SIMULTANEOUS CONDENSATION-EVAPORATION HEAT TRANSFER ON A VERTICAL TUBE, II-THEORETICAL STUDY

انتقال الحرارة بالتكثيف و التبخير الآنيين من أنبوب رأسى - II -دراسة نظرية

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خلاصة تم فى هذا البحث عمل تحليل نظرى لانتقال الحرارة المصاحبة لعمليتى التكثيف و التبخير الآنيين من أنبوب مكثف - مبخر رأسى. تم افتراض أن سريان المائع رقائقى لكل من حالات للتكثيف والتبخير حيث كان رقم رينولدز للماء الساقط يتراوح بين 100-800 فى حين كانت قيمة هذا الرقم للطبقة المتكاثفة تتراوح بين 200-600 وقد تم تحويل المعادلات الواصفة للسريان وانتقال الحرارة إلى صورة لابعدية عن طريق إدخال متغيرات مستقلة وتابعة فى صورة لابعدية مناسبة. تم حل هذه المعادلات عدديا باستخدام طريقة الفروق المحدودة وذلك بتصميم برنامج للحاسب الآلى. بحل هذه المعادلات الواصفة للسريان أمكن الحصول على توزيع درجات الحرارة والسرعة وذلك خلال طبقة المائع المتكاثف على جدار الأنبوب من الخارج وطبقة المائع المتبخر (طبقة السائل الرقيقة المحيطة بجدار الأنبوب من الداخل). وبذلك أمكن استنتاج معامل انتقال الحرارة ورقم نوسلت الموضعى والمتوسط وذلك عند قيم مختلفة لرقم رينولدز و لإختبار صلاحية النموذج الرياضى المقترح تم عمل مقارنة بين النتائج الحالية ونتائج الدراسات السابقة ووجد أن نتيجة هذه المقارنة مرضية. وكذلك تم عمل علاقة رياضية تربط بين رقم نوسلت ورقم رينولدز ورقم براندل

ABSTRACT

In the present work, condensation - evaporation heat transfer on a vertical tube is, theoretically, investigated. Evaporation takes place inside the tube and condensation is outside it. For such tube, the condensed steam outside the tube imparts its latent heat to the evaporated thin liquid film. The effect of Reynolds number on condensation-evaporation heat transfer process is, numerically, analyzed. To perform this study, a mathematical model describing the condensation- evaporation process is suggested. This model is, numerically, solved employing finite - difference technique. A computer program is developed to solve this model. Accordingly; for different values of Reynolds number, the condensation film thickness, evaporation film thickness, velocity distribution, temperature distribution, local and average Nusselt number, are determined. Throughout this work, the range of Reynolds number of falling water-film is from 100 to 800, while that of condensate (outside the tube) takes the values from 200-600. A comparison between the present results and those of the previous works is made. A correlation of problem parameters, based on the present obtained results, is derived.

Key words: Heat Transfer - Condensation – Evaporation - Vertical tube

NOMENCLATURE

c_p	Specific heat at constant pressure	J/kg K
d	Tube diameter	m
g	Gravitational acceleration	m/s^2
h	Heat transfer coefficient	W/m^2.K

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h_{fg}	Latent heat	J/kg
k	Thermal conductivity	W/m.K
ℓ	Tube height	m
L	Dimensionless tube height	--
Nu	Nusselt number; $Nu = \frac{h(Z)\,d}{k}$	--
p	Pressure	Pa
P	Dimensionless Pressure; $P = \frac{p}{\frac{1}{2}\rho V^{*2}}$	--
q	Heat flux	W/m^2
r	Radial coordinate	m
r_w	Tube raduis	m
R	Dimensionless radial coordinate; $R = r/r_w$	--
R_w	Dimensionless tube radius	--
Re^*	Reynolds number defined as; $Re^* = \frac{\rho V^* r_w}{\mu}$	--
T	Temperature	°C
T_w	Wall temperature	°C
u, w	Radial and axial components of velocity	m/s
U, W	Dimensionless radial and axial components of velocity	--
V^*	Characteristic velocity	m/s
z	Axial coordinate	m
Z	Dimensionless axial coordinate	--

Greek symbols

μ	Dynamic viscosity of water	kg/m.s
ν	Kinematics viscosity of water	m^2/s
ρ	Density	kg/m^3
θ	Dimensionless temperature; $\theta = \frac{T - T_e}{T_c - T_e}$	--

Subscripts

c	Condensation
e	Evaporation
l	Liquid
w	Wall

1. INTRODUCTION

Condensation- evaporation heat transfer process on a vertical tube has been widely employed in heat exchange devices such as desalination units, heat exchangers, refrigeration units, petroleum refinery and food industries. Accordingly and seeking for good understanding of simultaneous condensation-evaporation process, on vertical tube, more studies are needed to be carried out.

