THEORETICAL STUDY OF DIRECT CONTACT CONDENSATION OF LAMINAR SHEAR LIQUID FILM

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ABSTRACT

The present work is a theoretical study of the direct contact condensation process of saturated vapor on fully developed subcooled laminar liquid film flowing over thin liquid film on adiabatic and the inclined solid surface .A theoretical model based on momentum , continuity and energy equations . heat balance and thermal energy equation is developed to get approximate solution to describe the condensation performance of vapor on a thin liquid film . The obtained equations of solution are solved numerically using *Runge-Kutta* method and then plotting and the variation of must important parameters such as ; Reynolds , Prandtl , subcooling numbers and shear stress on the values of film thickness layer , bulk temperature , Nusselt number (heat transfer coefficient) and velocity of the flow . The result of variation shows that the major effect parameters is attributed to the Peclelt & Subcooling numbers , while the other parameters is less significant .

Keywords : direct contact condensation ; shear liquid film ; fully developed

الخـــلاصة

الدراسة الحالية تتضمن دراسة نظرية لعملية تكثيف التماس المباشر لبخار مشبع على طبقة رقيقة تامة التشكيل ذات جريان طباقي من السائل والذي يجري فوق سطح صلب مائل ومعزول حراريا . النموذج النظري للدراسة الذي تم تطويره أستند على المعادلات الأساسية للزخم والاستمرارية والطاقة . معادلات الموازنة الحرارية وموازنة الطاقة تم تطويرها للحصول على الحل التقريبي لوصف فعالية وسلوك البخار على الطبقة الرقيقة من السائل . المعادلات التي تم تطويرها للنموذج تم حلها عدديا وتم رسمها وأخذت بالحسبان تأثير أهم المتغيرات مثل عدد رينولد , عدد برانتل واجهاد القص على قيم سمك الطبقة المتاخمة ودرجة الحرارة ورقم نسلت (معامل انتقال الحرارة) و سرعة الجريان . اظهرت استنتاجات الدراسة ان رقم برانتل و رقم التكثيف لهما اكثر تأثيرا من غيرهما من المتغيرات على عملية تكثيف التماس المباشر .

Nomenclature

English symbols

A area [m^2]

b width of the liquid film [m]

C_p specific heat at constant pressure [J kg^{-1} K^{-1}]

Dcc direct contact condensation

g acceleration due to gravity [m s^{-2}]

h heat transfer coefficient [W m^{-2} K^{-1}]

h_{fg} latent heat of evaporation [J kg^{-1}]

k thermal conductivity [W m^{-1} K^{-1}]

$\dot{m}$ mass flow rate [kg s^{-1}]

Nu Nusselt number , hd/k

Pe Peclet number

Pr Prandtl number

q heat flux [W m^{-2}]

Re Ryenold number

S subcooling number

T temperature [C^o]

u velocity in direction od flow [m s^{-1}]

x coordinate in direction of flow [m]

y coordinate normal to the flow [m]

Greek-Symbols

δ liquid layer film thickness [m]

ε thermal diffusivity [m^2 s^{-1}]

μ dynamic viscosity [kg m^{-1} s^{-1}]

ν kinematic viscosity [m^2 s^{-1}]

τ shear stress [kg m^{-1} s^{-2}]

ρ density [kg m^{-3}]

ϖ, λ function defined by equation 20

θ angle of surface [degree]

Subscripts

b bulk

f liquid

o outlet

i Inlet

s vapor , steam

x local

w wall

Superscripts

+ dimensionless

1-INTRODUCTION

Condensation is the heat transfer process by which a vapor is changed in to a liquid by removing the latent heat of condensation . There is four basic types of condensation are generally recognized **[1]**: dropwise , filmwise , direct contact condensation and homogeneous .In the dropwise condensation the drops of liquid form the vapor at particular nucleation sites on a soild surface, and the drop remain separate during growth until carried away by gravity or vapour shear . In filmwise condensation , the drops initially formed and quickly coalesce to produce a continues liquid film on the surface . In direct contact condensation , the vapor condensate directly on the subcooled liquid surface . In homogeneous condensation , the liquid phase forms directly from supersaturated vapor , away from macroscopic surface **[2]** .

1.1 Direct contact condensation

Direct contact condensation process occurs when vapor contact with a moving liquid layer (in subcooled condition) along a solid surface with negligible heat transfer to the solid boundary , the liquid motion may be induced by the body force (e.g. gravity force) or surface forces due to second phase moving (e.g. shear stress) or pressure drop . The phenomenon of Direct contact condensation (**DCC**) has a large interest in industrial application , such as reflux condenser , tubular contractor , in cooling of rocket engines during the work of the last stage of steam turbines , in the chemical engineering industry (e.g. mixing type heat exchanger , degassers , sea water desalting by multiple distillation and by energy conversion application such as geothermal and solar system . In recent years the direct contact condensation has been of major importance in connection with nuclear industry (e.g . pressuize under normal operating conditions) **[3].** Several theoretical and experimental studies about film condensation pheanmena has been studied by many investigators , beginning with **Nusselt [4]** . who investigated the laminar film flow condensation under certain specified assumptions . later this model was modified by adding the contribution of the sensible heat term to a heat transfer coefficient . **Rohsenow [5].** Has included the effect of cross flow on heat transfer (convection in flow direction) within the film . **Hughes and Duffey [6]** introduced a "surface renewal theory" for DCC in turbulent separated flow, which points to an important role of the turbulence in the liquid layer. Experiments and models of DCC in a rectangular duct and rectangular tank were later described by **Lorencez et. al. Ramamurti et. al. [7]** performed a DCC experiment on a thick layer of moving water in the vessel with a stagnant vapour bubble and expressed the heat transfer coefficients in terms of Nusselt number as a function of liquid Reynolds and Prandtl number and the sub-cooling intensity .

2-ANALYTICAL APPROACH

Condensation in two- phase system causes variations in amount and distribution of each phase . this induce variation in the local heat transfer processes due to continuous change for all thermal and hydrodynamic properties . The **DCC** model of this work as shown in **Fig .1** with inlet flow rate $\dot{m}_o$ and subcooled inlet temperature T_o steam with saturation temperature T_s .

