An Integral Model for Prediction of Condensation Heat Transfer in Presence of Non-condensable under Forced Flow over a Flat plate

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This is to certify that the thesis entitled "**An Integral model for prediction of condensation heat Transfer in presence of non-condensable under forced flow over a flat plate**", which is being submitted by **Nilojendu Banerjee** in partial fulfilment of the requirements for the award of the degree of "**Master of Nuclear Engineering"** at the School of Nuclear Studies and Application, Jadavpur University, Kolkata700032, during the academic year 2018-2019, is the record of the student's own work carried out by him under our supervision.

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The foregoing thesis entitled "**An Integral model for prediction of condensation heat Transfer in presence of non-condensable under forced flow over a flat plate**" is hereby approved as a creditable study of an engineering subject carried out and presented in a satisfactory manner to warrant its acceptance as a prerequisite for the degree of "**Master of Nuclear Engineering"** at the School of Nuclear Studies and Application, Jadavpur University, Kolkata700032, for which it has been submitted. It is understood that by this approval the undersigned do not necessarily endorse or approve any statement made, opinion expressed or conclusion drawn there in but approve the thesis only for the purpose for which it is submitted.

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DECLARATION OF ORIGINALITY AND COMPLIANCE OF ACADEMIC ETHICS

It is hereby declared that the thesis entitled "**An Integral model for prediction of condensation heat Transfer in presence of non-condensable under forced flow over a flat plate**" contains literature survey and original research work by the undersigned candidate, as part of his degree of "**Master of Nuclear Engineering"** at the School of Nuclear Studies and Application, Jadavpur University, Kolkata700032.

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It is also declared that all materials and results, not original to this work have been fully cited and referred throughout this thesis, according to rules of ethical conduct.

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Abstract

Due to high heat transfer characteristics, condensation process finds its applications in many industrial processes such as power plants, refrigeration cycles, air-conditioning systems etc. In various physical circumstances this process always involves some amount of non-condensable gas as one of its phase. Due to non-diffusive nature of such gas, during condensation of pure vapour it accumulates near vapour-liquid interface region and creates a barrier to the heat transfer mechanism.

In this thesis, the non-condensable gas effect on condensation process over a flat horizontal isothermal plate has been investigated. An integral model is developed with the help of boundary layer approximation method to analyze the continuity, momentum, energy equations for both liquid and mixture and one species conservation equation for gas phase along with proper boundary and interfacial conditions. A set of ordinary differential equations are formed with the help of these equations. Using MATLAB these equations have been solved and the solutions are used to extract the various heat transfer parameters to analyze the actual situation. The results are validated with some performed experimental study [6]. From these results it was found that inlet gas concentration plays an important role on heat transfer characteristics. As the non-condensable gas mass fraction increases, it creates an additional interfacial resistance which significantly decreases the heat transfer rate. Similarly increasing free stream mixture velocity increases heat transfer taking other parameters as constant. Also for a fixed inlet gas concentration and inlet velocity, heat transfer coefficient decreases with rise in bulk temperature difference.

Key words: film condensation, non-condensable, interface, integral model,

Table and Contents

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Nomenclature

C_p	: Specific heat (J $\mathrm{kg}^{-1}\mathrm{K}^{-1}$)
D_m	: Mass-diffusion of mixture ($\mathrm{m}^2\mathrm{s}^{-1}$)
h_{fg}	: Latent heat due to condensation (J kg^{-1})
h_x	: Local heat transfer coefficient($\mathrm{Wm}^{-2}\mathrm{K}^{-1}$)
J	: Mass flux at interface between mixture and liquid film (kg $\mathrm{m}^{-2}\mathrm{s}^{-1}$)
Ja	: Jacob number
k	: Thermal conductivity (W $\mathrm{m}^{-1}\mathrm{K}^{-1}$)
L	: Length of the plate (m)
Nu	: Nusselt number
Pr	: Prandltl number
Pe	: Peclet number
P	: Pressure (bar)
q_x	: Local heat transfer (Wm^{-2})
R_v	: Gas constant (J/Kg .K)
M	: Molecular weight
Sc	: Schmidt number
T	: Temperature (K)
ΔT	: $(T_\infty - T_w)$
w	: Gas mass fraction
x	: Axial co-ordinate direction (m)
y	: Transverse co-ordinate direction (m)

Re : Reynolds number

Greek symbols

δ : Boundary layer thickness (m)

μ : Dynamic viscosity (N s m^{-2})

ρ : Density (kg m^{-3})

υ : Kinematic viscosity ($\mathrm{m}^{2}\mathrm{s}^{-1}$)

α : Thermal diffusivity ($\mathrm{m}^{2}\mathrm{s}^{-1}$)

η : Non-dimensional distance

Subscripts

∞ : Free stream condition

av : Average

c : Concentration

g : Gaseous phase

i : Liquid- mixture interface

l : Liquid

m : Vapour-gas mixture

nc : Non-condensable

sat : Saturation

t : Thermal

v : vapour

Chapter 1

Introduction

1.1 Condensation mechanism and its uses

Condensation process is very important thermo physical phase change process in nature. Due to high heat transfer characteristics this process finds its applications in refrigeration, air-conditioning system, many power engineering devices etc. In case of power plants like, nuclear, coal, and other plants, condensing steam at particular pressure and temperature is very important aspect regarding recycling of water and power production.

There are two forms of condensation processes observed in nature:

- Dropwise condensation
- Filmwise condensation

1.2 (a) Dropwise condensations

Dropwise condensation is generally shown when condensate does not wet the surface and some kinds of droplets are formed. In heat transfer point of view it gives much higher heat transfer rate compare to film condensation. In practical scenario it is found that heat transfer rate for dropwise condensation of pure steam at one bar pressure is 20 times more than that of film condensation. High heat transfer rate is achieved in this case because of offering less resistance to the surface. So dropwise condensation is desirable in case of industrial applications, but it is not easy to achieve such type of condensation. Special type of surfaces with non-wetting agents which is known as dropwise promoters are required to enhance this type of condensation.

1.2 (b*)* Filmwise condensations

In case of film condensation, the surface over which condensation initiates is wet-able in nature and as a result a thin condensate film is formed. This type of condensation process is commonly encountered in nature compare to dropwise condensation. Formation of film over the surface offers more resistance compare to dropwise condensation. As a result heat transfer rate drops significantly which is main disadvantage encounters in almost all industrial applications.

1.3. Applications

Due to its immense importance in thermodynamic and heat transfer field, condensation process finds its applications in many industrial devices for many years.

One of its realistic applications observed in heat exchanger design. The most common application is condenser used in refrigeration and air-conditioning systems. Also heat exchanger especially condenser is one of the essential part in energy sectors. In case of conventional coal based power plants, condensation of steam coming out from turbine is an integral part of work production from power cycle.

In case of nuclear based power plants condensation plays an important role for safety measurement systems. Water cooled reactors such as Boiling Water Reactor (BWR), Pressurized Water Reactor (PWR), one of the main accidental scenario that is observed is Loss Of Coolant Accident (LOCA). In such types of accidents, water vapour is released into reactor contaminant due to less water circulation and integrity of contaminant can be seriously threatened. Vapour of high temperature and pressure over pressurized the contaminant vessel. To maintain the pressure into a reasonable level steam has to be condensed as fast as possible. Various types of systems exist to provide rapid water vapour condensation in such situations. Contaminant water spray systems, Passive Contaminant Cooling Systems (PCCS) are examples of such processes.

In practical situation it is found that steam always contains some amount non-condensable gases such as air, helium etc. These gases play a negative role on transfer phenomenon. Even small amount of non-condensable gases strongly decrease condensation heat transfer. Due to heavy and impermeable nature, accumulation of these gases occure near the liquid-mixture interface region causing barrier to the heat transfer. As a result condensation rate is significantly decreased.

So the distribution and variation of non-condensable with vapour is the most important aspect in all industrial applications and safety systems. The main objective is to reduce the percentage of such gases from vapour –gas mixture to enhance heat and mass transfer processes but the most challenging task is to analyze such phenomenon due to its complexity. Justification of which effects are predominant in actual case is rather difficult task. No such concrete and convincing studies and experiments are performed to analyze all of the effects of such process even till today.

1.4. Literature review

This section contains a concrete study of literature survey of condensation mechanism in case of binary mixture of condensable and non-condensable gases which has been discussed in chronological order.

Sparrow et al [1] performed an analytical investigation of force convection condensation over a plate of constant temperature from vapour and non-condensable mixture. Liquid and mixture phase governing boundary layer equations have been solved using similarity method. Results illustrate that degradation of heat transfer is much lower compare to natural convection condensation case. This is because free stream velocity dictates increase of heat transfer coefficient as well as rate of heat transfer across the liquid-vapour interface region.

Sage et al. [2] performed an analytical investigation regarding vapour condensation along with non-condensable gases on a vertical surface. In this model they have considered quiescent flow of bulk mixture. This is because of considering only the free convection effect. For greater Lewis number the mixture phase behaves as supersaturated. In their analytical model they have used boundary layer equations for the individual species conservations for gas and liquid phase. In their analysis they have found that the effect of natural convection is dominant in case of a high molecular weight non-condensable which rises heat transfer through liquid-gas interface even for high Schmidt number. The rate of supersaturation in the boundary layer is determined by the Lewis number of gaseous mixture .For lower Lewis number of mixture, the boundary layer approaches to saturation along with increasing wall temperature. On the other hand for greater Lewis number, boundary always remains supersaturated. Also in case of ternary mixture like nitrogen-methane-water, the component for which the Schmidt number is higher will accumulate near the interface with respect to the bulk concentration.

Rose [3] performed an analytical investigation for condensation of mixture of vapour-gas flowing over the flat plate and in the normal direction in case of a horizontal tube. Theoretical equations are derived to analyze the physical situation for different free stream velocities, mixture non-condensable gas compositions and surface temperatures of the condenser. The equations are corrected for minimum and maximum condensation rate. For different value of Schmidt numbers, results related to flat plate and horizontal cylinder closely match with past numerical solutions for various range of applications.

Siddique et al. [4] studied condensation of steam in presence of air and helium separately in a vertical pipe geometry. According to their analysis it is observed that the heat transfer coefficient is a strong function of helium-air concentration ratios. If this ratio is nearly about unity, dominant effect of helium is observed on decrement of heat flux. On the other hand if the molar fraction of such gases tending to one, the air side dominance is observed.

Huhtiniemi et al. [5] have performed an experiment to study the surface orientation effect for steam condensation in presence of non-condensable gas. In their experiment they directed a mixture of steam and gas inside a rectangular duct over a condensing aluminum surface which is surface finished. In their successive experiments the aluminum surface orientation is varied from 0-90^0 and by doing so air-stream mass fraction is also varied from 0-0.87 with a mixture velocity 1-3 m/s. From their experimental study they have shown that the variation of heat transfer coefficient from 100 to 600 W/m^2K associated with 0.87-0.24 mass fraction variation. For pure vapour, maximum of value heat transfer coefficient is observed. Changing the position of the plate from horizontal to vertical position decreases heat transfer significantly. Similarly mixture velocities, non-condensable concentrations and system pressure directly affect the heat transfer rate.

Kang et al. [6] conducted an experimental investigation to study the non-condensable gas effect on condensation heat and mass transfer process. They also have analyzed this phenomenon in presence of wavy interface. For performing experiment they have used a plate which is nearly horizontal (4.1^0 w.r.t horizontal) which is made up of aluminum. In their investigation they have injected a wavy water film of steady thermal condition. The main objective is to observe the wavy interface effect on the condensation process. It was found from their analysis small percentage of non-condensable gas such as air can significantly decreased the heat transfer process across the smooth interface.

Park et al. [7] performed experiments to observe the wavy interface effect on pure steam condensation and steam with specific amount of non-condensable in case of a vertical wall.

Overall heat transfer coefficient, film thickness with respect to distance were obtained for different parameters such as non-condensable mass fraction (0.1-0.7),bulk mixture velocities and Reynolds number of condensate film (0-19,000).The experiment that they have conducted consisted of a long test section, auxiliary equipment, vapour supplying unit, condensate film supplying loop, to control the film Reynolds number and a cooling water loop which is provided to decrease the latent heat of condensation. From their experiment they have found that the local Nusselt number is significant function of condensate Reynolds number. In case of vapour consisting air overall heat transfer coefficient decreases and this decrement is a strong function of air concentration. It is clearly observed from their study increasing wave amplitude in the interface region increases overall heat transfer rate which directly affects condensation process.

