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COMPARATIVE STUDY OF WATER VAPOUR CONDENSATION FROM HUMID AIR FLOWING ON A VERTICAL PLATE AND A PIPE IN A LAMINA ALONG WITH CFD SIMULATION

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ABSTRACT

The Condensation of water vapor from humid air takes place in thermal systems. Theoretically the problem of water vapor condensation from humid air flowing on a vertical pipe is tackled and vertical plate in laminar flow is formulated. The vapor flow on the plate condenses at dew point temperature of the vapor air mixture. The vapor condensing will diffuse on to the wall of vertical plate and vertical pipe through a non-condensable gas film of high concentration air. The condensing vapor releases both the convection and latent heats to the wall of the plate and from inlet of pipe. Thus the problem is treated as a combined heat and mass transfer problem in both cases. The mass, momentum and energy balance equations for the vapor air mixture flowing on the plate and in the pipe and diffusion equation for the vapor species are considered. The flow of the condensate is laminar. The momentum and energy balance equations for the condensate film are also considered. The gas phase and liquid phase equations are solved by a finite difference method. The system parameters considered are temperature, gas phase Reynolds number and relative humidity of air at the inlet of plate and pipe. In both the cases, the numerical results are obtained to estimate the local condensation Nusselt number, condensate film thickness and gas-to-liquid interface temperature for different system parameters. The gas phase convection Nusselt and Sherwood numbers are also computed. The part modeling of plate and pipe is designed in CATIA V5 R19. The CFD simulation of plate and pipe is done with a comparative study.

Keywords: Condensation, non-condensable gas film, Humidity, Convection, CATIA, CFD.

INTRODUCTION

Condensation is defined as the phase change from the vapor state to the liquid state, and occurs when the temperature of the vapor is reduced below its saturation temperature corresponding to the system pressure. This is usually done by bringing the vapor into contact with a solid surface whose temperature is below the saturation temperature of the vapor. The latent energy of the vapor is released, heat is transferred to the surface, and the condensate is formed. This is called surface condensation. Another mode of condensation is direct contact condensation in which the vapor is brought into contact with a cold liquid. Surface condensation can be classified into two categories viz., film-wise and drop-wise condensation, depending on the condition of the surface.

Film wise condensation, in this, the liquid condensate wets the solid surface, spreads out and forms a continuous film over the entire surface .The liquid flows down the cooling surface under the action of gravity and the layer continuously grows in thickness because of newly condensing vapors. The continuous film offers thermal resistance and restricts further transfer of heat between the vapor and the surface. This type of condensation is generally characteristic of clean and uncontaminated surfaces.

Drop wise condensation, in this, the liquid condensate collects in the form of droplets and does not wet the solid cooling surface. The droplets develop in cracks on surface, grow in size and break away, and eventually run off the surface without forming a film. As there is no film barrier to heat flow, higher heat transfer rates are experienced. In this condensation it is a common practice to use surface coatings that inhibit wetting and hence stimulate drop-wise condensation. Silicones, Teflon and an assortment of waxes and fatty acids are often used for this purpose however such coatings gradually lose their effectiveness due to oxidation, fouling or outright removal, and film-wise condensation eventually occurs.

LITERATURE SURVEY

Othmer [1] was the first conducted the experiments to study the effect of presence of small amounts of air on the steam condensation rate on an isothermal condensing surface. He found that the heat transfer would be reduced by 50% or more due to the existence of a very small amount of air (0.5 % by volume) in the bulk of the vapor. Colburn and Hougen [2] were the first to tackle theoretically the effect of NCG during the condensation of steam and non-condensable mixture and developed theory for condensation mass transfer which was controlled by mass concentration gradient through non-condensable layer. Meisenberg et al. [3] performed experiments to investigate the effect of presence of air during the condensation of pure vapor on the outside of the vertical tube and found that with an increase in weight fraction of NCG the average heat transfer coefficient decreases. Hampson [4] performed experiments for the condensation of pure vapor with nitrogen as NCG on a vertical flat plate and found that the heat transfer coefficients decreases in the presence of nitrogen.

Sparrow and Eckert [5] solved theoretically the problem of effects of non-condensable gases on laminar film condensation for a stagnant vapor-gas mixture on an isothermal vertical plate and solved the mass, momentum and energy equations using a similarity transformation. Their analysis indicated that the condensing rate is dependent on the bulk gas mass fraction, the vapor-gas mixture Schmidt number, the viscosity ratio parameter, $\left(\rho_l\mu_l/\rho_g\mu_g\right)^{0.5}$ and sub-cooled boiling parameter, $C_{p,l}(T_0 - T_w)/\lambda\,Pr_l$.