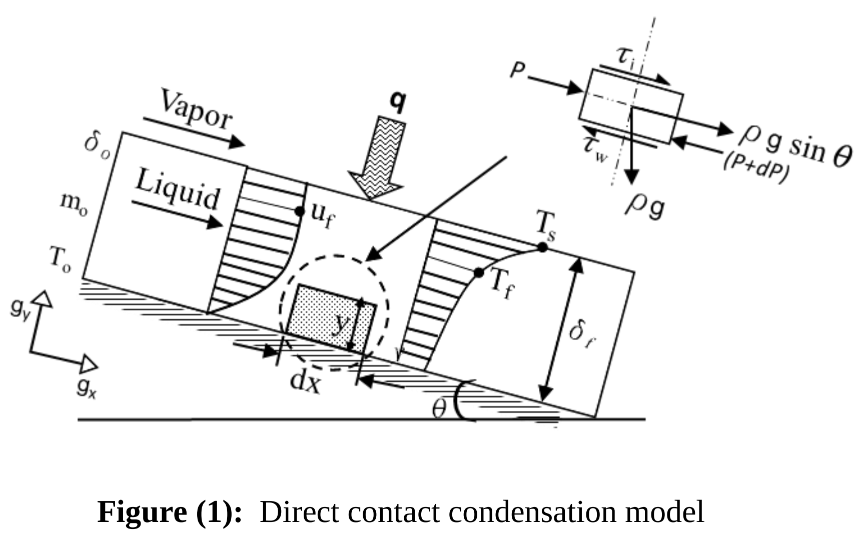

Figure (1): Direct contact condensation model

2.1 Analytic Assumptions

1. laminar and steady state flow of liquid layer
2. adiabatic solid surface
3. liquid inlet temperature was subcooled while the steam is in saturation temperature .
4. the gravity and shear stress are means of driving to liquid layer.
5. any instabilities or wave which may be present due to steam up flow are neglected .

2.2 Heat transfer model

The heat balance for model in **Fig .2**. can be described by term of latent heat of condensation vapor and sensible heat of interface as **[9]** :-

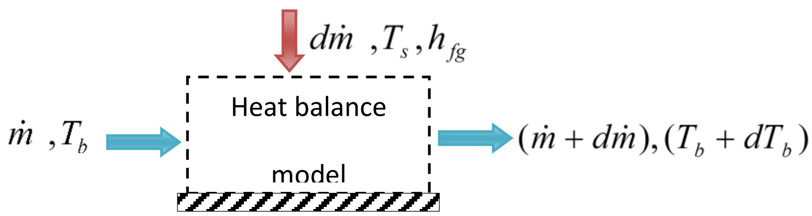

Figure (2): Heat balance control element model

Heat balance can be written as **[9] :**

$$d\dot{m}\ h_{fg} + d\dot{m}\ C_p\ T_s + \dot{m}\ C_p T_b = C_p(\dot{m}+d\dot{m})(T_b + dT_b) \qquad \text{.................................(1)}$$

$d\dot{m}\ C_p\ dT_b$: is neglected because it is small in comparison with value of (h_{fg})

Than the balance equation becomes as :

$$d\dot{m}\ [h_{fg} + C_p\ (T_s - T_b)] = \dot{m}\ C_p\ dT_b \qquad \text{...(2)}$$

For more convenient equation (2) can be written using dimensionless form , with [4]

$$x^+ = \frac{x}{\delta_o} \qquad \text{..(3a)}$$

$$\dot{m}^+ = \frac{\dot{m}}{\dot{m}_o} \qquad \text{..(3b)}$$

$$T_b^+ = \frac{T_s - T_b}{T_S - T_o} \qquad \text{...(3c)}$$

Equation (2) becomes :

$$\frac{dT_b^+}{dx^+} = \frac{1}{\dot{m}^+}\left[\frac{1}{S} + T_b^+\right]\frac{d\dot{m}^+}{dx^+} \quad \text{...}(4a)$$

Where ,

$$S = \frac{C_P(T_s - T_o)}{h_{fg}} \quad \text{...}(4b)$$

2.3 Hydrodynamic model

Equation (4a) cannot be integrated without evaluation of the $\dot{m}^+$, so we will use momentum and continuity equations with the model in Fig.1. to find the velocity distribution from the force balance on a segment of liquid flow

$$\tau = \left(\rho g \sin\theta - \frac{dP}{dx}\right)(\delta - y) + \tau_i \quad \text{..}(5a)$$

The shear stress at the wall is :

$$\tau_w = \left(\rho g \sin\theta - \frac{dP}{dx}\right)(\delta - y) + \tau_i \quad \text{..}(5b)$$

and

$$\tau = \mu\frac{du}{dy} \quad \text{...}(5c)$$

Using equations (5a) , (5b) & (5c)

$$\mu\frac{du}{dy} = \left(\rho g \sin\theta - \frac{dP}{dx}\right)(\delta - y) + \tau_i \quad \text{..}(6)$$

Integrating with following boundary conditions :

$$\tau_w = \mu \frac{du}{dy}\bigg|_{y=0} \qquad u = 0|_{y=0}$$

equation (6) can be written as :

$$u(y) = \frac{1}{\mu}\left(\rho g \sin\theta - \frac{dP}{dx}\right)\left(\delta y - \frac{y^2}{2}\right) + \frac{y\tau_i}{\mu} \quad \text{...(7)}$$

Equation (7) is velocity distirbution equation , for diminsionless becomes :

$$u^+ = \frac{\tau_w^+}{2\delta^+}\left[2\delta^+ y^+ - \left(1 - \frac{\tau_i}{\tau_w}\right)\left(y^{+2}\right)\right] \quad \text{...(8)}$$

Where ;

$$\tau_w^+ = \frac{\tau_w\,\delta_o}{\mu\,u_o}$$

The axial mass flow rate ($\dot{m}^+$) is given :

$$\dot{m}^+ = \int_0^{\delta^+} u^+ dy \quad \text{...(9)}$$

By integration , equation (9) becomes as :

$$\dot{m}^+ = \frac{\tau_w^+\delta^+}{3}\left[2 + \frac{\tau_i}{\tau_w}\right] \quad \text{...(9a)}$$

By derivative equation (9 a) ,

$$\frac{d\dot{m}^+}{dx} = \frac{\tau_w^+\delta^+}{3}\left[2 + \frac{\tau_i}{\tau_w}\right]\frac{d\delta^+}{dx} \quad \text{...(9b)}$$

Therefore substitution of equations (9 a) & (9 b) into equation (4 a) ,

$$\frac{dT_b^+}{dx^+} = -\left[\frac{12(1+S)}{(2+\frac{\tau_i}{\tau_w})S\tau_w^+\delta^{+3}}\right]\frac{d\delta^+}{dx^+} \quad \text{...(10)}$$

Integration equation (10) with boundary conditions at dimensionless :

$$T_b^+ = 1 \quad , \delta^+ = 1 \quad at \quad x^+ = 0$$

$$T_b^+ = 1 - \left[\frac{6(1+S)}{S\tau_w^+(2+\frac{\tau_i}{\tau_w})}\right]\left[\frac{\delta^{+2}-1}{\delta^{+2}}\right] \quad \text{...(11)}$$

It is appeared that T_b^+ cannot be determined from equation (11) . since there is no other relation to determine the δ^+. The following approach proposed to estimate δ^+ is considered to be the main point characterizing of this work . this was done through estimation of a new solution for δ^+ using a specific form of the energy equation instead of the heat balance equation used previously [4].

