FILM CONDENSATION ON A FLAT PLATE WITH ASSISTED VAPOR FLOW

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ABSTRACT The present study focuses on the problem of film condensation on a flat plate by taking into account the effect of flowing vapor. A general relation in the form of an ordinary differential equation is derived for calculating the liquid film thickness. However, due to its complexity, it is solved numerically with a forth order Runge Kutta method. The calculations for the liquid film thickness are followed by the Nusselt calculations and are discussed for various values of Reynolds number.

INTRODUCTION

The importance of film condensation on a flat plate is due to its great number of industrial applications such as heat exchangers and condensers. Several condensing systems, such as flat plates, circular cylinders, spheres, and non-circular cylinders have been extensively studied regarding the phase-change heat transfer problem of film-wise condensation.

Nusselt [1] first presented the basis for the problem of condensation and analyzed the local balance of viscous forces and the weight of the condensate. He showed that heat transfer in condensation depends on local film thickness. Following his work, many researchers have worked in this field [2-4]. The approaches have been along the same broad general lines, but since they use different assumptions and approximations, are of varying degrees of complexity. Esfahani and Ziaei-Rad [5,6] investigated an analysis of laminar film condensation on a vertical plate. They developed the work of Sparrow and Gregg [3] focusing on more details using dimensionless equations and parameters. As for the effects of interfacial shear forces on condensation, Yang and Hsu [7] developed a simple mathematical model for the study of mixed convection film condensation with downward flowing vapor onto a horizontal elliptical tube. Koh et al. [8] obtained an exact boundary layer solution, while the shear forces at the liquid-vapor interface were taken into account. Shekriladze and Gomelaury [9] analyzed the problem assuming that the shear stress at the liquid-vapor interface is equal to the loss of momentum of the condensing vapor. Dhir and Linhard [10] presented a similar solution for laminar film condensation on a two-dimensional isothermal surface.

Churchill [11] investigated the effects of the inertia and the heat capacity of the condensate on the vapor drag and the curvature of the surface. Hsu and Yang [12] and Rose [13] modified the Shekriladze and Gomelauri's model and took into account the pressure gradient effect by using the potential flow theory. Applying perturbation methods, Mendez et al. [14] studied the conjugate heat transfer condensation process of standard vapor, and the effect of longitudinal heat conduction on the thermal thick wall regime is taken into account. Adeyinka and Naterer [15] investigated the physical significance of entropy generation in plate film condensation. Using some simplifications, optimum conditions for high interfacial shear is introduced. In another entropy generation analysis, Esfahani and Koohi [16] studied the problem of film-wise condensation on a flat plate by considering shear forces at the liquid-vapor interface. But since the aim of their study was to discuss the optimum conditions for the problem, the first law aspect of the problem was not discussed in details.

In the present work, the problem of condensation on a flat plate with vapor free stream velocity is developed focusing on more details than the previous studies [16]. A general relation in the form of an ordinary differential equation is derived for calculating the liquid film thickness. The calculations for the liquid film thickness are followed by the Nusselt calculations and are discussed for various values of Reynolds number.

MATHEMATICAL MODEL

The physical model and coordinate system under consideration are shown in Fig.1. A saturated pure vapor with saturation temperature (T_{sat}) flows on a horizontal flat plate and imposes an assisted interfacial shear (τ) which varies over the interfacial length. The wall temperature (T_w) is uniform and below the saturation temperature. Thus, condensation occurs on the wall and a continues liquid film runs downward over the plate under the combined effects of gravity and the vapor interfacial shear force.

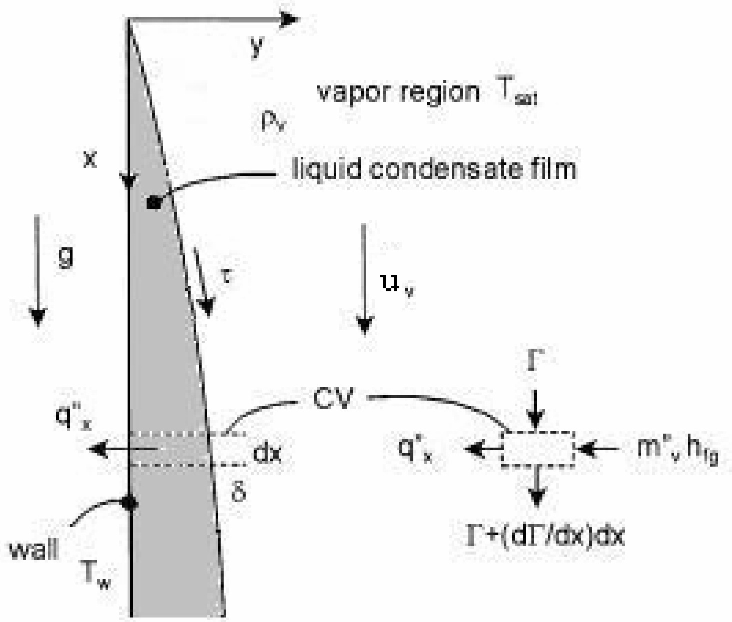

Figure 1. Physical model and coordinate system for condensate film flow on a flat surface

The assumptions employed in the formulation of the problem are
(1) The condensate film flow is laminar and steady state.
(2) The inertia effect of condensate is neglected due to the low velocity of the condensate.
(3) Viscous dissipation is ignored.
(4) Compared with the conduction in the y-direction, conduction in the x-direction in the condensate layer is negligible.
(5) A vapor boundary layer is assumed to exist on the condensate layer.

Therefore, for a laminar, steady state condensate film with constant fluid properties, the boundary layer equations governed by the basis of conservation principles (mass, momentum, and energy) are:

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

$$\mu\frac{\partial^2 u}{\partial y^2}=g_x(\rho-\rho_v) \tag{2}$$

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

It can be noticed that the momentum equation (Eq. (2)) reflects a balance of viscous and gravitational forces through its left and right terms, respectively. Note that the properties without a subscript refer to the liquid phase and properties with subscript *v* refer to the vapor phase.