Many investigators [1-4] analyzed theoretically, condensation of vapours in vertical and horizontal tubes. Lucas [1] studied film condensation occuring under two different entrance cases. In the first case, the condensation process starts at inlet cross section of the tube. In the other case, there is a dry inlet region, which allows the vapor to develop its parabolic velocity profile before condensation starts.

Dobran et al. [2] solved, analytically, the governing equations describing the process taking in consideration the hydrodynamic flow field. Rose [3] studied the improved theoretical results of heat transfer through individual drops and for the mean distribution of drop size and used it as the basis for assessing the validity of the basic assumption of dropwise condensation. Honda et al [4] studied a method for predicting the average heat transfer coefficient of condensation on a horizontal tube. Approximate equations based on the analytical solution of Nusselt are valid for finned horizontal tube

Yan [5] investigated laminar mixed convection associated with evaporation process in vertical channel under constant heat flux.

Many investigators [6-10] studied, theoretically, the performance of evaporator -condenser horizontal tube. Moalem et al [6] studied overall heat transfer coefficient in horizontal evaporator-condenser tube in case of low heat flux and laminar flow regime. Local evaporation heat transfer coefficient around the tube indicated a maximum value at an angle equals to $\pi/2$ from the top of the tube because the film thickness there is minimum. In laminar flow regime the average overall heat transfer eoefficient decreases with inereasing Reynolds number or increasing tube radius. Sideman et al [7] tried to enhance the heat transfer process in the horizontal evaporator-condenser tube, by cutting grooves of different shapes on the oute. surface. The studied groove shapes are the square-edged, triangular and circular grooves. The square-edged groove, with a straight or modified bottom, was the most efficient shape in the flow of range 300 <Re <1000.

Sideman et al [8] studied the effect of tube cross-section shape on heat transfer process. A Comparison of the overall heat transfer coefficient, based on identical heat transfer areas, indicated that the vertical elliptic eonduits yield higher values than those realized by corresponding circular tubes. Moalem et al [9] studied the heat transfer process assoeiated with the condensation-evaporation for the case of laminar flow of 60<Re<600. They assumed the velocity distribution in axial direction and solved the momentum and energy equations taking in account several assumptions. The velocity, temperature, film thickness and local heat transfer coefficient were evaluated around the tube periphery. The results showed that the film thickness was minimum at the tube sides (where the angle = $\pi/2$ from the top) and thus there the highest heat transfer coefficient is existed.

Mahgoub et al [10] studied the heat transfer in condenser- evaporator horizontal tube. The proposed model in that work, taking in consideration the effect of convective terms and pressure gradient in the flow governing equations. Local evaporation heat transfer coefficient around the tube indicated that the maximum value is existed at an angle equal to $\pi/2$ from the top of the tube, where the film thickness is minimum. In laminar flow regime, the average overall heat transfer coefficient decreases with increasing Reynolds number or increasing tube radius.

In the present work, heat transfer by simultaneous condensation- evaporation process on vertical tube is, theoretically, studied. The effect of convective terms in equations of motion is considered for both condensation and evaporation processes.

2. MATHEMATICAL FORMULATION OF THE PROBLEM

The simultaneous condensation-evaporation process can be described through the conservation laws of mass, momentum and energy. The physical description of the problem is presented in figure (1), showing the cylindrical coordinate system (r, ϕ, z), which is used to express the flow governing equations. The flow in both outside film and inside film, of the tube, is assumed to be steady and laminar.

