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Analytical and innovative solutions for heat transfer problems involving phase change and interfaces

Entropy generation of forced convection film condensation on a horizontal elliptical tube

Javad Abolfazli Esfahani *, Mahnaz Modirkhazeni

Department of Mechanical Engineering, Ferdowsi University of Mashhad, Mashhad, Iran

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ABSTRACT

The aim of this study is to analyze the entropy generation number in forced convection film condensation on a horizontal isothermal elliptical tube. A numerical approach has been used to investigate how parameters, including ellipticity of ellipse, Reynolds and Brinkman numbers affect the irreversibility. The results show that the minimization of total entropy generation number (N_S) can be achieved by reducing the values of Reynolds and Brinkman numbers. Also, it was concluded that the effect of ellipticity on total entropy generation number is significant for $e > 0.7$ and increasing ellipticity, increases the amount of N_S.

1. Introduction

Due to the practical importance of condensation in the design of condensers, air conditioning systems and many industrial process equipments, film condensation heat transfer of pure vapor flowing onto a body such as a plate, cylinder and sphere has been studied by many researchers, like Winkler et al. [1] and Yang and Hsu [2]. The basis of the laminar film condensation on a flat plate was first presented by Nusselt [3]. After him this field was considered by many researchers and they have used the assumption of the simple Nusselt theory for their works. Esfahani and Ziaei-Rad [4] worked on laminar film condensation phenomenon on a vertical flat plate. A heat transfer coefficient of a condensate film for a wide range of physical properties has been obtained and discussed by them. Analytical calculation on the problem of film condensation on a vertical flat plate by taking into account the shear forces at the liquid–vapor interface has been performed by Esfahani and Koohi-Fayegh [5]. Moreover Esfahani and Koohi-Fayegh [6] have done an analytical study considering the problem of film condensation of assisted flowing vapor on an isothermal vertical flat plate. Furthermore, Shekriladze and Gomelauri [7] considered forced convection film condensation from a vapor flowing downward to a horizontal circular tube, and obtained numerical solutions by utilizing the asymptotic shear stress at the interface. Rose [8], Memory et al. [9,10] and Michael et al. [11] investigated the forced convection laminar condensation on a horizontal tube in experimental data and theoretical analysis. In addition, several researchers such as Yang and Hsu [12] and Yang and Chen [13], Ali and McDonald [14], Karimi [15] and Memory and Rose [16] worked on the problem of the condensation heat transfer on cylinders of elliptical profiles with major axes aligned with gravity.

Conservation of useful energy depends on the design of efficient thermodynamic heat transfer processes, i.e. minimization of entropy generation due to heat transfer and flow friction. Entropy generation in thermal engineering systems destroys system available energy and reduces its efficiency. Thus, entropy generation minimization is of great concern in heat transfer problems associated with film condensation. Decreasing entropy generation rate needs a better understanding of how

* Corresponding author.
E-mail addresses: abolfazl@um.ac.ir (J. Abolfazli Esfahani), smm.modir@gmail.com (M. Modirkhazeni).

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Nomenclature

a	Semi-major axis of ellipse
b	Semi-minor axis of ellipse
Br	Brinkman number, $\mu\frac{u_0^2}{k\Delta T}$
c_p	Specific heat of condensate at constant pressure
D_e	Equivalent diameter of ellipsoid
e	Ellipticity of ellipse, $\sqrt{1-(b/a)^2}$
F	Dimensionless parameter, $\frac{Ra/Ja}{Re^2}$
g	Acceleration due to gravity
h	Condensing heat transfer coefficient angle ϕ
h'_{fg}	Latent heat of condensation corrected for condensate sub-cooling
Ja	Jakob number, $C_p\Delta T/h_{fg}$
k	Thermal conductivity of condensate
m''	Condensate mass flux (per unit area)
N_S	Total entropy generation number
Nu	Local Nusselt number, hD/k
p	Static pressure of condensate
P	Dimensionless pressure gradient parameter, $(\rho_v/\rho)Pr/Ja$
Pr	Prandtl number
Ra	Rayleigh number, $\rho(\rho-\rho_v)gPrD_e^3/\mu^2$
Re	Reynolds number, $\rho u_\infty D_e/\mu$
S''_0	Characteristic entropy generation rate
T_{sat}	Saturation temperature of vapor
T_w	Wall temperature
u_∞	Vapor velocity of free stream
u_e	Tangential vapor velocity at the edge of boundary layer
u	Velocity component in x direction
v	Velocity component in y direction

