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Film Condensation Behaviour of Steam on Isothermal Walls in Presence of Non-Condensable Gases -A Numerical Investigation

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ABSTRACT
Numerical modelling of condensation had been a challenging task for the researchers for many years due to its time scale and rapidity. Present work deals with the characterization of condensation in presence of non-condensable gases on isothermal surfaces which is a common situation in many condensing devices. The investigation includes the effect of different flow parameters on condensation of saturated steam-air mixture using a numerical approach. The study uses wall condensation model through ANSYS CFX solver. The effect of mass fraction of steam, operating pressure and mass flow rate of mixture is studied. Investigation provides some key characteristics about film condensation which normally remain absent in condensation without non-condensable gas. The findings of this study will provide valuable insight in thermal design process of components incorporating this phenomenon.

I. Introduction

Condensation is a thermodynamic, exo-energetic non equilibrium process of phase transition from vapor to liquid phase. It involves a large amount heat and mass transfer. In most of the practical applications, pure condensation is a rare phenomenon. Presence of non-condensable gases is quite common situation which hampers the condensation process in form of reduced condensation rate by imposing resistance to heat and mass transfer.

The study of reduction in heat transfer rates during condensation of steam in presence of non-condensable gases has been taken under investigation by several researchers in recent time; especially in the field of Nuclear Engineering. The first theory of film-wise condensation of pure stagnant steam on a wall was proposed by Nusselt with certain assumptions and limitations, which provided a firsthand correlation for local heat transfer coefficient along the length of a wall. Later on it was modified by researchers like Vierow and Schrock (1991) [11] and Siddique et al. (1994) [9], where they included some of the parameters - presence of non-condensable gases, interfacial shear of bulk gas in case of forced convection etc. which were not considered in Nusselt's theory. Viewrow et al. (1991) [11] proposed a modified empirical correlation of Nusselt number by correlating it with a degradation factor which was a function of local non-condensable gas mass fraction and local Reynolds number. In one of the earliest experimental study performed by Othmer (1929) [6], the presence of non-condensable gases was investigated. Researcher proposed an empirical correlation relating the heat transfer coefficient to the air/steam volume ratio and to the temperature difference between the cooling surface and bulk steam-air gas mixture. It was also reported that the heat transfer coefficient would reduce by 50% even if 0.5% air by volume was added to the gas mixture.

In the field of Nuclear Engineering, Uchida [7] carried out experiments on external surface of cylinder to investigate effect of type and amount of non-condensable gases on condensation. The operating pressures and the mass fractions of these gases were varied during these experiments. For these experiments, to test the effect of type and amount, different non-condensable gases like air, nitrogen and argon were used. The authors noted that the amount of non-condensable gas rather than the species of the gas determines the rate of heat transfer.

In 1991, Dehbi [2] and his team did experimental investigations of the natural convection condensation from steam-air and steam-air-helium mixtures on a tube using different pressures, mass fractions of vapour/ non condensable gases. The authors developed empirical correlations and reported that the heat transfer coefficient increased significantly with the increase in the total pressure. They also noted that the wall sub-cooling and length scale effects were of secondary importance. Another experimental study on air-steam mixture system Natural convection flow in tube was conducted by Kuhn and Schrock [3] in 1997. They correlated local heat transfer coefficient as a function of local gas mass fraction and Local Reynolds number. They also reported reduction in heat transfer and occurrence of instabilities at high air contents and concluded that the overall heat transfer coefficient value is augmented by forced convection and is reduced by presence of non-condensable gas.

Another significant experimental study was done by Ambrosini (2008) et al. [1] to study the effect of non-condensable mass fraction on condensation heat and mass transfer rates for a flat vertical plate. This experimental study is popularly known as CONAN Experimental Tests.

Along with the theoretical models, several researchers attempted to model the condensation phenomenon numerically under different assumptions. The pioneering numerical work for this phenomenon was carried out by Sparrow (1964) [10] by solving transport equations for the condensate film as well as the steam-gas concentration layer. Mimouni et al. (2011) [5] developed a numerical model for this phenomenon using a two phase flow approach and implemented in Neptune CFD code. Ranz-Marshall [8] correlation was used by him to model droplet condensation as bulk phenomena and wall phenomena both where re-evaporation was also accounted. Vyscocil et al. (2014) [12] proposed a simplified numerical model for volumetric as well as wall condensation where the complexity was reduced by considering heat transfer coefficient to represent the heat transfer through condensate film to wall. Zchaeck et al. (2014) [13] proposed a numerical model based on diffusion of steam through the non-condensable gas layer. In this model the concept of condensate film was discarded and it was assumed that the liquid condensate is removed from the domain by providing sink boundary condition on the cooling wall. The model was validated against the experimental data from Ambrosini (2008) et al. [1] and Kuhn (1997) [3].

It can be summarized that considerable efforts has been made by researcher for complete understanding of condensation mechanism in presence of non-condensable greases using different methodology. A thorough understanding addressing all dimensions of problem still missing and needs deeper and wider investigation under enhanced computing capability. Condensation being a rapid mechanism of small time scale needs numerical approach for its deeper investigation. The problem under investigation addresses this issue. The objective is to understand the behavior of condensation in presence of non-condensable gases under different operating conditions like pressure, amount of non-condensable, sub-cooling temperature difference.

II. Problem Definition and Objective

In some accidental cases of nuclear reactors, large amount of steam and hydrogen gas (produced from high temperature electrolysis of steam) are released into the nuclear containment volume because of failures of steam lines shown in Figure 1. The steam mixes with air present in the containment and raises the pressures and temperature to critical magnitudes which can lead to self-detonation of released hydrogen gas. Condensation of this released steam by means of external cooling mechanisms has been found to be a promising way to reduce the temperature and pressure inside the containment to safer levels.

So as to effectively design the external cooling systems, it is necessary to study the nature of heat transfer behaviour occurring during condensation of steam in presence of non-condensable gases and effect of various parameters on it.