2.1 The Condensation-Evaporation Processes Describing Equations

The condensation-evaporation, outside and inside the tube is, properly, taken axi- symmetrical. In order to simplify the describing equations of momentum and energy through the condensate film and falling thin water film, some assumptions are made. The condensation and evaporation are assumed to take place at the free surface of water film. The flow is taken two-dimensional laminar and steady. Moreover, the physical properties of water are assumed to be constant. According to the foregoing assumptions, the continuity, momentum and energy equations in cylindrical coordinates can be written as:

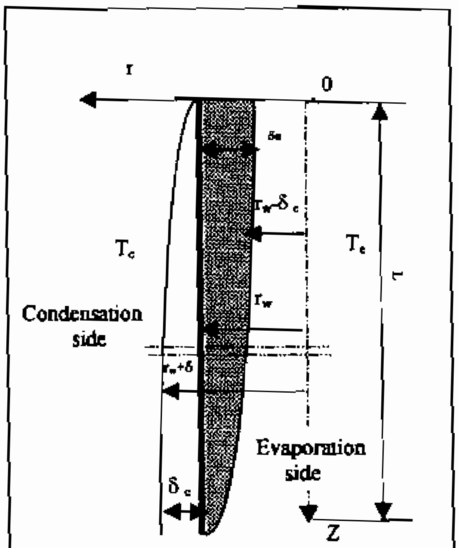

Fig. (1) The Used Coordinate System and Physical Description of the Problem

$$\frac{1}{r}\frac{\partial}{\partial r}(ru)+\frac{\partial w}{\partial z}=0.0 \quad (1)$$

$$u\frac{\partial u}{\partial r}+w\frac{\partial u}{\partial z}=-\frac{1}{\rho_l}\frac{\partial p}{\partial r}+\nu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right)+\frac{\partial^2 u}{\partial z^2}\right] \quad (2)$$

$$u\frac{\partial w}{\partial r}+w\frac{\partial w}{\partial z}=-\frac{1}{\rho_l}\frac{\partial p}{\partial z}+\frac{\Delta\rho}{\rho_l}g+\nu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial w}{\partial r}\right)+\frac{\partial^2 w}{\partial z^2}\right] \quad (3)$$

$$u\frac{\partial T}{\partial r}+w\frac{\partial T}{\partial z}=\frac{k}{\rho_l c_p}\left[\frac{\partial^2 T}{\partial r^2}+\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial z^2}\right] \quad (4)$$

The foregoing equations are valid for both condensation and evaporation processes. The velocity components in radial and axial directions are denoted by u and w respectively. As it is clear, the flow of both inner and outer film is two-dimensional flow, since the flow is assumed axe-symmetrical. Considering the energy equation; equation (4), the conductive heat transfer in tangential direction is neglected compared with those in radial and axial directions. Equations (1-4) must satisfy the following boundary conditions:.

- ***For condensation process***

for $0 \le z \le \ell$ *and* $r = r_w$: $u = w = 0, \; T = T_w$,

for $0 < z \le \ell$ *and* $r = r_w + \delta_c(z)$: $\dfrac{\partial u}{\partial r} = \dfrac{\partial w}{\partial r} = 0, \; T = T_c$ (5)

Where $\delta_c(z)$ is the condensate film thickness at any axial position z. By solving the governing equations. (1-4 and 5), one can obtain the velocity and temperature distributions throughout the flow field and, in turn, the local heat transfer coefficient and Nusselt number can be determined according to the following definitions:

$$h_c(z) = \frac{q_w}{T_c - T_w} = \frac{-k \left.\frac{\partial T}{\partial r}\right)_{r=r_w}}{T_c - T_w} \quad \text{and} \quad Nu_c = \frac{h_c(z)\, d}{k} \tag{6}$$

Equations (6) express local heat transfer coefficient $h_c(z)$ as a function of wall heat flux (q_w), the wall temperature (T_w) and the saturation temperature of the heating steam (T_c), while the local Nusselt number is expressed as a function of local heat transfer coefficient $h_c(z)$, thermal conductivity of the condensate film k and tube diameter d.

