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Liquid film condensation from water vapour flowing downward along a vertical tube

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ABSTRACT

This paper focuses on a numerical study of the liquid film condensation from the vapour-gas mixtures inside a vertical tube. The model uses an implicit finite difference method to solve the governing equations for liquid film and gas flow together including the boundary and interfacial matching conditions. The external wall of the tube is subjected to a constant temperature or uniform heat flux. The influence of the inlet conditions of the gas mixtures and the tube length on the heat and mass transfer are analyzed. The results indicate that an increase of the relative humidity and the inlet-to-wall temperature enhance the condensation process. Increasing the tube length produces a high amount of condensed vapour along the tube. Additionally, non-condensable gas is a decisive factor in diminishing the efficiency of the heat and mass exchanges. For a fixed heat flux, increasing the inlet temperature substantially increased the accumulated condensation rate. Overall, these parameters are relevant factors to improve the effectiveness of the thermal and desalination systems.

1. Introduction

Condensation from water vapour is widely encountered in practice such as refrigeration, heat exchangers, chemical industries, and desalination units. The shortage of drinking water is expected to be a serious problem for many countries in the world [1]. Faced with the decrease in drinking water, the desalination of seawater or brackish water is one of the promising techniques to satisfy the needs of the growing population [2]. According to the International Association of Desalination, since 2015, more than 18000 desalination plants in 150 countries are producing 86.8 million m^3 per day and covering the daily need of water for more than 300 million people. Nowadays, the thermal desalination processes such as multi-effect distillation (MED) and multi-stage flash (MSF) both have about 28% of the market share in the world, whereas 65% of reverse osmosis [3]. Therefore, the necessity to improve the thermal technologies (which are based on the phase change phenomenon of evaporation/condensation) continues to receive great interest.

Heat and mass transfer during the liquid film condensation from the vapour-gas mixtures have been the subject of a large number of works. An experimental study of condensation from water vapour was carried out by Lebedev et al. [4]. Their results confirm that the condensation heat transfer is enhanced by increasing the relative humidity. The phenomenon of condensation with the presence of non-condensable gas was experimentally realized inside a vertical tube [5–8]. No and Park [9] conducted a theoretical study in the case of steam condensation in a vertical tube. They showed that non-iterative model improves the condensation process. Revankar and Pollock [10] found from filmwise condensation of vapour that the accumulation of air especially near the gas-liquid interface reduces the condensation heat transfer.

Numerical study for laminar film condensation inside horizontal channel with a constant temperature at the wall was proposed by Siow et al. [11]. They analyzed the effect of the inlet conditions on the film thickness and the local Nusselt number. Later Siow et al. [12,13] studied the liquid film condensation from the vapour-gas mixtures in vertical, then in inclined channels. Their results show that the film thickness is reduced with a high non-condensable gas. They also found that the angle of declination has a strong effect on the condensation process. Dharma et al. [14] developed a model to study the condensation of water vapour along a vertical tube. Their results show that the condensation heat transfer coefficient decreases with the presence of a high percentage of non-condensable gas. Numerical and experimental studies of condensation from the steam-air mixtures with the presence of non-condensable gas in various geometries are also available in literature [15–18].

In the thermal desalination unit, the condenser is used for producing freshwater from the saline water sources. Actually, the liquid films condensation with the presence of non-condensable gas in seawater desalination have been studied by many authors. Semiat and Galperin [19] have shown from steam condensation that even a weak

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Nomenclature

C_P	Specific heat $[\mathrm{J \cdot kg^{-1} \cdot K^{-1}}]$
D	Diffusion coefficient $[\mathrm{m^2 \cdot s^{-1}}]$
d	Diameter of the tube ($d = 2R$) [m]
g	Gravitational acceleration $[\mathrm{m \cdot s^{-2}}]$
h_{fg}	Latent heat of vaporization $[\mathrm{J \cdot kg^{-1}}]$
h_M	Mass transfer coefficient $[\mathrm{m \cdot s^{-1}}]$
h_T	Heat transfer coefficient $[\mathrm{W \cdot m^{-2} \cdot K^{-1}}]$
J_I''	The mass flux at the interface $[\mathrm{kg \cdot m^{-2} \cdot s^{-1}}]$
L	Tube length [m]
m_0	Inlet gas mass flow rate $[\mathrm{kg \cdot s^{-1}}]$
M_a	Molar mass of air $[\mathrm{g \cdot mol^{-1}}]$
M_v	Molar mass of vapour $[\mathrm{g \cdot mol^{-1}}]$
Mr	Condensation rate
Nu_x	Overall Nusselt number
P	Atmospheric pressure $[\mathrm{N \cdot m^{-2}}]$
Q_L	Latent heat flux $[\mathrm{W \cdot m^{-2}}]$
Q_S	Sensible heat flux $[\mathrm{W \cdot m^{-2}}]$
Q_T	Total heat flux $[\mathrm{W \cdot m^{-2}}]$
Q_W	Uniform heat flux at the tube wall $[\mathrm{W \cdot m^{-2}}]$
R	Radius of the tube (d/2) [m]
r	Radial coordinate [m]
Re	Reynolds number
Sh	Interfacial Sherwood number
T	Temperature [K]
T^*	Dimensionless temperature ($(T - T_W)/(T_0 - T_W)$)
u	Axial velocity $[\mathrm{m \cdot s^{-1}}]$
v	Radial velocity $[\mathrm{m \cdot s^{-1}}]$
w	Mass fraction of vapour
w^*	Dimensionless mass fraction of vapour (w/w_0)
w_I	Mass fraction of vapour at gas-liquid interface
X	Transformed coordinate along the tube ($X = x/L$)
Y_i	Molar fraction of species i vapour