It is further assumed that a shear force exists at the interface. Thus, the boundary conditions subjected to the above equations are:

$$at \quad y=0 \quad , \quad u=0 \quad T=T_w \tag{4}$$

$$at \quad y=\delta \quad , \quad \frac{\partial u}{\partial y}=\frac{\tau}{\mu} \quad T=T_{sat} \tag{5}$$

The shear stress τ in the velocity boundary condition at the liquid vapor interface can be replaced by its definition due to the characteristics of the vapor flow as below:

$$\tau=\frac{1}{2}C_f\rho_v(u_v-u_i)$$

where C_f, u_i and u_v are the coefficient of friction, interfacial and potential flow velocities, respectively. The coefficient of friction, C_f, can be gained as

$$C_f=0.664\,\mathrm{Re}_{v,x}^{-1/2} \tag{6}$$

where

$$\mathrm{Re}_{v,x} = \frac{(u_v - u_i)x}{\upsilon_v} \tag{7}$$

where $\mathrm{Re}_{v,x}$ is the vapor Reynolds number. Note that when $u_v = u_i$, the problem is simplified to the classical theory of Nusselt for condensation [1] where no vapor shear is considered to exert upon the condensate.

THERMAL ANALYSIS

For the prescribed boundary conditions, the dimensionless temperature and velocity profiles are obtained as below:

$$\frac{T - T_w}{T_{sat} - T_w} = \frac{y}{\delta} \tag{8}$$

$$\frac{u}{u_v - u_i} = \frac{Ar_L}{\mathrm{Pr}\,\mathrm{Re}_L}\bar{\delta}^2\left(\frac{y}{\delta} - \frac{1}{2}\frac{y^2}{\delta^2}\right) + 0.332\,\mathrm{Re}_{v,L}^{1/2}\,\bar{\delta}\bar{x}^{-1/2}\left(\frac{\mu_v}{\mu}\right)\left(\frac{y}{\delta}\right) \tag{9}$$

where

$$\bar{x} = \frac{x}{L};\ \bar{\delta} = \frac{\delta}{L};\ \mathrm{Re}_L = \frac{(u_v - u_i)L}{\upsilon};\ Ar_L = \frac{g(\rho - \rho_v)L^3}{\rho\upsilon\alpha} \tag{10}$$

In the above relations, L, Re_L and Ar_L are the plate length, *relative Reynolds* number due to the velocity difference of the vapor and *Archimedes* number, respectively. Note here that Re_L presented above is the liquid Reynolds number which can be converted to vapor Reynolds number ($\mathrm{Re}_{v,L}$) by substituting υ by υ_v. The Archimedes number is similar to the Rayleigh number in free convection which is a representative of the relative magnitude of the buoyancy and viscous forces in the fluid. Equation (9) can be written in the form of:

$$\frac{uL}{\upsilon} = \frac{Ar_L}{\mathrm{Pr}}\bar{\delta}^2\left(\frac{y}{\delta} - \frac{1}{2}\frac{y^2}{\delta^2}\right) + 0.332\,\mathrm{Re}_{v,L}^{3/2}\,\bar{\delta}\bar{x}^{-1/2}\left(\frac{\mu_v}{\mu}\right)\left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{y}{\delta}\right) \tag{11}$$

The first term in the right hand side of Eq. (11) accounts for the gravitational forces. As can be noticed, assuming an assisted vapor flow for the problem adds a second term to the velocity profile. Therefore, higher values of the velocity profile are expected for the case of having a vapor flow and the higher is the value of vapor free stream velocity (higher $Re_{v,L}$), the higher are the values of the velocity profile across the liquid film thickness.

Also, the term corresponding to the gravitational forces is of second degree of y/δ and the term corresponding to the vapor velocity effects is linear. Consequently, for the limiting case of high Ar_L number compared to $Re_{v,L}$, the effect of the vapor free stream velocity (the second term) becomes very small and the velocity profile versus y/δ tends to take a polynomial form while for the inverse case of low Ar_L number compared to $Re_{v,L}$, it will take a linear form.

Furthermore, the relative Reynolds number can be expressed in the form of

$$\mathrm{Re}_L = \mathrm{Re}'_L \left(1 - \frac{u_i}{u_v}\right) \tag{12}$$

where

$$\mathrm{Re}'_L = \frac{u_v L}{\upsilon} \tag{13}$$

It is seen that for very low values of interfacial velocity (u_i) compared to vapor free stream velocity (u_∞), the term u_i / u_v in Eq. (12) can be counted as negligible. Therefore, one can have

$$\mathrm{Re}_L \approx \mathrm{Re}'_L \tag{14}$$

Film Thickness

What remains undefined in the above equations is the liquid film thickness (δ) which can be obtained by writing a balance of energy between the latent heat released at the interface through condensation and the heat flux conducted through the liquid layer to the wall surface. Since the liquid temperature is linear, Fourier's law may be used to express the surface heat flux.

$$\dot{m}''_v h_{fg} = k \frac{\Delta \mathrm{T}}{\delta} \tag{15}$$

In the above relation, $\dot{\mathrm{m}}''_{\mathrm{v}}$ is the condensate mass flow rate per unit area and can be calculated using the continuity equation (see Fig.1):

$$\dot{m}''_v = \frac{\partial \Gamma}{\partial x} \tag{16}$$

where Γ is the condensate mass flow rate per unit width and is calculated by replacing the velocity profile from Eq. (9):

$$\Gamma(x) = \int_{y=0}^{y=\delta(x)} \rho u dy = \mu \frac{Ar_L}{3\,\mathrm{Pr}} \bar{\delta}^3 + 0.166\,\mathrm{Re}_{v,L}^{3/2}\, \bar{\delta}^2 \bar{x}^{-1/2} \left(\frac{\upsilon_v}{\upsilon}\right) \mu_v \tag{17}$$