2.4. Energy Equation

The appororiate energy equation has the following form :

$$u(y)\frac{dT}{dx} = \alpha\frac{d^2T}{dy^2} \quad \text{...(12)}$$

It should be noticed here that the left hand side of the above equation is a function of x . so the assuming is done to simplification the equation as :

$$\frac{\partial T_{(x,y)}}{\partial x} \cong \frac{dT_{(x)}}{dx} \cong \frac{dT_b}{dx} = f(x) \quad only$$

Using the non dimensional form :

$$u^+ = \frac{u}{u_o} \quad \text{...(12a)}$$

$$T^{+}=\frac{T_s-T}{T_S-T_o} \quad \text{...}(12b)$$

and

$$Pe=\frac{u_o\delta_o}{\alpha} \quad \text{..}(12c)$$

Equation (12) becomes :

$$\frac{\partial^2 T^{+}}{\partial y^{+2}}=u^{+}Pe\,\frac{dT_b^{+}}{dx^{+}} \text{..}(13)$$

Substituting for u^+ from equation (8) in equation (13) lead to :

$$\frac{\partial^2 T^{+}}{\partial y^{+2}}=Pe\,\frac{dT_b^{+}}{dx^{+}}.\left[\frac{\tau_w^{+}}{2\delta^{+}}\left(2\delta^{+}y^{+}-\left(1-\frac{\tau_i}{\tau_w}\right).y^{+2}\right)\right] \text{.....................}(14)$$

In order to find T^+ equation (14) is integrated with following boundary conditions for (insulation surfaces) :

$$\frac{\partial T^{+}}{\partial y}=0 \quad at \quad y^{+}=0$$

$$T^{+}=0 \quad at \quad y^{+}=\delta^{+}$$

Equation (14) becomes :

$$T^{+}=Pe\,\frac{dT_b^{+}}{dx^{+}}.\left[\frac{\tau_w^{+}\delta^{+3}}{24}\left(4\left(\frac{y^{+3}}{\delta^{+3}}-1\right)-\beta\left(\frac{y^{+4}}{\delta^{+4}}-1\right)\right)\right] \text{...............}(15)$$

Where $\qquad \beta=\left(1-\frac{\tau_i}{\tau_w}\right)$ ....(15a)

In order to find the temperature T^{+}_{b} that defined by equation (3c) the definition of the bulk temperature may be used :

$$T_b^+ = \frac{\int_0^{\delta^+} u^+ T^+ dy^+}{\int_0^{\delta^+} u^+ dy^+} \quad \text{...(16)}$$

Now , substitution of equations (8) ,(10) and (15) in to equation (16) with integration result in

$$T_b^+ = \frac{3Pe(1+S)}{2(2+\frac{\tau_i}{\tau_w})} \frac{d\delta^+}{dx^+} \left[\frac{\beta^2}{21} - \frac{\beta}{5} + \frac{3}{5} \right] \text{.......................................(17)}$$

Now , comparison of equation (17) that result from energy equation with equation (11) that result from heat balance to find the value of $\frac{d\delta^+}{dx^+}$

$$\frac{d\delta^+}{dx^+} = \frac{\left[1 - \frac{6(1+S)}{S.\tau_w^+ \left(2 + \frac{\tau_i}{\tau_w} \right)} \right] \left[\frac{\delta^{+2} - 1}{\delta^{+2}} \right]}{\frac{3}{2} \left[\frac{Pe(1+S)}{S\left(2 + \frac{\tau_i}{\tau_w} \right)^2} \right] \left[\frac{\beta^2}{21} - \frac{\beta}{5} + \frac{3}{5} \right]} \quad \text{..................................(18)}$$

Equation (18) will solved numerically using *Runge-Kutta method* [10 ,11] in order to calculate the thickness δ^+ for any value of x^+ .

2.5. Heat transfer coefficient calculation

The local heat transfer coefficient may be obtained from heat balance

$$h_x = \frac{h_{fg}}{b.(T_s - T_b)} \frac{d\dot{m}}{dx} \quad \text{..(19)}$$

Used equations (11) ,(9a) & (18) with dimensionless form both h_x calculated as :

$$h_x = \frac{\varpi.\tau_w^+.\delta^+.\lambda}{\frac{3}{2}\left[\frac{Pe(1+S)}{S.\lambda^2}\right]\left[\frac{\beta^2}{21}-\frac{\beta}{5}+\frac{3}{5}\right]} \quad \text{.......................................(20)}$$

Where $\varpi = \left(\frac{h_{fg}.\dot{m}_o}{\delta_o.(T_s - T_o)}\right)$, $\lambda = \left(2 + \frac{\tau_i}{\tau_w}\right)$

And the heat transfer coefficient in the form of Nusselt number is obtained :

$$Nu_x = \frac{h_x.b}{k} \quad \text{...(21)}$$

$$Nu_x = \frac{\dfrac{\varpi.\tau_w^+.\delta^+.\lambda}{\frac{3}{2}\left[\frac{Pe(1+S)}{S.\lambda^2}\right]\left[\frac{\beta^2}{21}-\frac{\beta}{5}+\frac{\beta}{5}\right]}.b}{k} \quad \text{.................(21)}$$