- ***For evaporation process***

$$\begin{aligned} &for \quad 0 \le z \le \ell \quad and \quad r = r_w: \quad u = w = 0 \quad \text{and} \quad T = T_w \\ &for \quad 0 \le z < \ell \quad and \quad r = r_w - \delta_e(z): \quad \frac{\partial u}{\partial r} = \frac{\partial w}{\partial z} = 0 \quad \text{and} \quad T = T_e \\ &At\ z = 0.0 \quad and \quad r = r_w \quad T_w = T_c \end{aligned} \tag{7}$$

Where $\delta_e(z)$ is the thickness of the inside film, which is found as a function of the axial position (z) and must be known to make the solution of the foregoing governing equations (1-4) possible. The evaporation temperature T_e is taken at the free surface of the film. The temperature of the tube wall is denoted by T_w, and r_w is the radius of the tube.

As a result of the solution of the foregoing governing equations (1-4 and 7), one can obtain the velocity and temperature distributions throughout the flow field of the heated water-film. Consequently, the local heat transfer coefficient and the local Nusselt number can be determined according to the following definitions:

$$h_e(z) = \frac{q_w}{T_w - T_e} = \frac{-k \left.\frac{\partial T}{\partial r}\right)_{r=r_w}}{T_w - T_e} \quad \text{and} \quad Nu_e = \frac{h_e(z)\ d}{k} \tag{8}$$

Referring to equation (8), the local heat transfer coefficient $h_e(z)$ is expressed as a function of wall heat flux (q_w), the wall temperature (T_w) and the evaporation temperature (T_e). In addition, the local Nusselt number is expressed as a function of the local heat transfer coefficient $(h_e(z))$, the thermal conductivity of the heated water-film (k) and the tube diameter (d).

With the aid of equations (6 and 8), one can derive the local overall heat transfer coefficient, for both inside and outside films, and hence the local and the average Nusselt number as:

$$h(z) = \frac{h_e(z) \times h_c(z)}{h_e(z) + h_c(z)}, \quad Nu(z) = \frac{h(z)\ d}{k} \quad \& \quad \overline{Nu} = \frac{1}{\ell}\int_0^{\ell} Nu(z)\ dz \tag{9}$$

Where $\overline{Nu}$ is the average Nusselt number. In the above equations, the effect of thermal resistance of the tube wall is neglected compared with that of inside and outside water-films.

2.2 Dimensionless Form of The Governing Equations

To put the flow describing equations in dimensionless form, one introduces the following definitions of the dependent and independent variables as:

$$\theta = \frac{T - T_e}{T_c - T_e},\ U = \frac{u}{V^*},\ W = \frac{w}{V^*},\ R = \frac{r}{r_w}\ \&\ \ Z = \frac{z}{r_w} \tag{10}$$

Where θ, U and W are the dimensionless temperature, the dimensionless velocity components in radial and axial directions. V^* is the characteristic velocity, which is found to be:

$$V_c^* = \frac{g\ (\Delta\rho)_c \cdot r_w^2}{\mu} \quad \text{and} \quad V_e^* = \frac{g\ (\Delta\rho)_e \cdot r_w^2}{\mu}$$

Where $(\Delta\rho)_c$ and $(\Delta\rho)_e$ are the change in density of liquid and vapour at the corresponding saturation temperature in the condensation and evaporation sides; respectively. As it is clear from the above two equations that, the characteristic velocity V^* has two definitions, based on the properties of the corresponding film (outside or inside film).

Substitution with the equations (10) in the governing equations (1-4) yields to the dimensionless form of the continuity, momentum and energy equations. They can be written as:

$$\frac{1}{R}\frac{\partial(RU)}{\partial R} + \frac{\partial W}{\partial Z} = 0.0 \tag{11}$$

$$Re^*.R\left[U\frac{\partial U}{\partial R} + W\frac{\partial U}{\partial Z}\right] = -R\frac{\partial P}{\partial R} + R\frac{\partial^2 U}{\partial R^2} + \frac{\partial U}{\partial R} + R\frac{\partial^2 U}{\partial Z^2} \tag{12}$$

$$Re^*.R\left[U\frac{\partial W}{\partial R} + W\frac{\partial W}{\partial Z}\right] = -R\frac{\partial P}{\partial Z} + R\frac{\partial^2 W}{\partial R^2} + \frac{\partial W}{\partial R} + R\frac{\partial^2 W}{\partial Z^2} + R \tag{13}$$

$$Pe\ R\left[U\frac{\partial \theta}{\partial R} + W\frac{\partial \theta}{\partial Z}\right] = R\frac{\partial^2 \theta}{\partial R^2} + \frac{\partial \theta}{\partial R} + R\frac{\partial^2 \theta}{\partial Z^2} \tag{14}$$