Hence, by differentiating the mass flow rate per unit width in the x direction, $\dot{m}''_v$ is obtained:

$$\dot{m}''_v = \frac{\mu}{L} \left\{ \left[\frac{Ar_L}{\mathrm{Pr}} \bar{\delta}^2 + 0.332\,\mathrm{Re}_{v,L}^{3/2}\, \bar{x}^{-1/2} \bar{\delta} \left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right) \right] \frac{d\bar{\delta}}{d\bar{x}} - 0.083\,\mathrm{Re}_{v,L}^{3/2}\, \bar{x}^{-3/2} \bar{\delta}^2 \left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right) \right\} \tag{18}$$

Substituting $\dot{m}''_v$ in Eq. (15) will result in the following relation for the liquid film thickness (δ):

$$\frac{d\bar{\delta}}{d\bar{x}} = \frac{D + B(\bar{x}, \bar{\delta})}{A(\bar{\delta}) + C(\bar{x}, \bar{\delta})} \tag{19}$$

where

$$A(\bar{\delta}) = \frac{Ar_L}{\Pr}\bar{\delta}^3$$

$$B(\bar{x},\bar{\delta}) = 0.083\,\mathrm{Re}_{v,L}^{3/2}\left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right)\bar{x}^{-3/2}\bar{\delta}^3 \tag{20}$$

$$C(\bar{x},\bar{\delta}) = 0.332\,\mathrm{Re}_{v,L}^{3/2}\left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right)\bar{x}^{-1/2}\bar{\delta}^2$$

$$D = Ja\Pr$$

and

$$Ja = \frac{C_p\left(T_{sat} - T_w\right)}{h_{fg}} \tag{21}$$

The dimensionless liquid film thickness ($\bar{\delta}$) can be gained numerically via the forth order Runge Kutta method from Eq. (19).

Nusselt Calculation

As in Nusselt theory [1], interpreting a local heat transfer coefficient gives:

$$h_x = \frac{k}{\delta} = \frac{k}{\bar{\delta}L} \tag{22}$$

Therefore

$$Nu_L = \frac{1}{\bar{\delta}} \tag{23}$$

Having the numerical values of Eq. (19), Nu_L can be obtained. The average Nusselt number can also be obtained as:

$$\overline{Nu} = \frac{\overline{h_L}L}{k} = \int_0^1 \frac{d\bar{x}}{\bar{\delta}} \tag{24}$$

Where

$$\bar{h}_L = \frac{1}{L}\int_0^L h_x dx \tag{25}$$

Flow Rate

The dimensionless mass flow rate (local Reynolds number) in the condensation phenomena may be expressed in terms of a Reynolds number defined as:

$$\mathrm{Re}_\delta = \frac{4\Gamma}{\mu} \tag{26}$$

Putting the liquid mass flow rate per unit width (Γ) presented in Eq. (17), the dimensionless mass flow rate may be expressed simply as:

$$\mathrm{Re}_\delta = \frac{4Ar_L}{3\,\mathrm{Pr}}\bar{\delta}^3 + 0.664\,\mathrm{Re}_{v,L}^{3/2}\,\bar{\delta}^2\bar{x}^{-1/2}\left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right) \tag{27}$$

As it is seen the mass flow rate per unit width (Γ) is affected by both gravitational effects of the fluid (first term on the right) and the vapor free stream velocity (second term on the right). In other words, unlike the case of Nusselt theory where the only important factor in the flow regime was the gravitational effect, in the case of considering the vapor flow, another factor due to these forces should be taken into account ($\mathrm{Re}_{v,L}$). As one can notice, for high values of Ar_L compared to $\mathrm{Re}_{v,L}$, Re_δ can be obtained as

$$\mathrm{Re}_\delta = \frac{4Ar_L}{3\,\mathrm{Pr}}\left(\frac{\delta}{L}\right)^3 \tag{28}$$

which means the only effective factor on the fluid regime is the gravitational force. This is expected as it is the result gained by the Nusselt theory. But as one might expect, for the opposite case of having vapor flow with a high velocity, the gravitational force does not have as big a role on the mass flow rate as the vapor free stream velocity does. In this case Re_δ can be obtained as

$$\mathrm{Re}_\delta = 0.664\,\mathrm{Re}_{L,v}^{3/2}\,\bar{\delta}^2\bar{x}^{-1/2}\left(\frac{\upsilon_v}{\upsilon}\right)\left(\frac{\mu_v}{\mu}\right) \tag{29}$$

This case is similar to the case of condensation on a horizontal flat plate with forced vapor flow.

RESULTS AND DISCUSSION

The numerical solution of Eq. (19) for calculating the liquid film thickness is carried out for $Ar_L/Pr=10^8$, $Re_L=10^6$, $Re_{v,L}=10^4$, $Ja=0.1$ and $(\upsilon_v/\upsilon)(\mu_v/\mu)=3.46$ except for the cases where different values are introduced.

The effect of $Re_{v,L}$ on the ratio of the velocity at the liquid vapor interface to the vapor free stream velocity is shown in Fig. 2. As it is shown, the values of the interfacial velocity are very low compared to the vapor free stream velocity (about 1%). Therefore, approximating this velocity ratio (u_i/u_∞) to zero will simplify the numerical calculations with not adding a big deal of error (less than 1%).

