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Numerical analysis of thermodynamic behaviour of falling film outside a horizontal tube

Luopeng Yang, Chen Xue, Bo Zhang*, Kun Zhang, Shen Tao

School of Energy and Power Engineering, Dalian University of Technology, Dalian 116024, P.R. China
Tel. +86 411 84707963; email: zhangbo@dlut.edu.cn

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ABSTRACT

Falling film evaporation and condensation on horizontal tubes have been widely employed in the field of chemical, petroleum refining and desalinization. A numerical investigation on the thermodynamic behaviour of falling film outside a horizontal tube is presented. Based on the analysis of falling film state around a horizontal tube, a mathematical model is proposed, which takes into account the effect of jet impingement at the tube top on falling film behaviour. The numerical local heat transfer coefficients agree well with the experimental, which proves the reliability of the developed model. The variation of the local heat transfer coefficients and film thickness with flow density and evaporation temperature is simulated and analysed.

Keywords: Flow density; Falling film; Thermodynamic behaviour; Jet impingement; Horizontal tube

1. Introduction

Horizontal-tube falling film evaporation is an important condition in many common applications, including the refrigeration, chemical, petroleum refining and desalinization industries. The horizontal-tube falling film evaporator enjoys wide utilization because of its numerous advantages, which include high heat transfer coefficient at low film flow rate, small liquid inventories, significantly reducing the scaling and corrosion problems and effectively mitigating non-condensable gas effect on the heat transfer process [1–5].

The horizontal-tube falling film evaporator is a shell-and-tube type heat exchanger, which primarily consists of bundles of horizontal tubes connected by headers at the front end and the rear. Falling film evaporation on horizontal tubes is a heat and mass transfer process from liquid to steam. Heat transfer mainly depends on conduction and convection, while mass transfer depends on diffusion of molecules and convection. The liquid is introduced through spray nozzles to impinge upon the top of tube and flow is equally on either side of the tube. The film rejoins at the bottom and falls from tube to tube. The liquid film will be heated to the evaporative temperature by the heating steam inside the tube. When the film reaches the saturation temperature, liquid molecules will enter the steam boundary layer next to the film layer which has the same saturation temperature as liquid film. With the heat transfer to the liquid film, the liquid molecules enter into the steam layer continuously and partial pressure of steam will be enhanced. The steam

*Corresponding author.

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then enters into the condenser to keep the fit pressure, which assures that the process can be carried out.

A number of models have been found concerning the flow and heat transfer topic. These models were classified in terms of an assessment of the flow regime and the falling film flow mode. The early research on falling film heat transfer was done by Nusselt [6] who considered the formation of falling film evaporation being similar to thin laminar film of condensate on vertical surfaces. That was based on the assumption that the flow was ready with a flat, smooth and shear free gas–liquid interface. Fletcher et al. [7,8] experimentally studied the heat transfer coefficients of saturated seawater film on electrically heated horizontal tubes. Fletcher et al. [9] further investigated their work and found that heat transfer coefficients of external evaporation at non-boiling condition increased with an increase in film flow rate. Based on the assumption of a fully developed velocity boundary, a constant film thickness and initial temperature profile, a theory was put forward by Fletcher to predict the heat transfer coefficient. Chyu and Bergles [10] defined the regions of the falling film as the impingement region, the thermal developing region and the fully developed region. Local thermodynamic film performance was obtained by the continuity, momentum and the energy equations for each of these regions. There were two models which were based on the Nusselt's analysis and experimental correlations, respectively. Also, there seemed to be an obvious deviation between the calculated results [10,11] and the experimental results [12]. A possible reason for the deviation is that the strengthening heat transfer effect near the tube bottom was not considered in the models.

This investigation based upon the previous work aims at providing a design tool applying to a two-dimension vertical column of horizontal tubes. Compared with the previous mathematical models, the model in the stagnation region is taken into account and the area of the stagnation region is presented. The influence of various parameters for practical conditions in a low temperature multi-effect distillation desalination plant on the heat and mass transfer process is analysed. This study is done to provide a better understanding of the characteristics of liquid film falling down a horizontal tube and will lay groundwork for an improved design of the horizontal-tube falling film evaporators.

2. Mathematical models

It is shown in Fig. 1 that saturated liquid on the top of a horizontal tube forms a thin film flow on the tube surface. The flow is considered in the four regions including stagnation region, impingement region, thermally developing region and fully developing region. Because of the effect of jet impingement, the heat transfer coefficients are remarkably high in both stagnation and impingement regions although the two regions are short. The subsequent region is a thermally developing region where flow is being heated. A fully developing region, where evaporation occurs at the free surface of the film, follows the thermally developing region.

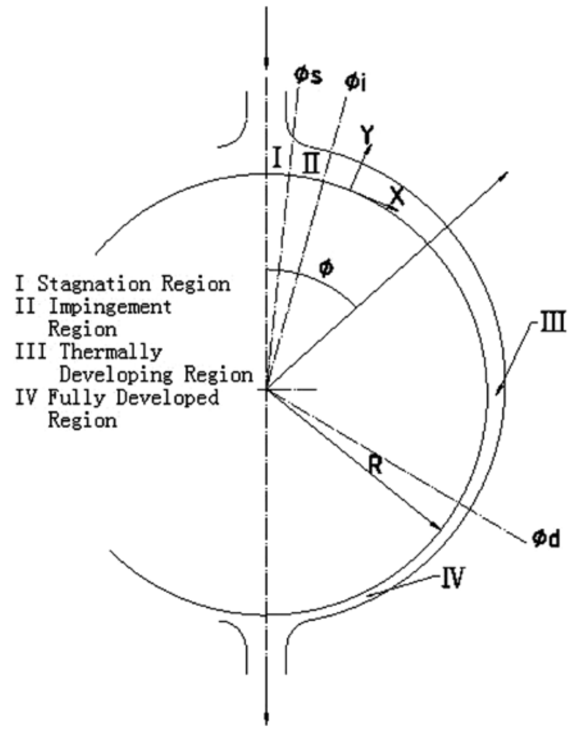

Fig. 1. Schematic of flow region of horizontal-tube evaporation falling film along the perimeter.