Greek symbols

δ	Thickness of condensate film
δ^*	Dimensionless thickness of condensate film
θ	Angle measured from top of the tube
μ	Absolute viscosity of condensate
ρ	Density of condensate
ρ_v	Density of vapor
ϕ	Angle between the tangent to tube surface and the normal to direction of gravity
ψ	Irreversibility distribution ratio
Θ	Dimensionless temperature difference, $\Delta T/T_w$
τ_b	Interfacial vapor shear stress

Subscripts

0	Reference value
sat	Saturation
v	Vapor
w	Tube wall

Superscripts

$*$	Indicates dimensionless

and where the entropy is generated. Bejan [17] presented the basis for the entropy generation minimization method. In his book [18], Bejan also conducted the second law analysis of thermodynamics via the minimization of entropy generation for the single-phase convection heat transfer. Bejan [19] devised concrete methods for minimizing entropy generation in engineering equipment for heat transfer. More recently, Saouli and Aiboud-Saouli [20] performed second law analysis of laminar liquid falling film along an inclined heated plate and found that fluid friction irreversibility dominates over heat transfer irreversibility. Dung and Yang [21] investigated the entropy generation of free convection film condensation on a horizontal tube and they found that minimization of entropy generation number can be achieved by reducing the values of Brinkman parameters without losing condensation heat transfer coefficient. Entropy generation minimization of free convection film condensation on an elliptical cylinder was done by Li and Yang [22]. The entropy generation number was found to be a function of the Rayleigh, Brinkman numbers and geometrical parameters (ellipticity). Moreover, Yang et al. [23] have studied thermodynamic optimization of free convection film condensation on a horizontal elliptical tube with variable wall temperature. Their results indicated that the optimal entropy generation number is proportional to amplitude of non-isothermal wall temperature variation. Li and Yang [24] investigated thermodynamic analysis of free convection film condensation on an elliptical cylinder. Their results expressed that the effect of surface tension forces on the entropy generation number is significant for the larger values of ellipticities (around $e > 0.7$). Chen et al. [25] have considered entropy generation of forced convection film condensation from downward flowing vapors onto a horizontal circular tube. They deduced that the minimization of entropy generation rate can be achieved by reducing the values of Brinkman parameters.

As was mentioned before the problem of entropy generation of free convection film condensation on an elliptical tube is considered by many researchers. Also, analysis of entropy generation of forced convection film condensation for circular tubes was done before but no attempt has been performed to study the entropy generation of forced convection film condensation on the outer surface of horizontal elliptical tube. So the main objective of this study is to analyze the mechanism of entropy generation on an elliptical tube encountered in forced convection film condensation heat transfer. To achieve complete thermodynamic analysis, including first and second law, an expression has been derived for entropy generation number, which considers both heat transfer and flow friction irreversibility. In order to investigate the effect of geometry on the process, parameter ellipticity has been introduced. As stated in earlier works such as Yang and Chen [26,27], ellipticity specifies different elliptical cylinders from $e = 0$, circular cylinder to $e = 1.0$, vertical plate.

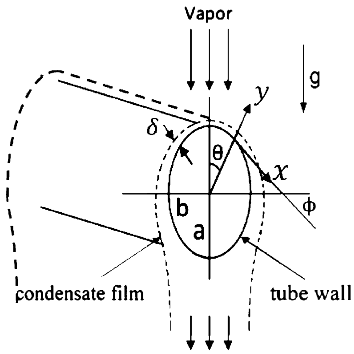

Fig. 1. Physical model and coordinate system for condensate film flow on an elliptical tube.

2. Thermal analysis

Physical model and coordinate system for condensate film flow on a horizontal elliptical tube with major axis “2a” in the gravitational direction and minor axis “2b” are shown in Fig. 1. Outer surface of the tube is subjected to a downward flowing pure vapor that is at its saturated temperature T_{sat} and moves at uniform velocity u_∞. The uniform wall temperature T_w is below the saturation temperature. Thus, vapor condenses on the surface of the tube and forming a thin film of liquid that flows down the tube surface under the influence of gravity. The thickness of the film, δ, and the local mass flow rate increase with distance down the tube as condensate forms continuously along the entire film/vapor interface.

According to the proposed physical model, the boundary layer equations governed by the basis of conservation of mass, momentum and energy for a laminar, steady state condensate film with constant fluid properties are respectively as:

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

$$\mu \frac{\partial^2 u}{\partial y^2} = \frac{\mathrm{d}P}{\mathrm{d}x} - (\rho - \rho_\upsilon) g (\sin\phi) \tag{2}$$

$$k \frac{\partial^2 T}{\partial y^2} = 0 \tag{3}$$

where $\phi = \phi(x)$ is the angle between the tangent to the tube wall and the horizontal direction at position (r, θ). The related boundary conditions are:

$$y = \delta; \qquad \tau_\delta = \mu \frac{\partial u}{\partial y}; \qquad T = T_{sat} \tag{4}$$

$$y = 0; \qquad u = 0; \qquad T = T_w \tag{5}$$

In writing the above equations, inertia effect, viscous dissipation, and axial conduction were neglected. Furthermore, it was assumed that the condensate film thickness is much smaller than the curvature diameter.