Sparrow and Lin [6] extended the theoretical model developed by Sparrow and Eckert [5] and predicted condensation heat transfer in the presence of a NCG on a vertical cooled surface. The predicted results of Sparrow and Lin [6] agreed with the experimental results given by Othmer [1]. Minkowycz and Sparrow [7] extended the model of Sparrow and Lin [6] considering the interfacial resistance, superheating, free convection due to temperature and concentration gradients, mass diffusion and thermal diffusion, and variable properties in both the liquid and gas-vapor regions. and concluded that the effect of interfacial resistance, thermal diffusion and property variation in the condensate film and on the vapor gas mixture are to have a little effect on heat transfer. The numerical solutions of theoretical models developed by Minkowycz and Sparrow [7] and Sparrow and Lin [6] require extensive computation.

To reduce the computation time, Rose [8] presented an approximate integral boundary layer solution assuming uniform properties except for density in the buoyancy term of the momentum equation. He used plausible velocity and concentration profiles for the vapor gas boundary layer and assumed that these two layers had equal thickness.

Taitel and Tamir [9] analyzed the effect of a NCG present in the bulk of a vapor in direct contact condensation. The model considered was a free laminar stream that is exposed to its own vapor atmosphere containing the inert gas. They observed reduction in the heat transfer rates near the leading edge, which may be significant depending on the concentration of the NCG and the temperature driving force. The reduction in heat transfer accentuated at lower pressures.

The reduction in heat transfer coefficients in the presence of NCG obtained from experiments of Meisenberg et al. [3] and Hampson [4] were far less compared to the heat transfer coefficients obtained from the numerical results of Minkowycz and Sparrow [7]. To provide additional experimental results Sleger and Seban [10] conducted experiments for the condensation of steam in the presence of air on a vertical surface with saturated mixtures of air and steam at sub atmospheric pressure. They compared experimental results obtained with the numerical results given by Minkowycz and Sparrow [7]. It was found that the measured condensation rates were about 20 percent above the predictions given by Minkowycz and Sparrow [7].

Felicione and Seban [11] conducted experimental and theoretical study on laminar film condensation of stagnant vapor in the presence of a NCG on a vertical surface. The effect of solubility of gas in the liquid for four different gases viz., air, argon, carbon-dioxide and krypton was included in the analysis. Felicione and Seban [11] found that for gases heavier than the vapor, the effect of gas solubility in condensate was not appreciable.

Al-Diwany and Rose [12] reported heat transfer measurements for condensation of steam in the presence of air, argon, neon and helium. The experimental data for steam-air, steam-argon and steam-neon showed satisfactory agreement with the predicted theoretical values of Minkowycz and Sparrow [7]. The results obtained for steam-helium mixture showed the same trend as those of steam-

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air, steam-argon and steam-neon, but the heat flux for the case of helium was slightly less compared to higher molecular weight gases of air, argon and neon.

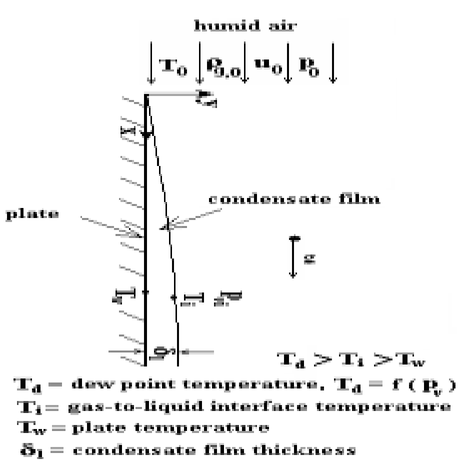
Mori and Hijikata [13] tackled theoretically the problem of free convective condensation on an isothermal vertical surface in the presence of non-condensable gas by solving the equations of liquid film and boundary layers adjoining the liquid film. Rutunaprakam and Chen [14] studied quiescent condensation of a Freon-l2-nirogen mixture on a vertical plate. The assumptions and the set of equations that were used are the same as in Minkowycz and Sparrow [7]. At the interface they used continuity of velocity, mass flow, and shear stress. They obtained Nusselt numbers from numerical results. They observed that the larger shear force at the interface, caused by the buoyant effect, may introduce instability in both vapor layer and at the interface.