Liu et al. [8] investigated the condensation heat and mass transfer through liquid vapour interface for vertical parallel plate geometry. They have performed transient simulations inside a small two dimensional parallel plates. VOF method is used to analyze the vapour liquid interfacial behavior. To keep the interface sharp Piecewise-Linear-Interface (PLIC) model is employed. This means two fluids interface is linear in nature in each control volume. The governing equations (continuity, momentum, and energy) and VOF equations are solved explicitly with presence of proper source term. For calculating the source terms related to phase change process, they used their developed source code. The effect of surface tension is also considered in this model. In two phase separation region, the surface tension plays a dominant role to minimization of surface energy. In this modeling they have used CSF approach to calculate interfacial surface tension force with the help of interfacial curvature (grad of volume fraction of liquid) and pressure across the interface.

According to their numerical results it is found that a thin laminar film is formed at the topmost portion of the wall. A wavy regime is observed as the film moving downward. This happens due to the combined effect of shear and gravity. These are playing very important role regarding heat and mass transfer through interface as condensation occurs. The condensation rate sharply increases with increase of vapour to wall temperature difference. The thickness of boundary layer decreases as inlet velocity increases. for a higher inlet velocity , the effect of wavy nature in the interface leads to a higher heating resistance .Several research have been done experimentally to determined wave properties values in laminar as well as turbulent flows . It has been shown experimentally wavy region appears even well defined laminar flow due to surface irregularities and other resisting parameters. This result strongly influences the heat and mass transfer across the interface.

Siow et al [9] developed an analytical model for condensation in case of declining parallel plate geometry in presence of non-condensable gas. The results related to different declination angles have been analyzed for different Froud numbers. Full forms of governing equations for both phases have been solved using this model. Local velocities, temperatures and gas mass fractions variations for different axial locations are obtained to analyze the actual flow, heat and mass transfer behaviors. Also they have incorporated the effects of various declination angles for different free stream conditions.

Siow et al [10] investigated the non-condensable gas effect on condensation in case of an internal flow. Full forms of governing equations are used to analysis of such flows. Developed model includes the two dimensional governing equations of both air and vapour phases. Fully coupled approach is used to solve this model. The results generated from these solutions give a fair idea about the effect of non-condensable on condensation process. The accuracy of these results is very good in case of high inlet gas concentration which directly affects condensation process.

Chen et al [11] have developed an analytical model which predicts the laminar film condensation of downward steam-air mixture flows around a horizontal tube of circular cross section. Complete two phase two dimensional governing equations have been solved using boundary layer approximation method. The numerical results show the variation of film thickness and heat transfer with different free stream conditions. In their analysis they have assumed that mixture at free stream is fully saturated with vapour corresponding to .local partial vapour pressure, wall is isothermal in nature, all the properties are uniform in case of both phases, surface tension effect at liquid-vapour interface is negligible and film thickness is very much thin with respect to tube radius. Fully coupled approach is used to solve the two dimensional conservation equations. Their results predicts fair values of heat transfer coefficients in case of pure vapour and vapour associated with non-condensable gas. They have also found that decrease of wall temperature significantly deteriorate the heat transfer rate.

Du et al. [12] have investigated the condensation mechanism on vertical surfaces covered with metal foams taking gravity into account. They have established a numerical simulation in case of film condensation mechanism in a metallic foam in which nonlinear temperature distribution was solved in details. They have also studied the porosity and pore density effect on the film thickness. The domain extension method is used outside the film layer. Governing equations are solved using simple algorithm and convective equation is solved using discretized power law model. The predicted results show that existence of foam near the downstream deviates the linearity of temperature distribution significantly. Metal foams create a resistance barrier to the t heat transfer across the interface. The film thickness becomes thinner in case of low permeability by either increasing or decreasing porosities.

Li [13] have studied water vapour condensation for turbulent flow in case of vertical cylinder condenser tube. Full form of governing equations with proper boundary conditions are used to analyzed the actual situation. To account turbulent behavior they have used $K - \epsilon$ model. For solution of above equations they have used CFD technique with commercial software ANSYS FLUENT. From their analysis first they have successfully implemented CFD simulation of heat and mass transfer in the gas mixture and heat transfer of coolant flowing inside the cooling channel of vertical condenser tube. The results clearly show that average axial velocity decreases with increase in water-vapour condensation and this decrement is very rapid in nature. Also it is observed that gas mixture density increases along gas-liquid interface. For low Reynolds number at the inlet section of the tube, the axial velocity at the interface is larger compare to average velocity of the mixture.

Das et al. [14] developed a model which predicts the effect of film boiling over a vertical geometry taking mixed convection into account. The developed model is valid over wide rangr of applications under different parameters. By achieving this, they have used boundary layer integral method to solve the various conservations equations for both of liquid and vapour phases. The cooling of liquid due to forced as well as natural convection and radiation effect between plate and the interface have been discussed in details. Also they have analyzed different types of instability effects on boiling process.

Tang et al. [15] have developed a mathematical model of double boundary layer to analyze the effect of film condensation mechanism on the outer surface of a horizontal tube in presence of air. This model is appropriate to solve simultaneous equations of mass, momentum, energy and diffusion to simulate the condensation heat transfer. They have mainly used dimensionless transformation which plays an important role to solve the boundary layer equations. Normalized velocity, temperature, concentration profile of both mixture and liquid layer and at interface have been obtained using finite difference method. It has been seen even in small amount of non-condensable can strongly deteriorate the heat transfer process.

Li and Peng [16] have investigated laminar film condensation in presence of non-condensable over a horizontal tube. They developed a mathematical model using double-layer and Prandlt boundary layer theory. For calculation they have mainly used C++ program with appropriate flow chart. In their paper they have mainly focused how gas-liquid separations influences heat and mass transfer phenomenon along liquid mixture interface. In their paper three important cases have been discussed for different values of mixture free stream velocity, such as both gas-liquid film non separating, gas film separating with non-separated liquid film and gas film separating with separated liquid film. It has been found that separation of gas-liquid film strongly influences condensation heat and mass transfer. Separation of this film causing increase in heat transfer coefficient which will ultimately lead to enhance vapour-gas condensation heat transfer.

Qiujie et al. [17] have numerically done their research on condensation of water vapour with air over a vertical plate of constant temperature considering air as non-condensable gas. They have selected inlet mass fraction for water vapour as 0.05 to 0.5. The analysis has done using VOF of fluent software. They have defined a UDF function which is useful for calculating all the properties of air and water vapour as a function of temperature. They have simulated this type of model by generating two interfaces, liquid water and vapour gas interface. The calculation results from the condensation simulation that they have used find reasonable agreement with the experimental data of Dehbi [18] and Tagamia[19] .The calculation is based on second order upwind scheme with $K-\in$ model. The predicted results show decrement of condensation an adverse effect on tangential velocity. Normal velocity is increasing close to the wall because of the concentration gradient. In the similar way they have observed that decrement of mass fraction along the flow direction of air steam mixture with respect to the wall. Mass fraction increases normal direction to the wall. It can also be seen that higher the concentration of air smaller the absolute velocity. The reason for this as the air concentration increases, condensation rate is smaller as a result driving force related to condensation and normal velocities are reduced. In the similar way it is found that as the thickness condensate film is higher, it creates an additional resistance and due to this driving potential of the condensation is decreased which lower the value of heat transfer coefficient as well as rate of condensation.

Punetha and Khander [20] have investigated complex phenomenon such as effect of non-condensable gas like air, helium or other gas spices on condensation heat transfer of pure vapour. The main objective is to analyze this mechanism in case of nuclear reactor safety when Loss Of Coolant Accident (LOCA) is taken place. To study this phase change phenomenon they have used CFD approach using Ansys CFX commercial software. The thermodynamic and hydraulic mechanism of such phenomenon is solved using full form of governing equations of both liquid and mixture phase. Pressure is coupled with mass and momentum. Buoyancy term is also included with momentum term of mixture and solved with respect to a reference density. Turbulence effect is modeled with the help of $k - \omega$ shear stress transport model. They have not used any wall function to predict the accurate heat transfer near wall region. Developed model works well for higher concentration of non-condensable gas, typically higher than 6-8% which is very important in case of contaminant structure analysis. For large scale of helium air mixture extension, diffusion mixture of lighter component is more dominant compare to buoyancy force. These trends are also seen for large scale experiments of steam condensation in presence of air helium mixture.

Wu et al. [21] studied analytical investigation to get approximate equations for film condensation in presence non-condensable gases over a horizontal tube and a vertical plate. Their study develops a theory which gives a simplify solution of such complex phenomenon. They have used thermodynamic and heat transfer analysis to relate interface behavior with free stream parameters. The developed theory predicts well the decrement of heat flux with increase of non-condensable gas in case of both condensations over horizontal tube as well as vertical plate. In addition, their analysis also shows that non-condensable plays more dominant role in case of free convection.

Datta et al. [22] numerically investigated vapour bubble condensation within a subcooled liquid. They have used volume of fluid method to analyze this phenomenon. Their works mainly deal with modeling of interfacial jump effect on heat and mass transfer processes. Comparison of interfacial jump approach and proposed empirical model help to understand the condensation mechanism in terms of collapse rate and other heat transfer parameters.

Datta et al. [23] performed condensation induced water hammer (CIWH) phenomenon in a horizontal pipe geometry. The effect of different inlet parameters such as inlet water temperature, steam-water section pressure difference, superheating of steam and quality were analyzed on above mentioned process. Their work show that location peak value of pressure is significantly influenced by opening of different vales. Also this investigation revel that CIWH process could be avoided by superheating of wet steam.

1.5. Scope of work and organization of the thesis

The literature survey reveals that there are significant numbers of experimental and numerical studies associated with condensation of vapour non-condensable gas mixture. Most of these observations show that these works are mainly performed in case of vertical geometries, some ducts etc. As such there is a lack of comprehensive analysis for film condensation of binary mixture of vapour and non-condensable gases in case of horizontal geometries. Moreover all of these works are either experimentally or numerically performed with the help of CFD analysis. These techniques are very time consuming and also complex in nature.

In this present work, the geometry under consideration is horizontal plate. A generalized analytical approach known as "Boundary layer integral method" is used here to observe the various effects of non-condensable gas on condensation under forced flow condition with different gas mass fractions, free stream mixture velocities and degree of sub cooling.

In this chapter, a general outline regarding this thesis is presented. Second chapter contains the boundary layer integral formulation to analyze the non-condensable gas effect on condensation. In third chapter, the model validation with experimental data and the prediction of various results are presented and analyzed in details. Final chapter represents the conclusion of the work that is being presented in this thesis.

1.6. Objective:

The main objective of the present study is to analyze the effects of non-condensable gas on film condensation over a flat surface. In case of nuclear studies condensation heat transfer in presence of non-condensable is a primary concern in passive systems used in advanced plants to confirm the inherent safety such as the Passive Contaminant Cooling System (PCCS), the Isolation Condensing System (ICS), Passive Heat Removal System (PHRS) in various reactor cores.

The working behaviors of these types of systems based on heat exchanger which is connected with a pool located over the contaminant. In these systems condensation heat transfer in the outer surface of the tubes plays a major role in main heat transfer mechanism. Non-condensable gases can be present during accidental conditions. The increase of non-condensable gas concentration at the interface between liquid and gas reduces the condensation heat transfer rate due to poor thermal conductivity of gas. As a result performance of heat exchanger deteriorates significantly along with heat removal capacity in accident conditions and impact on plant safety. Many experimental and numerical researches have been done to analyze these types of behavior but still lot of unexplored zones makes this phenomenon very complex as well as interesting.

In the present study we have focused on the nature of heat and mass transfer in the presence of non-condensable gas such as air mixed with vapour in pure form on a horizontal flat plate maintained at constant surface temperature. Full forms of governing equations for both mixture and liquid phases have been used with proper boundary layer approximation. Different profiles related to velocity, temperature and concentration for both phases have been assumed across each layer. Finally with the help integral analysis various types ordinary differential equations have been obtained. Solving these equations in MATLAB gives different layer thickness and other parameters which help us to understand and analyze different parameterssuch as non-condensable concentrations, mixture velocities, rate of sub-cooling on condensation process.

