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Research Paper

Numerical investigations on two-phase flow modes in evaporative condensers

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HIGHLIGHTS

- The falling film evaporation has a crucial importance in evaporative condensers.
- A numerical model of the falling film evaporation is here presented.
- Two different flow modes have been investigated.
- The flow modes depend on the water-to-air mass flow ratio and tubes arrangement.
- A trade-off curve for a specific geometry was obtained.

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ABSTRACT

Falling film evaporation over horizontal tubes consists of simultaneous heat and mass transfer processes: in an evaporative condenser it improves the heat rejection from the condensing refrigerant to the air. The liquid flow is generally influenced by viscous, gravity, tension effects, liquid mass flow rate, tube diameter and spacing and distance from the feeding system.

In this work, a two-dimensional numerical model of the falling film evaporation on horizontal tubes is presented. The temporal change characteristics of the film flow process were studied and different types of flow (stable film and drops mode) were investigated, by varying the ratio between the water-to-air mass flow ratio.

The effect of the tubes arrangement on the flow mode was analyzed too: an increase of 73% of the longitudinal pitch corresponds to an increase of 66.7% of the minimum water mass flow rate that prevents the film break-up. The trade-off curve for a given geometry was obtained: at a specific air mass flow rate, a transition zone between the stable film to the drops mode conditions was individuated, with an uncertainty of 10% referring to a water mass flow rate variation of 10%.

1. Introduction

Due to its interest in many fields (chemical and food industries, refrigeration equipment and so on), many researchers in the past have investigated the falling film evaporation over horizontal tubes, that involves simultaneous and complex heat and mass transfer processes. When carried out in evaporative condensers it improves the heat rejection from the condensing refrigerant to the air.

Chyu [1] computed the film thickness at any angular position θ as function of the flow rate over half a tube of unit length, water and air density and water dynamic viscosity:

$$\delta = \left[\frac{3\mu_w \Gamma}{g\rho_w(\rho_w - \rho_{m,a})\sin\theta} \right]^{1/3} \quad (1)$$

Rogers [2] calculated the laminar film thickness on horizontal tubes as function of the film Reynolds and Archimedes numbers, solving the motion and energy equations. The film Reynolds number is expressed as function of the water flow rate over half a tube per unit length:

$$Re_w = \frac{4\Gamma}{\mu_w} \quad (2)$$

The Archimedes number is:

$$Ar = \frac{g\rho_w{}^2 d_{ext}{}^3}{\mu_w{}^2} \quad (3)$$

Then he obtained an empirical relationship for the film thickness [3]:

$$\delta = 1.186\left(\frac{Re_w}{Ar} \right)^{1/3} \quad (4)$$

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Armbruster et al. [4] investigated the falling film flow mode transitions for plain tubes, observing that the flow pattern depends on the film Reynolds number and the tube spacing.

Hu et al. [5] proposed a flow mode transition depending on the relationship between the film Reynolds number and the Galileo number. The Galileo number is defined as:

$$Ga = \frac{\rho_w \sigma^3}{g \mu_w{}^4} \tag{5}$$

At each relationship between these two dimensionless quantities corresponds a different flow mode.

Yung et al. [6] expressed the mass flow rate per unit length at the stable film-drops mode transition as:

$$\Gamma_{trans} = 0.81 \frac{\rho_w}{\lambda} \frac{\pi d_p{}^3}{6} \left(\frac{2\pi\sigma}{\rho_w \lambda^3} \right)^{1/2} \tag{6}$$

λ is the stability wavelength on horizontal tubes and d_p is the diameter of the primary drop written as:

$$d_p = C \sqrt{\frac{\sigma}{g \rho_w}} \tag{7}$$

The empirical constant for water C is equal to 3. Honda et al. [7] defined a coefficient K as follows, whose value range corresponds to different flow modes:

$$K = \frac{\Gamma}{\sigma^{3/4}} \left(\frac{g}{\rho_w} \right)^{1/4} \tag{8}$$

All the cited works were focused on the falling film evaporation in still air. Actually in the previous empirical correlations no dependence on the air velocity was present, while the influence on the flow formation and heat transfer has a great interest in real evaporative condenser configuration, where water falls down against a countercurrent air flow provided by a fan.