Where Re^* and Pe are Reynolds and Peclet numbers. Their definitions depend on the physical properties of considered film (condensation or evaporation film). In case of condensation (outside film), they are defined according to the following relations:

$$Pe_c = \frac{\rho_c\, c_p\, V_c^*\, r_w}{k}, \quad and \quad Re_c^* = \frac{\rho_c\, V_c^*\, r_w}{\mu}$$

They have the following definitions in case of evaporation process (inside film):

$$Pe_e = \frac{\rho_e\, c_p\, V_e^*\, r_w}{k}, \quad and \quad Re_e^* = \frac{\rho_e\, V_e^*\, r_w}{\mu}$$

The dimensionless forms of the boundary conditions can be derived in the same previously used manner, they can be expressed, corresponding to equations (5 and 7), as follows:

- ***For condensation process***

  At the tube wall: $R = 1,\ \ U = W = 0\ \ \&\ \ \theta = 0_w$

  At the film edge: $R = 1 + \frac{\delta_c}{r_w},\ \ \frac{\partial U}{\partial R} = \frac{\partial W}{\partial Z} = 0\ \ \&\ \ \theta = 1$ (15)

Solving equations (11-15), the dimensionless velocity and temperature distributions across the condensate film, at different values of axial position (Z), can be obtained. Consequently and according to the definition of local heat transfer coefficient and Nusselt number equations (6), one can derive the following expressions of local heat transfer coefficient and Nusselt number:

$$h_c\,(Z) = -\frac{k}{r_w}\,\frac{1}{\theta_w}\,\frac{\partial\theta}{\partial R}\bigg)_{R=1} \quad \text{and} \quad Nu_c\,(Z) = \frac{h_c(Z)\;d}{k} \tag{16}$$

- ***For evaporation process***

The dimensionless boundary conditions of the governing equations (11 - 14), in case of evaporated film, can be written as:

At the tube wall: $R = 1,\;\; U = W = 0 \;\;\&\;\; \theta = \theta_w$

At the film edge: $R = 1 - \frac{\delta_e}{r_w},\;\; \frac{\partial U}{\partial R} = \frac{\partial W}{\partial Z} = 0 \;\;\&\;\; \theta = 0$ (17)

Solving equations. (11 – 14 and 17), the dimensionless velocity and temperature distributions across the evaporated film at different positions (different values of Z) along the axis can be obtained. In accordance, and referring to equation (8), one can derive the following expression of local heat transfer coefficient and local Nusselt number, in terms of the previously defined dimensionless variables; equation (10) as:

$$h_e\,(Z) = -\frac{k}{r_w}\,\frac{1}{\theta_w}\,\frac{\partial\theta}{\partial R}\bigg)_{R=1} \quad \text{and} \quad Nu_e(Z) = \frac{h_e(Z)\;d}{k} \tag{18}$$

With the aid of equations (16 and 18), one can derive the local overall heat transfer coefficient h, for both inside and outside films, and hence, the associated local and average Nusselt number as:

$$h(Z) = \frac{h_e(Z)\times h_c(Z)}{h_e(Z)+h_c(Z)},\quad Nu(Z) = \frac{h(Z)\;d}{k},\;\text{and}\;\; \overline{Nu} = \frac{1}{L}\int_0^L Nu(Z)\;dZ \tag{19}$$

Where $\overline{Nu}$ is the average Nusselt number of simultaneous condensation-evaporation heat transfer.

2.2 The Used Technique for the Numerical Solution

The dimensionless form of the hydrodynamic and thermal flow fields for both outside and inside films; equations (11-15) are solved, numerically, using finite divided difference technique. According to this technique, the partial differential equations describing the flow are transformed to eight sets of linear algebraic equations. The mesh size is 200 × 200 for condensation and evaporation sides. The relative error in Nusselt number is $\pm$ 0.0001.