PHYSICAL MODEL AND FORMULATION

The work is about water vapor condensation from humid air flowing in a lamina. The comparison is done between numerical calculations between a vertical plate and a Pipe. The CFD simulation is also done to both vertical plate and pipe. For this research, the physical model and formulation is done as follows in both cases.

Case-1: (Vertical Plate) The physical model is shown in Fig.1. Air flows with a velocity u_0 and at temperature T_0 over a vertical flat plate, which is maintained at a constant temperature T_w, and $T_w < T_0$. The air is in laminar flow. The air entering at x = 0 contains water vapor with a relative humidity R_H, where $R_H = p_{v0} / p_{vs0}$, p_{v0} is the partial pressure of water vapor in the vapor- air mixture at inlet and p_{vs0} is the saturation vapor pressure corresponding to the temperature T_0.

Figure-1:

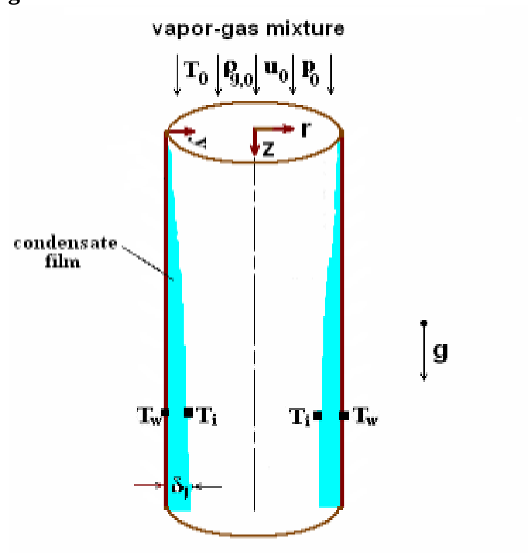


Physical model and configuration

The vapor density ρ_{v0} at the inlet is related to p_{v0} by the equation that $\rho_{v0} = p_{v0}/(R_v T_0)$, where R_v is the vapor constant. $R_v = 8314 / (18x10^5)$ bar-m^3-kg^{-1}-K^{-1}. When the flowing air comes in contact with the cold plate, water vapor from the gas phase condenses on the plate. The condensation takes place by the diffusion of the water species onto the plate through an air film, which forms a cover over the plate.

Case-2: (Vertical Pipe) Steady, laminar, film-wise condensation of vapor from a mixture of a vapor and a non-condensable gas in high concentration flowing in a vertical tube with down flow is considered. The inner diameter of the tube is D and length of the tube is L. The flow of the vapor-gas mixture is laminar. The physical model under consideration is illustrated schematically in Fig.2.

Figure-2:



Laminar flow of vapour gas-mixture in a vertical pipe

The flow is two-dimensional and symmetrical about the center line of the pipe. The u and v are the velocity components in z-direction (flow direction) and radial direction respectively. The water vapor-gas mixture enters at inlet of the pipe with a Reynolds number, temperature, total pressure and relative humidity. The tube surface is maintained at a constant temperature.

CFD SIMULATION

Computational Fluid Dynamics (CFD)

CFD is acronym of the Computational Fluid Dynamics. Computational refers to do with the applications of mathematics, computation and Fluid Dynamics for which it refers to the things which flows in concern with dynamics. CFD is a commercial tool which is computational technology and enables us to study things which had a flow behavior. It is not only predicts the fluid flow

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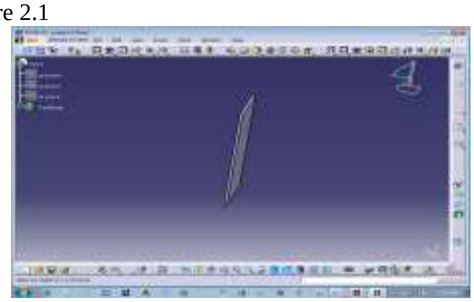
behavior, but also mass, heat transfer, chemical reaction, change in phase, mechanical movement and stress or the deformation that relates to the solid structures.

CFD is focused in obtaining the numerical solutions to the problems of fluid flows by using the computers. The advantages of the large-memory and high-speed computers had enabled CFD to obtain the solutions to many fluid flow problems which also includes incompressible or compressible, turbulent or laminar, chemically non-reacting or reacting. CFD is a science of predictions that involves chemical reactions, mass transfer, heat transfer and related phenomenon which solves the mathematical equations that governs the processes by using the computational methods.