For this parametric study, it is assumed that the saturated steam-air gas mixture flows along and condenses over a vertical flat surface maintained at a lower temperature. The geometry of the computational domain in this study is taken similar to CONAN experimental tests Ambrosini et.al. (2008) [1].

In the present study, saturated mixture of steam and non-condensable gas with predefined concentration flows in the downward direction through a square channel of dimension (0.34m × 0.34m × 2m). One of the sides of this channel is maintained at isothermal condition well below saturated temperature for condensation to take place. Steam tries to condense on the isothermal wall under the influence of non-condensable gases under different parametric variations. This may lead to a variable condensation heat flux along the length of condensing surface which is expected to get affected by mass fraction of non-condensable gas, mass flow rate of the mixture and operating pressure. Effect of these parameters on condensation heat flux can characterise the condensation behaviour to great extent. To complete the characterization, the parameter is proposed to be varied in the following range:

1. Mass fraction of non-condensable gas :0.10 to 0.90
2. Operating pressure :0.25 bar to 10 bar
3. Mass flow rate of mixture : 0.00261 m/s to 2.615 m/s

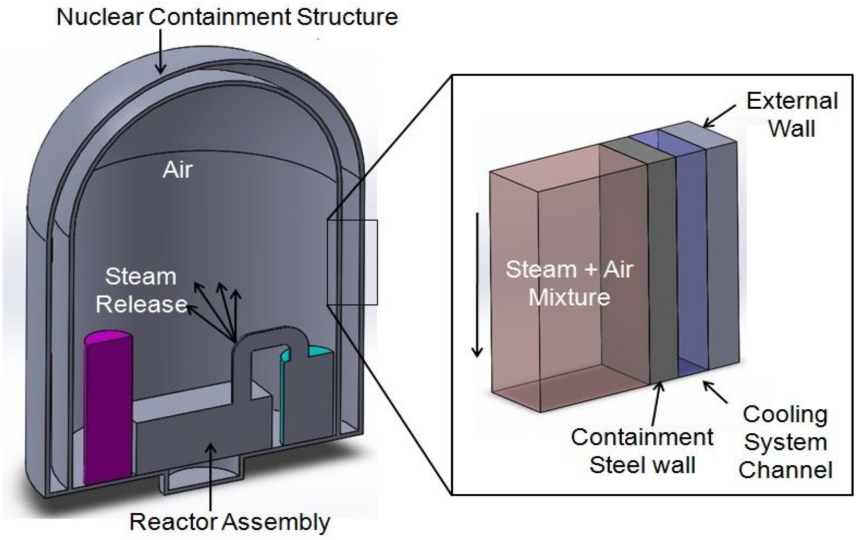

Figure 1. Problem case –nuclear containment structure

The effect of nature of non-condensable gas (i.e. Nitrogen, Hydrogen and Helium) has been excluded from the scope of study.

The objective of the proposed work is to characterize the condensation behaviour of steam air mixture by investigating the value of local condensation heat flux and overall condensation rate by using a numerical approach.

III. Mathematical Model

The fluid medium of the problem under investigation is a multi-component gas mixture of steam and air which are condensable and non-condensable respectively. The flow of the mixture is assumed to be steady, incompressible and follow the characteristics of Newtonian fluid. The formation of condensation film and related changes is expected to be take place under the limits of the boundary layer formed due to mixture viscosity playing a significant role here. Condensate film is expected to spread in laminar and turbulent flow conditions both. The effect of gravity is negligible due to low order of temperature and small mass of the condensate film.

Governing Equations

Mass balance equation

$$\nabla.(\rho\,\vec{V}) = 0 \qquad (1)$$

Momentum balance equation

$$\nabla.(\rho\,u\,\vec{V}) = -\Delta \mathrm{p} + \mu\,(\nabla^2 u) \qquad (2)$$

Energy balance equation

$$\nabla.\left(\vec{V}\,(\rho\ c_p\ \nabla T)\right) = \nabla.(k_{eff}\ \nabla\,T) - \textstyle\sum_j h_i M_i \qquad (3)$$

Where, k_{eff} is effective thermal conductivity, $h_i M_i$ is the energy transfer due to mass diffusion of species, h_i is the enthalpy of condensation and M_i is diffusive mass flux for condensable species *i*.

Species balance equation

$$\nabla.(\rho\,v\,Y_i) = -\nabla.(\overrightarrow{m_i}) + S_i \qquad (4)$$

Where, $Y_i = \frac{mass\ of\ i^{th} component\ \ in\ mixture}{total\ mass\ of\ mixture}$ is mass fraction of i^{th} species, $\overrightarrow{m_i}$ is the diffusive mass flux and S_i is the rate of creation or addition from dispersed phase or any user-defined source term.

Turbulence Model: Investigation by other researchers recommends the use of SST k-ω model for such application. Therefore it is chosen to capture the turbulence occurring in the system.

Transport equation turbulence of kinetic energy (k)

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_i}\left(\gamma_k \frac{\partial k}{\partial x_i}\right) + G_k - Y_k \qquad (5)$$

Transport equation of frequency (ω)

$$\frac{\partial}{\partial x_i}(\rho \omega u_i) = \frac{\partial}{\partial x_i}\left(\gamma_\omega \frac{\partial \omega}{\partial x_i}\right) + G_\omega - Y_k \qquad (6)$$

Where, γ_k and γ_ω represent effective diffusivity of k and ω, G_k, G_ω, Y_k and Y_ω represent generation and dissipation of k and ω due to turbulence respectively.

Wall Condensation Model. The condensation heat mass transfer in presence of non-condensable gases is calculated with help of a numerical model known as wall condensation model. The assumptions under this model are mentioned below.
-The multi-component gas mixture contains at least one condensable species and at least one non-condensable gas species.
-The condensation rate is controlled by the concentration boundary layer.
-The partial pressure of the condensable component at the wall is equal to its saturation pressure evaluated at the interface temperature.
-The wall condensation model is a single phase model, i.e. the transport of condensate film is not modelled. The mass of gaseous phase which is condensed is removed from the system and hence re-evaporation is not taken under consideration.
-Wall functions are not influenced by wall suction.
-In case of conjugate heat transfer, it is assumed that the latent heat released during condensation is absorbed by the solid material at the interface.
In this model, the equations representing mass diffusion of steam through the non-condensable gas for laminar and turbulent boundary layer region are solved along with the governing equations as follows.