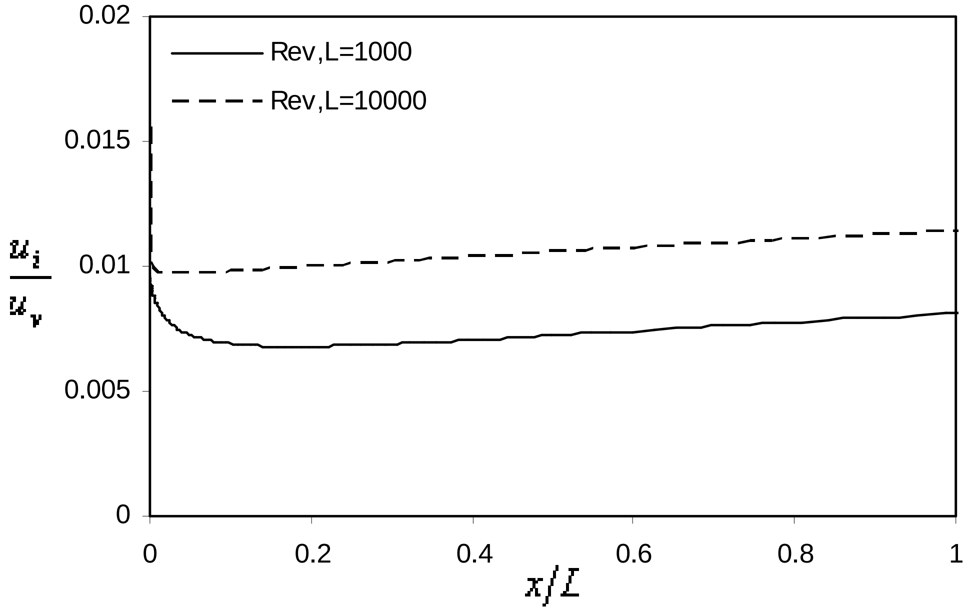

Figure 2. The effect of $Re_{v,L}$ on the ratio of the velocity at the liquid vapor interface to the vapor free stream velocity

The effect of $Re_{v,L}$ on the dimensionless liquid film thickness is shown in Fig. 3. As it is expected, marching along the plate will have an increase in the dimensionless liquid film thickness. Another characteristic of the diagram is the increase in the value of the dimensionless liquid film thickness for low values of $Re_{v,L}$ (lower than 1000). So, it is concluded that when a vapor with downward direction is forced past a cool wall (high Reynolds number), it will drag the liquid film along and will make it thinner. Also, it is observed that for low values of $Re_{v,L}$ ($Re_{v,L}=10^3$ when $Ar_L/Pr=10^8$) representing low values of vapor free stream velocity, the numerical results overlap the results of Nusselt theory were no shear stress is assumed to exert upon the condensate layer. Furthermore, it is seen that for all vapor Reynolds numbers, there is a sudden growth of the condensate layer (89°) at the beginning of the plate ($x/L \to 0$) which may be reliable due to neglecting the advection terms in the momentum equation.

Figure 4 indicates the effect of $Re_{v,L}$ on the velocity profile. It is seen that for constant values of Ar_L/Pr ($Ar_L/Pr=10^8$), the increase in $Re_{v,L}$ will lead to an increase in the value of uL/υ. As it was shown in Fig. 3, high vapor free stream velocities will result in thinning the condensate layer. Therefore, assuming constant properties for the liquid (constant υ), Fig. 4 shows that the velocity profile has higher values in cases with higher vapor free stream velocity (higher $Re_{v,L}$) compared to the Nusselt case. This is expected as the thinner condensate layer will have an enhanced heat transfer coefficient and consequently an enhanced condensate mass flow rate (Eq. (15)). Due to the relation for the condensate mass flow rate and having a thinner condensate layer, the increased mass flow rate for high vapor free stream velocity is possible only when the velocity profile have higher values which is illustrated in Fig. 4. Furthermore, the velocity profile is the sum of two terms corresponding to gravitational and vapor free stream velocity effects (Eq. (11)). Due to the previous discussions made on Eq. (11) and as it is shown in the figure, the velocity profile tends to take a polynomial form for the limiting case of low $Re_{v,L}$ number ($Re_{v,L}=10^3$) compared to Ar_L, while it will take a linear form for the inverse case of high $Re_{v,L}$ number ($Re_{v,L}=10^5$) compared to Ar_L.

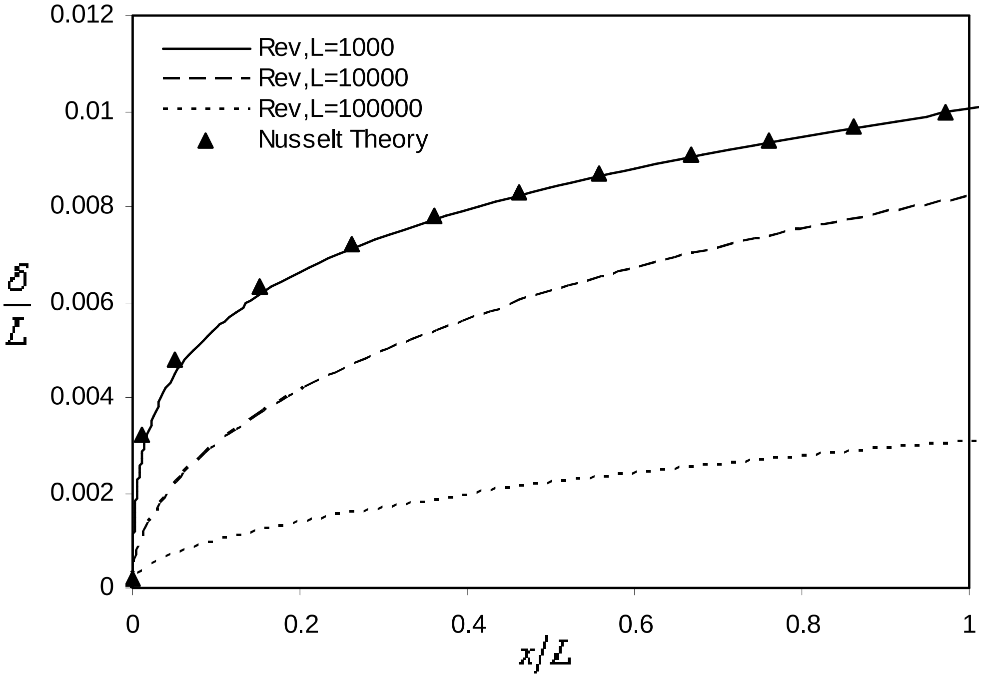

Figure 3. The effect of $Re_{v,L}$ on the dimensionless liquid film thickness (δ/L)