2.6. Calculation Algorithm

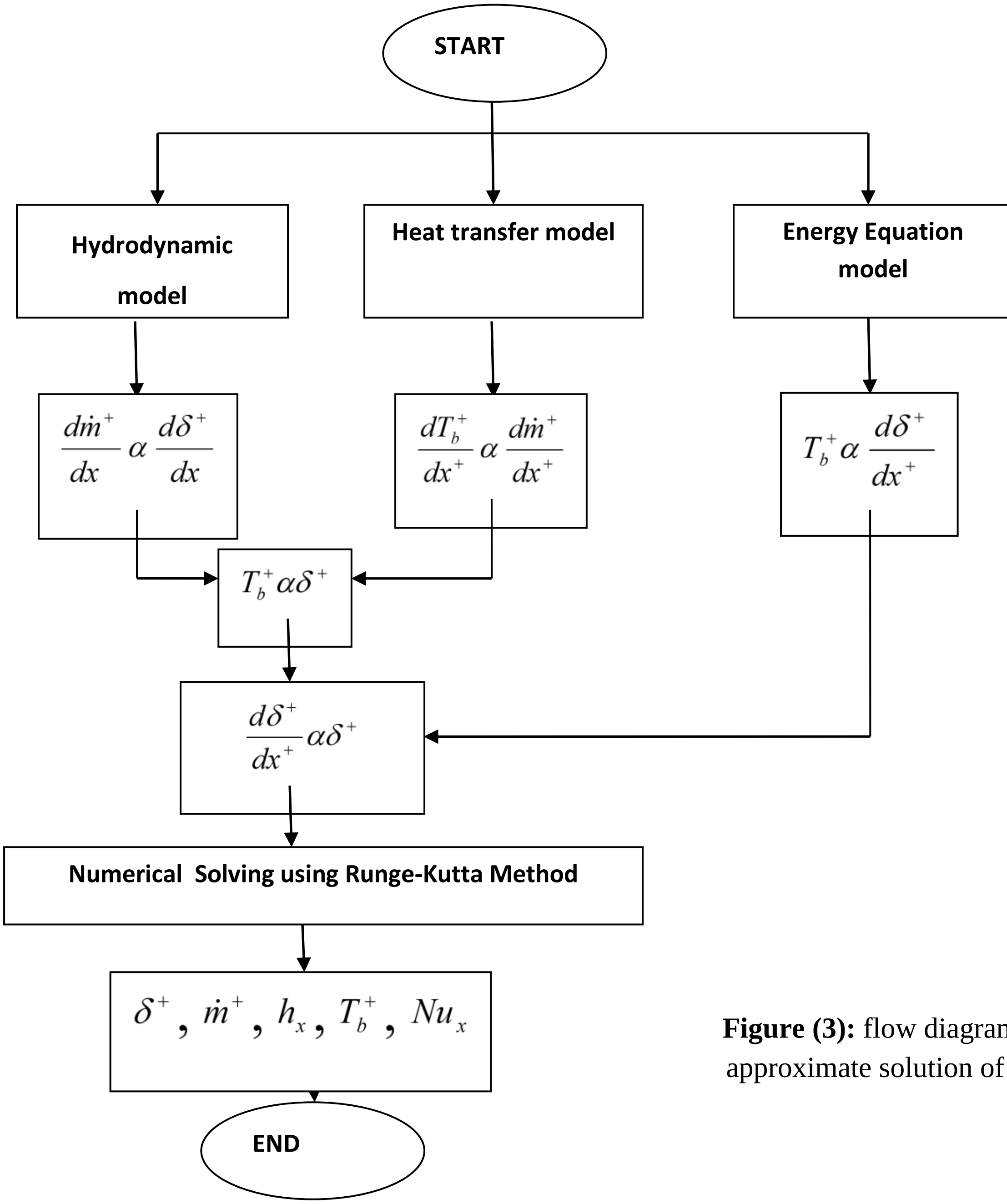

Figure (3): flow diagram of a approximate solution of study

3. Result & discussion :

In order to investigate the direct contact condensation process the obtained equation are solved and the important parameters of this process suh as , bulk temperature , local film thickness and local nusselt number in the flow direction , the evalution of these parameters is essential for design and application of the **DCC** systems . the effect of some parameters like Reynold , prandtl , subcooled and shear stress is shown in the following points :-

1- <u>Effect of Reynold Number</u>

Fig.4. shows of dimensionless thickness (δ^+) against the axial distance it is clear from figure increasing of (δ^+) with increment of (x^+) due to the continuous condensation at the liquid – vapour interface . Increasing of Reynold number value and fixed others parameters Pr , shear stress ratio and **S** lead to decrease the values of film thickness due to increase of liquid velocity and that mean less time to transfer heat with vapour at the interface surfaces.

Fig.5. graph between the dimensionless bulk temperature against axial distance , the observed decreasing in bulk temperature with axial distance due to bulk temperature is (T_b , T_s , T_o) that mean in the difference (Ts-To) between vapour and liquid when is small that mean faster and good heat transfer in interface surface . so the increase of Reynold number value lead to large (Ts-To) because of high velocity in liquid flow .

Fig .6. graph between the local Nussult number and axial distance , can observed from figure that Nu_x is increasing with distance until reach to value near to constant due to heat transfer coefficient (h_x) the decreasing in Nu_x or (*hx*) with increasing of Reynold number is due to increasing of velocity and that lead to increase of heat transfer resistance additionally there is no turbulent flow the only laimnar flow is available .

2- <u>Effect of Peclet Number</u>

Fig.7. drawn for various values of Peclet number while other parameters are kept constant , the effect of variation of Peclet number on (δ^+) shows a significant decrement in (δ^+) with increment of Peclet number , this due to increasing in velocity due to increasing of Reynold number (Pe=Re *Pr) and there is another reason may changing of liquid properties because of Pr changes (Pr = n/ a) .

Fig. 8. Shown the effect of various values Peclet number on bulk temperature , is clear from the plot that bulk temperature decreasing with decrease of peclet number value because the difference (Ts-To) with value of (Pe=1000) has rapid decreasing and that mean T_b decrease too .

3- <u>Effect of Subcooling Number</u>

The influence of changes in subcooling number **S** on the liquid thickness is illustrated in **Fig .9.** this figure indicates that the decrease in **S** is accompanied by a decrease in (δ^+) this is due to the corresponding decrease in the heat transferred through the free liquid surface . other parameters which may have an effect on (δ^+) are related to the means of driving the liquid layer .

Fig.10. explain that increasing in subcooling number means small increasing of bulk temperature due to increase the difference (Ts-To) where ($S = \dfrac{C_P(T_s - T_o)}{h_{fg}}$) and that lead to increasing of T_b .

Fig.11. shows the variation of subcooling number vs. axial distance and clear from the figure the increasing of subcooling number leads to increasing of difference (Ts-To) due to increase (**S**) and that lead to increase of bulk temperature and means decrease in heat transfer coefficient due to large difference (Ts-To) and that accompanied with decease in Nux value due to (*Nux= f(hx)).*

4- Effect of Shear stress ratio Number

The effect of increasing shear ratio (interface shear to wall shear stress) leads to increase of liquid layer due to increase the friction in interface surface between the vapor and condensate liquid due to exposed to more heat transfer area .

Fig . 12 . plotting between bulk temperature against axial distance , as it shown the increase of shear ratio leads to decrease the bulk temperature due to increase heat transfer rate at the interface surface because of frication forces and that can be seen also in **Fig.13.** of increasing of Nusselt number for same reason mentions above .

Fig . 14. Shows the relation between film thickness layer (δ^{+} /y) vs. velocity profile , as natural behavior the velocity increase when we far from the wall . so , the increase in shear ratio leads to decrease of velocity due to friction between two surfaces .

4. CONCLUSIONS

The main points which can be drawn from above analysis and discussion are :

(1) An adequate solution from the direct contact condensation was obtained .

(2) The solution considers only a few parameters controlling the process – Reynold number – the Subcooling number – the Peclet number and the shear ratio .

(3) The major effect on the liquid layer thickness is atttributed to the Peclelt & Subcooling numbers , while the other parameters is less significant .

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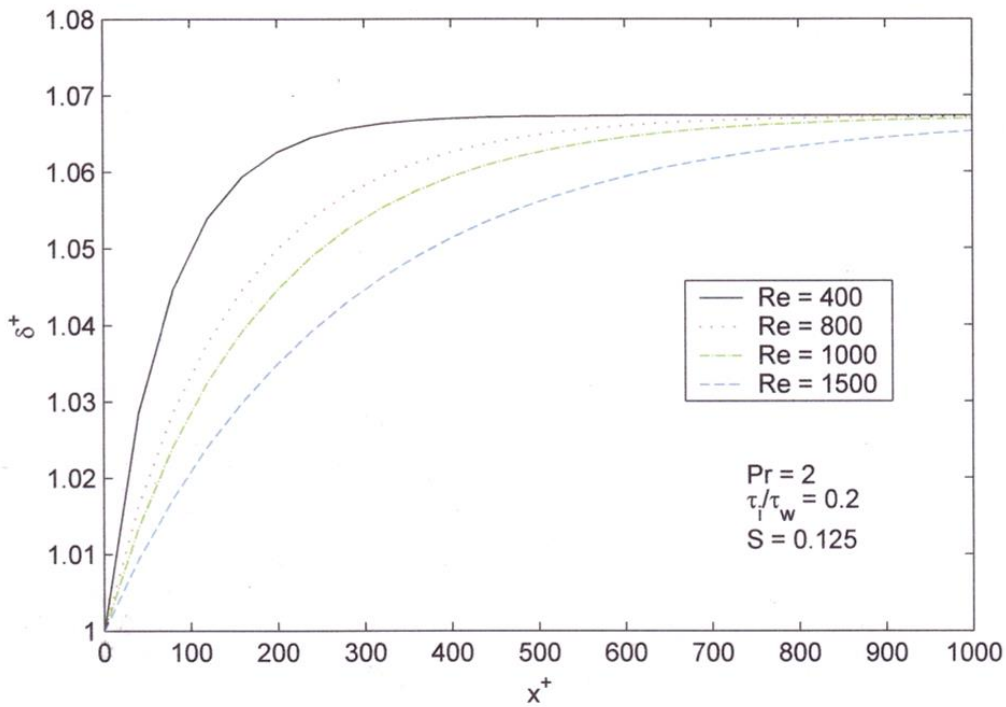