(a) Mass Diffusion in Laminar Boundary Layer. In the laminar boundary layer, for the binary gaseous mixture, the mass flux of the non-condensable component is zero and the mass flux for condensable component through the plane parallel to wall and at a distance of x is given by Zchaeck et.al. (2014) [13]

$$M_i = W_i \vec{J_i} = \frac{W_i D_{ij} \rho_m}{W_m \delta} \ln\left(\frac{1-X_i(\delta)}{1-X_i(0)}\right) \quad (7)$$

Where, M_i is condensate mass flux, W_i and W_m are molecular weights of condensable component *i* and gas mixture respectively, J_i molar flux, D_{ij} is the mass diffusivity of condensable component *i* through non-condensable component *j*, and X_i is the molar fraction of condensable component *i* and δ is the distance normal to the cooling wall.

(b) Mass Diffusion in Turbulent Boundary Layer. For turbulent boundary layer, the concentration boundary layer is modelled using turbulent wall functions and the equation for mass flux of condensable component in near wall region is given as follows,

$$M_{iw} = -\frac{D_{ij} W_i \rho_m}{W_m \delta}\left(\frac{Y_{iP} - Y_{iW}}{1-Y_{iW}}\right) \quad (8)$$

Where, Y_i is the molar fraction of condensable species (steam), the subscripts w are refer to wall quantities, and P refers to near wall mesh points. M_i is condensate mass flux, W_i and W_m are molecular weights of condensable component *i* and gas mixture respectively, J_i molar flux, D_{ij} is the mass diffusivity of condensable component *i* through non-condensable component *j*.

3. Numerical Implementation

As depicted in the figure 2, the flow domain can be modeled as 2D x-y plane length wise perpendicular to the isothermal cooling wall of physical flow region shown in figure 1. It is assumed that condensation of steam takes place on a vertical flat isothermal surface in presence of air. The 2D computational domain is of a size 0.38m × 2m which is shown in figure 2 along with mesh used.

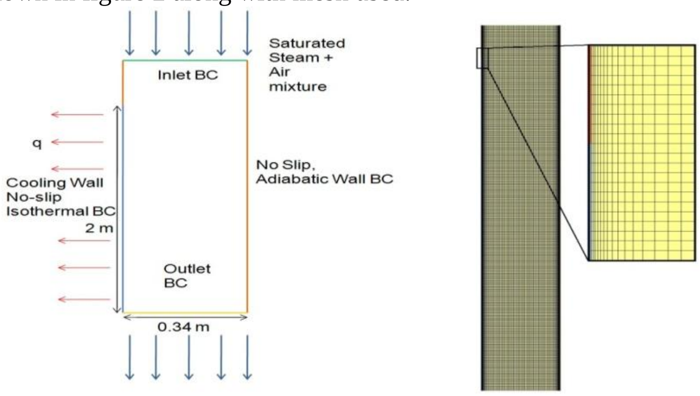

Figure 2. Computational domain with boundary condition details and Meshing

In the region of expected condensation the mesh is refined to capture the variation in flow parameters. Far region is expected not to effect on the results significantly. So they are coarse**.** Quadrilateral cells are used with boundary layer meshing arrangement near isothermal cooling wall.

Numerical Setup. The numerical investigation is carried out in ANSYS CFX commercial software where wall condensation model is used to capture condensation effect. As the flow domain is assumed to incompressible, steady state pressure based solver is used where effect of density variation is neglected. The working fluid is a mixture of saturated vapor and air, where air is a non-condensing medium. The boundary condition imposed on the four boundaries is specified below.
Inlet: steam–air mixture at saturation temperature with velocity inlet;
Outlet: pressure outlet Boundary condition
Cold wall: no-slip flow condition with wall temperature below saturation
Outer wall: no-slip flow condition with adiabatic thermal condition
The saturation temperature of the gas mixture at inlet is calculated from the partial pressure of the steam which is function of steam mass/molar fraction which is formulated as follows

$$T_{saturation} = f(p_{steam}) \tag{9}$$

$$f(p_{steam}) = f(X_{steam} \ or \ Y_{steam}) \tag{10}$$

Where, p, X and Y are the partial pressure, molar fraction and mass fractions respectively. The solver was set up with higher resolution scheme for advection as well as turbulence numeric. A false transient time stepping method is implemented with automatically computed timescale option for the steady state calculation. The convergence criterion is set 1×10^{-5} for all residues.

IV. Results and Discussions

This section deals with grid independence test, validation of result and other cases investigated for the characterization of condensation in presence of non-condensable gases. The testimonial of the present investigation is benchmark CONAN experimental test done by Ambrosini et al. (2008) [1]. In his experimental facility the steam-air mixture allowed to enter into the channel from top in saturated condition with predefined concentration and leaves from the outlet at the bottom. The steam from gaseous mixture condenses on the wall and the condensate is collected at the bottom. Details of few result is tabulated below which is proposed to be used for the mesh dependence work and validation of present numerical work.