In case of this CFD, the art of replacing differential equations which governs Fluid Flows and with the set of algebraic equations which is a process called discretization and in turn it is solved with aid of digital computers and got approximate solutions. The discretization methods that used in it are Finite Element Method, Boundary Element Method, Finite Volume Method and Finite Difference Method and is the most common method in the applications of CFD. In past decade, CFD had been widely used in the industries such as in the fields of mechanical, aerospace, chemical, civil and electrical engineering and also as well as the chemistry and physics.

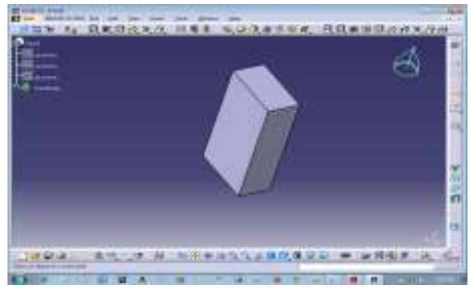
The ANSYS-CFX 15.0 simulation is done for both vertical plate and vertical pipe by modeling the plate and pipe and also creating domain in CATIA V5 R19. The simulation procedure and humid air velocity and temperature in the form of streamlines for the both cases are represented.

Figure 2.1



The part modeling of vertical plate in CATIA V5 R19 is as follows.

Figure 2.2



The plate in domain is as follows.

Figure 2.3

The part modeling of plate is saved in .stp format and exported to Ansys CFX.

Figure 2.4

After exporting the meshing takes place.

Figure 2.5.1

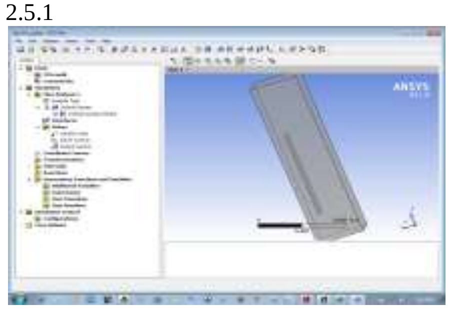
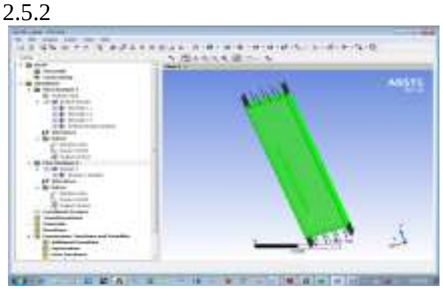


Figure 2.5.2



After meshing, boundary conditions are applied.

The solution takes place and simulation results are as follows.

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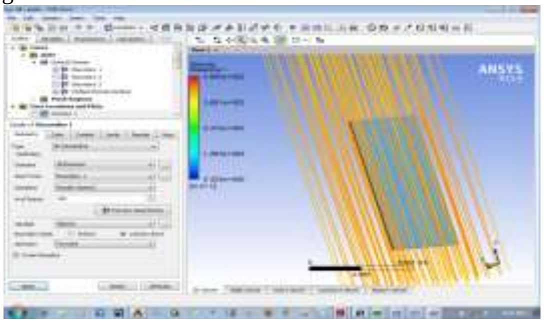
Figure 2.6



The velocity flow of fluid.

The temperature parameter is as follows.