Chapter 2

Problem Definition and Mathematical Modeling

Introduction

In this chapter a complete mathematical description of given problem is described. Firstly governing equations of continuity, momentum, energy for both liquid and mixture and one concentration equation related to non-condensable gas are analyzed. In the following section liquid-mixture interfacial behaviors have been written in most generalized form. These equations are simplified using order of magnitude analysis and boundary layer approximation. Using boundary layer integral method, each of these equations is integrated throughout boundary layers with proper velocity, temperature and concentration profiles. Chapter ends with eight ordinary differential equations which are generated from the above analysis. Finally with the help of these analysis local and average heat transfer coefficient, Nusselt number have found out.

2.1 Problem description

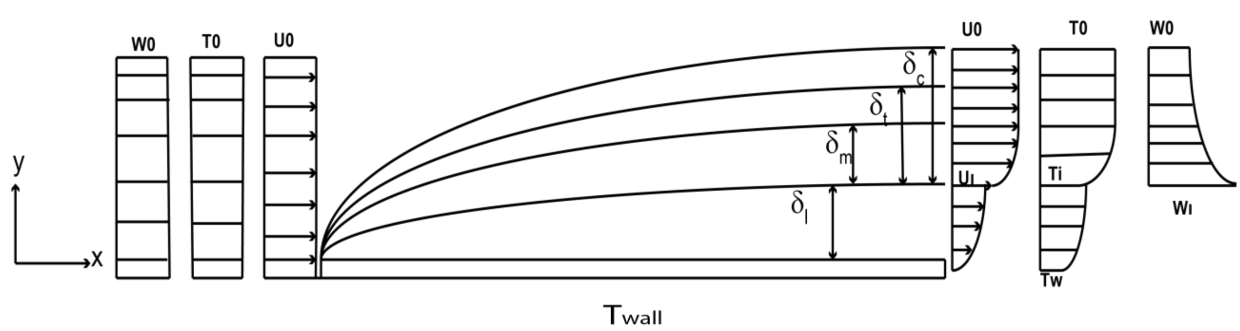

Figure 1 Geometry and Coordinate system

The geometry is described in Fig. 1. A saturated vapour and a non-condensable gas (air) mixture enter a horizontal plate of length L which is maintained at a uniform temperature T_{wall}

such that it is less than the saturation temperature of vapour corresponding to free straam vapour pressure. The mixture flowing over the plate has uniform velocity $u_{\infty,}$ temperature T_{∞}, pressure $P_{\infty,}$ and gas concentration w_{∞},. As the mixture flows along the plate due to heat rejection from vapour to liquid, a liquid layer of thickness δ_l is formed which is very thin compare to dimension of the plate. In case of non- condensable gas this liquid vapour interface is impermeable so gas cannot diffuse through the interface. As a result concentration of gas with respect to vapour in mixture region is continuously increased and affects interfacial heat and mass transfer phenomenon. Beyond liquid hydrodynamic boundary layer (δ_l) there are several mixture boundary layers like mixture hydrodynamic boundary layer (δ_m), mixture thermal boundary layer (δ_t), gas concentration boundary layer (δ_c) that are formed along the vertical direction of the plate. In this model we made the following assumptions to simplify our analysis.

2.2 Assumptions associated with the liquid film

- Liquid flow is laminar in nature
- slip is neglected between liquid and mixture phase
- liquid temperature is constant and same as wall temperature
- film interface is impermeable to non-condensable gas

Assumptions associated with the mixture phase

- homogeneous binary mixture of water vapour and air
- mixture is maintained at saturated condition in vapour-liquid interface
- mixture flow is laminar
- only diffusion of water vapour is important in case of condensation heat and mass transfer process across interface
- each component of homogeneous mixture are in dynamic and thermal equilibrium at each and every point of the mixture
- all the properties of the mixture are constant and evaluated at average of free stream and wall temperature
- free stream value of mixture is considered as non-viscous flow

Assumptions associated with plate the geometry

- horizontal, smooth surface with uniform wall temperature

To analyze this physical situation conservation equations in liquid and mixture phases, boundary layer approximation method is used. Some terms in the boundary layer can be neglected by using order of magnitude analysis without loss of generality. This process is very useful to analyze these types of complicated problems

2.3 Property Averaging

Whenever we analyze these types of multicomponent mixture condensation processes, property averaging plays a very important role, because as condensable species condense the mass fraction of such component decreases and mass fraction of other non-condensable species continuously increases. As a result thermo physical property of each component varies locally. So as a whole we cannot treat the properties to be constant. Taking such variable properties, analytical treatment of conservation equations of each component is very difficult. To overcome such difficulties, in our analysis we assume, the air-vapour mixture is homogeneous in nature that means mixture is in thermal equilibrium at each and every point. On the basis of above assumptions we take weighted averaged property (McAdams relationship) at a mean mixture temperature which is defined as the arithmetic mean of free stream and wall temperature.

If ε be any species property, then averaged property of mixture can be expressed as,

$$\frac{1}{\varepsilon_m} = \frac{w_{nc}}{\varepsilon_{nc}} + \frac{1 - w_{nc}}{\varepsilon_v} \tag{1}$$

2.4. Governing equations

Based on the assumptions stated in article (2.2), the governing equations are written in the Cartesian coordinate as follows:

For liquid film layer $(0 < y < \delta_l)$

Continuity Equation

$$\frac{\partial u_l}{\partial x} + \frac{\partial v_l}{\partial y} = 0 \tag{2}$$

Momentum Equation

$$\frac{\partial ({u_l}^2)}{\partial x} + \frac{\partial (u_l v_l)}{\partial y} = \upsilon_l \frac{\partial^2 (u_l)}{\partial y^2} \tag{3}$$

Energy Equation

$$\frac{\partial (u_l T_l)}{\partial y} + \frac{\partial (v_l T_l)}{\partial y} = \alpha_l \frac{\partial^2 (T_l)}{\partial y^2} \tag{4}$$

For vapour – gas mixture layer

Continuity Equation

$$\frac{\partial u_m}{\partial x} + \frac{\partial v_m}{\partial y} = 0 \tag{5}$$

Momentum Equation $(\delta_l < y < \delta_l + \delta_m)$

$$\frac{\partial ({u_m}^2)}{\partial x} + \frac{\partial (u_m v_m)}{\partial y} = \upsilon_m \frac{\partial^2 (u_m)}{\partial y^2} \tag{6}$$

Energy Equation $(\delta_l < y < \delta_l + \delta_t)$

$$\frac{\partial (u_m T_m)}{\partial x} + \frac{\partial (v_m T_m)}{\partial y} = \alpha_m \frac{\partial^2 (T_m)}{\partial y^2} \tag{7}$$

Species (gas) concentration equation $(\delta_l < y < \delta_l + \delta_c)$

$$\frac{\partial(u_m w_{nc})}{\partial x} + \frac{\partial(v_m w_{nc})}{\partial y} = D_m \frac{\partial^2(w_{nc})}{\partial y^2} \tag{8}$$

$$w_v + w_{nc} = 1 \tag{9}$$

So, only one species equation (gas) is sufficient.

To simplify the situation, thermal diffusion, viscous dissipation and compressible heating effects have been neglected. To confirm that the mixture maintains thermal and mechanical equilibrium at each and every point the steam and air are assumed to have the same values of $u, v\ , T$ at all positions position within the mixture region i.e. mixture treated as homogeneous in nature. This assumption is very important in thermodynamic point of view. If we not consider this assumption we have to solve different governing equations separately for each vapour and gas layer. Which are very complex and difficult in practice. We have to determine δ_l, δ_m, δ_t, δ_c, T_i, w_i by boundary layer integral method [14, 24] and the final forms are expressed as some ordinary differential equations. Calculation of these thicknesses gives us fair amount of heat and mass transfer rates predictions across the interface.

2.5. The boundary and interfacial conditions

$$(u_l)_{y=0} = 0 \tag{10a}$$

$$(u_l)_{y=\,\delta_l} = (u_m)_{y=\,\delta_l} = u_i \tag{10b}$$

$$(u_m)_{y=\,\delta_l+\,\delta_m} = u_\infty \tag{10c}$$

$$(T_l)_{y=0} = T_w \tag{10d}$$

$$(T_l)_{y=\delta_l} = (T_m)_{y=\delta_l} = T_i \qquad (10e)$$

$$(T_m)_{y=\delta_l+\delta_t} = T_\infty \qquad (10f)$$

$$(w_{nc})_{y=\delta_l+\delta_c} = w_\infty \qquad (10g)$$

$$(w_{nc})_{y=\delta_l} = w_i \qquad (10h)$$

Interface mass balance is given by

$$j = \rho_l \left\{ u_i \frac{d\delta_l}{dx} - (v_l)_{y=\delta_l} \right\} = \rho_m \left\{ u_i \frac{d\delta_l}{dx} - (v_m)_{y=\delta_l} \right\} \qquad (10i)$$

Interface shear stress balance is given by

$$\mu_l \left(\frac{\partial u_l}{\partial y} \right)_{y=\delta_l} = \mu_m \left(\frac{\partial u_m}{\partial y} \right)_{y=\delta_l} \qquad (10j)$$

Interface energy balance is given by

$$-k_l \left(\frac{\partial T_l}{\partial y} \right)_{y=\delta_l} + j h_{fg} = -k_m \left(\frac{\partial T_m}{\partial y} \right)_{y=\delta_l} \qquad (10k)$$

Interface non-condensable gas mass balance is given by

$$\rho_m D_m \left(\frac{\partial w_{nc}}{\partial y} \right)_{y=\delta_l} + j w_i = 0 \qquad (10l)$$

Interface temperature at the interface can be calculated using Clausius-Clapeyron equation:

$$In \left(\frac{P_{vi}}{P_\infty} \right) = \frac{h_{fg}}{R_v} \left(\frac{1}{T_i} - \frac{1}{T_\infty} \right) \qquad (10m)$$

Interface temperature at the interface can be calculated using Gibbs-Dalton equation

$$w_i = \frac{P_{vi} M_{vi}}{P_{vi} M_{vi} + (P - P_{vi}) M_{nc}} \qquad (10n)$$

2.6. Integral approach with proper profile assumptions

Integrating Eq. (2) from 0 to δ_l and applying Leibniz rule, we get

$$j = \frac{\mathrm{d}}{dx}\int_0^{\delta_l} \rho_l u_l dy \tag{10o}$$

Integrating of Eq. (3) from 0 to δ_l

$$\int_0^{\delta_l} \frac{\partial}{\partial x}(u_l{}^2)\, dy + \int_0^{\delta_l} \frac{\partial}{\partial x}(u_l v_l)\, dy = \upsilon_l \int_0^{\delta_l} \frac{\partial^2 u_l}{\partial y^2} dy \tag{11}$$

Applying Leibniz rule for expanding the above equation along with the Eqs. (10b), (10i) and (10o) it is obtained that

$$\frac{d}{dx}\int_0^{\delta_l} (u_l{}^2)dy - (u_i \frac{j}{\rho_l}) = \upsilon_l \left(\left\{\frac{\partial u_l}{\partial y}\right\}_{y=\delta_l} - \left\{\frac{\partial u_l}{\partial y}\right\}_{y=0} \right) \tag{12}$$

Similarly the integration of liquid energy equation, Eq. (4) and applying the Leibniz rule along with proper boundary conditions Eqs. (10e), (10i) and (10o) results in

$$\frac{d}{dx}\int_0^{\delta_l} (u_l T_l)dy - (T_i \frac{j}{\rho_l}) = \alpha_l \left(\left\{\frac{\partial T_l}{\partial y}\right\}_{y=\delta_l} - \left\{\frac{\partial T_l}{\partial y}\right\}_{y=0} \right) \tag{13}$$

Integral mixture momentum Eq. (6) becomes

$$\frac{d}{dx}\int_{\delta_l}^{\delta_l+\delta_m} u_m(u_m-u_\infty)dy+\frac{j}{\rho_m}(u_i-u_\infty) = -\upsilon_m\left\{\frac{\partial u_m}{\partial y}\right\}_{y=\delta_l} \tag{14}$$

Integral mixture energy Eq. (7) becomes

$$\frac{d}{dx}\int_{\delta_l}^{\delta_l+\delta_t} u_m(T_m-T_\infty)dy+\frac{j}{\rho_m}(T_i-T_\infty) = -\alpha_m\left\{\frac{\partial T_m}{\partial y}\right\}_{y=\delta_l} \tag{15}$$

Integral non-condensable concentration Eq. (8) becomes

$$\frac{d}{dx}\int_{\delta_l}^{\delta_l+\delta_c} u_m(w_{nc}-w_\infty)dy+\frac{j}{\rho_v}(w_i-w_\infty)+\frac{d}{dx}\int_{\delta_l+\delta_m}^{\delta_l+\delta_c} u_\infty(w_{nc}-w_\infty)dy = -D_m\left\{\frac{\partial W_{nc}}{\partial y}\right\}_{y=\delta_l} \tag{16}$$

In case of liquid layer we define a non-dimensional length

$$\eta=\frac{y}{\delta_l} \tag{17}$$

We assume a cubic velocity profile between the interface and isothermal wall, which is in the form

$$\frac{u_l}{u_i} = a_1+ b_1\eta+ c_1\eta^2+d_1\eta^3 \tag{18}$$

Here, a_1, b_1, c_1, d_1 are constants which are function of *x*.