Table 1. CONAN test parameters

Test name	V in [m/s]	T in [k]	Non Condensable Mass Fraction	T wall [K]
P10-T30-V25	2.57	348.6	0.707	304
P15-T30-V25	2.61	356.5	0.5722	303.5
P20-T30-V25	2.59	364.5	0.359	305.25
P25-T30-V25	2.6	366.8	0.2789	305.95

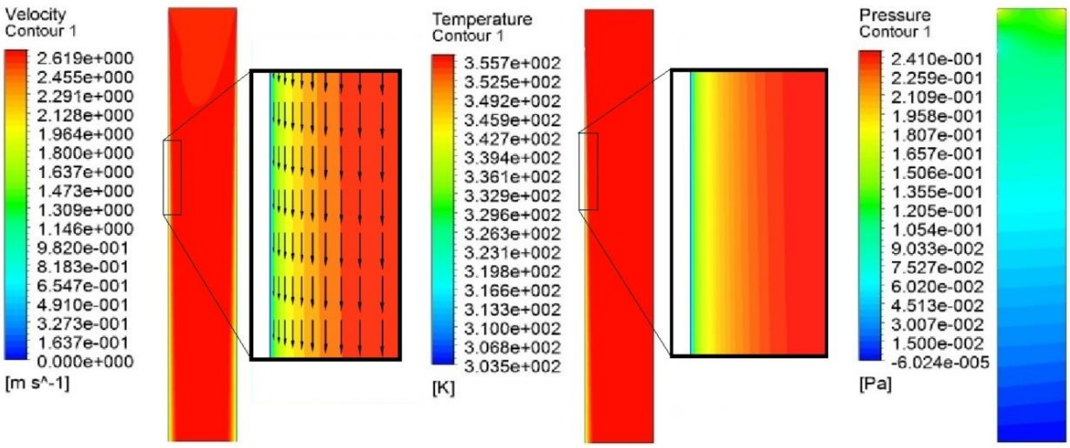

Figure 3. Velocity, temperature and pressure variation in flow domain

Figure 3 shows the variation of velocity, temperature and pressure, in the flow domain during condensation in presence of non-condensable gases for test P15-T30-V25 when steam mass fraction is 0.428.

4.1 Mesh Dependence Test

To know the effect of mesh size on results, numerical investigation with six different wall mesh arrangement is used as represented in table 2. The growth ratio of mesh thickness normal to the wall is considered to be 1.2. CONAN test results of P10-T30-V25 are used to investigate the mesh effect.

Table 2. Mesh details

Sr. No.	1	2	3	4	5	6
1st layer thickness [mm]	0.5	0.75	1	2	5	10

Local condensation heat flux values at a distance of 1 m along the cooling wall for each mesh case is shown in the figure 4.

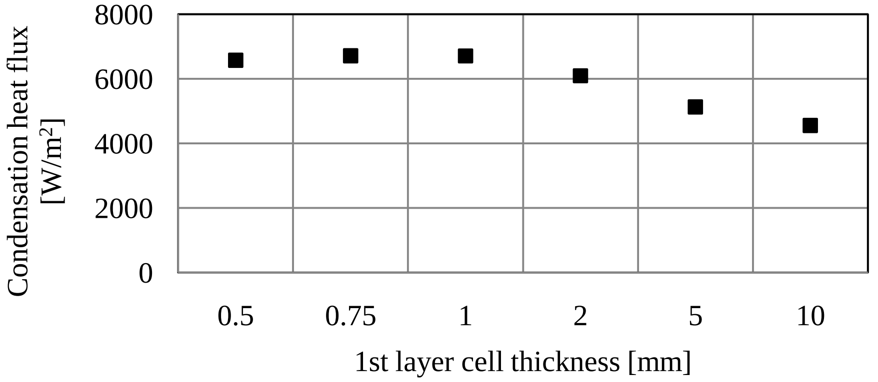

Figure 4. Effect of different mesh arrangement on condensation heat flux

It was observed that the mesh with 1st cell thickness of 1 mm to 0.75 mm is sufficient to provide mesh independent result. So, 1st cell thickness of 0.75 mm has been considered for further investigations.

4.2 Parametric Study

In order to characterize the condensation on the vertical wall, in presence of condensable gases present work investigates the effect of following parameters.

a) Mass fraction of non-condensable gas,
b) Operating pressure and
c) Mixture mass flow rate

The condensation heat flux and condensation rate along condensing surface being significant characterizing parameters, have been used in following sections to present and explain the results. These parameters are interdependent. The characterizing non-dimensional parameter for condensation, Sherwood number (*Sh*), has also been used to illustrate the investigation results. It represents the non-dimensional condensation mass transfer. *Sh* is known to be the function of Reynolds number (*Re)* and Schmidt number *(Sc).*

$$Sh_{0,x,molar} = \frac{\dot{m}''}{M_v * \frac{c\,D}{x} * \ln\left(\frac{X_{n,bulk}}{X_{n,wall}}\right)} \qquad (11)$$

Where, $\dot{m}''$ is local condensate mass flux value (kg/s), M_v is molecular weight of condensate (H_2O =18), c is the molar concentration of steam (mol/m^3), D is diffusion coefficient (m/s), x is the distance along the wall (m), and $X_{n,bulk}$ and $X_{n,wall}$ are the mole fractions of non-condensable gas in bulk flow and at wall respectively as given by Ambrosini et al., (2008) [1].

As the condensation is a surface phenomenon, the condensation effect is expected to be limited to the condensing surface. Although the flow in present problem is in closed conduit but the presence of other surfaces is expected not to effect the surface condensation. Under this assumption the flow Reynolds number is defined by,

$$Re = \frac{\rho\ v\ x}{\mu} \qquad (12)$$

Where, ρ is the density (kg/m^3), v is bulk flow velocity (m/s), x is distance along the condensing wall and μ is the dynamic viscosity (kg/m.s). All these values are calculated by averaging the local properties of mixture at wall and in the bulk flow.

Mass Fraction of Non-Condensable Gas. The mass fraction of steam entering the computation domain is varied from 0.1 to 0.9 in the step of 0.1 with a bulk inlet velocity of 2.6 m/s where the condensing surface is maintained at 305K where steam –air mixture is in saturated condition. Four additional cases similar to the experimental investigation done by Ambrosini et al. 2008 [1] in CONAN tests under similar conditions. A comparison of the results from numerical and experimental investigation is shown in figure 5. It can be observed that numerically investigated results agree well with the experimental work for the four different mass fractions of non-condensable gases. The heat flux values are slightly under predicted in the region near entrance and are slightly over predicted near the exit region this may be due to entry and exits effect of the computational domain. In all predictions the variation never exceeds more than 15%. A comparison of the condensation rates is also tabulated in below in table 3.