Figure 2.6.1

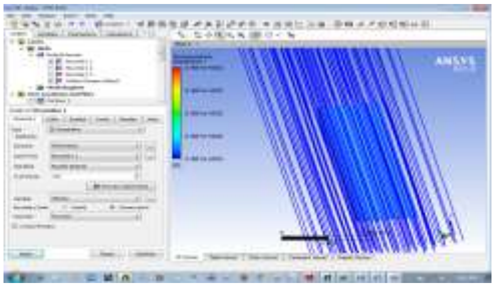


Figure 2.6.2

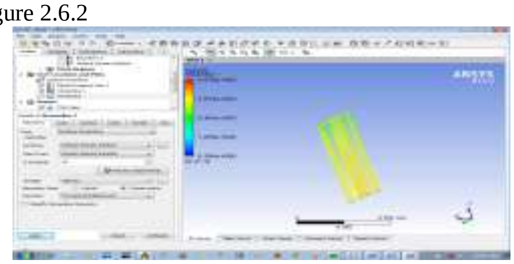
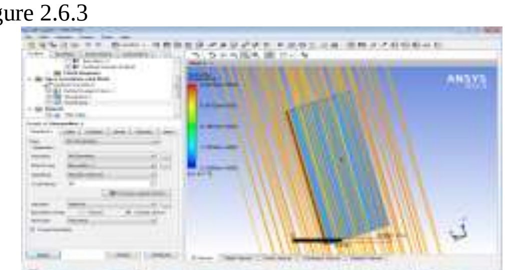
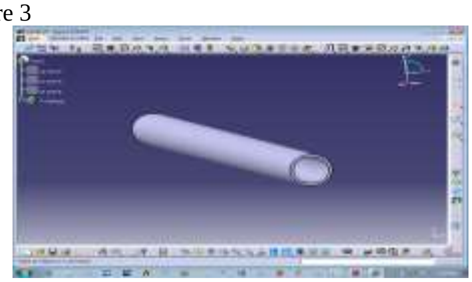


Figure 2.6.3



The part modeling of vertical pipe in CATIA V5 R19 is as follows.

Figure 3



The part modeling is saved in .stp file and exported to CFX.

Figure 3.1

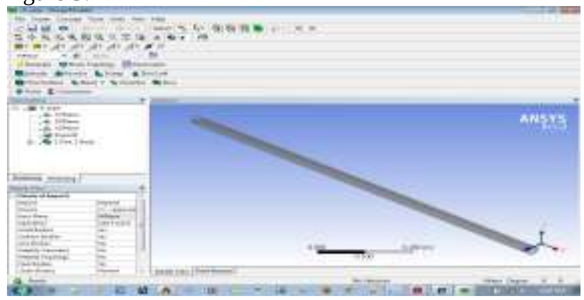
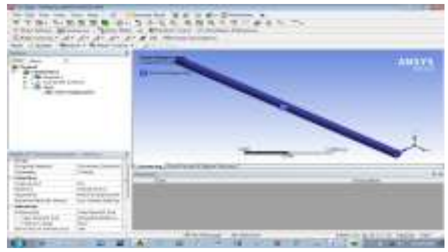


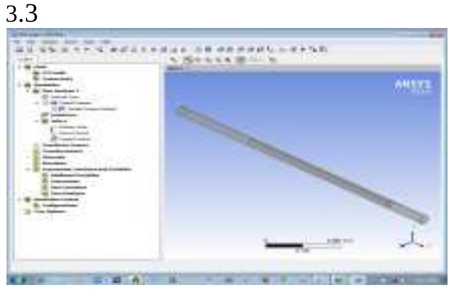
Figure 3.2



The .stp file is generated and undergone meshing is as follows.

The boundary conditions are applied for further procedure.

Figure 3.3



The simulation results are as follows. The velocity flow of the fluid is as follows.

Figure 3.4.1

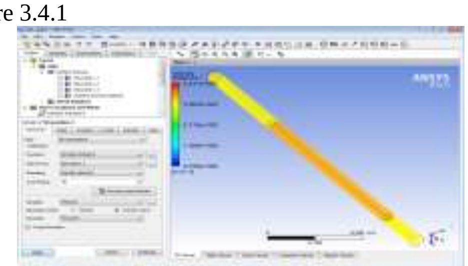
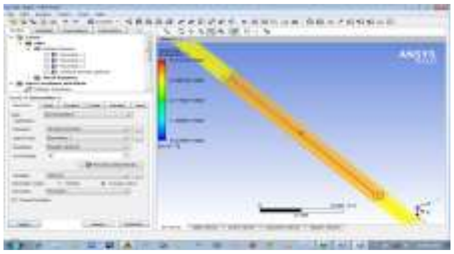
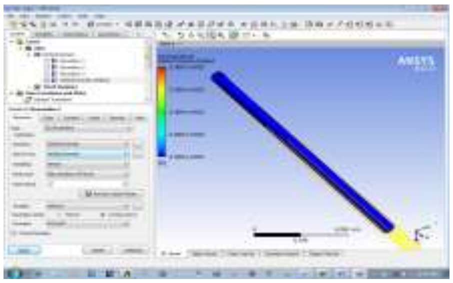