Seeking for the linearity of the foregoing equations, one replaces the values of the unknown velocities, appear in some coefficients of these equations, with the values of the corresponding velocities obtained from the previous iteration. The Gauss-Seidel iteration method is applied to solve, numerically, the eight sets of linear algebraic equations. To make this solution possible, the film thickness of both condensation and evaporation must be, approximately, estimated. The temperature profile is assumed to be linear and the heat transfer of evaporation process is due to conduction only. Equations[(20) and (21)] are the thickness of the films of condensation and evaporation. These equations are obtained by neglecting the viscous force in radial direction and the inertia terms appear in the momentum equations (2 - 3):

$$\delta_c(Z) = \frac{1}{r_w}\left[\frac{2\,k\,\mu\,(\Delta T)_c\,z}{\rho_c\,g\,(\Delta\rho)_c\,\pi\,h_{fg}}\right]^{\frac{1}{2}} \tag{20}$$

$$\delta_e(Z) = \frac{1}{r_w}\left[\frac{2\,k\,\mu\,(\Delta T)_e\,z}{\rho_e\,g\,(\Delta\rho)_e\,\pi\,h_{fg}}\right]^{\frac{1}{2}} \tag{21}$$

Where ρ_c and ρ_e are the liquid density at the condition of the outer and inner films; respectively.

A computer program is developed to solve the previously described theoretical model to analyze such present flow.

3. RESULTS AND DISCUSSIONS

The solution of the proposed model of simultaneous condensation– evaporation process on a vertical tube, makes the determination of the present problem variables possible. These flow variables are, such as, the film thickness, velocity distribution, temperature distribution, and local and average Nusselt number. The dimensionless film thickness for both condensation and evaporation sides along the tested tube axis is shown in figure (2). It is observed that, at entrance length up to $Z\sim5$, the dimensionless film thickness for condensation δ_c is small. Thereafter it increases at higher rate as Z increases. This film thickness, generally, is higher for higher Reynolds numbers. On the other hand, the inner film thickness δ_e for all values of Reynolds number, decreases sharply at the entrance region of the tube and then at slower rate in down stream direction. The reason of such behavior may be explained with the aid of the temperature distribution, across both films, shown in figure (3). As Z increases, δ_c increases and this causes a decrease in wall temperature as shown in the figure. This leads to a slowly decrease in δ_e. Also, it is observed that wall temperature decreases with increasing Reynolds number, because the increase of Reynolds number causes an increase in δ_c and consequently the thermal resistance increases.

Heat flux for different axial position is shown in figure (4). For different values of Reynolds number, it has a maximum value near the tube inlet due to that the film thickness for condensation δ_c is small. It decreases sharply until $Z\sim5$ and then it decreases slowly and tends to approach an asymptotic value near the lower end of the tube.

Dimensionless radial and axial velocity profiles inside the condensation and evaporation films at certain axial position $(Z=10)$ for different values of Reynolds number (Re) are shown in figures (5). It is observed that, at the wall; the velocity takes zero value, then the velocity increases with increasing δ for both condensation and evaporation sides. Radial and axial velocity components *(U and W)* have the same behavior.

Nusselt number along the tube axis for evaporation and condensation sides is shown in figures (6). As it is shown in figure (6-a), Nusselt number of evaporation (Nu_e) increases with increasing Z due to the reduction of evaporation film thickness. One the other hand, referring to figure (6-b), condensation Nusselt number (Nu_c) decreases with the increase of axial distance (Z), referring to the equation (16). In addition, Both Nusselt number of evaporation and of condensation decreases with increasing Reynolds number.

Figure (7) shows simultaneous condensation-evaporation average Nusselt number for different values of Reynolds number in laminar flow regime. Average Nusselt number takes its maximum value at the smallest tested value of Reynolds number and then it decreases rapidly for lower values of Reynolds number, thereafter, decreases slowly until it reaches an asymptotic value.

To check the validity of the present model, a comparison between the present results and that the previous results by Yan [5] is made, as shown in figure (8). The examined range of Reynolds number by Yan [5] is very small compared with that of the present work. Although the present results exhibit higher values of *Nu*, for all values of Reynolds number, both of the compares two cases seem to have the same trend. With the aid of the obtained present results, the following correlation is proposed for simultaneous condensation – evaporation on vertical tube in laminar flow regime:

$$\overline{Nu} = 46.7\, Re^{-0.34}\, Pr^{2.6}$$

4. CONCLUSIONS

A theoretical model is proposed for simultaneous condensation and evaporation heat transfer on a vertical tube in laminar flow regime. This model is, numerically, solved using finite difference technique. This model is valid for all vertical tubes whatever it is dimensions. A correlation for the average Nusselt number as a function of Reynolds and Prandtl numbers is proposed.