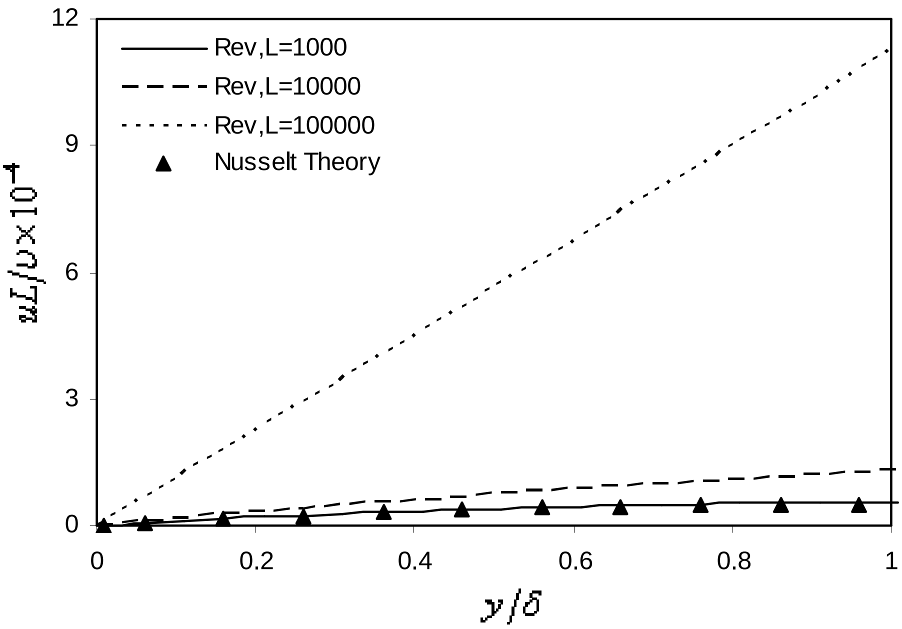

Figure 4. The effect of $Re_{v,L}$ on the velocity profile for $x/L=1$

The growth of the velocity profile along the plate length is shown in Fig. 5. It is seen that marching through x direction will have an increase in the values of the velocity profile. This is expected due to the increase in mass flow rate along the plate.

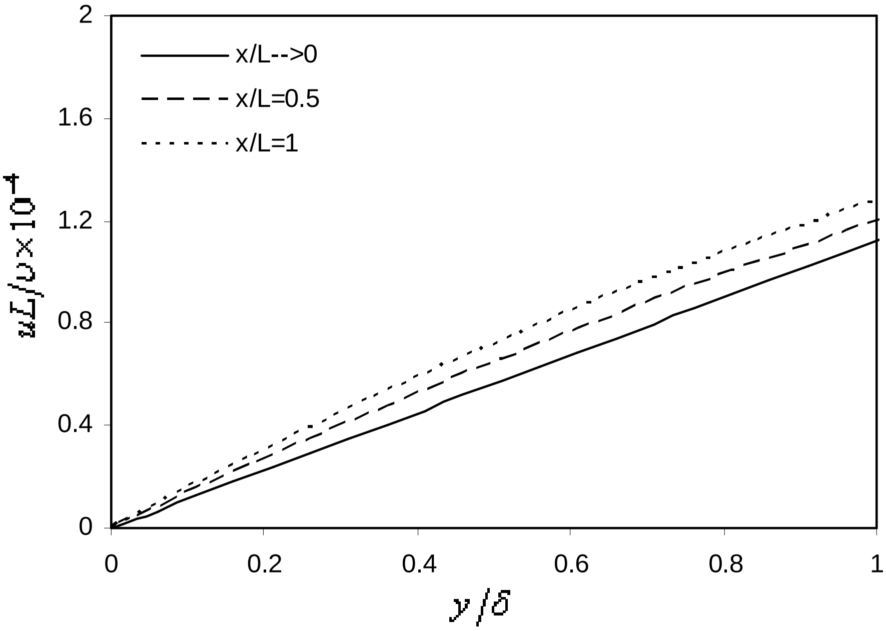

Figure 5. The growth of the velocity profile along the plate length when $Re_{v,L}=10^4$

Figure 6 displays the effect of $Re_{v,L}$ on Nu_x number. It is observed that for low values of $Re_{v,L}$ ($Re_{v,L}$=1000) representing low values of vapor free stream velocity, the numerical results overlap the results of Nusselt theory. This it is expected from the discussion made on Fig. 3; thinner liquid film thickness corresponding to higher vapor free stream velocity will enhance the heat transfer coefficient.