Table 3. Condensation rates for steam mass fractions

Mass fraction	Condensation rates (gm/s)	
	Numerical (Present work)	CONAN Experiment
0.279	6.40	6.70
0.559	5.30	5.70
0.572	3.20	3.40
0.707	2.03	2.10

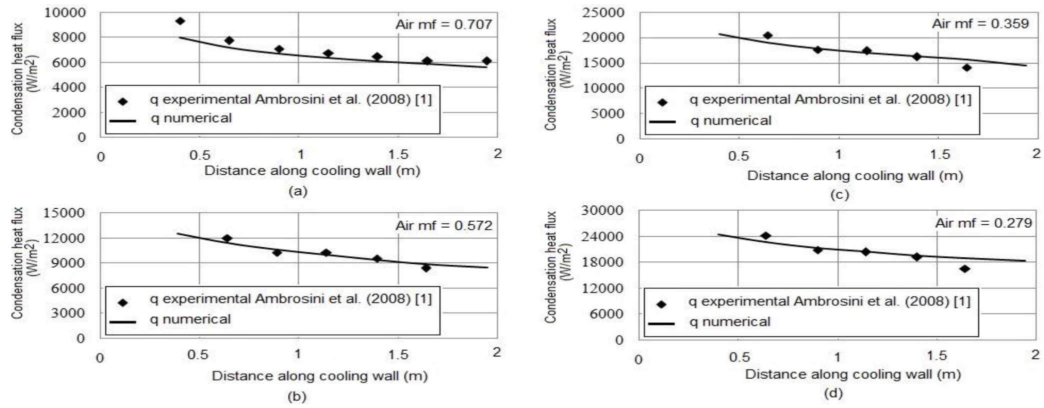

Figure 5. Validation of results

Saturated mixture of steam and air coming in contact with a sub-cooled surface is expected initiate the condensation which may lead to change in mass fraction of steam and air along the condensing plane. Figure 6 shows the contours of steam and air mass fraction near condensing wall for steam mass fraction of 0.5. An increasing air mass fraction can be observed towards wall in a thin region along wall. Also thin region along condensing wall appears to be thickening on downstream side of the flow. This is due to increasing non-condensable gas concentration in downward direction.

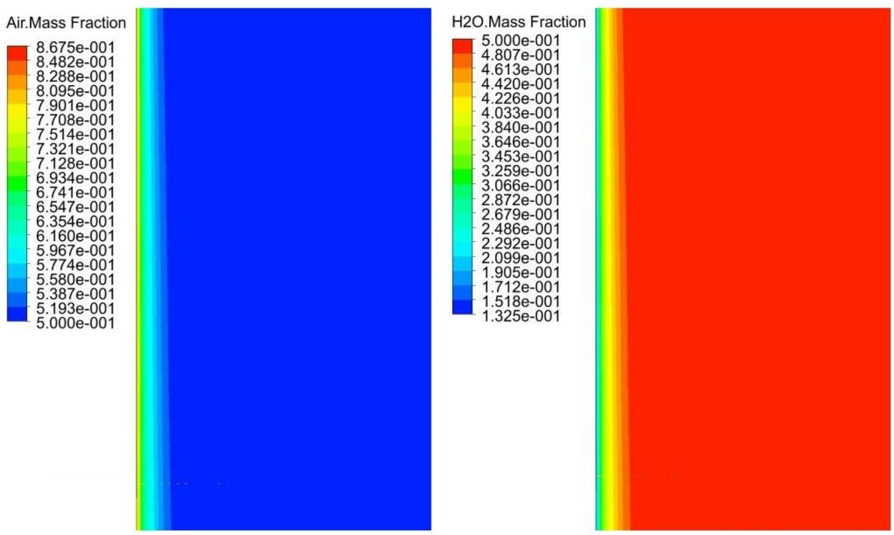

Figure 6. Contours of air and steam mass fraction

Figure 7 shows the variation of steam mass fraction normal to condensing plane for different steam mass fraction. Steam mass fraction drops drastically closer to wall due to rapid condensation of steam of the mixture. The mass fraction gradient is observed to be increasing with increasing stream mass fraction. The condensation appears to be more effective at higher mass fraction of steam. This leads to higher mass transfer and condensation heat flux. The presence of non-condensable gases hinders the condensation more effectively at lower steam mass fraction. This may be due to the presence of more non-condensable mediums reducing the probability of condensation.

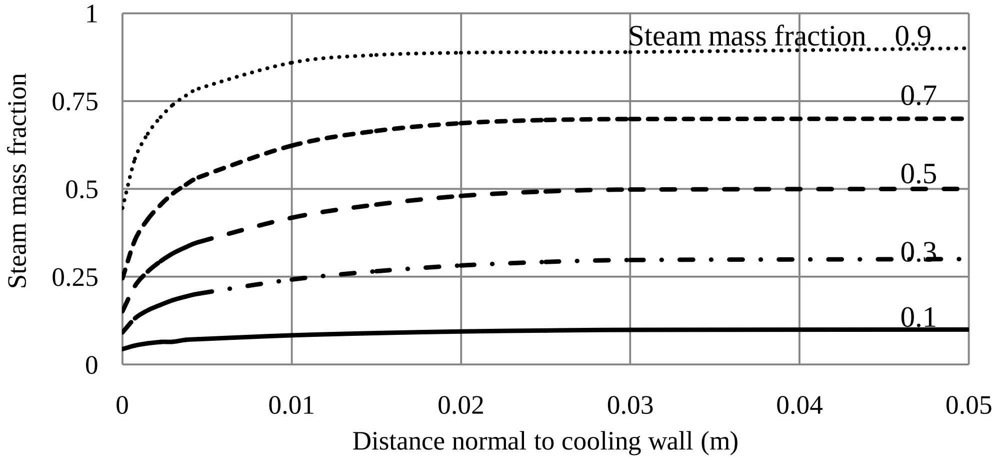

Figure 7. Variation of steam mass fractions normal to condensing surface

The condensation heat flux at the wall along the length of condensing surface is expected to change due to increasing presence of non-condensable gases which have tendency to hamper the condensation of steam component of mixture. Figure 8 shows the variation of condensation heat flux at the wall along the length of condensing surface for various mixtures.