Additionally, an energy balance can be written for an element of the condensate film of height δ and width dx. This balance is between the latent heat released at the interface and heat flux conducted to the tube wall.

$$h'_{fg} m'' = k \frac{\partial T}{\partial y} = k \frac{\Delta T}{\delta} \tag{6}$$

where $h'_{fg} = h_{fg}(1 + 0.68 C_p \frac{\Delta T}{h_{fg}})$ is the latent heat of condensation that considers the effect of condensate sub-cooling which was pioneered by Rohsenow [28]. The local mass flow rate can be obtained by inserting the velocity profile into the following equation as:

$$m'' = \rho \frac{\mathrm{d}}{\mathrm{d}x} \int_0^\delta u \, \mathrm{d}y \tag{7}$$

By assuming that film thickness is very thin compared with the radius of surface curvature, the pressure gradient which has been appeared in momentum equation can be approximately expressed by Bernoulli’s equation:

$$\frac{\mathrm{d}P}{\mathrm{d}x} = -\rho_v u_e \frac{\mathrm{d}u_e}{\mathrm{d}x} \tag{8}$$

Next applying potential flow theory to a flow with uniform velocity u_∞, the vapor velocity at the edge of the boundary layer can be derived as [1]:

$$u_e = u_\infty\left(1+\sqrt{(1-e^2)}\right)\sin(\phi) \tag{9}$$

where $e = \frac{\sqrt{a^2-b^2}}{a}$ and is the ellipticity of ellipse. It should be noted that ellipticity is different from eccentricity which is defined as b/a. The Chain derivative should be used to compute velocity gradient. By inserting u_e into pressure gradient and then substituting the pressure gradient into Eq. (2), the final momentum equation can be gained as follows:

$$\mu\frac{\partial^2 u}{\partial y^2} = -(\rho-\rho_\upsilon)g(\sin\phi) + \left(-\frac{\rho_\upsilon u_\infty^2 \sin(2\phi)(1+\sqrt{(1-e^2)})^2\sqrt{(1-e^2\sin^2(\phi))^3}}{2a(1-e^2)}\right) \tag{10}$$

The vapor shear which has been used in boundary condition should be modeled. A good approximation for high condensate rates is given by the Shekriladze and Gomelauri model [7] and it is used in the present study. This method has more usefulness and simplicity than a numerical approach, as mentioned in Jacobi [29].

$$\tau_\delta = m''u_e = m''u_\infty\left(1+\sqrt{(1-e^2)}\right)\sin(\phi) \tag{11}$$

By integrating Eqs. (3) and (10) with respect to the boundary conditions the following velocity and temperature profiles will be achieved as:

$$\begin{aligned} u ={}& \frac{(\rho-\rho_\upsilon)g}{\mu}(\sin\phi)\delta^2\left(\frac{y}{\delta}-\frac{1}{2}\left(\frac{y}{\delta}\right)^2\right) + \frac{1}{2\mu}\frac{\rho_\upsilon u_\infty^2(1+\sqrt{(1-e^2)})^2\sqrt{(1-e^2\sin^2(\phi))^3}}{a(1-e^2)} \\ &\times \sin(2\phi)\delta^2\left(\frac{y}{\delta}-\frac{1}{2}\left(\frac{y}{\delta}\right)^2\right) + \frac{m''u_\infty(1+\sqrt{(1-e^2)})}{\mu}\sin(\phi)\times y \end{aligned} \tag{12}$$

$$T(y) = (T_{sat}-T_W)\frac{y}{\delta}+T_W = \Delta T\frac{y}{\delta}+T_W \tag{13}$$

From Eq. (6), m'' can be derived as:

$$m'' = \frac{k}{h'_{fg}}\frac{\Delta T}{\delta} \tag{14}$$

Eq. (14) is identical to Eq. (7), so m'' will be eliminated. By substituting velocity profile into Eq. (7) and integrating the updating equation, also introducing the dimensionless parameter, the following relation will be resulted:

$$\begin{aligned} &2\delta^*\frac{d}{d\phi}\left[\frac{F}{3}\delta^{*3}(\sin\phi)+\frac{P}{6}\delta^{*3}D_e\frac{(1+\sqrt{(1-e^2)})^2\sin(2\phi)\sqrt{(1-e^2\sin^2(\phi))^3}}{a(1-e^2)}+\frac{1}{2}\delta^*\left(1+\sqrt{(1-e^2)}\right)\sin(\phi)\right] \\ &\quad = \frac{2}{D_e}\frac{a(1-e^2)}{\sqrt{(1-e^2\sin^2(\phi))^3}} \end{aligned} \tag{15}$$