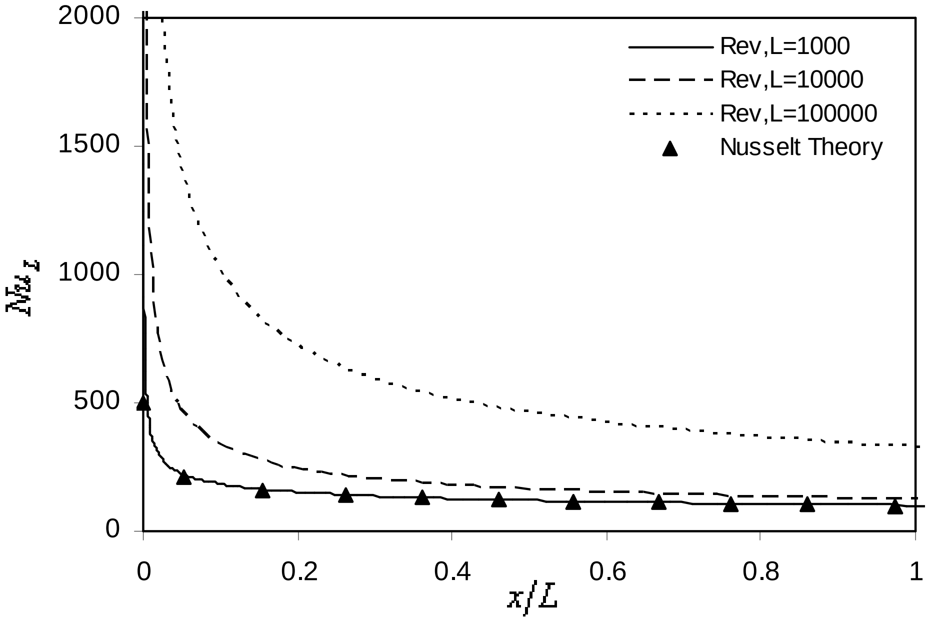

Figure 6. The effect of $Re_{v,L}$ on Nu number

The effect of Ja on Nu number is shown in Fig.7. It is shown that higher values of Ja number will result in lower values of Nu numbers. As mentioned previously, the values of dimensionless liquid film thickness vary with Ja (Eqs. (19) and (20)). Similar to the Nusselt's calculations for the liquid film thickness where the liquid film thickness varies with $Ja^{1/4}$, here as well, higher values of *Ja* will lead to higher values of dimensionless liquid film thickness and consequently a lower heat transfer coefficient (decrease from $(Nu_L)_{Ja=0.1}=127$ to $(Nu_L)_{Ja=0.4}=84$ at $\bar{x}=1$).

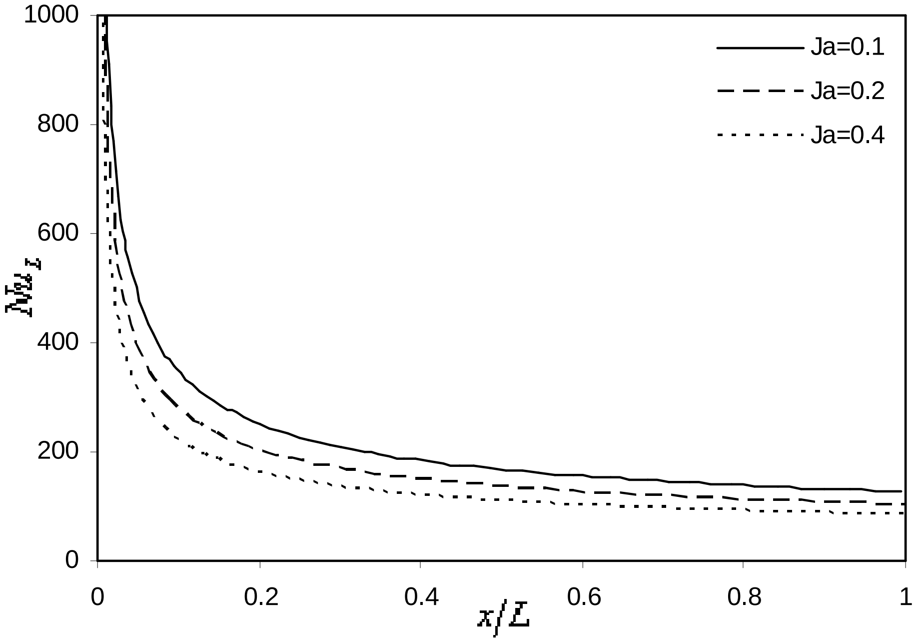

Figure 7. The effect of *Ja* on *Nu* number

Figure 8 depicts the variation of *Nu* number with Ar_L number. As it is shown, for constant values of $Re_{v,L}$ number, increasing the Ar_L number will result in an increase in the *Nu* number. This is expected, as higher values of Ar_L number corresponding to high gravitational forces will drag the liquid film down, making it thinner. Consequently, the thinner liquid film results in the higher heat transfer coefficient (increase from $(Nu_L)_{Ar/Pr=1e6}=103$ to $(Nu_L)_{Ar/Pr=1e10}=313$ at $\bar{x}=1$).