To evaluate these constants, we use boundary conditions (10a), (10b) along with wall condition

$$v_l \frac{\partial^2(u_l)}{\partial y^2} = 0 \tag{19}$$

Finally we get the required velocity profile

$$\frac{u_l}{u_i} = \eta\,(1 - d_1) + d_1\eta^3 \tag{20}$$

Similarly, assuming a quadratic temperature profile in case of liquid layer

$$\frac{T_l - T_i}{T_w - T_i} = a_2 + b_2\eta + c_2\eta^2 \tag{21}$$

Applying boundary conditions (10d), (10e) we get required temperature profile

$$\frac{T_l - T_i}{T_w - T_i} = 1 - (1 + c_2)\eta + c_2\eta^2 \tag{22}$$

Similarly, in case of mixture momentum layer we define a non-dimensional length

$$\eta_m = \frac{y - \delta_l}{\delta_m} \tag{23}$$

We, assume quadratic velocity profile in the mixture layer which is defined as

$$\frac{u_m - u_\infty}{u_i - u_\infty} = a_3 + b_3\eta + c_3\eta^2 \tag{24}$$

Applying boundary conditions (10b), (10c) and zero velocity gradients at mixture momentum layer edge, we get

$$\frac{u_m - u_\infty}{u_i - u_\infty} = \left(1 + \frac{\delta_l}{\delta_m} - \frac{y}{\delta_m}\right)^2 \tag{25}$$

Considering, quadratic temperature profile within mixture energy layer, incorporating zero temperature and using boundary conditions (10e), (10f) we get

$$\frac{T_m - T_\infty}{T_i - T_\infty} = \left(1 + \frac{\delta_l}{\delta_t} - \frac{y}{\delta_t}\right)^2 \tag{26}$$

Following, similar approach we get quadratic non-condensable profile within mixture concentration layer which is defined as

$$\frac{w_{nc} - w_\infty}{w_i - w_\infty} = \left(1 + \frac{\delta_l}{\delta_c} - \frac{y}{\delta_c}\right)^2 \tag{27}$$

Putting these profiles related to Eqns.(20),(22),(25),(26),(27) in the equations (10o),(12),(13),(14),(15),(16) we get

eight ordinary differential equations consisting of $\delta_l, \delta_m, \delta_t, \delta_c, T_i, w_i, d_1, c_2$.

2.7. Non-dimensional equations

Now to non-dimensionalized these equations, some non-dimensional parameter are introduced as

$$\bar{x} = \frac{x}{L} \tag{28}$$

$$\bar{\delta} = \frac{\delta}{L}, \tag{29}$$

$$\bar{u} = \frac{u}{u_\infty}, \tag{30}$$

$$\theta = \frac{T_i}{T_w - T_i}, \tag{31}$$

$$Re_l = \frac{u_\infty L}{\upsilon_l}, \tag{32}$$

$$Re_m = \frac{u_\infty L}{\upsilon_m}, \tag{33}$$

$$Pr_l = \frac{\upsilon_l}{\alpha_l}, \tag{34}$$

$$Pr_m = \frac{\upsilon_m}{\alpha_m}, \tag{35}$$

$$Pe_l = Re_l Pr_l \tag{36}$$

$$Pe_m = Re_m Pr_m \tag{37}$$

$$Sc_m = \frac{\upsilon_m}{D_m} \tag{38}$$

$$\theta_{li} = \frac{T_i - T_w}{T_\infty - T_w} \tag{39}$$

$$\theta_{mi} = \frac{T_\infty - T_i}{T_\infty - T_w} \tag{40}$$

$$Ja_{in} = \frac{C_{pm}(T_\infty - T_w)}{h_{fg}} \tag{41}$$

$$Ja_l = Ja_{in}\theta_{li} \tag{42}$$

$$Ja_m = Ja_{in}\theta_{mi} \tag{43}$$

In terms of variables $(\bar{\delta}_l, \bar{\delta}_m, \bar{\delta}_t, \bar{\delta}_c, \bar{T}_i, w_i, d_1, c_2)$ a system of ode is formed as

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\left\{\frac{2-d_1}{4}\right\}\frac{2A\{1+a_0\}}{B}\right] - \frac{d\bar{\delta}_m}{d\bar{x}}\left[\left\{\frac{2-d_1}{4}\right\}\frac{2AFa_0}{B}\right] - \frac{dd_1}{d\bar{x}}\left[\frac{A}{B}\bar{\delta}_l\left\{0.5 + \left(\frac{2-d_1}{4}\right)\right\}\right]$$
$$= \frac{2Ja_m}{\bar{\delta}_t Pe_m}\left(\frac{\rho_m}{\rho_l}\right) - \frac{Ja_l(1-c_2)}{\bar{\delta}_l Pe_l} \tag{44}$$

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\left\{\frac{8{d_1}^2 - 28d_1 + 35}{105}\right\}\left\{\frac{4Aa_0}{B} + \frac{2A}{B}\right\}\right] - \frac{d\bar{\delta}_m}{d\bar{x}}\left[4\left\{\frac{8{d_1}^2 - 28d_1 + 35}{105}\right\} - Aa_0\frac{\bar{F}}{B}\right]$$

$$+\frac{dd_1}{d\bar{x}}\left[\frac{8}{105}\left\{8{d_1}^2-28d_1+35\right\}\frac{\bar{\delta}_l A}{B^2}\right]=\left[3Re_l\frac{d_1}{\bar{\delta}_l}+2\frac{Ja_m}{\bar{\delta}_l Pe_m}\left\{\frac{\rho_m}{\rho_l}\right\}-\frac{Ja_l\{1-c_2\}}{\bar{\delta}_l Pe_l}\right] \tag{45}$$

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\frac{2Aa_0}{B}\left\{\left(\frac{10-7d_1+3d_1c_2-5c_2}{60}\right)+\left(\frac{2-d_1}{4}\right)\theta\right\}+2\frac{A}{B}\theta\left(\frac{2-d_1}{4}\right)\right]$$

$$-\frac{d\bar{\delta}_m}{d\bar{x}}\left[\frac{2AFa_0}{B}\left\{\left(\frac{10-7d_1+3d_1c_2-5c_2}{60}\right)+\left(\frac{2-d_1}{4}\right)\theta\right\}+\theta\left(\frac{2-d_1}{4}\right)\right]$$

$$+\frac{dd_1}{d\bar{x}}\left[\frac{2A\bar{\delta}_l}{B}\left\{\frac{(3c_2-7)}{60}-\frac{\theta}{4}\right\}-4A\frac{\bar{\delta}_l}{B^2}\left\{\left(\frac{10-7d_1+3d_1c_2-5c_2}{60}\right)+\left(\frac{2-d_1}{4}\right)\theta\right\}\right]$$

$$+\frac{dc_2}{d\bar{x}}\left[\frac{2A\bar{\delta}_l}{B}\left(\frac{3d_1-5}{60}\right)\right]+\frac{d\bar{T}_l}{d\bar{x}}\left[\frac{2A\bar{\delta}_l}{B}\left\{\left(\frac{2-d_1}{4}\right)-\left(\frac{10-7d_1+3d_1c_2-5c_2}{60}\right)\right\}\right]$$

$$=\left[\frac{2c_2}{\bar{\delta}_l Pe_l}+\frac{2Ja_m\theta}{\bar{\delta}_t Pe_m}\left(\frac{\rho_m}{\rho_l}\right)-\frac{Ja_l(1-c_2)\theta}{\bar{\delta}_l Pe_l}\right] \tag{46}$$

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\frac{2\left(\frac{\mu_m}{\mu_l}\right)a_0}{B}\left(0.4-\frac{0.33}{a_0}\right)\right]+\frac{d\bar{\delta}_m}{d\bar{x}}\left[(0.33-0.2a_0)-\frac{0.66Aa_0}{B}\left(0.4-\frac{0.33}{a_0}\right)\right]+$$

$$\frac{dd_1}{d\bar{x}}\left[4\frac{\bar{\delta}_l\left(\frac{\mu_m}{\mu_l}\right)}{B^2}\left(\frac{0.33}{a_0}-0.4\right)\right]=\left[\frac{2}{\bar{\delta}_m Re_m}+\frac{2Ja_m}{\bar{\delta}_t Pe_m}\left(\frac{\rho_m}{\rho_l}\right)-\frac{Ja_l(1-c_2)}{\bar{\delta}_l Pe_l}\right] \tag{47}$$

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\frac{2\left(\frac{\mu_m}{\mu_l}\right)a_0}{3B}\left(\bar{\psi}-\frac{\bar{\psi}^2}{2}+\frac{\bar{\psi}^3}{10}\right)\right]+\frac{d\bar{\delta}_m}{d\bar{x}}\left[\frac{a_0}{3}\left(\frac{\bar{\psi}^3}{5}-\frac{\bar{\psi}^2}{2}\right)+0.66A\frac{a_0}{3B}\left(\frac{\bar{\psi}^2}{2}-\psi-\frac{\bar{\psi}^3}{10}\right)\right]$$

$$+\frac{dd_1}{d\bar{x}}\left[1.33\frac{\left(\frac{\mu_m}{\mu_l}\right)\bar{\delta}_l}{3B^2}\left(\frac{\bar{\psi}^2}{2}-\psi-\frac{\bar{\psi}^3}{10}\right)\right]+\frac{d\bar{\delta}_t}{d\bar{x}}\left[0.33+\frac{a_0}{3}(\bar{\psi}-1-0.3\bar{\psi}^2)\right]$$

$$+\frac{d\bar{\bar{T}}_i}{d\bar{x}}\frac{1}{\bar{\bar{T}}_i}\left[a_0\frac{\bar{\delta}_t}{3}\left(\frac{\bar{\psi}}{2}-\frac{\bar{\psi}^2}{10}-1\right)+\frac{\bar{\delta}_t}{3}\right]=\left[\frac{2}{\bar{\delta}_t Pe_m}-\frac{2Ja_m}{\bar{\delta}_l Pe_m}-\frac{Ja_l\{1-c_2\}}{\bar{\delta}_l Pe_l}\left(\frac{\rho_l}{\rho_m}\right)\right] \tag{48}$$

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\frac{2\left(\frac{\mu_m}{\mu_l}\right)a_0}{3B}\left(\frac{1}{10\bar{\bar{\emptyset}}^2}-\frac{1}{2\bar{\bar{\emptyset}}}+1\right)\right]+\frac{d\bar{\delta}_m}{d\bar{x}}\left[\frac{a_0}{3}\left(\frac{1}{\bar{\bar{\emptyset}}}-1-\frac{3}{10\bar{\bar{\emptyset}}^2}\right)-0.66A\frac{a_0}{B}\left(\frac{1}{10\bar{\bar{\emptyset}}^2}-\frac{1}{2\bar{\bar{\emptyset}}}+1\right)\right]$$

$$\frac{dd_1}{d\bar{x}}\left[1.33\frac{\left(\frac{\mu_m}{\mu_l}\right)\bar{\delta}_l}{3B^2}\left(\frac{1}{2\bar{\bar{\emptyset}}}-\frac{1}{10\bar{\bar{\emptyset}}^2}-1\right)\right]+\frac{d\bar{\delta}_c}{d\bar{x}}\left[0.33-0.33a_0\left(\frac{1}{2\bar{\bar{\emptyset}}^2}-\frac{1}{5\bar{\bar{\emptyset}}^3}\right)\right]$$

$$+\frac{\frac{dw_i}{d\bar{x}}}{(w_i-w_\infty)}\left[\bar{\delta}_c\left\{0.33a_0\left(\frac{1}{2\bar{\bar{\emptyset}}^2}-\frac{1}{10\bar{\bar{\emptyset}}^3}-\frac{1}{\bar{\bar{\emptyset}}}\right)+\frac{1}{3}\right\}\right]$$

$$=\left[\frac{2}{\bar{\delta}_c}\frac{1}{Sc_m Re_m}-\frac{2Ja_m}{\bar{\delta}_l Pe_m}-\frac{Ja_l\{1-c_2\}}{\bar{\delta}_l Pe_l}\left(\frac{\rho_l}{\rho_m}\right)\right] \tag{49}$$