Figure 3.4.2



The temperature on the pipe is shown below.
Figure 3.5



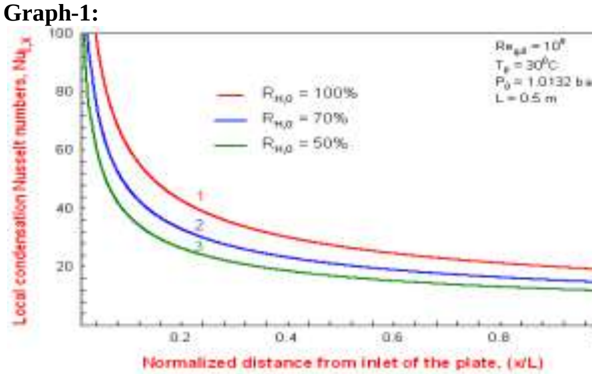
RESULTS AND DISCUSSIONS

Results in case of Vertical Plate: The input system parameters of the present study are the $R_{H,0}$, the relative humidity of the air entering the plate, $Re_{g,0}$, the gas phase Reynolds number at the plate inlet, and T_0, temperature of the air at plate inlet. The numerical results are obtained for the chosen common parameters of constant wall temperature, $T_w = 5°C$, the length of the plate, L = 0.5 m, and the total pressure of the air at the plate inlet, P_0 =1.0132 bar. From the numerical results the local values of Nusselt number, condensate film thickness and gas-liquid interface temperature are estimated for different values of the system parameters. The average condensation Nusselt number, $Nu_{l,av}$ and average gas phase convection Nusselt number are also computed from numerical results for different values of system parameters.

In forced convection condensation, the total heat transferred to the wall is the sum of the heat transferred by convection through the gas film and the latent heat of condensation of vapor. The condensation Nusselt number reported in this work is calculated from the total heat transfer rate. Graph.1 show the effect of $R_{H,0}$ on $Nu_{l,x}$ for prescribed values of T_0 and $Re_{g,0}$. A comparison of curves 1, 2 and 3 shows a decrease in local condensation Nusselt number with a decrease in the value of $R_{H,0}$, due to an increase in the percentage of non-condensable gas in the vapor-air mixture at inlet and a consequent decrease in the vapor content of the mixture. Graph.2 show the effect of T_0 at fixed $R_{H,0}$ and $Re_{g,0}$ on the local condensation Nusselt number, $Nu_{l,x}$. The saturation vapor pressure increases with an increase in T_0, resulting in an increase in the humidity of water vapor in the gas phase. The heat transfer by convection also increases due to an increase in the temperature difference between the vapor-gas mixture core and wall with an increase in T_0. Hence $Nu_{l,x}$ increases with an increase in T_0, due to increase in total heat transfer to the condensate film. The heat and mass transfer during condensation of vapor from vapor-air mixture is also controlled by flow hydrodynamics. As $Re_{g,0}$, the gas phase Reynolds number at inlet increases, the flow rate of vapor increases due increase in the flow rate of vapor-gas mixture. The higher the flow velocity, the higher the heat transferred to the condensate film.

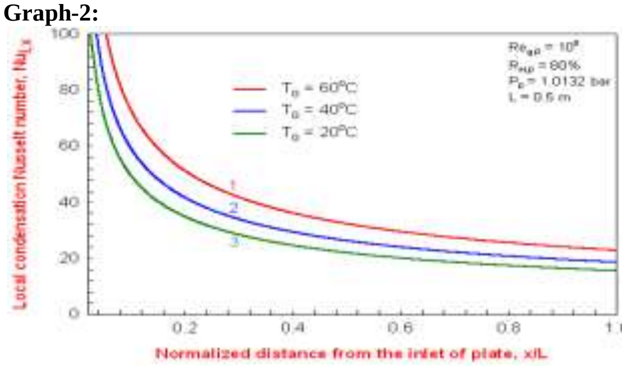
In the above graph.3 shows the effect of the gas phase Reynolds number at the plate inlet ($Re_{g,0}$) on average gas phase Nusselt number ($Nu_{g,av}$). The common parameters chosen are P_0 = 1.0132 bar, $R_{H,0}$ = 100% and T_w = 5 °C, and T_0 = 30°C. The $Re_{g,0}$ varies from 0.5 x 10^5 through 3x10^5. The Nusselt and Sherwood numbers are close enough to one other, since the Prandtl and Schmidt numbers are nearly the same for the water vapor-air mixture.

Graph-1:



Effect of inlet relative humidity ($R_{H,0}$) on $Nu_{1,X}$.

Graph-2:



Variation of local condensation Nusselt number with T_0.