Figure (4): Effect of Reynolds number on Film Thickness

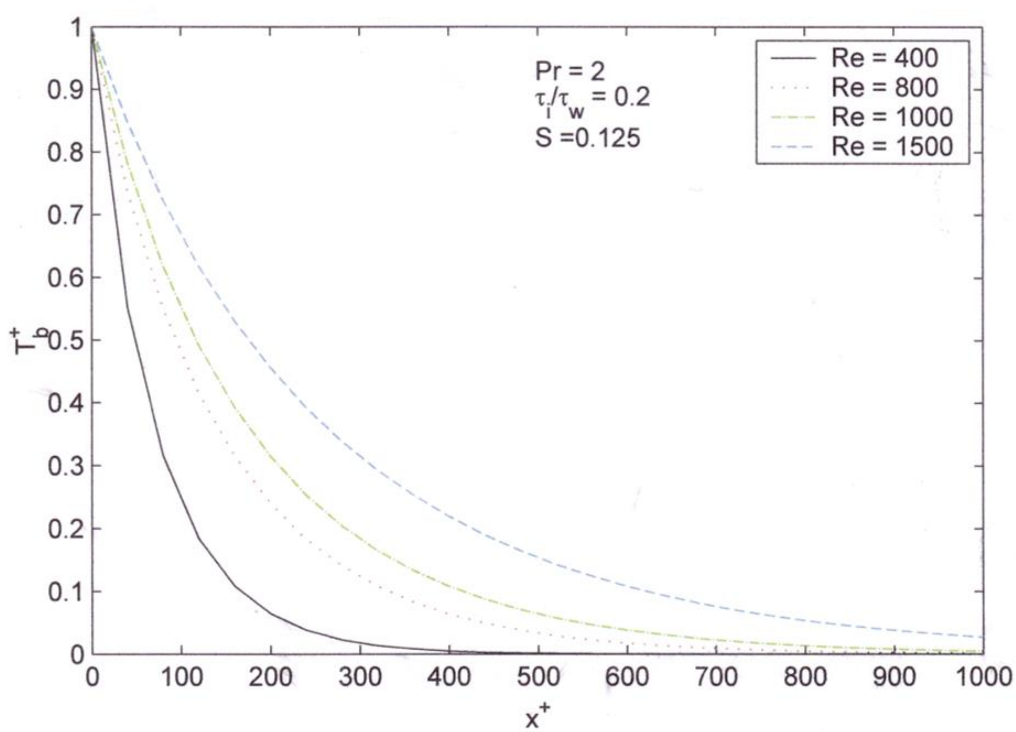

Figure (5): Effect of Reynolds number on Bulk Temperature

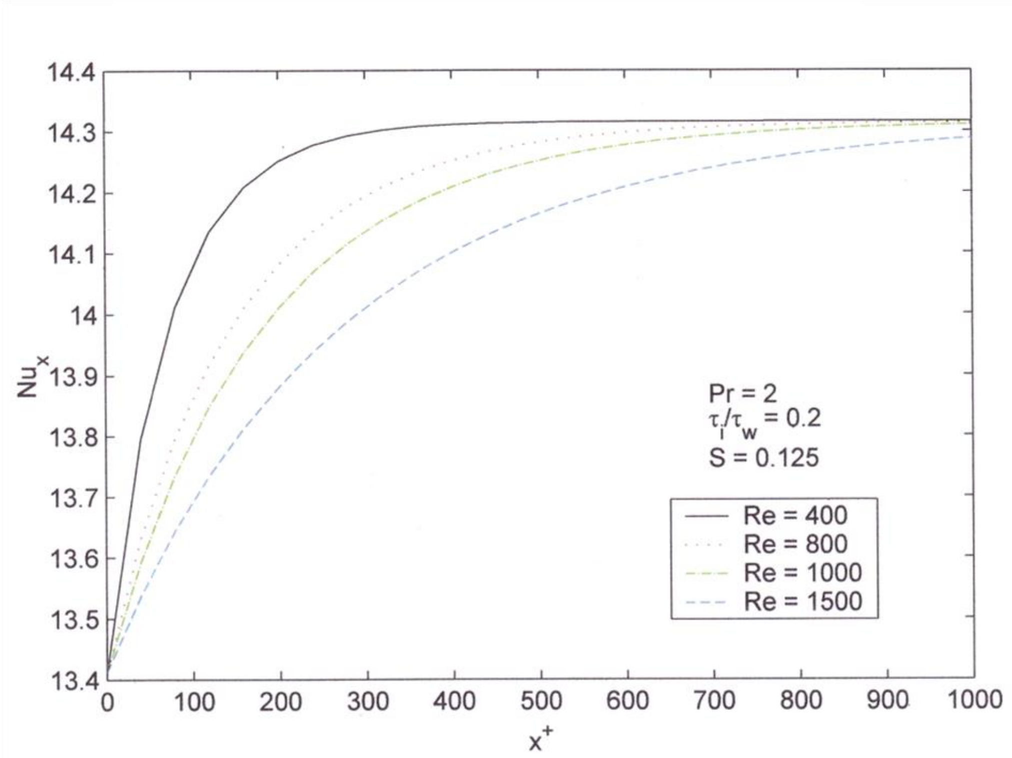

Figure (6): Effect of Reynolds number on Nusselt

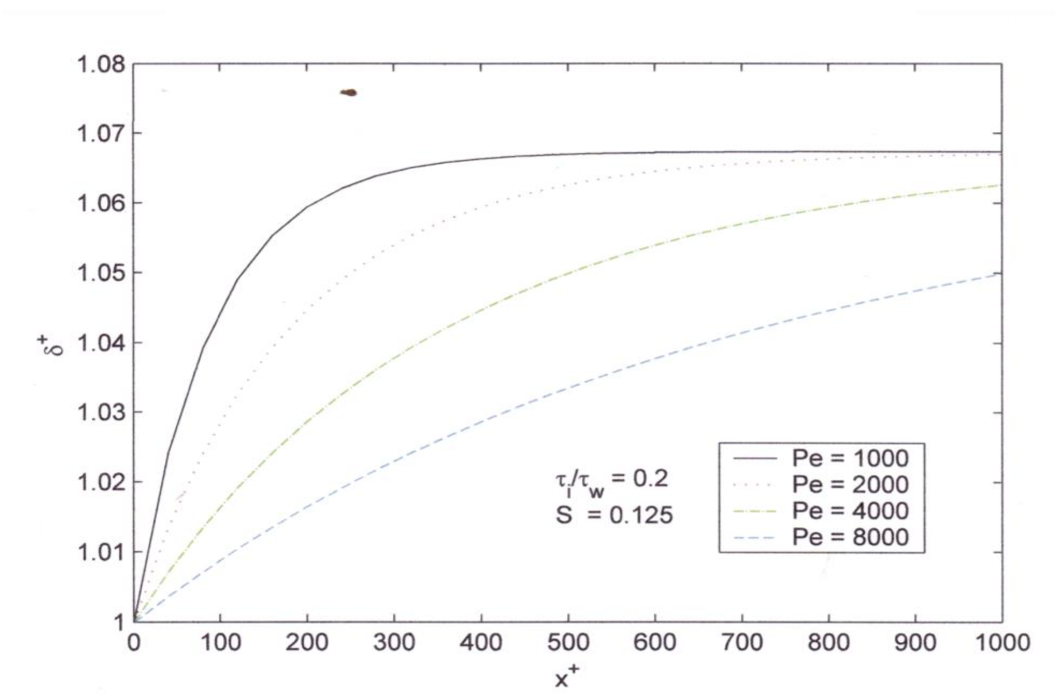