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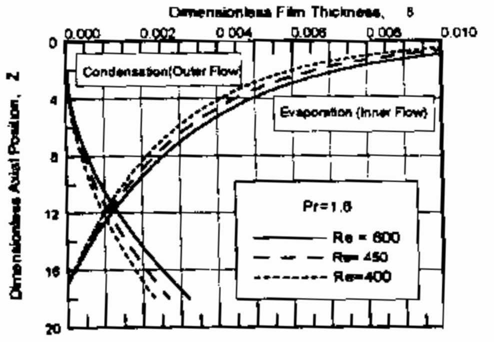


Fig. (2) Dimensionless Film Thickness for Condensation and Evaporation Along The Dimensionless Tube Axis

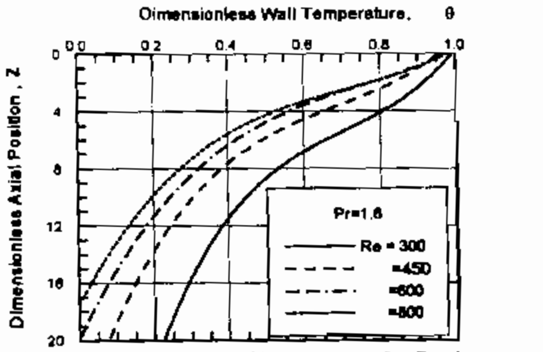


Fig. (3) Dimensionless wall Temperature for Condensation and Evaporation Along Dimensionless Tube Axis

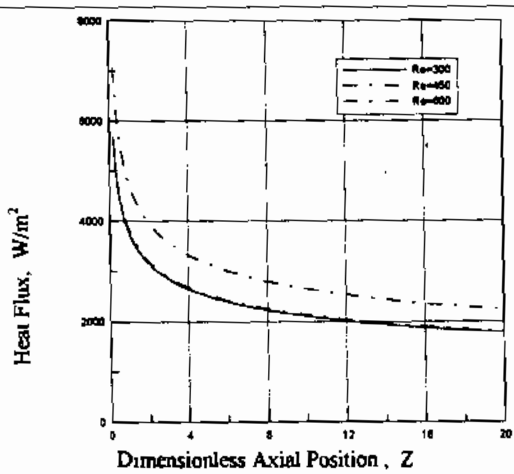
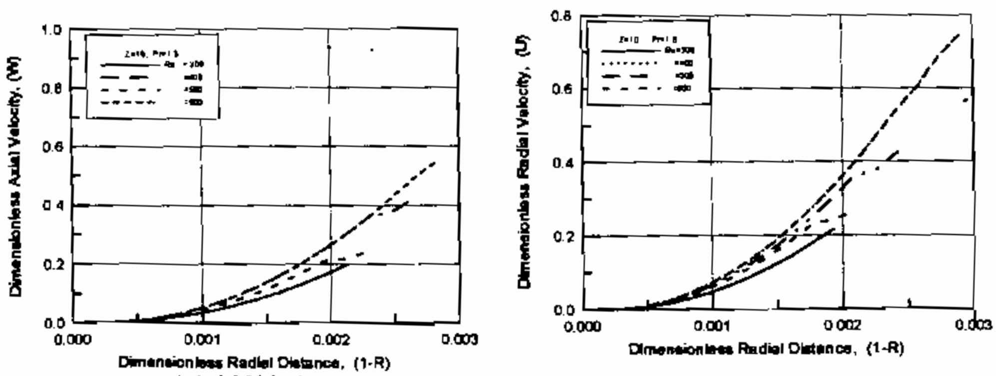