First closer Equation

$$\frac{d\bar{\bar{T}}_i}{d\bar{x}}\left[(w_i-1)(1.61-0.61w_i)\frac{C_{pm}}{R_v Ja_m}\frac{1}{(\zeta-\bar{\bar{T}}_i)^2}\right]-\frac{dw_i}{d\bar{x}}=0 \tag{50}$$

Second closer Equation

$$\frac{d\bar{\delta}_l}{d\bar{x}}\left[\frac{\left(\frac{k_l}{k_m}\right)}{\bar{\delta}_l^{\,2}}\bar{\delta}_l(1-\bar{\bar{T}}_i)(1-c_2)\right]+\frac{d\bar{\delta}_t}{d\bar{x}}\left[\frac{2\bar{\delta}_c\bar{\bar{T}}_i}{\bar{\delta}_l^{\,2}}\right]-\frac{dc_2}{d\bar{x}}\left[\frac{\left(\frac{k_l}{k_m}\right)}{\bar{\delta}_l}\bar{\delta}_c(1-\bar{\bar{T}}_i)\right]$$

$$\frac{d\bar{\delta}_c}{d\bar{x}}\left[\frac{2\bar{\bar{T}}_i}{\bar{\delta}_t}-\frac{\left(\frac{k_l}{k_m}\right)}{\bar{\delta}_l}(1-\bar{\bar{T}}_i)(1-c_2)\right]-\frac{d\bar{\bar{T}}_i}{d\bar{x}}\left[\frac{2\bar{\delta}_c}{\bar{\delta}_t}-\frac{\left(\frac{k_l}{k_m}\right)}{\bar{\delta}_l}\bar{\delta}_c(1-c_2)\right]$$

$$-\frac{dw_i}{d\bar{x}}\left[\frac{2w_\infty}{w_i^{\,2}}\frac{Pr_m}{Ja_m}\right]=0 \tag{51}$$

$$A = \left(\frac{Re_m}{Re_l}\right)\left(\frac{\bar{\delta}_l}{\bar{\delta}_m}\right), a_0 = \frac{(1+2d_1)}{\{(1+2d_1)+2A\}}, \theta = \frac{T_\infty + \bar{T}_i(T_w - T_\infty)}{(T_w - T_\infty)(1-\bar{T}_i)}, \bar{\psi} = \frac{\bar{\delta}_t}{\bar{\delta}_m}, \bar{F} = \frac{\bar{\delta}_l}{\bar{\delta}_m},$$

$$\bar{\emptyset} = \frac{\bar{\delta}_c}{\bar{\delta}_m}, \varsigma = \frac{T_\infty}{(T_\infty - T_w)}, \bar{T}_i = \frac{T_i - T_\infty}{T_w - T_\infty}$$

These systems of equation (Eqs. 44-51) have been solved with the help of MATLAB ordinary differential equation solver. The numerical solutions has been used to generate the results to demonstrate the actual process.

2.8. Evaluation of local heat transfer coefficient and Nusselt number

$$h_x = \frac{q_x}{(T_\infty - T_w)} \tag{52}$$

Where

$$q_x = -k_l\left(\frac{\partial T_l}{\partial y}\right)_{y=0} = \frac{(1+c_2)}{\delta_l}\left(\frac{T_i - T_w}{T_\infty - T_w}\right) \tag{53}$$

$$h_{av} = \frac{\int_0^L h_x dx}{L} \tag{54}$$

$$Nu_{av} = h_{av}\frac{L}{k_l} \tag{55}$$

Summary

Chapter contains in detail studies of boundary layer governing equations in both liquid and homogeneous vapour-gas mixture phases. These equations have been solved analytically with the help of boundary layer integral method and obtained ordinary differential equations in non-dimensional form. Finally with the help of these equations heat transfer coefficient, Nusselt number and other heat transfer parameters have been obtained

Chapter 3

Results and Discussion

Introduction

This chapter contains analytical result obtained from the boundary layer integral method applies on governing equations of both liquid and mixture phases presented in previous chapter. The obtained results mainly deal with forced condensation on an isothermal horizontal plate. The mixture considers here is the homogeneous mixture of vapour and air with different air concentration. Next analytical solution that is obtained from the model discussed previously is compared with experimental data. Furthermore effects of various non-condensable mass fractions on heat transfer coefficient have been presented in detailed manner. At the end of this chapter relation between various non-dimensional numbers regarding heat and mass transfer for different air mass fraction, mixture free stream velocities and rate of sub cooling have been summarized.

In the present study calculations are performed for steam-air mixture with three different air concentrations 0%, 5% and 20%. Inlet gas-steam mixture entries the plate with saturated condition (humidity$H_r = 100\%$) . The geometry of the plate (L=0.05 mm) is taken as flat and isothermal to avoid any complexity. System pressure is maintained at one atmospheric pressure.

3.1. Validation of model

The developed model discussed in the previous chapter has been validated with those of Kang [6] (Fig.2) and (Fig.3) for two different air mass fraction (0.41 and 0.78). Free stream mixture is considered as saturated condition which enters the plate with a free stream velocity of 3m/s. The average errors between numerical results presented in this work and those of Kang [6] are approximately in the range of 20 to 25%. The main reasons behind the differences between the numerical results presented in this study and finding for the experimental analysis are: the plate temperature of the given experiment is not constant; it is varied along the length of the plate. To maintain the plate isothermal cooling water is fed from the bottom of the plate but due to natural reasons constant temperature is not fully achieved. The other reason is that in the present study the properties are taken as constant with respect to the average temperature taken between free stream and wall but in experiment it is very difficult to achieve such condition.

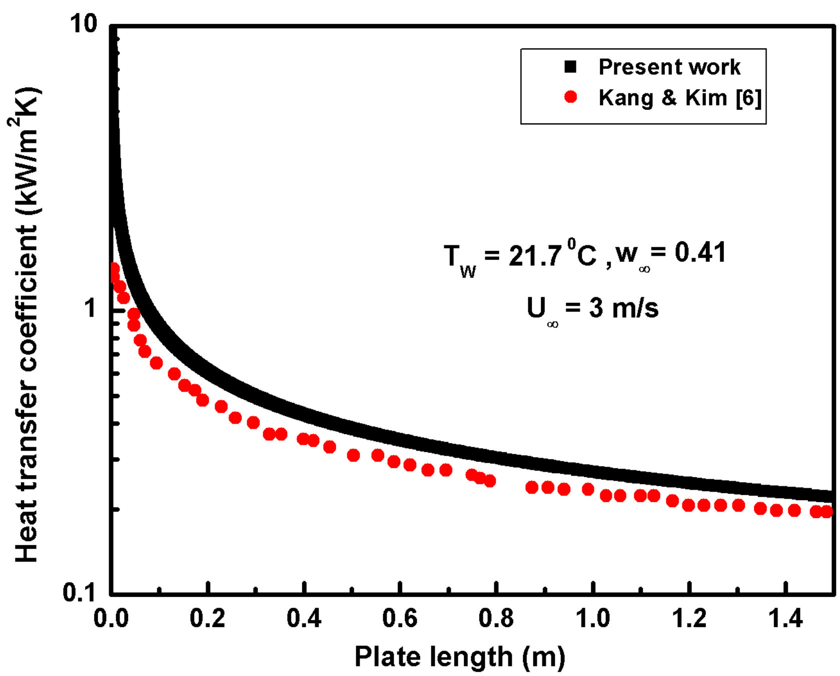

Fig 2: Comparative study of result with those of Kang [6].

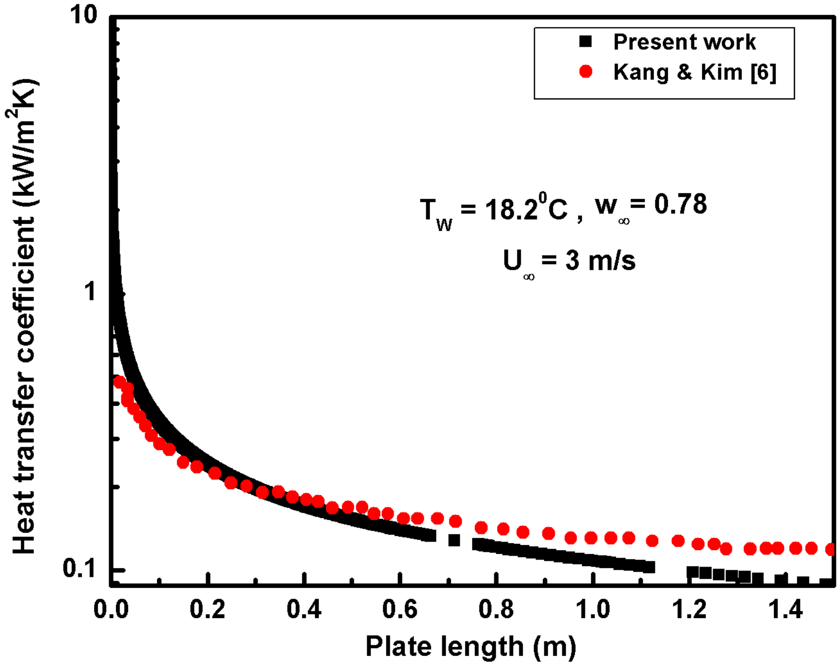

Fig 3: Comparative study of result with those of Kang [6].

3.2 Velocity, Temperature and Concentration profiles for different air mass fraction

3.2 (a) Velocity profiles

Typical velocity profiles at different axial positions of 0.001m and 0.05 m for two different air mass concretions of 0.05 and 0.2 have been presented in Fig. 4(a) and Fig. 4(b). These profiles are evaluated correspond to free stream conditions of $P_\infty = 1\ bar, \Delta T = 20K\ and\ U_\infty = 1m/s$. Velocities of mixture and liquid usually follow the boundary layer variation i.e. at the liquid vapour interface mixture has interfacial velocity which also equal to the liquid interface velocity to maintain the continuity, then as mixture momentum layer increases, mixture velocity also increases and reaches free stream condition at edge of the mixture momentum boundary layer. The variation of the mixture velocity is quadratic in nature because we take quadratic mixture velocity profile in our boundary layer integral analysis. Similarly in case of liquid layer, velocity is decreased from interfacial value to zero at the wall to match the no slip condition. According to the profile assumption that we have taken in our analytical model, velocity variation is cubic in liquid boundary layer. The film thickness is so small that the cubic variation of velocity is not encountered if we take linear velocity scale. So to capture actual distribution of velocity we take logarithmic velocity scale in our analysis.