Figure (7): Effect of Peclet number on Film Thickness

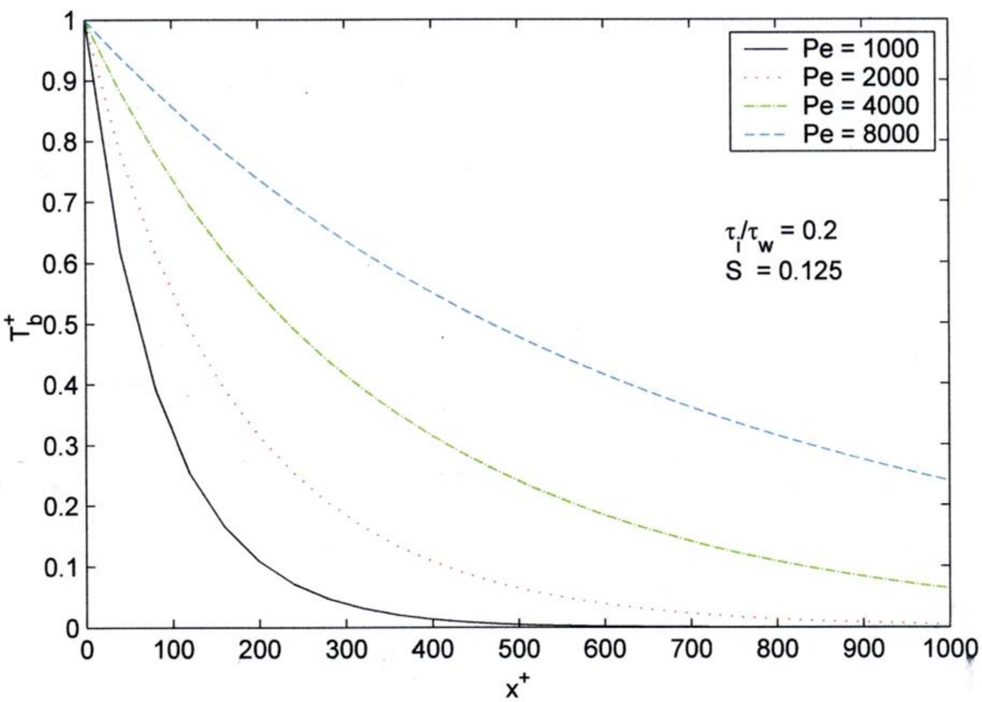

Figure (8): Effect of Peclet number on Bulk Temperature

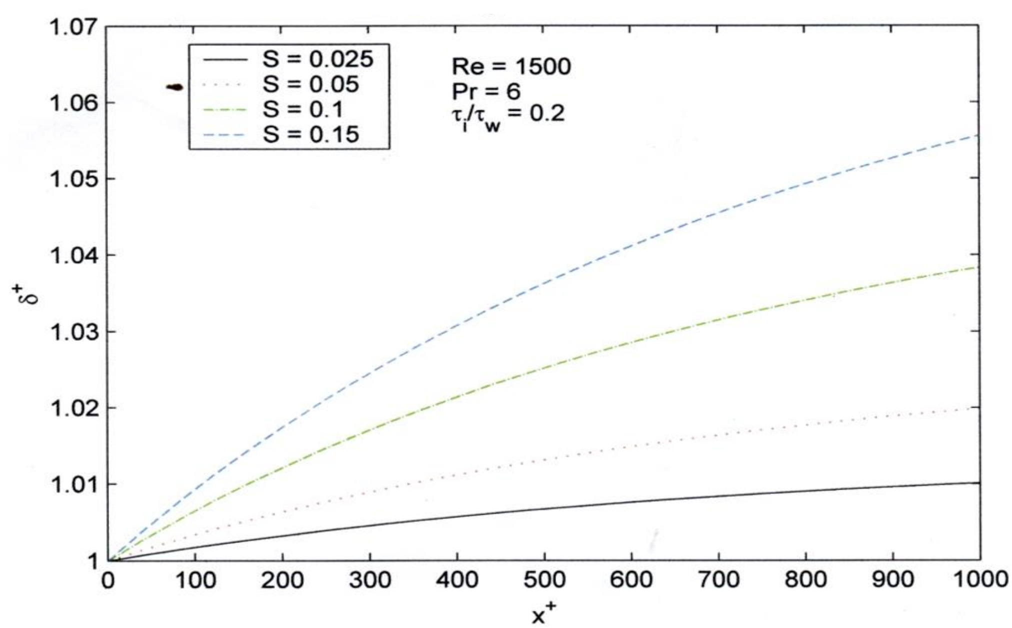

Figure (9): Effect of Subcooling number on Film Thickness

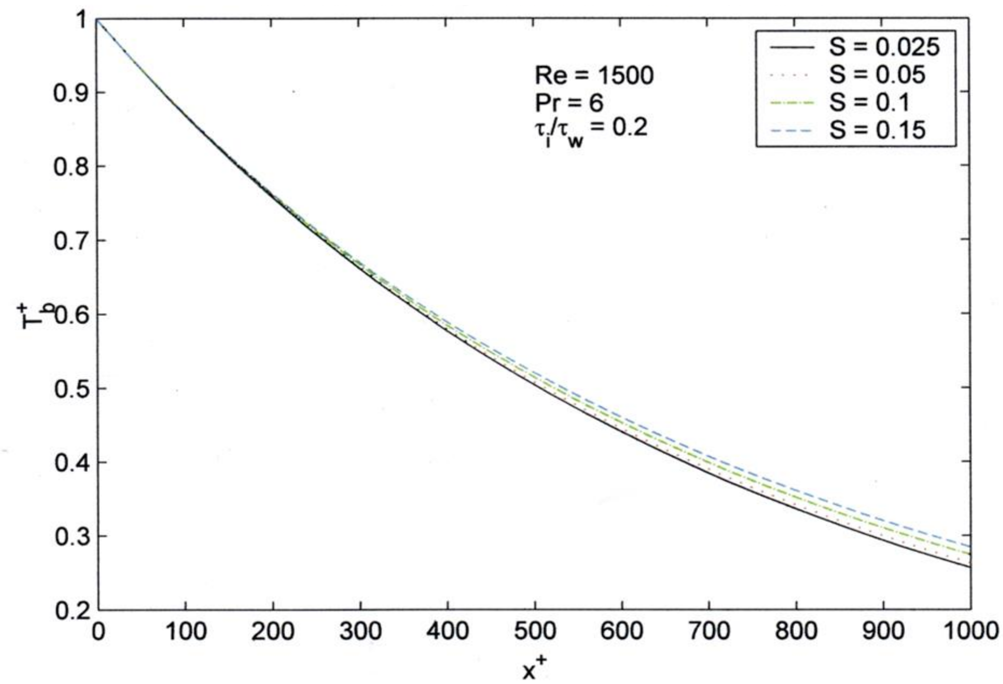