Due to the symmetry of the problem, the initial condition is given by:

$$\frac{d\delta^*}{d\phi} = 0 \quad \text{at } \phi = 0 \tag{16}$$

where

$$\begin{aligned} &\delta^* = \frac{\sqrt{Re}}{D_e}\delta, \qquad F = \frac{\frac{Ra}{Ja}}{Re^2}, \qquad P = \frac{\rho_\upsilon}{\rho}\frac{Pr}{Ja}, \qquad Pr = \frac{\mu C_p}{k}, \qquad Ja = \frac{C_p\Delta T}{h'_{fg}}, \qquad Re = \frac{\rho u_\infty D_e}{\mu} \\ &Ra = \frac{\rho(\rho-\rho_\upsilon)gPrD_e^3}{\mu^2} \end{aligned} \tag{17}$$

while the equivalent diameter of ellipsoid is defined as:

$$D_e = 2\frac{a}{\pi}\int_0^\pi \frac{1-e^2}{\sqrt{(1-e^2\sin^2\phi)^3}}\,d\phi \tag{18}$$

By applying the condition from Eq. (16) into Eq. (15), an expression for condensate film thickness at $\phi = 0$, δ_0^* can be obtained. Thus, the film thickness can be approached by applying a numerical method of fourth-order Runge–Kutta integration.

As in Nusselt theory [3], the dimensionless local heat transfer coefficient can be expressed as:

$$Nu = \frac{hD_e}{k} = \frac{D_e}{\delta} = \frac{\sqrt{Re}}{\delta^*} \tag{19}$$

On the other hand, based on the second law method in Bejan [18], the volumetric entropy generation rate for convection heat transfer can be derived as:

$$S'''_{gen} = \frac{k}{T_w^2}\left(\frac{\partial T}{\partial y}\right)^2 + \frac{\mu}{T_w}\left(\frac{\partial u}{\partial y}\right)^2 \tag{20}$$

The first term in Eq. (20) represents entropy generation due to heat transfer irreversibility and the second part indicates flow friction losses. Inserting Eqs. (12) and (13) into Eq. (20) yields:

$$\begin{aligned} S'''_{gen} = {} & \frac{k}{T_w^2}\left(\frac{\Delta T}{\delta}\right)^2 + \frac{\mu}{T_w}\left(\frac{(\rho-\rho_\upsilon)g}{\mu}(\sin\phi)\delta\left(1-\frac{y}{\delta}\right) + \frac{1}{2\mu}\frac{\rho_\upsilon u_\infty^2(1+\sqrt{(1-e^2)})^2\sqrt{(1-e^2\sin^2(\phi))^3}}{a(1-e^2)} \right. \\ & \left. \times \sin(2\phi)\delta\left(1-\frac{y}{\delta}\right) + \frac{m'' u_\infty(1+\sqrt{(1-e^2)})\sin(\phi)}{\mu}\right)^2 \end{aligned} \tag{21}$$

Volumetric entropy generation number, N'''_S, is of great concern here and is defined as the ratio between the volumetric entropy generation rate (S'''_{gen}) and a characteristic entropy generation rate (S'''_0). N'''_S and S'''_0 are expressed as:

$$N'''_S = \frac{S'''_{gen}}{S'''_0}, \qquad S'''_0 = \frac{k(\Delta T)^2}{D_e^2 T_w^2} \tag{22}$$

Further, by introducing the following parameters

$$u_0 = \frac{(\rho-\rho_\upsilon)gD_e^2}{2\mu}, \qquad Br = \mu\frac{u_0^2}{k\Delta T}, \qquad \eta = \frac{y}{\delta} \tag{23}$$

the entropy generation number per unit length of the tube can be expressed as:

$$\begin{aligned} N''_S = {} & \left(\frac{\sqrt{Re}}{\delta^*}\right)^2 + \frac{1}{\sqrt{Re}}\frac{Br}{\Theta}\left(2\delta^*(\sin\phi)(1-\eta) + \frac{2\delta^* P}{F}\left(1+\sqrt{(1-e^2)}\right)^2\sin(2\phi)(1-\eta) \right. \\ & \left. + \frac{2}{\delta^* F}\left(1+\sqrt{(1-e^2)}\right)\sin(\phi)\right)^2 \end{aligned} \tag{24}$$

