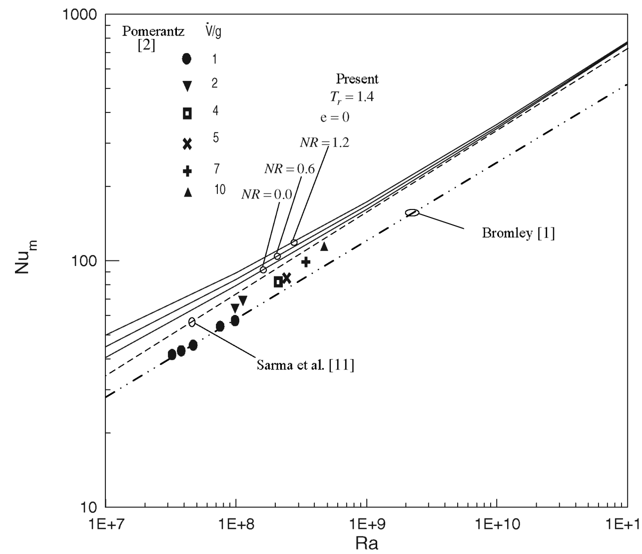

Fig. 8 Comparison of the present results with previous theory data: effects of Rayleigh number on mean Nusselt number.

$$Nu_m = 0.272Ra^{1/3}(T_r - 1)^{1.25} \tag{31}$$

where Ra is the modified Rayleigh number, $Ra = Gr_{D_e}Pr[0.5 + 1/S(T_r - 1)]$. Observation of Fig. 8 shows that the mean Nusselt number calculated in the present study is broadly similar to the results generated by Sarma et al. [10]. In addition, the increase of the Rayleigh values lead to an increase of the mean Nusselt number.

Furthermore, the figure also compares the present results with the theoretical data [1] and experimental results [2] presented in earlier studies. It can be seen that there is a similar trend between the two sets of results at both low and high Rayleigh numbers for all regions. However, because the present paper considered the effects of the eddy diffusivity and the radiation, the present results have higher Nusselt numbers than the former results. In addition, the increase of the Rayleigh values will lead to an increase of the mean Nusselt number.

Conclusions

The following conclusions can be drawn from the results of the present theoretical study:

1) It is noted that an increase in the values of the eccentricity e, radiation effects NR, or temperature ratio Tr leads to an increase of the heat transfer. And the increase of the heat transfer brings out an increase of the vapor-film thickness.

2) A higher temperature ratio increases the mean Nusselt number. In addition, the radiation is also one of the dominant factors, and therefore the increase in the radiation parameter also leads to an increase in the mean Nusselt number.

3) For the higher eccentricity values ($e > 0.3$), the eccentricity and the surface tension have a slight influence on the mean Nusselt number. However, for the lower eccentricity values ($e \leq 0.3$), the influence of eccentricity and surface-tension effects on the mean Nusselt number can almost be neglected.

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