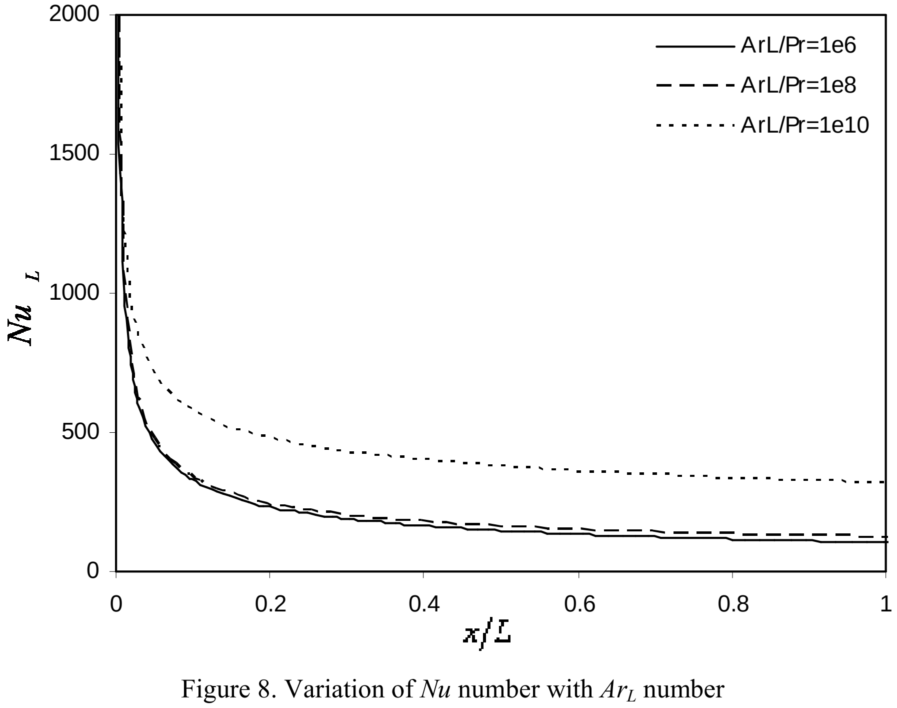

Figure 8. Variation of *Nu* number with Ar_L number

Figure 9 shows the variation of liquid mass flow rate per unit width (Γ) with Reynolds number ($Re_{v,L}$). It is seen that for a constant value of Archimedes number, increasing the vapor free stream velocity ($Re_{v,L}$) will result in an increase in the dimensionless liquid mass flow rate per unit width. Although the vapor free stream velocity have a decreasing effect on the liquid film thickness, the increase in the condensate velocity values will have an overall increasing effect on the liquid mass flow rate.

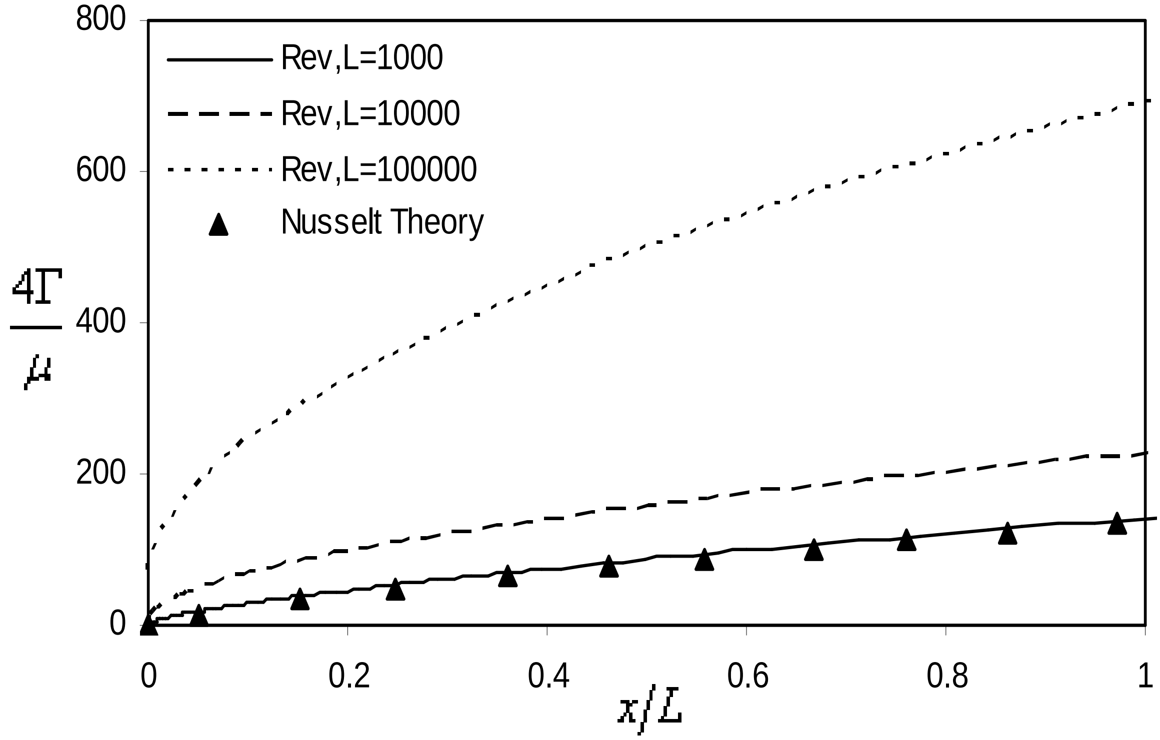

Figure 9. Variation of liquid mass flow rate per unit width (Γ) with Reynolds number

Variation of liquid mass flow rate per unit width (Γ) for low and high vapor Reynolds numbers ($Re_{v,L}=100000$ and 1000) is shown in Figs. 10 and 11, respectively. As it is shown in Eq. (27), the mass flow rate per unit width (Γ) is affected by both gravitational effects of the fluid and the vapor free stream velocity. It is seen in Fig. 10 that for low values of vapor Reynolds number, the mass flow rate per unit width is mostly affected by the gravitational term and the term related to vapor free stream velocity can be counted as negligible (Eq. (28)). As it is seen in Fig. 11, for the case of having a high vapor free stream velocity ($Re_{v,L}=100000$), the effect of the gravitational force on the value of mass flow rate is negligible compared to the effect of vapor free stream velocity. Therefore, in this case, Eq. (29) can be used with excellent degree of accuracy.