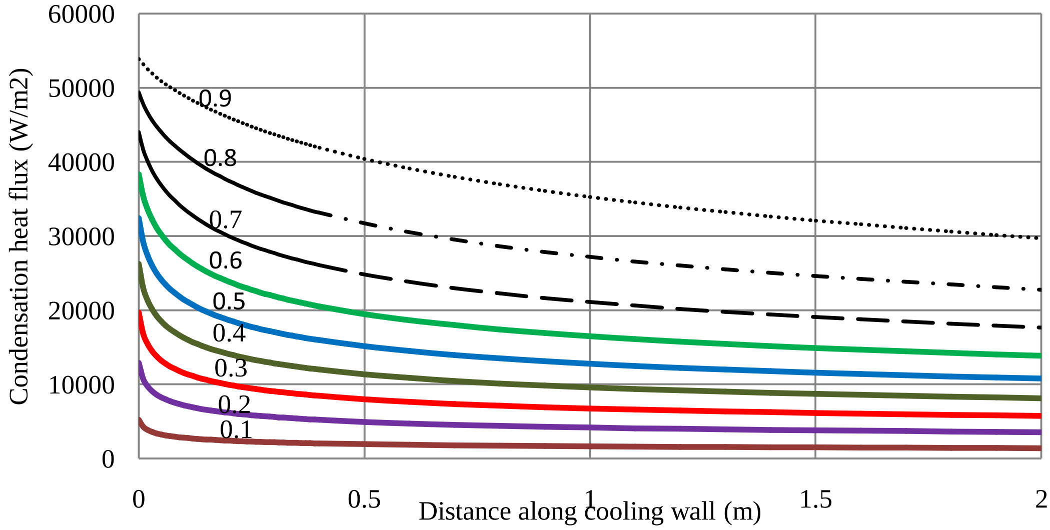

Figure 8. Condensation heat flux along condensing wall for different steam mass fractions

From Figure 8 it is clearly seen that the amount of non-condensable gas strongly affects the rate condensation heat and mass transfer. It is also observed that gradient of condensation heat flux with respect to length is higher in case of higher steam mass fraction. The variation in condensation heat and mass transfer rate along the length of the condensing surface is converted into non-dimensional form using Sherwood number and Reynolds number. The following Figure 9 shows the graph of calculated Sherwood number plotted against the Reynolds Number for different steam-air mixture cases.

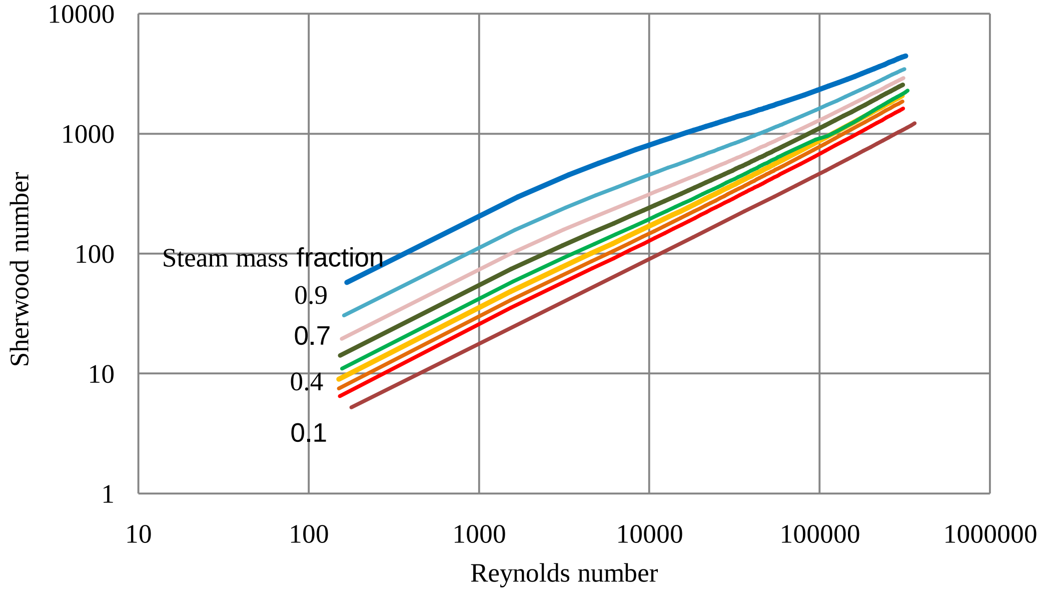

Figure 9. Variation of Sherwood number with local Reynolds number.

Along with the condensation heat flux values, overall condensation rates over the condensing surface were calculated for the all the gas mixture configurations. The variation of overall condensation rate with steam mass fraction is represented in Figure 10.