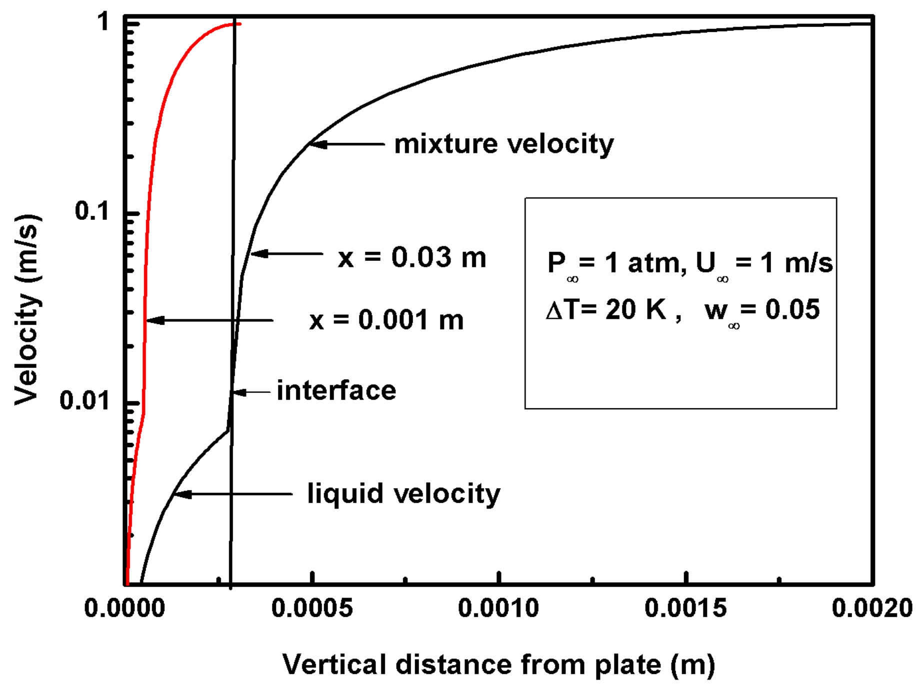

Figure 4(a): Velocity profiles for air mass fraction 0.05

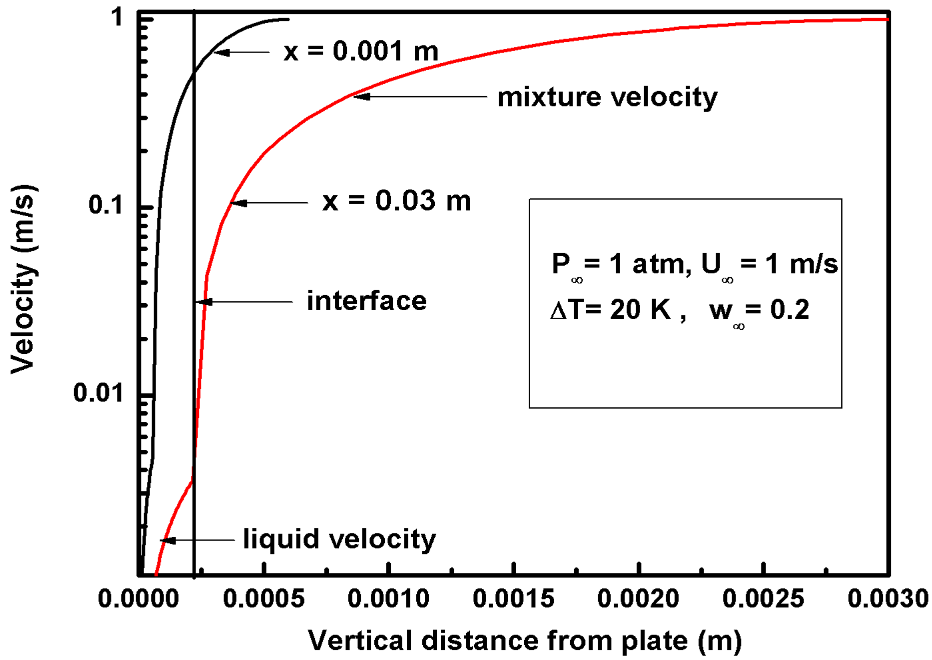

Figure 4 (b): Velocity profiles for air mass fraction 0.2

3.2 (b) Temperature profiles

Fig.5(a) and Fig.5(b) shows temperature distribution for both mixture and liquid phases along mixture energy and liquid layer. These profiles are evaluated for same free stream conditions discussed in 3.2 (a). Temperature distribution for mixture layer varies from interfacial to free stream value at the edge of the thermal boundary layer. This variation follows parabolic distribution as assumed in analytical model. Similarly in case of liquid, a temperature gradient is observed from interface to wall. The profile of the liquid is almost close to a linear shape. This indicates diffusion term plays more dominant role compare to convective term. It is important to note that due to impermeable nature of the non-condensable, it accumulates along the interface. This will decrease vapour pressure along the interface and as a result saturation temperature corresponding vapour pressure continuously decrease along the length of the plate. This implies that rate of condensation will continuously decrease along horizontal direction.

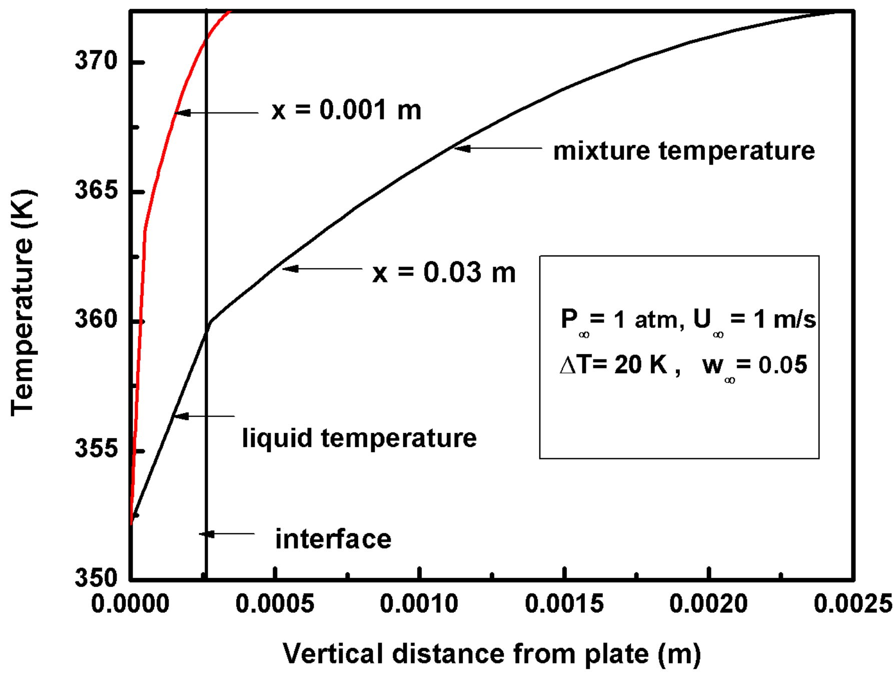

Figure 5 (a): Temperature profiles for air mass fraction 0.05

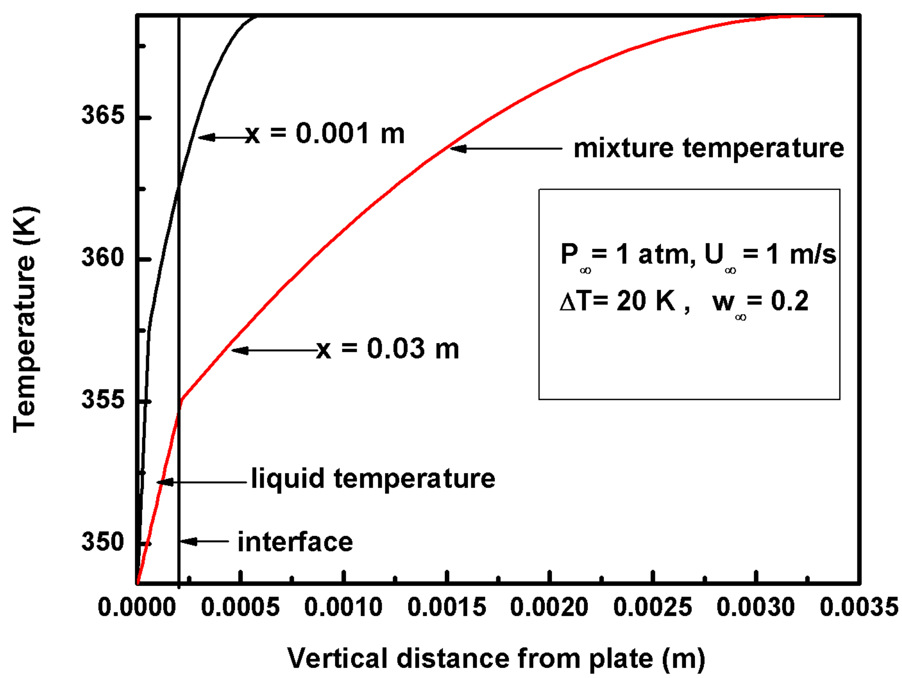

Figure 5 (b): Temperature profile for air mass fraction 0.2

3.2 (c) Air-concentration profiles

Fig. 6(a) and Fig. 6(b) describe the variation air-concentration for two different mass fractions of 0.005 and 0.2. The free stream conditions are maintained same as previously stated articles. Due to non-diffusive nature of non-condensable gas such as air, it creates a resistive barrier to vapour condensation and continuously accumulates near the vapour-liquid interface region. As a result a non-linear increasing concentration profile of air is observed within the gas concentration layer. According to analytical model this non-linear profile is parabolic in nature. The reduction in vapour mass fraction due to condensation from free stream to interface increases gas concentration, due to this vapour pressure at each and every point continuously decreases within the concentration layer, as a result saturation temperature corresponding to vapour pressure is locally decreased. At the interface, temperature reaches saturation state corresponding interface vapour concentration.

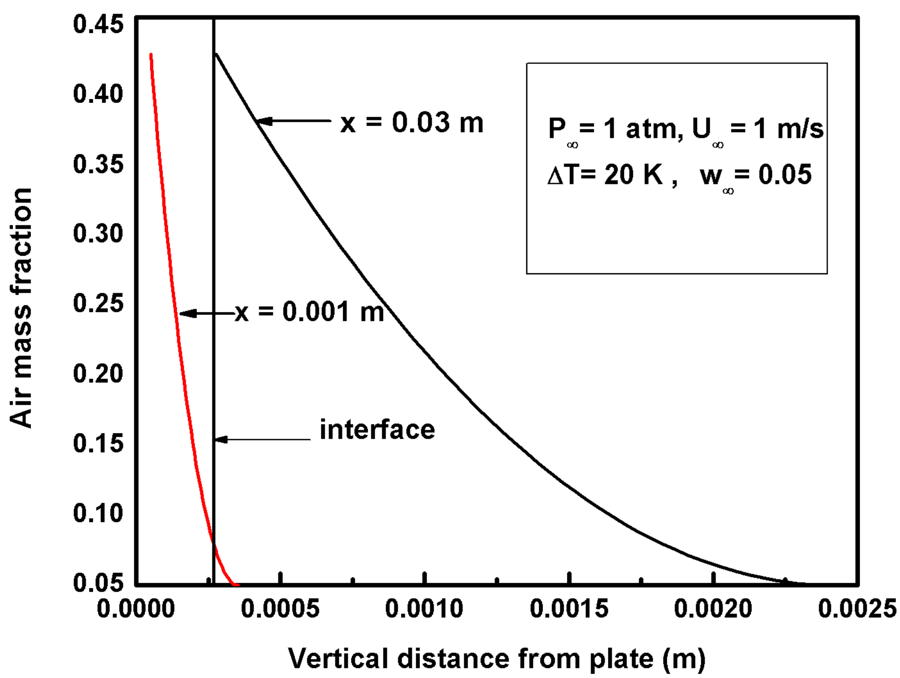

Figure 6 (a): Mixture air concentration profiles for air mass fraction 0.05

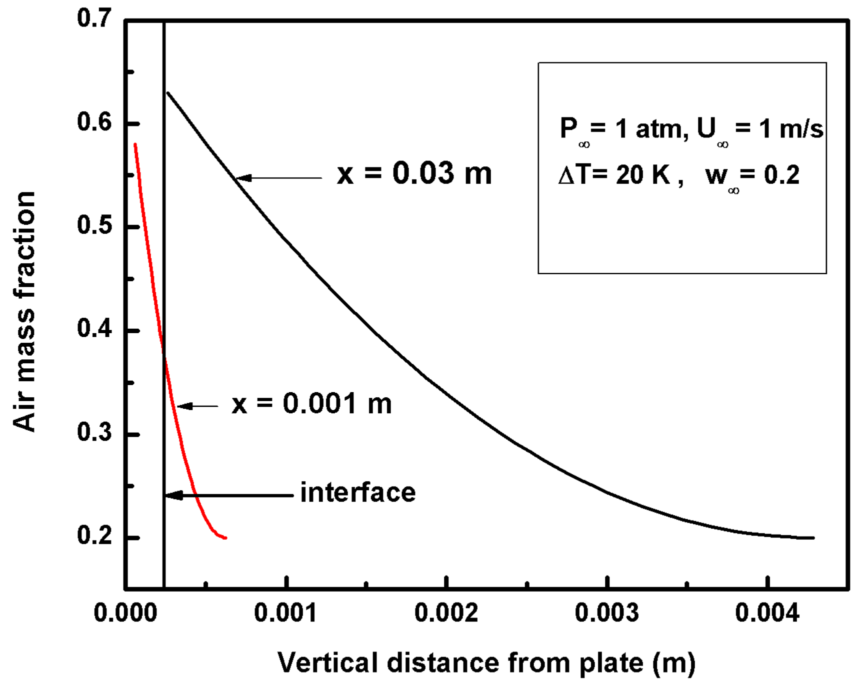

Figure 6 (b): Mixture air concentration profiles for air mass fraction 0.2

3.2 (d) Variation of interface temperatures

Fig.7(a) and 7(b) shows the variation of interface temperature along length of the isothermal plate. Due to non-diffusive nature of the non-condensable gas, even small amount of this has a significance influence on the resistance to the condensation heat and mass transfer process. Accumulation of the gas increases partial pressure of the gas stream at the interface region. Since the total pressure is constant at each and every point of the mixture, partial pressure of the vapour is lowered compare to bulk mixture vapour pressure. This difference in pressure creates a driving force for diffusion of vapour near interface. On the other hand due to increase in gas partial pressure, saturation pressure of vapour decrease at the interface which leads to lowering the interface saturation temperature. This decrement is continuous along the axial direction of the plate. So as a whole temperature difference between interface and wall is lowered compare to pure vapour condensation process. As a result rate of condensation is decreased significantly.