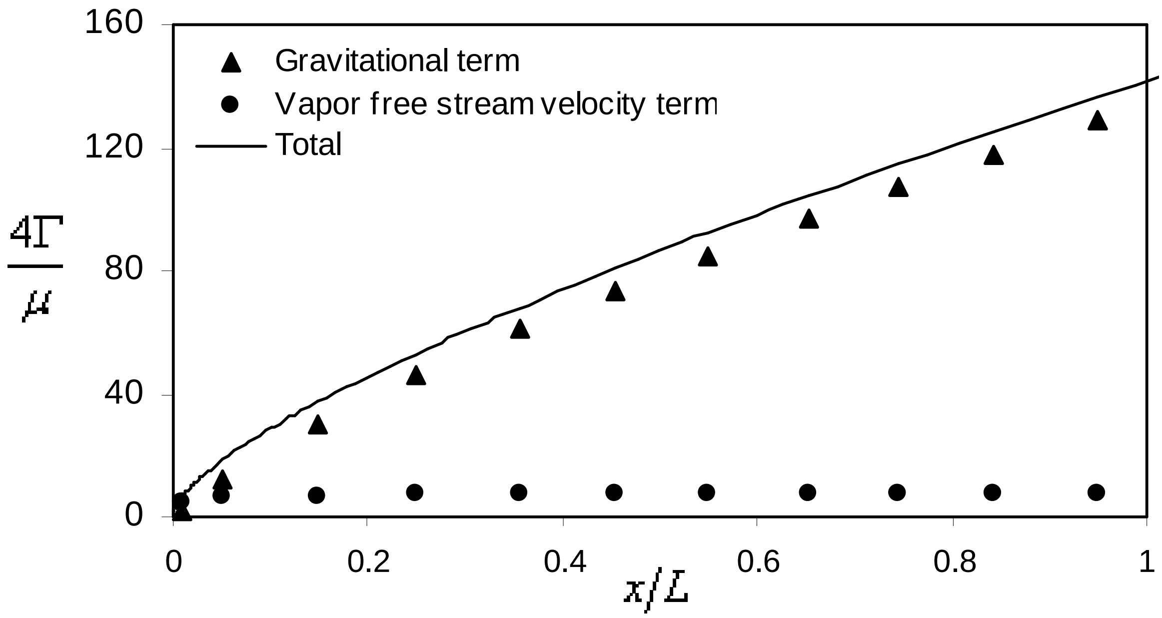

Figure 10. Variation of liquid mass flow rate per unit width (Γ) for $Re_{v,L}$*=1000*

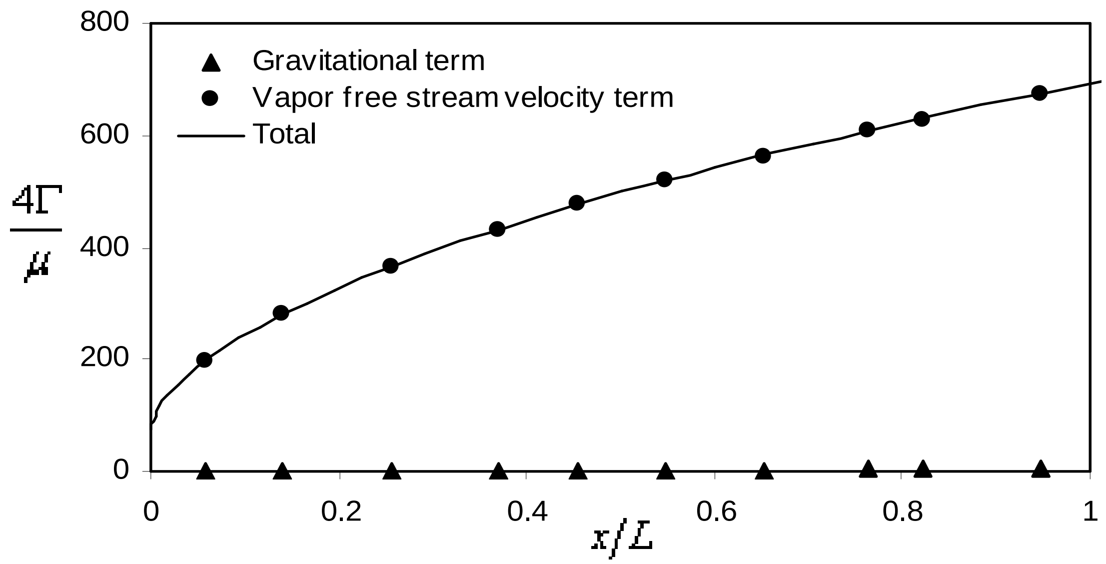

Figure 11. Variation of liquid mass flow rate per unit width (Γ) for $Re_{v,L}$*=100000*

CONCLUSIONS

An analytical study was performed considering the problem of film condensation of assisted flowing vapor on an isothermal flat plate. Based on the analysis on this paper the following conclusions can be drawn:

- The values of the interfacial velocity are very low compared to the vapor free stream velocity (about 1%)
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- The velocity profile has greater values in the cases with higher values of vapor free stream velocity compared to the Nusselt case were the vapor free stream velocity is assumed to be zero and no shear stress is assumed to exert upon the condensate layer.
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- When a vapor with downward direction is forced past a cool wall (high Reynolds number), it will drag the liquid film along and will make it thinner.
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- Assuming constant properties for the liquid (constant υ), it is resulted that the velocity profile has greater values in the cases with higher vapor velocities compared to the Nusselt case.
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- Thinner liquid film thickness resulted from higher values of Reynolds number will enhance the heat transfer coefficient.
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- Higher values of *Ja* will lead to higher values of the dimensionless liquid film thickness and consequently a lower heat transfer coefficient.
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- For constant values of $Re_{v,L}$ number, increasing the Ar_L number will result in an increase in the *Nu* number.
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- The vapor free stream velocity will result in an increase in the liquid mass flow rate.

This work was able to develop the problem of film condensation on a flat plate by taking into account the effects of vapor free stream velocity which is its key aspect compared to many pervious studies. The results of this study can be applied in any work of condensation on a flat plate which consists of the shear effects of the flowing vapor.

NOMENCLATURE

Ar Archimedes number
C_p specific heat at constant pressure, J/Kg.K
g gravity acceleration, m/s^2
h_{fg} enthalpy, J/Kg.K
Ja Jacob number
k thermal conductivity, W/m.K
L plate length, m
Pr Prandtl number
Re Relative Reynolds number
Re' Reynolds number
T temperature, K
u x-velocity component, m/s
v y-velocity component, m/s

x vertical coordinate
y horizontal coordinate

Greek Letters
δ thickness of the condensate layer, m
ρ density, Kg/m^3
μ dynamic viscosity, Kg/m.s
υ kinematic viscosity, m^2/s
τ interfacial shear stress

Subscripts
L plate length
sat saturated
v vapor
w wall

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