Graph-3:

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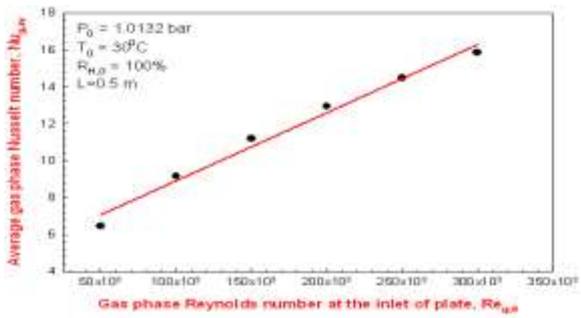


Variation of average convection Nusselt number with $Re_{g,0}$

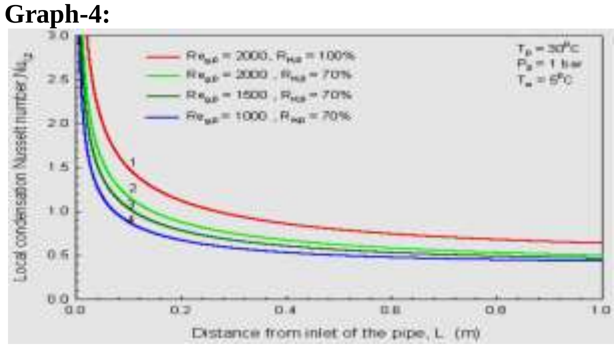
Results in case of Vertical Pipe:

Numerical results are obtained for local values of condensation Nusselt number, Nul,z, condensate Reynolds number, Rel,z, gas-to-liquid interface temperature, Ti, convection Nusselt number, Nug,z, gas phase Sherwood number, Shg,z and gas-vapor mixture temperature as functions of normalized downstream distance, z/L, for different values of system parameters. The system parameters considered in the present study are the RH,0, the relative humidity of the air at the inlet of the pipe, Reg,0, the gas phase Reynolds number at the pipe inlet, T0, temperature of the air at pipe inlet, and P0, the total pressure of the air in the pipe at inlet. The chosen common parameters are the constant wall temperature, Tw = 5oC, length of the pipe, L = 1 m, and the diameter of the pipe, D = 0.025 m.

Effect of system parameters on local values:
Graph.4 shows the effects of the system parameters RH,0 and Reg,0 on the local condensation Nusselt numbers, Nul,z. The heat transfer from humid air during condensation depends on two interdependent effects, viz., sensible heat transfer through the gas film due to the temperature difference between air stream and wall, and the latent heat transfer due to the water vapor mass fraction difference. At prescribed values of P0, T0 and Reg,0, with decrease in RH,0 the water vapor content of the mixture at inlet decreases. The latent heat transfer and consequently the total heat transfer to the condensate film decreases with decrease in RH,0. Therefore, the local condensation Nusselt numbers decrease with a decrease in RH,0. The same trend can be observed from the curves 1 and 2 of Graph.4.

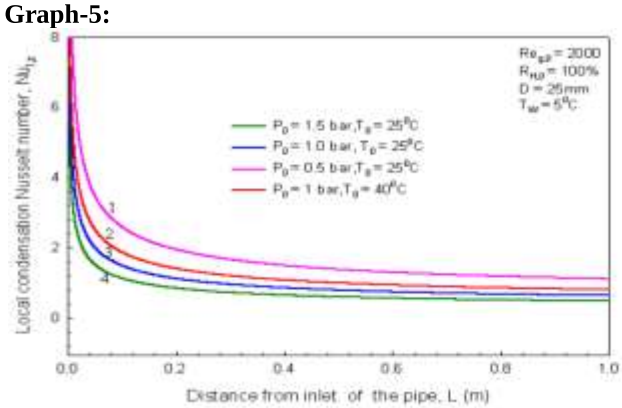
The curves 2, 3 and 4 of Fig.2 show the effect of Reg,0 on Nul,z when the other system parameters remain the same. At a constant value RH,0 an increase in the gas phase Reynolds number at inlet (Reg,0) results in an increase in the flow rate of the vapor. Hence the local condensation Nusselt numbers increases with an increase in Reg,0. Graph.5 shows the effects of the system parameters T0 and P0 on the local condensation Nusselt numbers, Nul,z. The curves 1 and 2 of Fig. 3 show the variation of local condensation Nusselt number, Nul,z with inlet temperature of air, T0. At a given system pressure, P0, relative humidity, RH,0, and mixture Reynolds number, Reg,0, the latent heat transfer to the liquid film increases with an increase in T0 due to increase in the saturation vapor pressure and a resulting increase in the vapor density. The heat transfer by convection also increases due to an increase in the temperature difference between the gas vapor mixture core and wall with an increase in T0.