Greek symbols

δ	Liquid film thickness [m]
λ	Thermal conductivity $[\mathrm{W \cdot m^{-1} \cdot K^{-1}}]$
μ	Dynamic viscosity $[\mathrm{kg \cdot m^{-1} \cdot s^{-1}}]$
ϕ	Relative humidity
ρ	Density $[\mathrm{kg \cdot m^{-3}}]$

Subscripts

0	Condition at inlet of the tube
a	Referring to the air
b	Bulk
G	Referring to the vapour-gas mixture
I	Interface
L	Referring to the liquid
W	Condition at wall of the tube

concentration of non-condensable gas reduces the heat transfer coefficient in seawater desalination plant. Later, an experimental study achieved by Al Shammari et al. [20] indicates that non-condensable gas has a negative effect on the heat transfer. Belhadj Mohamed et al. [21] carried out a numerical study to improve the process of condensation from the water vapour in a vertical channel. Recently, Hassaninejadfarahani et al. [22] numerically investigated the condensation of the liquid film from a high amount of non-condensable gas in a vertical tube. Their study has shown that the thickness of the liquid film is higher with a large Reynolds number and relative humidity.

It is important to note that the laminar film condensation of the water vapour inside a vertical tube subjected to a uniform heat flux has not been investigated, in spite of their practical significance. This motivates the current study to analyze simultaneous heat and mass transfer during condensation for both the constant wall temperature and uniform heat flux. In order to improve the effectiveness of condensing vapour in the presence of non-condensable gas, special attention is addressed to investigate the effects of the inlet relative humidity, the inlet pressure, the inlet temperature and the tube length on the condensation process.

2. Mathematical analysis

The problem under consideration is a vertical tube with length L, and radius R (Fig. 1). A mixture of vapour and non-condensable gas enters the tube with a uniform velocity u_0, pressure P_0, temperature T_0 and relative humidity ϕ_0. The wall is maintained at T_W or Q_W. The thickness δ_x of liquid film increases according to the amount of condensed vapour along the tube. The governing equations are written in the coordinate system (r, x), x is positively measured in the direction of downward flow, while r is the radial distance positively counted from the axis of the symmetry of the tube.

In order to formulate mathematically our problem, a lot of simplifying assumptions have been used in the analysis:

- The flow in both the gas and liquid phases is considered to be incompressible, laminar, steady and axisymmetric.
- The interface between the two phases is in thermodynamic equilibrium [23]. This assumption means that the phase change occurs only under the conditions of saturation.
- The humid air is assumed to be a perfect gas.
- The heat transfer by the radiation, the Soret and Dufour effects are negligible.
- It was also assumed that the axial diffusion of mass, heat and momentum are negligible [24].

2.1. Governing equations

2.1.1. Liquid film equations

The governing equations for the liquid phase are

- Continuity equation:

$$\frac{\partial}{\partial x}(\rho_L u_L) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho_L v_L) = 0 \tag{1}$$

- Momentum equation:

$$\frac{\partial}{\partial x}(\rho_L u_L u_L) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho_L v_L u_L) = -\frac{\mathrm{d}p_d}{\mathrm{d}x} + \frac{1}{r}\frac{\partial}{\partial r}\left(r\mu_L\frac{\partial u_L}{\partial r}\right) + (\rho_L - \rho_0)g \tag{2}$$

- Energy equation

$$\frac{\partial}{\partial x}(\rho_L C_{P,L} u_L T_L) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho_L C_{P,L} v_L T_L) = \frac{1}{r}\frac{\partial}{\partial r}\left(r\lambda_L\frac{\partial T_L}{\partial r}\right) \tag{3}$$

2.1.2. Gas phase equations

The governing equations for the gas flow are:

- Continuity equation:

$$\frac{\partial}{\partial x}(\rho_G u_G) + \frac{1}{r}\frac{\partial}{\partial r}(r\rho_G v_G) = 0 \tag{4}$$