By integrating the following equation with respect to η from zero to one, the final forms of the entropy generation number can be gained as:

$$\begin{aligned} N'_S = {} & \left[\left(\frac{\sqrt{Re}}{\delta^*}\right)^2\right] + \left[\frac{Br}{\Theta}\left(\frac{2\delta^*(\sin\phi)}{\sqrt{Re}} + \frac{2P\delta^*(1+\sqrt{(1-e^2)})^2\sin(2\phi)}{F\sqrt{Re}} + \frac{2(1+\sqrt{(1-e^2)})(\sin\phi)}{F\delta^*\sqrt{Re}}\right)\right. \\ & \left. \times\left(\frac{2(1+\sqrt{(1-e^2)})(\sin\phi)}{F\delta^*\sqrt{Re}}\right) + \frac{1}{3}\frac{Br}{\Theta}\left(\frac{2\delta^*(\sin\phi)}{\sqrt{Re}} + \frac{2P\delta^*(1+\sqrt{(1-e^2)})^2\sin(2\phi)}{F\sqrt{Re}}\right)^2\right] \\ = {} & \left[N'_H\right] + \left[N'_F\right] \end{aligned} \tag{25}$$

where N'_H represents entropy generation number due to heat transfer and N'_F is flow friction entropy generation number. The Brinkman number, Br, which has appeared in N'_F in the above equation is a measure of the importance of the viscous heating against the conductive heat transfer. As it is concluded in previous works such as [12] and [30], according to F, two regions can be introduced. For high values of F or in other words when $F \to \infty$, it is said that free convection region is dominant and no effect of vapor shear is considered. Furthermore, for low F ($F \to 0$), the region is called forced convection and the effect of gravity is neglected. So the influence of free or forced convection is expressed by the value of F parameter. In this paper, the attention is on forced convection film condensation and $F = 0.1$ was adopted. Moreover, the pressure gradient term, P, is attributable to potential flow and practical values of this dimensionless parameter are assumed to be in the range of 0.01–1 ($0.01 < P < 1$). Consequently $P = 1$ was chosen for this study.

It is useful to introduce irreversibility distribution ratio which can be used to recognize which part of irreversibility, heat transfer and friction flow, is dominated. It is defined as:

$$\psi = \frac{N'_F}{N'_H} \tag{26}$$

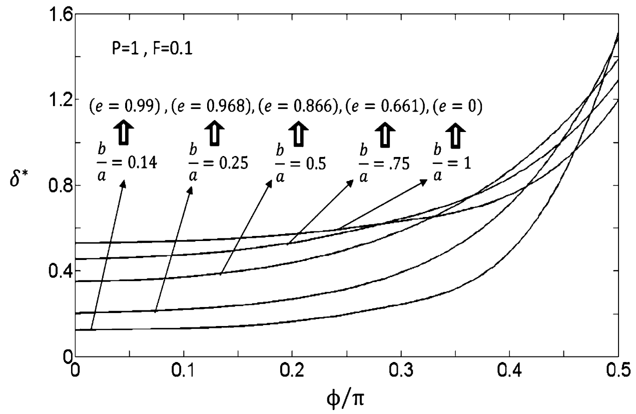

Fig. 2. Variation of film thickness versus ϕ for various eccentricities.

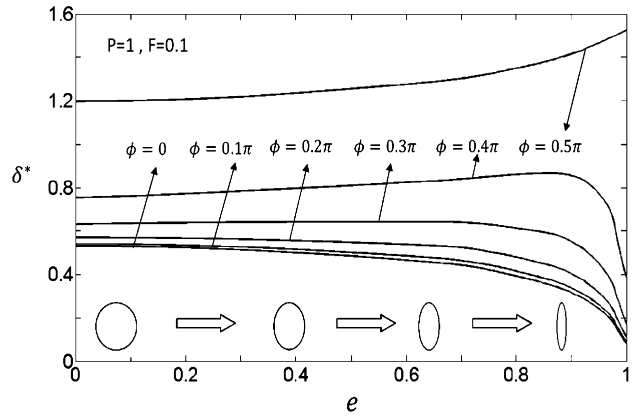

Fig. 3. Variation of film thickness versus ellipticity for some selected ϕ.

3. Results and discussion

It is proved that in the forced convection region, the higher the ellipticity, e, is, the larger the critical angle is [12]. Critical angle relates to the angle at which separation occurs. So by increasing the ellipticity, the location of the separation shifts more downstream for the elliptic tube. Consequently, in order to compare different parameters for various ellipticities, the range of $0 < \phi < \frac{\pi}{2}$ has been chosen.