Graph-4:



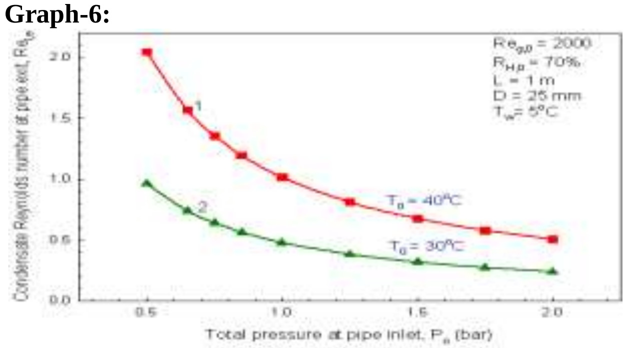
Variation of local condensation Nusselt number with $Re_{g,0}$ and $R_{H,0}$

Graph-5:



Variation of local condensation Nusselt number with P_0 and T_0.

Graph-6:



Effect of P_0 and T_0 on condensation rate ($Re_{1,e}$)

Graph.6 shows the variation of Rel,e, the condensate Reynolds number at the pipe exit with P0, ranging from 0.5 to 2 bar for T0 = 30oC and T0 = 40oC. The chosen common parameters are Reg,0 = 2000, RH,0 = 70%. The Rel,e changes from 0.96275 to 0.23786 for T0 = 30oC and changes from 2.04212 to 0.50534 for T0 = 40oC. As the total pressure at inlet increases from 0.5 to 2 bar the Rel,e decreases due to decrease in saturation temperature of mixture.

CONCLUSIONS

The following conclusions can be drawn from the present work of condensation of water vapor from air-water vapor mixture flowing on a vertical plate and a vertical pipe.

1. A theoretical model is formulated for the case of laminar film condensation of water vapor from a mixture of vapor and high concentration non-condensable gas flowing on a vertical plate under laminar forced flow. And also a theoretical model is developed for the case of in-tube laminar film condensation of water vapor in the presence of high concentration non-condensable gas, when the gas vapor mixture flows inside a vertical tube under laminar forced flow conditions.
2. Numerical results are obtained for different values of system parameters controlling the process, such as $R_{H,0}$, the relative humidity of the air entering the plate, $Re_{g,0}$, the gas phase Reynolds number at the plate inlet, and T_0, temperature of the air at plate inlet. The salient system parameters controlling the process are found to be inlet temperature T0, gas phase Reynolds number at inlet Reg,0, Relative Humidity RH,0 and inlet pressureP0.
3. From the numerical results the local condensation Nusselt numbers, condensate film thickness and gas-liquid interface temperature are estimated for wide range of system parameters in case of vertical plate. And from the numerical results obtained the local and average condensation Nusselt numbers, condensate Reynolds number and gas-liquid interface temperature are estimated. The variation of local gas-vapor mixture temperature along the length of the pipe is also estimated in case of vertical tube.
4. The average condensation Nusselt number and average gas phase convection Nusselt number are also computed for different values of system parameters in case of vertical plate. The local and average gas phase Nusselt and Sherwood numbers are also computed for wide range of gas phase Reynolds number at inlet in case of vertical tube.
5. In case of vertical plate, the presence of non-condensable gas in high percentage is found to have significant effect on the process of condensation. The condensation Nusselt number and condensate film thickness are decreased significantly. In case f vertical tube the condensation heat transfer coefficients and the rate of condensation decreases considerably with the decrease in the rate of heat transfer in the presence of non-condensable gas in high percentage along the length of the pipe.
6. The best example in case of vertical plate of water vapor-air mixture is considered in view of its wide practical utility. However this theoretical model can be applied to any vapor-gas combination with suitable substitution of property values. And also in case of vertical pipe the example is of water vapor-air mixture is considered in view of its wide practical utility. However this theoretical model can be applied to any vapor-gas combination with suitable substitution of property values.
7. Numerical results are obtained for wide range of the system parameters.
8. The stagnation of fluid velocity to the vertical plate is more when compared to the vertical pipe.
9. The temperature in both cases is minimum with respect to atmospheric conditions.
10. A comparative study is done between vertical plate and vertical tube in case of water vapor condensation from humid air flowing in a lamina.

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