Figure (10): Effect of Subcooling number on Bulk Temperature

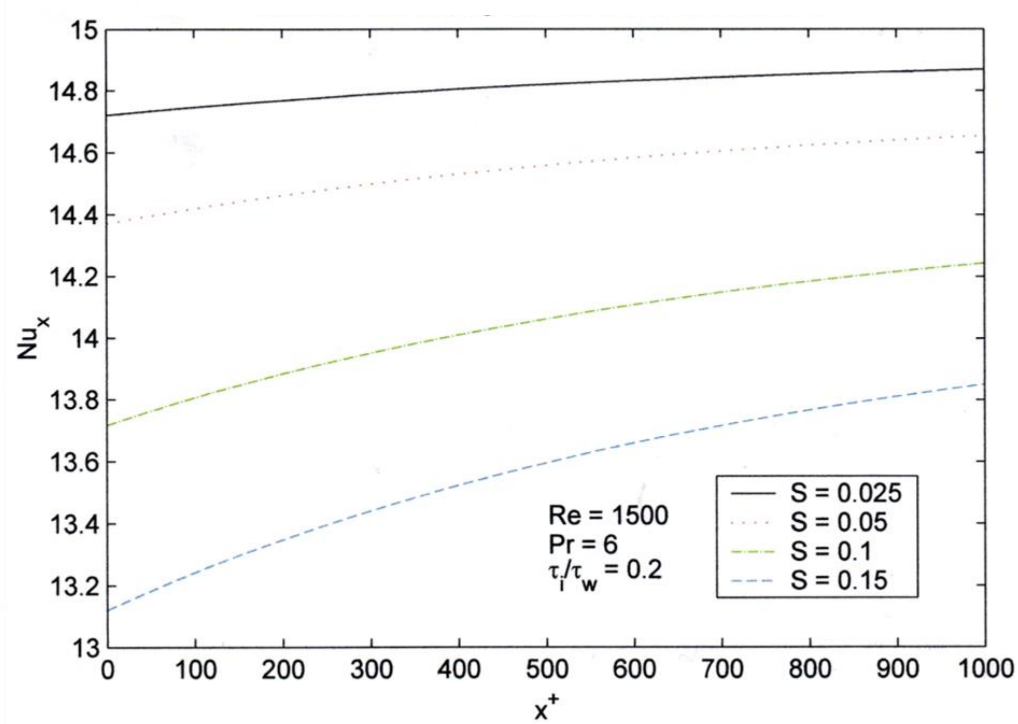

Figure (11): Effect of Subcooling number on Film Nusselt Number

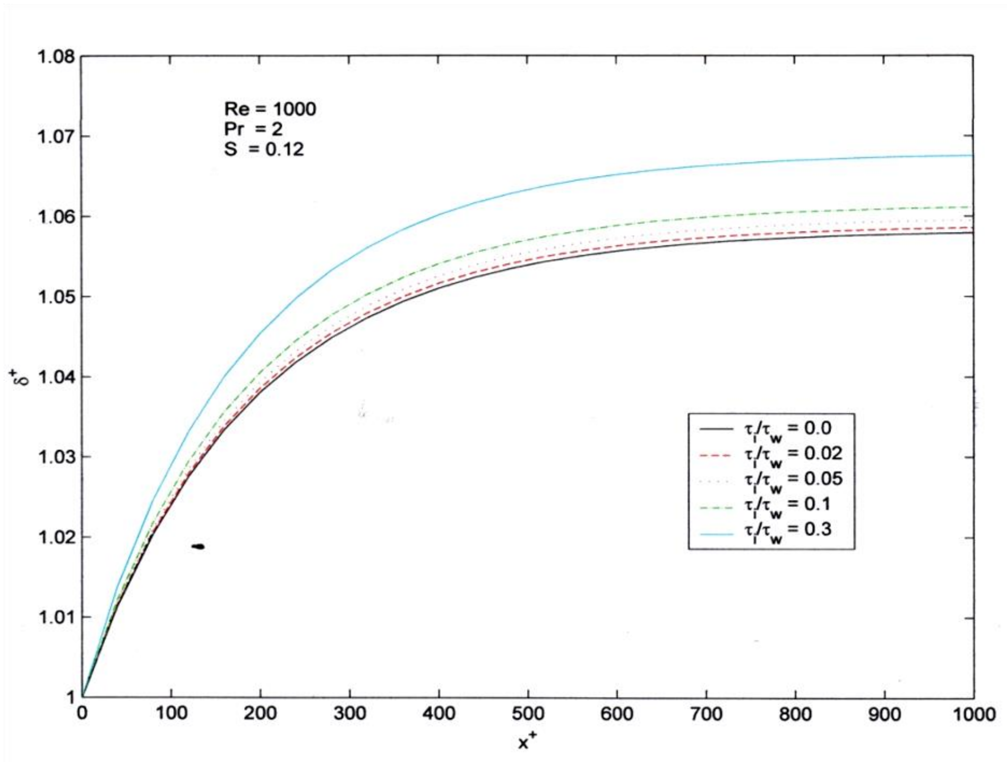

Figure (11): Effect of shear stress ratio number on Bulk Temperature