In Fig. 2 variation of non-dimensional thickness of condensate layer via ϕ, for various eccentricities has been plotted. It demonstrates that for a specified eccentricity, film thickness increases as ϕ increases until it separates from the surface of the tube. Also, the figure shows that how the parameter eccentricity influences the condensate layer thickness. It can be seen that for the front portion of the tube, decreasing eccentricity will cause reduction in the value of film thickness but this trend is not true for the other section of the tube surface. As it has been observed in Eq. (25), heat transfer and flow friction entropy generation numbers are related to the condensate thickness and increasing or decreasing the film thickness will affect them. Moreover, in Fig. 3, the curves display that how ellipticity of ellipse influences δ^* at each ϕ. It can be concluded that for low values of ϕ, increasing ellipticity decreases the film thickness while at $\phi = \pi/2$, by increasing ellipticity, δ^* has increased.

In Fig. 4(a), the effect of $\frac{Br}{\Theta}$ and ϕ on N'_F and the variation of N'_H versus ϕ for a specified eccentricity have been studied. It demonstrates that increasing ϕ and therefore increasing δ^* causes reduction in the value of N'_H. Physically we know that condensate film acts as a resistor and reduces the amount of heat transfer coefficient, consequently the amount of heat transfer irreversibility decreases. On the other hand, it is seen in Fig. 4(a) that for a specified eccentricity and $\frac{Br}{\Theta}$, by increasing ϕ, the amount of N'_F increases until reaching a specific angle and after that it has a descending behavior. Also, Fig. 4(a) shows that flow friction entropy generation number and consequently entropy generation number increases as value of $\frac{Br}{\Theta}$ increases. This confirms that the parameter $\frac{Br}{\Theta}$ plays an important role in influencing irreversibility induced by condensate flow friction. Thus, the minimization of entropy generation number can be achieved by reducing the value of $\frac{Br}{\Theta}$.

Influence of Reynolds on irreversibility has been considered in Fig. 4(b). As it is seen in this figure, the heat transfer entropy generation number increases as Reynolds number increases while increasing Reynolds has an inverse effect on

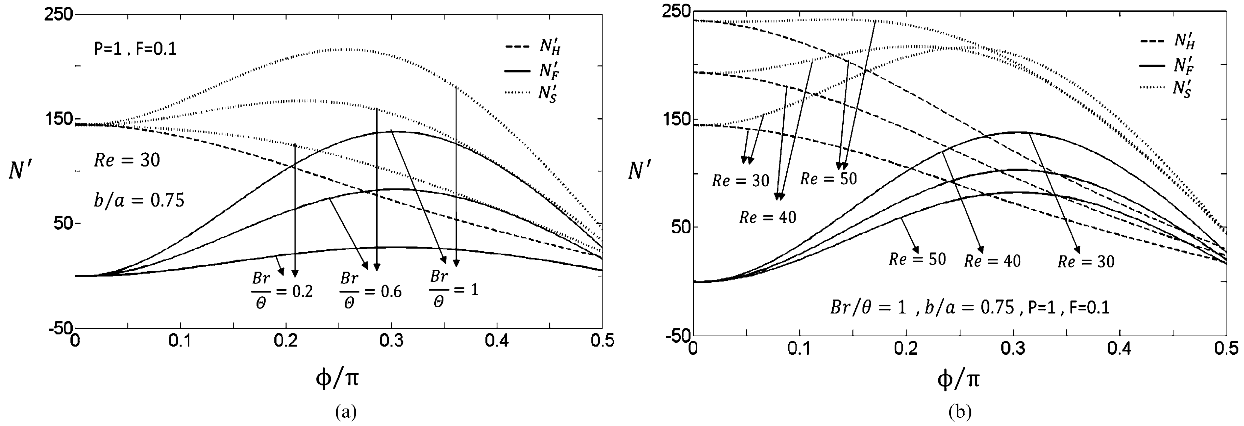

Fig. 4. Variation of entropy generation number versus ϕ for (a) different Br/Θ and (b) different Re.