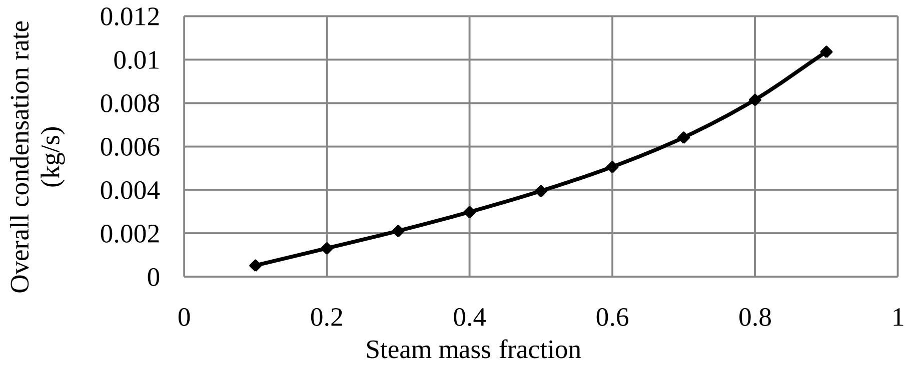

Figure 10. Effect of steam mass fraction on overall condensation rate

Effect of Operating Pressure. Condensation under different pressure condition is very common in the presence of non-condensable gases. To investigate its effect on condensing heat and mass transfer rates, it is varied from very low (0.2 bar) to high (10 bar) which prevails in many mechanical systems. The gas mixture of 0.5 steam mass fraction enters the domain at saturated state corresponding to imposed pressure where condensing wall maintained at 305 K. At an imposed condensing pressure condensing heat flux slowly drops to a steady state along the length of condensing surface as shown in figure 11. This effect can be attributed to decreasing temperature gradient in the entry region only. The condensation rate is observed to be increasing with condensing pressure. It may be due to increasing saturation temperature of entering mixture. The linear trend of increase in overall condensation rate with increase in operating pressure is shown in figure 12.

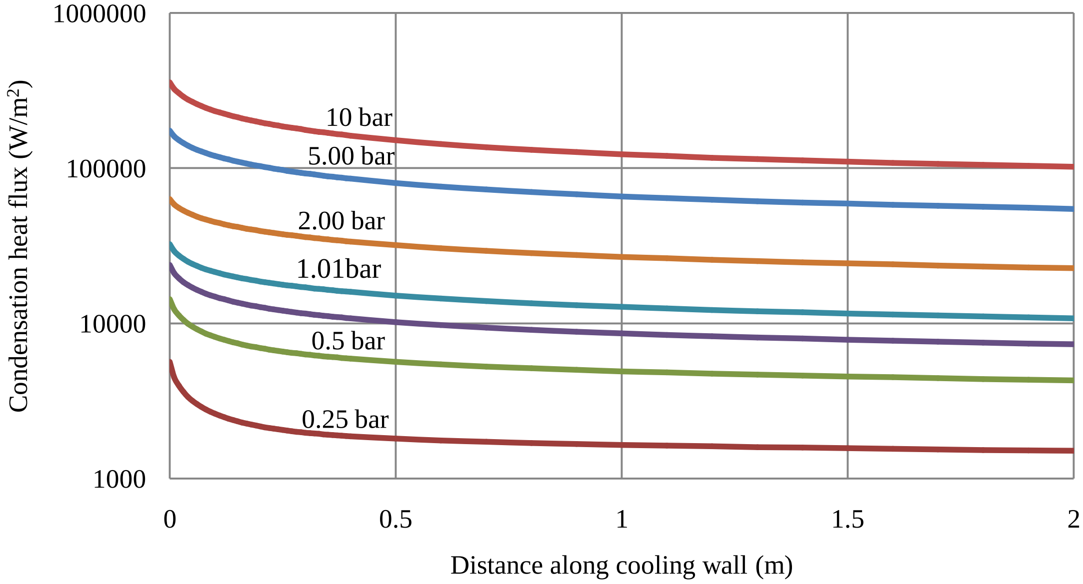

Figure 11. Effect of operating pressure on condensation heat flux

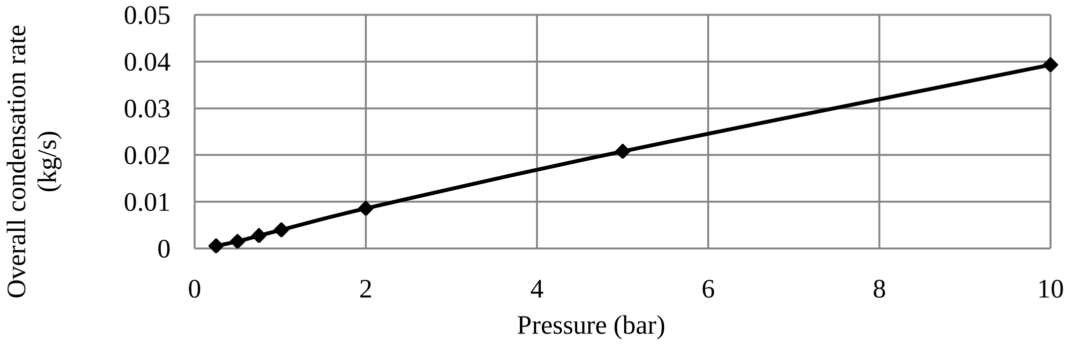

Figure 12. Effect of operating pressure on overall condensation rate

Effect of Mixture Mass Flow Rate. The mass flow rate of the gas mixture is expected to affect the condensation rate significantly. This is due to variation of residence time of vapor in contact with condensing surface. The presence of non-condensable gases may create a significant effect of condensation due to shearing effect induce in the flow of mixture due their density. Shearing effect on the non-condensable gas tends to reduce the thickness of non-condensable gas concentration boundary layer. To investigate the effect of non-condensable gases, the flow of mixture with steam mass fraction 0.50 entering to domain, is varied with different mass flow rates from 0.002611 kg/s to 2.615 kg/s in 9 different cases keeping other conditions same as per the boundary condition.

Table 4. Mass flow rate case details

Case no.	1	2	3	4	5	6	7	8	9
Mass flow rate [kg/s]	0.00261	0.01044	0.161	0.2262	0.2698	0.3498	0.784	1.5709	2.615
V_b [m/s]	0.03	0.12	1.85	2.6	3.1	4.02	9.01	18.03	30

The graph in Figure 13 shows the condensation heat flux versus vertical length along cooling wall for all the 9 cases and can be observed that the condensation heat and mass transfer is enhanced greatly with increase in mixture mass flow rate. From Figure 13 it can be clearly seen that the trend of decrease in condensation heat flux along the length of the cooling wall remains almost similar for every mass flow rate magnitude. Also, it was observed that the rate of increase in condensation heat and mass transfer with increase in mass flow rate goes on reducing with further increase in mixture mass flow rate. Figure 14 shows the nature of change in overall condensation rate with the mass flow rates. The Figure 15 shows the non-dimensional representation of above results with the help of Sherwood number and Reynolds number.