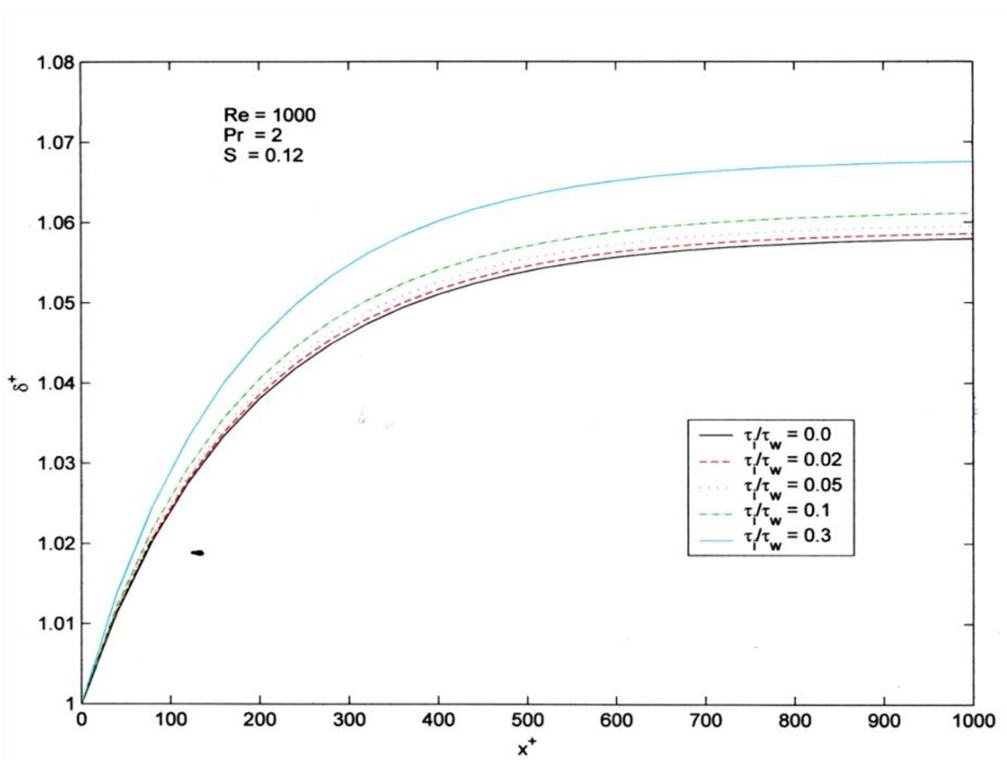

Figure (12): Effect of shear stress ratio on Film Nusselt

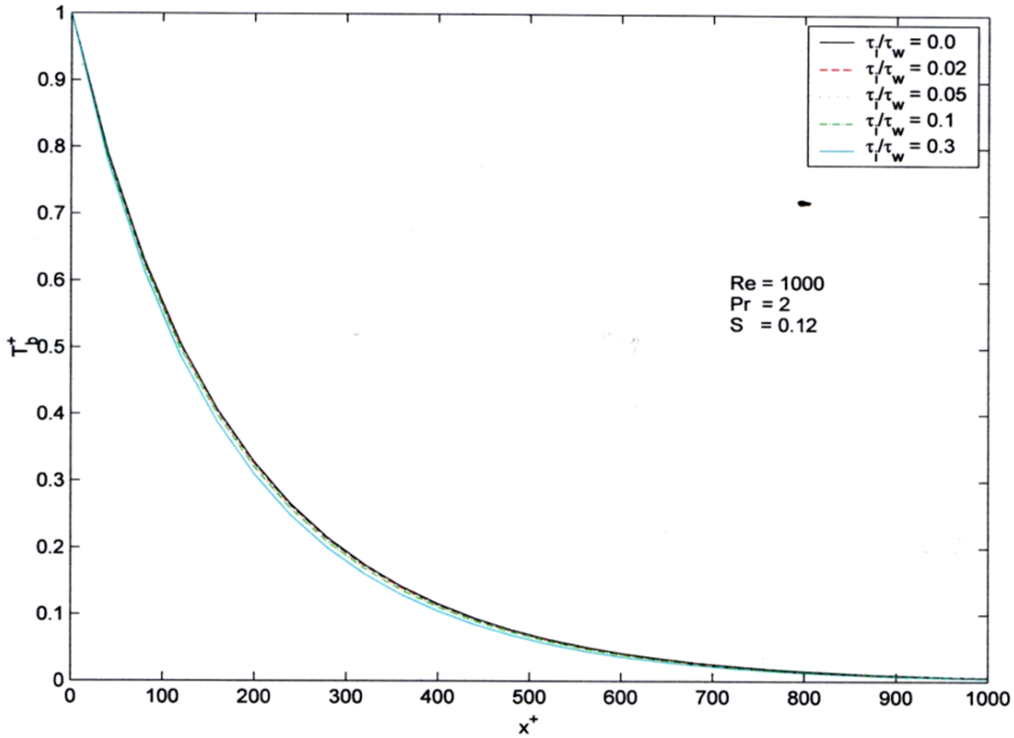

Figure (13): Effect of shear stress ratio on Bulk

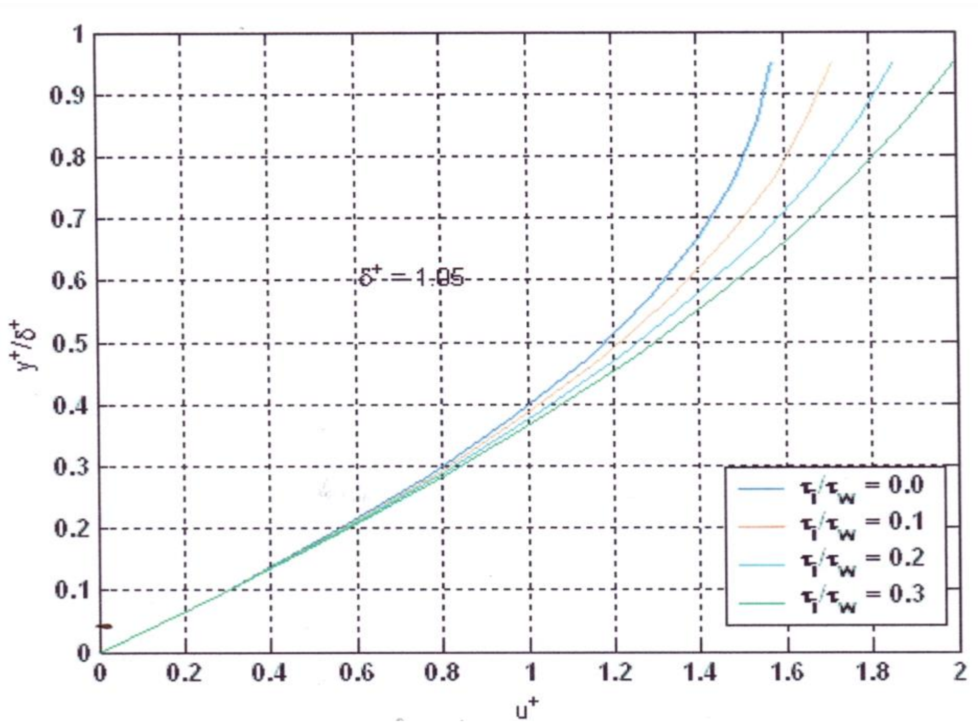

Figure (14): Effect of shear stress ratio on velocity profile