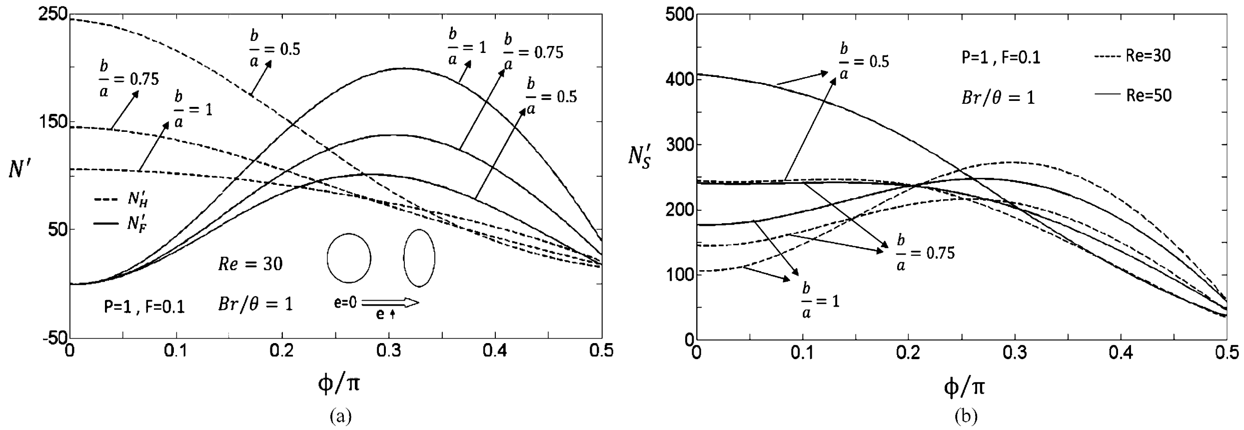

Fig. 5. Effect of eccentricity on (a) heat transfer and flow friction entropy generation numbers and (b) entropy generation number.

flow friction entropy generation number and high Reynolds number leads to low entropy generation due to flow friction irreversibility. So parameter Reynolds affects the entropy generation number which its variation is also plotted in this figure.

The effect of eccentricity for a specified amount of Re and $\frac{Br}{\Theta}$ on N'_F and N'_H is shown in Fig. 5. Due to the inverse dependence of N'_H to δ^*, for a condensate film which is moving downward the tube, wherever decreasing eccentricity causes reduction in the amount of film thickness, heat transfer irreversibility increases and inverse trend happens for region where decreasing eccentricity increases the value of δ^*. So as it is seen in Fig. 5(a), for $\phi < 0.3\pi$, N'_H increases as eccentricity decreases but after that, decreasing eccentricity slakes down the heat transfer entropy generation number. Also, the curves show that the effect of eccentricity on N'_H is more significant for $\phi < 0.3\pi$. Moreover, this figure represents that N'_F decreases as eccentricity decreases and the effect of eccentricity on flow friction entropy generation number is more obvious for high values of ϕ. According to the above statements, eccentricity will affect the entropy generation number. To study the effect of Reynolds and eccentricity on N'_S, Fig. 5(b) has been plotted. The figure confirms that decreasing eccentricity has a non-uniform influence on entropy generation number and this trend is true for all Reynolds.

Variation of N_S which can be obtained by integrating N'_S with respect to ϕ from zero to $\pi/2$, versus different ellipticity of ellipse has been plotted in Fig. 6. It can be concluded from this figure that the effect of ellipticity on total entropy generation number is significant for $e > 0.7$ and for this range of ellipticity, increasing e, increases the amount of total entropy generation number. Also, the figure confirmed that Reynolds and Brinkman numbers influence the N_S and as Re and $\frac{Br}{\Theta}$ increase, N_S increases. Furthermore, it can be observed that for a specific Reynolds, the effect of Brinkman number is more significant for $e < 0.8$.

The irreversibility distribution ratio for a specified eccentricity is illustrated in Fig. 7(a). When $\psi < 1$, heat transfer irreversibility dominates over the flow friction irreversibility and vice versa trend happens for $\psi > 1$. The figure shows that as condensate layer moves downward the tube the irreversibility ratio increases until reaching a specific angle and after that it decreases. Also, it can be observed from this figure that as $\frac{Br}{\Theta}$ increases, the irreversibility distribution ratio increases and the contribution of flow friction entropy generation number in ψ becomes more than heat transfer entropy generation number but increasing Reynolds number causes reduction in the amount of irreversibility ratio.

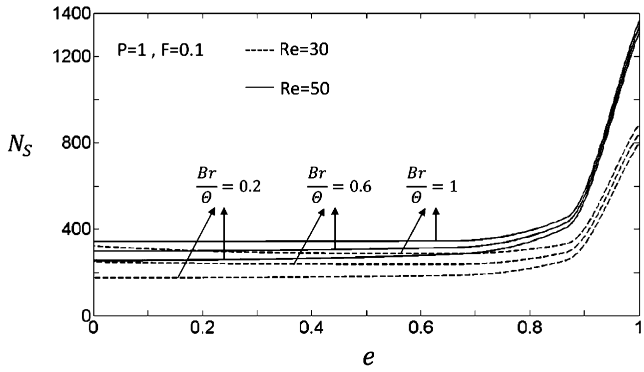

Fig. 6. Total entropy generation number versus ellipticity of ellipse for different Reynolds and Brinkman numbers.