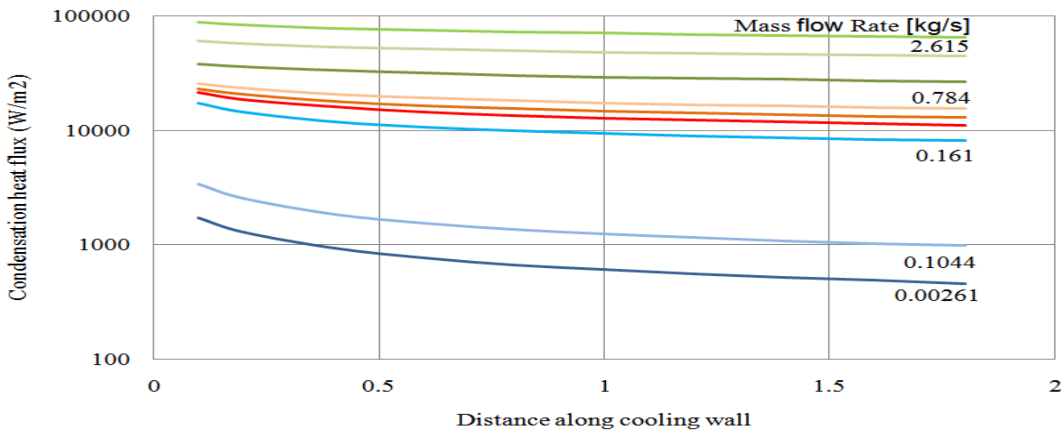

Figure 13. Effect of mass flow rate on condensation heat transfer rate

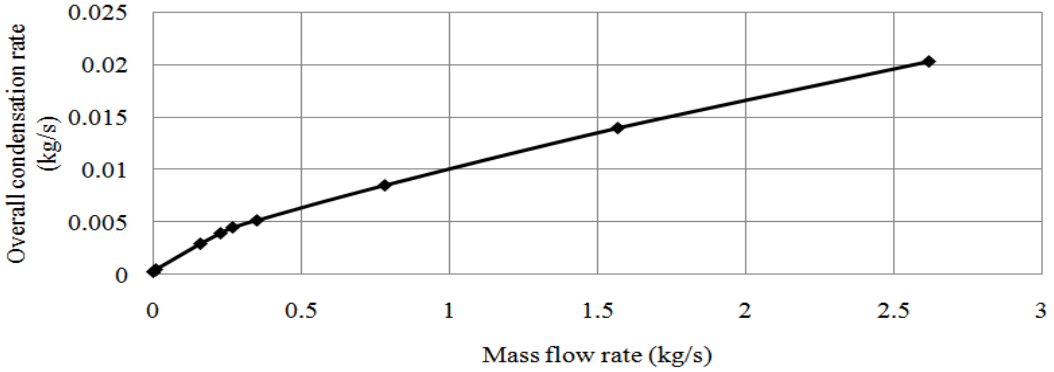

Figure 14. Effect of mass flow rate on overall condensation rate

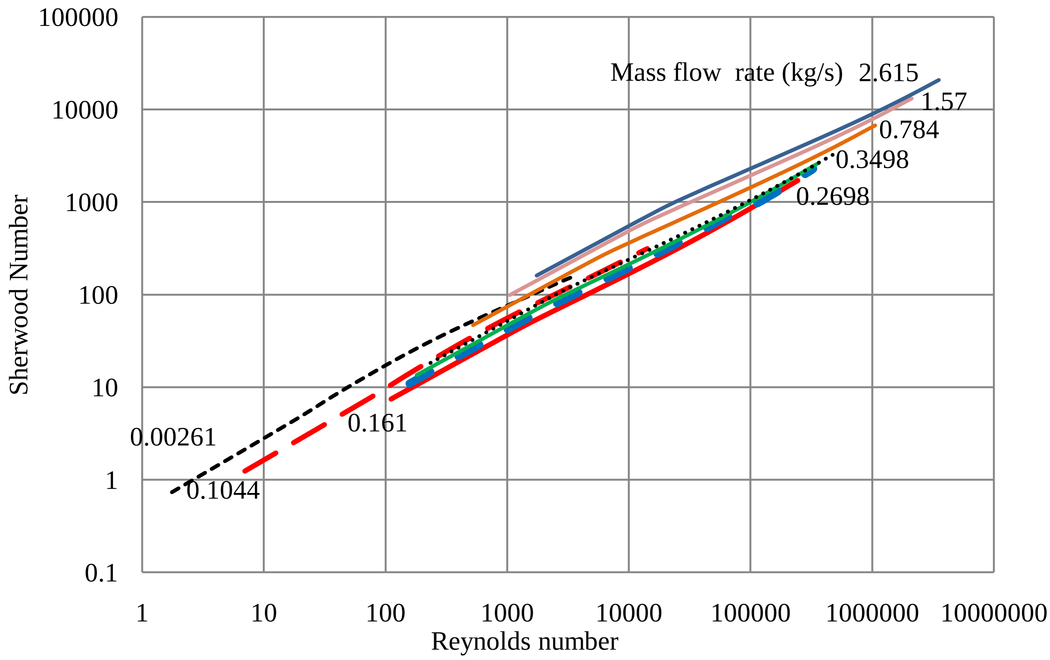

Figure 15. Variation of Sherwood number with local Reynolds number over cooling wall for different mass flow rates

IV. Conclusion

It can be concluded from the present study that the condensation in presence of non-condensable gases increases with increasing steam mass fraction which become more dominating at higher mass fractions. The condensation rate is found to be varying linearly with operating pressure. The condensation rate is observed to have linear relationship also with the mass flow rate of the mixture.

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