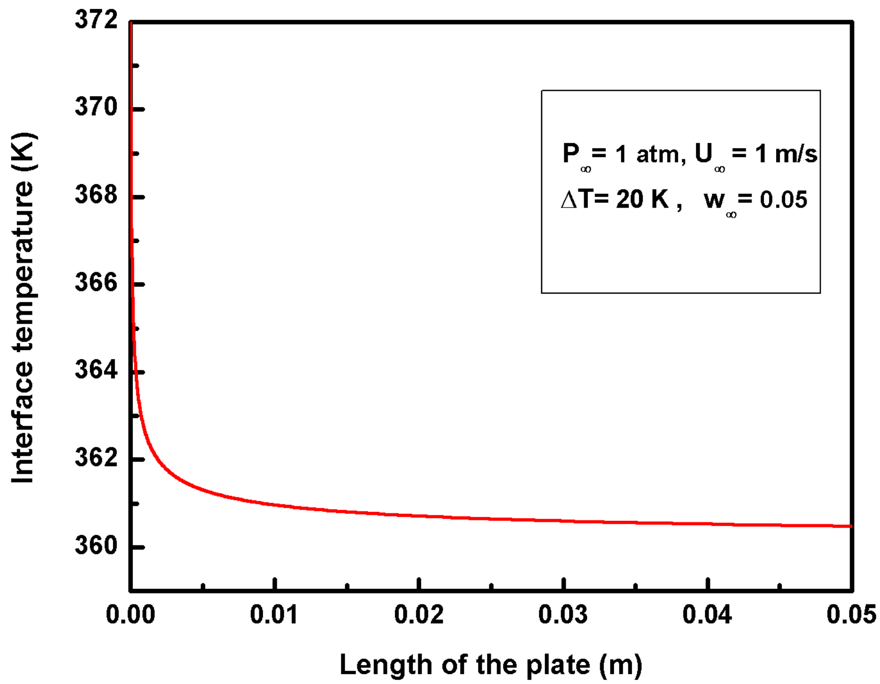

Figure 7 (a): Interface temperature profile for air mass fraction 0.05

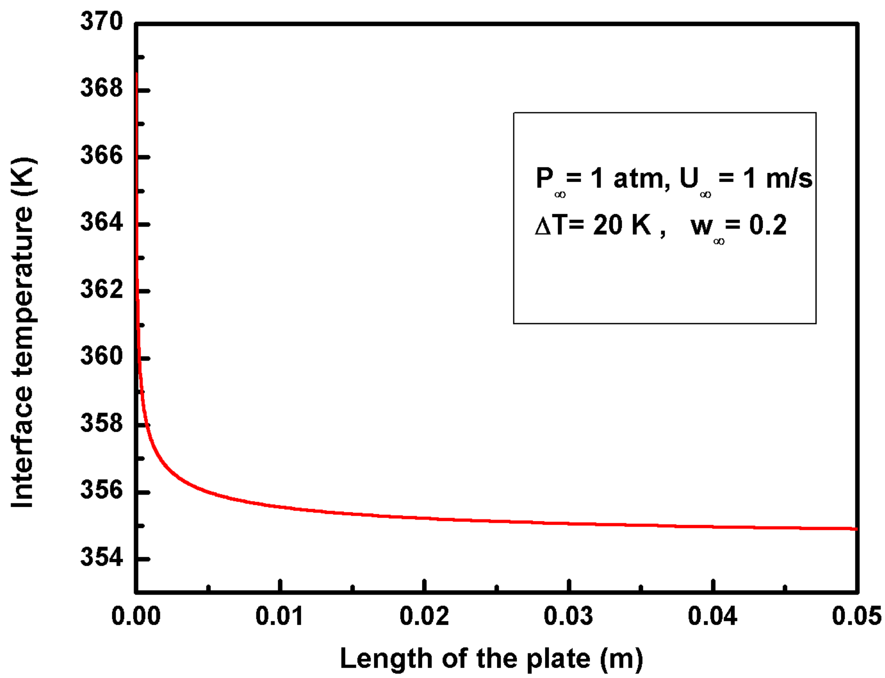

Figure 7 (b): Interface temperature profile for air mass fraction 0.2

3.3 Heat transfer characteristics of forced convection condensation in presence of non-condensable gas

3.3(a) Effect of free stream gas concentration

The variation of free stream gas concentration on condensation process has been analyzed for two different air mass fractions of 0.05 and 0.2. These analyses were performed for $U_\infty = 1m/s$, $P_\infty = 1\ atm$ and $\Delta T = 20k$. Increasing w_∞ creates an additional resistance near vapour-liquid interface region. This decreases vapour concentration at the interface to maintain total mass fraction as unity. This leads to decrement of saturation temperature of vapour at the interface. As a result interface to wall temperature decreases continuously along the axial position. Due to this condensation process as well as liquid mass transfer through interface decreases significantly. Fig. 8(a)-(b) describe such behaviors.

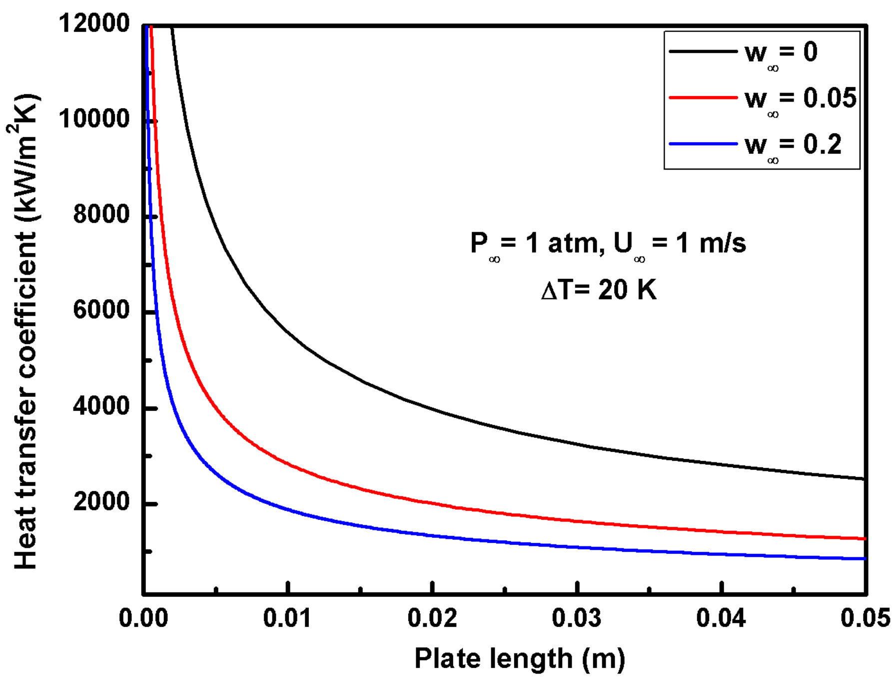

Figure 8(a): Variation of heat transfer coefficients for different air mass fractions

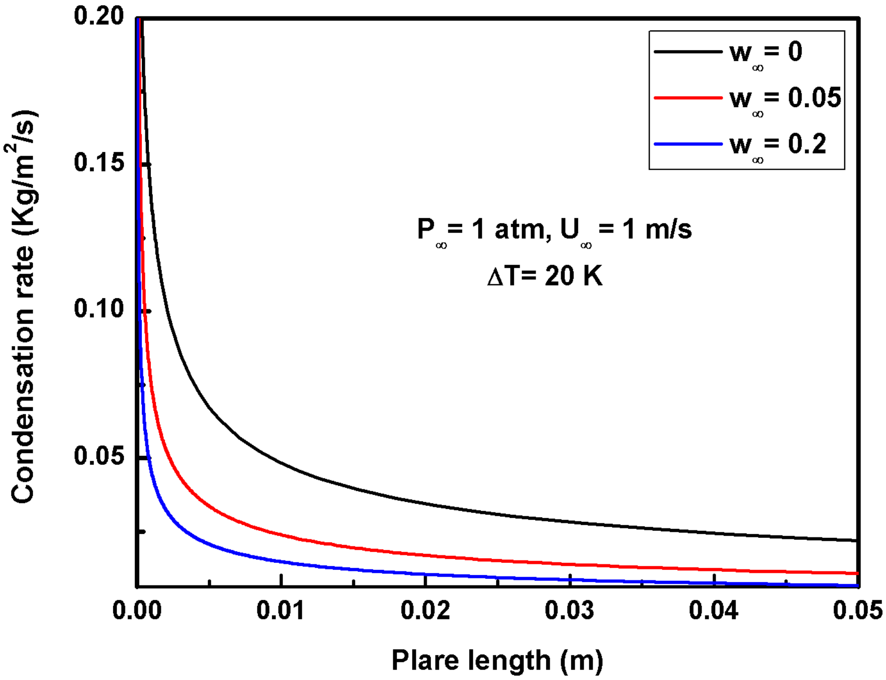

Figure 8(b): Variation of condensation rates for different air mass fractions

3.3 (b) Effect of mixture Reynolds number

The effect of inlet mixture Reynolds number is observed for two different air-mass fractions of 0.05 and 0.2. The analysis is performed at 1atm pressure for two free streams to inlet temperature difference, $\Delta T = 10K$ and $\Delta T = 20K$. Fig.9(a) demonstrates the variation of average heat transfer coefficient with different free stream velocities for $w_\infty = 0.05$. This behavior is illustrated in terms of three non-dimensional numbers such as Nu_{av}, Re_m,Ja_{in}. Increase in Re_m means forced convection effect is pronounced and as a result average heat transfer coefficient and Nusselt number are increased. For a fixed value of Reynolds number increasing ΔT causes reduction in heat transfer coefficient. This is because with ΔT, increase in rate of wall heat flux is much slower compare to increase in bulk temperature difference itself. Due to this average heat transfer coefficient also decreases. Fig.9 (b) shows the same variations of Nu_{av} with respect to Re_m for $w_\infty = 0.2$. with two same ΔT. The only difference is lowering the average Nusselt number because of higher air concentration.