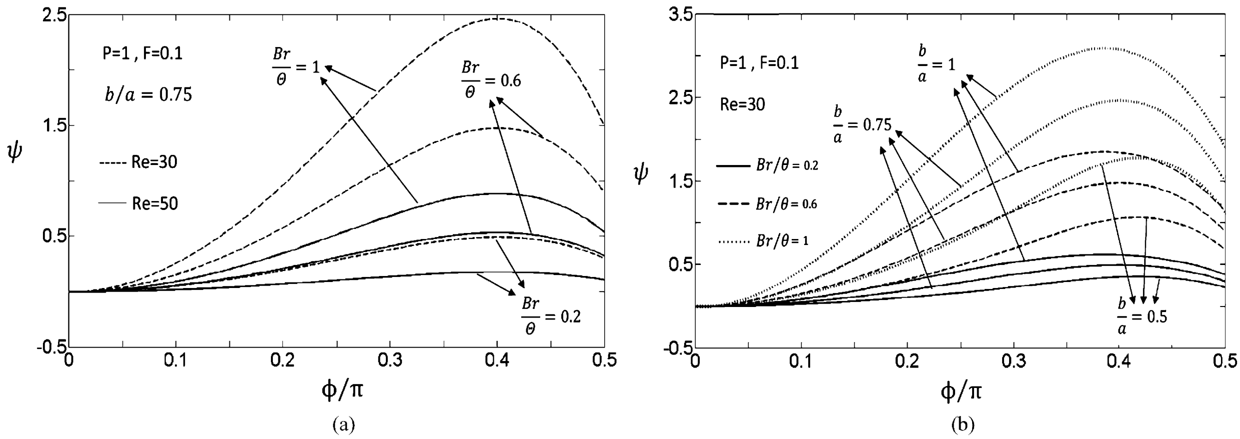

Fig. 7. Irreversibility distribution ratio versus ϕ for (a) various $\frac{Br}{\Theta}$ and *Re*, (b) various parameters $\frac{Br}{\Theta}$ and eccentricity.

In Fig. 7(b), the influence of eccentricity on irreversibility distribution ratio has been studied. This figure shows that by decreasing the eccentricity, the irreversibility ratio decreases. Also, comparing the values of ψ at a specific *Re* but different Brinkman numbers shows that the curves of irreversibility distribution ratio at different $\frac{Br}{\Theta}$ are more close to each other at lower eccentricities so the effect of $\frac{Br}{\Theta}$ on ψ is less pronounced for lower amounts eccentricities.

4. Conclusions

In this study, thermodynamic second law analysis was used to study the mechanism of entropy generation in forced convection film condensation from saturated vapor flowing onto an isothermal horizontal elliptical tube. To achieve the desired goal, an expression has been derived for entropy generation number, which considered both heat transfer and flow friction irreversibility. The results of the solution are as follows:

- Increasing ellipticity or decreasing eccentricity influences the condensate film thickness which was explained in Fig. 2;
- Increasing ϕ and thus increasing condensate film thickness causes reduction in the values of N'_H;
- By increasing ϕ, the amount of N'_F increases until reaching a specific angle and after that it has a descending behavior;
- For a specified Reynolds and eccentricity, N'_F and consequently N'_S increase as value of $\frac{Br}{\Theta}$ increases. So, entropy generation minimization can be applied by reducing the Brinkman number while no losing in condensation heat transfer coefficient happens;
- For a specified eccentricity and $\frac{Br}{\Theta}$, N'_H increases as Reynolds number increases while increasing Reynolds has an inverse effect on N'_F;
- Effect of ellipticity on N_S is significant for $e > 0.7$ and for this range of ellipticity, increasing e, increases the amount of N_S;
- Increasing *Re* and $\frac{Br}{\Theta}$ cause increscent in the value of N_S;
- For a specified eccentricity, as $\frac{Br}{\Theta}$ increases, the irreversibility distribution ratio increases but increasing Reynolds number has an inverse effect and reduces the amount of irreversibility ratio;
- By decreasing eccentricity for a specified Reynolds and $\frac{Br}{\Theta}$, the irreversibility ratio decreases.

Totally it can be said that in forced convection film condensation, Re, $\frac{Br}{\Theta}$ and ellipticity or eccentricity affect the entropy generation number and the minimization of N_S can be achieved by reducing the value of ellipticity of ellipse, Brinkman and Reynolds numbers.

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