The authors who dealt with the modeling of evaporative condensers and cooling towers aimed to evaluate the thermal performance of the whole system.

Parker and Treybal [8] suggested a design method for countercurrent evaporative coolers, based on Merkel hypothesis (Lewis number equal to unity). They took into account the water temperature variation in the heat transfer process, while previous authors referred to a constant mean value.

Mizushina [9] experimentally studied evaporative coolers and obtained empirical correlations for heat and mass transfer coefficients.

Kreid [10], Leidenforst and Korenic [11] focused on finned evaporative condensers.

Bykov et al. [12] studied heat and mass transfer as well as fluid flow characteristics in evaporative condensers. They detected three different zones: (a) the area above the tube bundle; (b) the tube bundle; (c) the area between the coil and the bottom sump; they investigated on water temperature and air enthalpy changes depending on the elevation above the sump level.

Webb [13] developed a unified mathematical model for cooling towers, evaporative coolers and condensers.

Dreyer et al. [14] studied evaporative coolers and condensers and compared empirical correlations provided by different authors to determine heat and mass transfer coefficients. They observed that expressions obtained by Mizushina [9] were valid over a wider range of operating conditions.

Zalewski and Gryglaszewski [15] developed a mathematical model based on analysis carried out by Poppe et al. [16]. They applied expression suggested by Bykov [12] for the heat transfer coefficient and modified the expression suggested by Bosnjakovic and Blackshear [17] for the mass transfer coefficient with a correction factor.

Ettouney et al. [18] investigated on evaporative condensers performance by varying the condensing temperature and the water to air mass flow rates ratio.

Qureshi and Zubair [19] carried out a study on the influence of fouling in evaporative coolers and condensers performance. In order to predict the device behavior, they used the model developed by Dreyer [14] adding the fouling factor.

Then Qureshi and Zubair [20] obtained an empirical relation to evaluate water evaporation rate and observed a maximum deviation of 2% between calculated and experimental values.

In the most recent works the phenomena involved in evaporative condensers were modeled at the tube scale.

Jahangeer and Tay [21] developed a model using finite difference technique, to simulate a single straight tube wet by the water film and invested by air in a cross flow scheme.

They studied the influence of many boundary conditions, like condensing temperature, dry air bulb temperature and relative humidity on heat transfer coefficient.

In [22] the evaporative condenser at tube scale was modeled with Fluent under stable film condition and the computed overall heat transfer coefficients values compared with those coming from empirical relationships.

Islam and Jahangeer [23] carried out experimental and theoretical analyses on an evaporatively cooled bare tube condenser: a good agreement between experimental and numerical results was obtained.

In this work, the different flow modes as influenced by the working fluids mass flow rates values are shown as a result of numerical investigations on the falling film evaporation phenomenon over horizontal tubes.

2. Mathematical model

Evaporative condenser geometry consists of staggered straight tubes, whose outer surface is wet by a liquid film while air flows in a countercurrent configuration.

The computational domain was set to represent the portion of fluid between two staggered tubes whose horizontal and vertical distances are equal to half of the transversal and to the longitudinal pitch, respectively. The geometry characteristics are summarized in Table 1 and the computational domain with the boundary conditions are shown in Fig. 1.

The model is based on the following assumptions:

I. The refrigerant condenses inside the tubes, and causes a nearly constant wall temperature (in the boundary condition setting, small temperature differences on the tube outer walls were neglected).
II. The air conditions at a specific distance from the inlet section were assumed the same over the tube length (i.e. along the z-direction); thus the problem becomes two-dimensional.
III. Since air and water flows have opposite directions (are not crossed) and there is no reason why water would be not equally distributed by the feeding system, a symmetric boundary condition was applied to the vertical surfaces of the computational domain.

Table 1
Geometrical parameters of the tube banks.

d_{ext}	mm	25
P_l	mm	50
P_t	mm	100