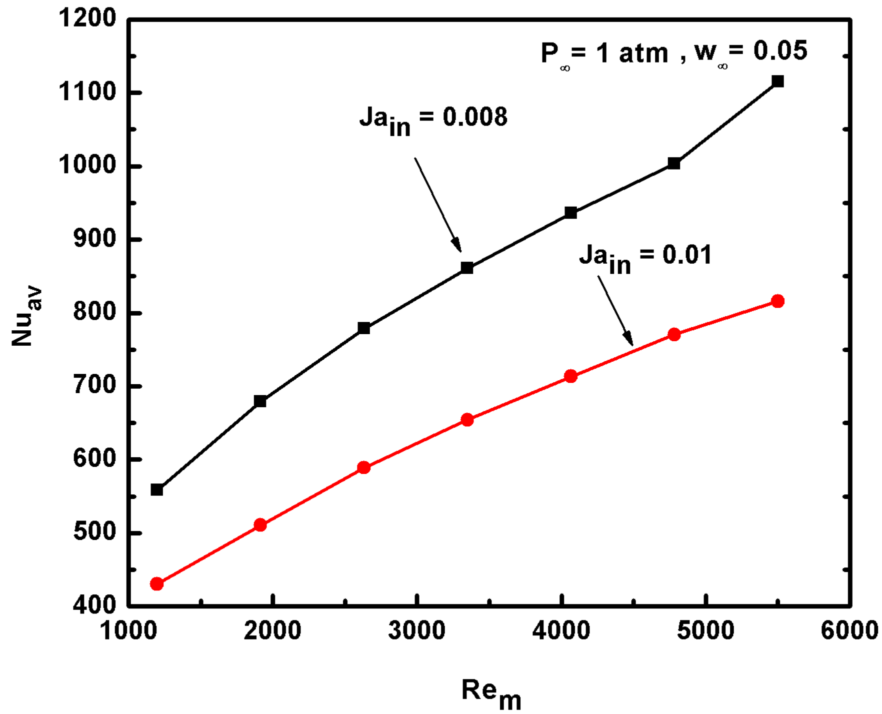

Figure 9 (a): Variation of average Nusselt number with different mixture Reynolds number for air mass fraction 0.05

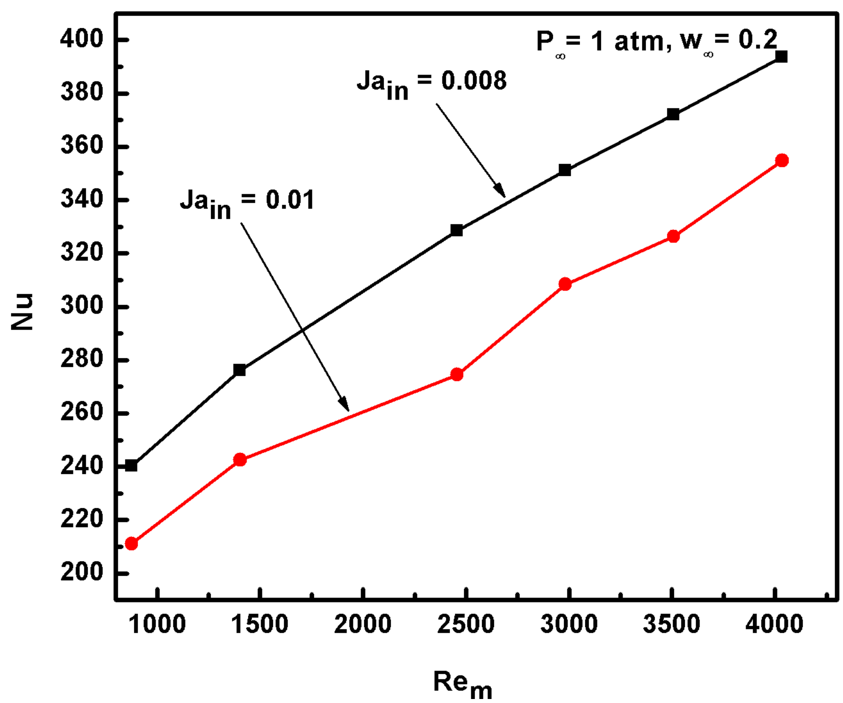

Figure 9 (b): Variation of average Nusselt number with different mixture Reynolds number for air mass fraction 0.2

3.3 (c) Effect of variation of free stream to wall temperature difference

The results show in Fig. 10(a) and Fig. 10(b) represent effect of variation of ΔT on average heat transfer coefficient and Nusselt number. These analyses are performed for two different air concentrations of 5% and 20% at $P_\infty = 1\, atm$ for $Re_m = 800\; and\; 610$. For a fixed free stream pressure and air concentration inlet temperature of mixture is fixed. So increasing ΔT relates decreasing wall temperature. Due to this film thickness increases along the length of the plate. Also ΔT plays an adverse effect on average heat transfer coefficient. As ΔT increase wall heat flux is also increased due to rise of $(T_i - T_w)$ but rate of increasing of wall heat flux is much slower compare to increase of $(T_\infty - T_w)$. As a result overall heat transfer coefficient decreases. For a large value of ΔT, rate of decrement of heat transfer coefficient and average Nusselt number is slower compare to early case, this is because now wall heat flux effect is significant along with ΔT. The trends are same for both in 0.05 and 0.2 air mass fraction but decreasing average Nusselt number is more prominent in second case due to higher air concentration.

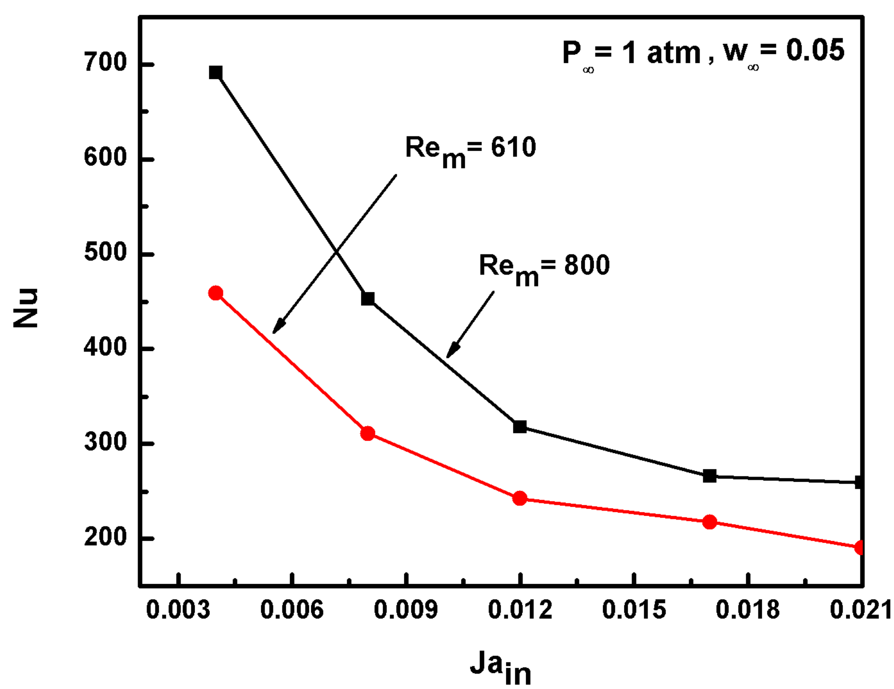

Figure 10 (a): Variation of average Nusselt number with different inlet mixture Jacob number for air mass fraction 0.05

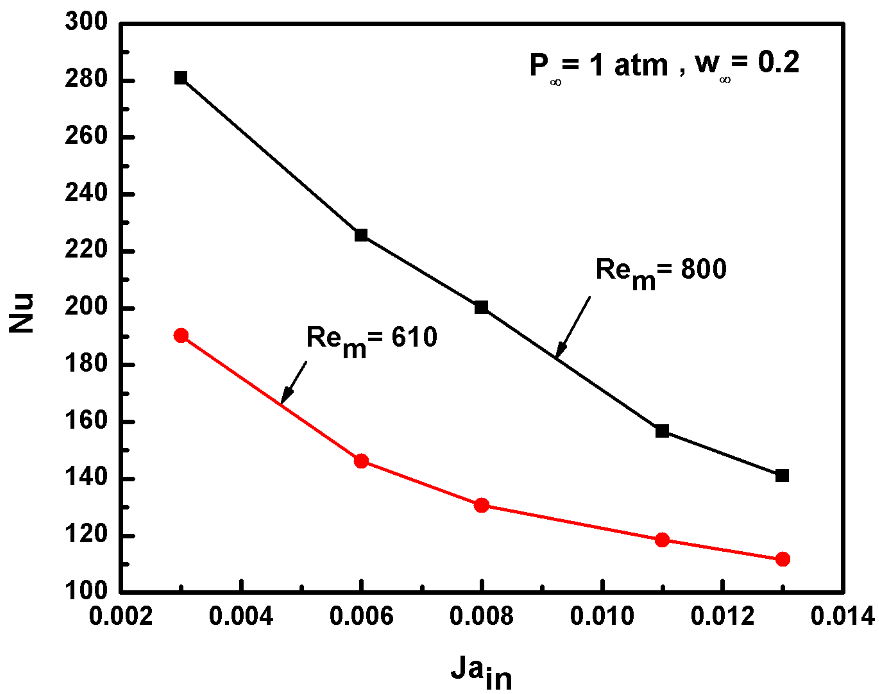

Figure 10 (b): Variation of average Nusselt number with different inlet mixture Jacob number for air mass fraction 0.2

Summary

Results have been obtained for three different air concentrations of 0%, 5%and 20%. Different governing parameters are used at inlet conditions such as: mixture Reynolds number, temperature and air mass concentration to analyze such phenomenon. Based on the following investigations, we get some important conclusions, presented here –

Increasing air mass fraction plays an important role on heat and mass transfer process. As the air concentration increases, due to non-diffusive nature of non-condensable, it accumulates near the vapour-liquid interface region and due to this lowering of interface saturation temperature has been observed. This decreases interface to wall temperature. As a result condensation rate decreases.

For a fixed value of air concentration and bulk temperature difference if we increase mixture Reynolds number as a function free stream velocity, plays a favorable effect on condensation process. This is because rise in free stream velocity increases force convection effect which ultimately leads to greater rate of condensation.

Increasing ΔT, taking other parameters as constant also decreases heat transfer rate. This is because rise in bulk temperature difference results increase in liquid film thickness and wall heat flux but this increment did not match the increase of inlet to wall temperature difference. As a result overall heat transfer coefficient and average Nusselt number.

Chapter 4

Conclusion and Future Scope

In the present thesis a detailed analytical study has been performed to investigate the effect of non-condensable gas on condensation process. In most of the practical situation vapour always contains some amount non-condensable gas. Due to its adverse effect on condensation process this phenomenon plays an important role in many industrial applications such as in various power plants, refrigeration systems etc. Especially in case of nuclear power plants this process has important implementations on safety performance. To understand various complex mechanisms regarding this process a concrete theoretical study and better modeling approach is required. This research mainly focuses on above mentioned areas.

This thesis mainly deals with forced condensation over a horizontal flat isothermal surface. An analytical model is developed with the help of boundary layer integral method. Mass, momentum, energy equations for both mixture and liquid phases and one species conservation equation for mixture phase have been used and simplified using boundary layer approximation. Integration are performed across each boundary layer and finally with the help of free stream and interfacial conditions several Odes are formed which have been solved using MATLAB. This model has been validated against experimental results obtained by Kang and Kim [6]. The results obtained from this model include various parameters such as film thickness of each layers, interfacial temperature and air-mass fraction variation along liquid-vapour interface region. Further processing of these results velocity, temperature and concentration profiles across each layers, heat transfer coefficient as a function of axial position have been obtained. In this analysis we have considered two different air mass fractions such as 0.05 and 0.2.

According to the results -

- Heat transfer coefficient is decreased with increase in air mass fraction due to more accumulation of non-condensable near the interface region. This directly affects rate of condensation.
- Velocities for both liquid and mixture follow the parabolic variation across the boundary layer.
- Liquid temperature varies almost linearly within the film but in mixture layer it follows the non-linear distribution.
- Due to non-diffusive nature air mass fraction is continuously increased from free stream to interface.
- Interface temperature decreases along the length of the plate as a result condensation process is also decreased.
- With increase in mixture Reynolds number average Nusselt number also increases. This is quite expected because increase in mixture Reynolds number free stream velocity increases which increases the forced convection effect.
- Average Nusselt number is decreased with increase in inlet mixture Jacob number. This is due to rise in Jacob number film thickness increases but increment of wall heat flux is much slower compare to bulk temperature difference.

Future Scope

Results show that this integral model works well to analyze forced convection of vapour air mixture in case of a flat horizontal isothermal plate. So this method extends further to study mixed convection film condensation of vapour along with air or other different non-condensable gases in case of various geometries.

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