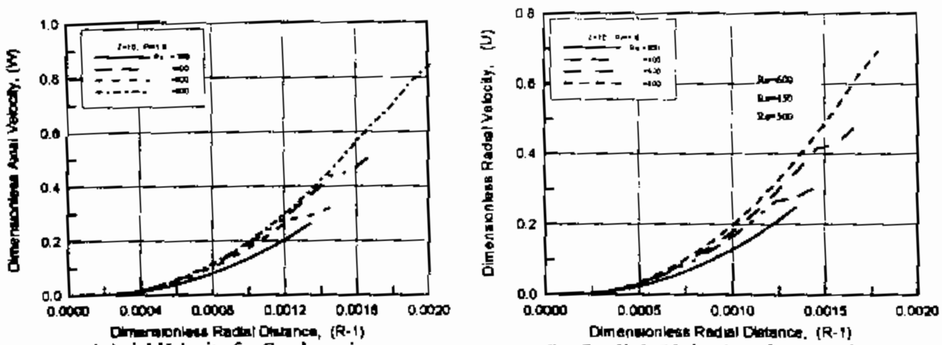
Fig. (4) Heat Flux along Dimensionless Tube Axis



a) Axial Velocity

b)Radial Velocity

Fig.(5-a) Dimensionless Axial and Radial Velocity Profile in Evaporation Film



c) Axial Velocity for Condensation

d) Radial Velocity for condensation

Fig.(5-b) Dimensionless Axial and Radial Velocity Profile in Condensation Film

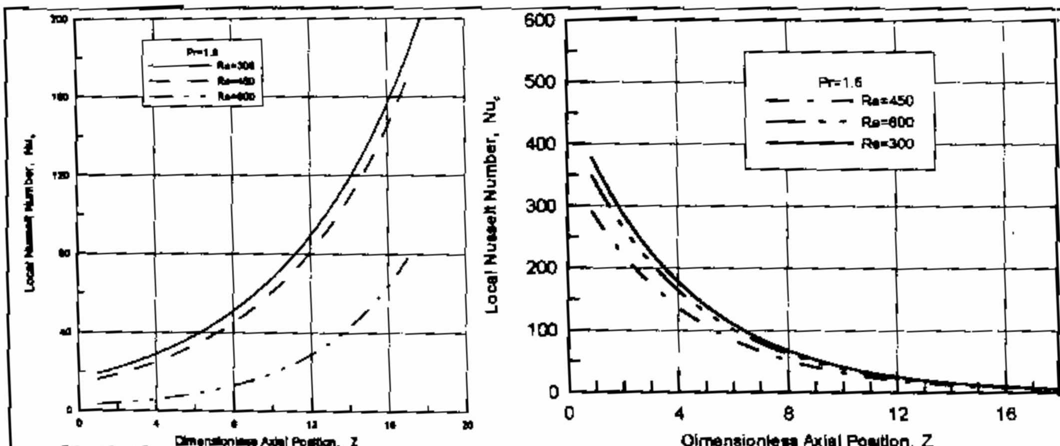


Fig.(6.a) Local Nusselt Number for Different Dimensionless Axial Position Along Tube (in case of evaporation)

Fig.(6.b) Local Nusselt Number for Different Dimensionless Axial Position Along Tube (in case of condensation)

Fig.(6) Local Nusselt Number along The Tube Axis in both Evaporation and Condensation Sides

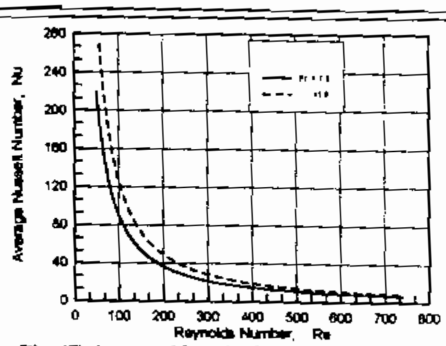


Fig. (7) Average Nusselt Number for Different Reynolds Number

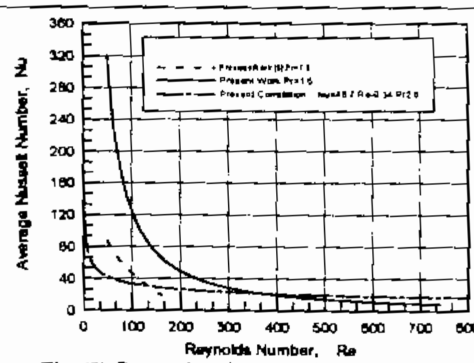


Fig. (8) Comparison between Average Nusselt Number for Different Reynolds Number for Present Study and Yan [5]