The assumptions are made as follows: the film is thin compared with the diameter of the tubes, the flow mode of the falling film along the circumference of the tube is laminar and only the body force caused by gravity is considered.

2.1. Stagnation and impingement regions

Since the film thickness is much smaller than the tube radius and both the stagnation and impingement regions are short, the condition at the top of a horizontal tube may be considered as a liquid jet impinging on a flat plate.

Local heat transfer coefficient in the stagnation region can be correlated by the following equation [13]:

$$\mathrm{Nu_s} = 0.826\mathrm{Pr}^{0.46}\mathrm{Re}_B^{0.5} \tag{1}$$

$$h_\mathrm{s} = \frac{\mathrm{Nu}\cdot k}{B} \tag{2}$$

The stagnation region covers the range of $0 < x < B$.

Local heat transfer coefficient in the impingement region is given as [14]

$$Nu_i = 0.7263(B/x)^{0.24}\mathrm{Re}_B^{0.5}\mathrm{Pr}^{1/3} \tag{3}$$

$$h_\mathrm{i} = \frac{k\cdot Nu_\mathrm{i}}{x} \tag{4}$$

The impingement region is within the range of $B < x < 2B$.

2.2. Thermally developing region

The tangential velocity along the perimeter, obtained by solving the equation of motion, is expressed as follows:

$$u = \frac{g(\rho_\mathrm{f}-\rho_\mathrm{g})\delta^2}{\mu}\sin(\phi)\left[\left(\frac{y}{\delta}\right)-\frac{1}{2}\left(\frac{y}{\delta}\right)^2\right] \tag{5}$$

The film thickness is calculated as:

$$\delta_\phi = \left[\frac{3\mu\Gamma}{g\rho_\mathrm{f}(\rho_\mathrm{f}-\rho_\mathrm{g})\sin\phi}\right]^{1/3} \tag{6}$$

Local heat transfer coefficient can be given as:

$$h_\mathrm{d} = \frac{k}{\delta_\mathrm{d}(\phi)} \tag{7}$$

The angle at which the developing region ends is calculated as:

$$\phi_\mathrm{d} = \frac{1}{\pi\alpha R}\left(\frac{3\mu\Gamma^4}{g\rho^5}\right)^{1/3} \tag{8}$$

2.3. Fully developed region

The film thickness is obtained as [10] follows:

$$\delta_\mathrm{fd}(\phi) = \left\{\left[\frac{3\mu_\mathrm{f}\Gamma_\mathrm{i}}{g\rho_\mathrm{f}(\rho_\mathrm{f}-\rho_\mathrm{g})}\right]^{4/3} - \frac{4e}{3\sin^{4/3}\phi}\int_{\phi_d}^{\phi}\sin^{1/3}\phi' d\phi'\right\}^{1/4} \tag{9}$$

Based on the film thickness solution, local heat transfer coefficient is calculated as:

$$h_\mathrm{fd} = \frac{k}{\delta_\mathrm{fd}(\phi)} \tag{10}$$

The average heat transfer coefficient can be obtained by summing heat transfer contributions from each of the regions:

$$\overline{h} = \overline{h_\mathrm{s}}\left(\frac{\phi_\mathrm{s}}{\pi}\right) + \overline{h_\mathrm{i}}\left(\frac{\phi_\mathrm{i}-\phi_\mathrm{s}}{\pi}\right) + \overline{h_\mathrm{d}}\left(\frac{\phi_\mathrm{d}-\phi_\mathrm{i}}{\pi}\right) + \overline{h_\mathrm{fd}}\left(1-\frac{\phi_\mathrm{d}}{\pi}\right) \tag{11}$$

3. Results and discussion

This study presents a solution for calculating the local evaporating heat transfer coefficients and the local film thickness along the circumference allowing for different flow density and saturated temperature of the liquid film. In the following analysis, the selected tube made of copper is smooth, 25.4 mm in diameter and 0.7 mm thick.

A comparison of the local evaporating heat transfer coefficients around the circumference, between the calculated values in this paper and the calculation values of mathematical models presented in the literature [10,15] for a particular condition, is shown in Fig. 2. It can be seen that both results are almost identical. The agreement between the present calculation and the reported result proves that the developed models are reliable.

Fig. 3 shows in detail the variation of local evaporation heat transfer coefficients at different heights H of jet nozzle in the stagnation and impingement regions with different distances x from the stagnation point. The local evaporation heat transfer coefficient reduces sharply as the film flows away from the stagnation point to impingement region. The strong effect of impingement results in the high local heat transfer coefficients in the stagnation. This effect ends at $x = 1.0$. It can be seen that the weight of jet impingement on heat transfer performance enhances with an increase in H.

Figs. 4 and 5 illustrate how the flow density affects the distribution of local heat transfer coefficient and local film thickness. In Fig. 4, with an increase in peripheral angle, the local evaporating heat transfer coefficient increases and reaches the maximum at the peripheral angle of 90°. The opposite variation of local film thickness with the peripheral angle is observed in Fig. 5. It also can be observed in Fig. 4 that the local

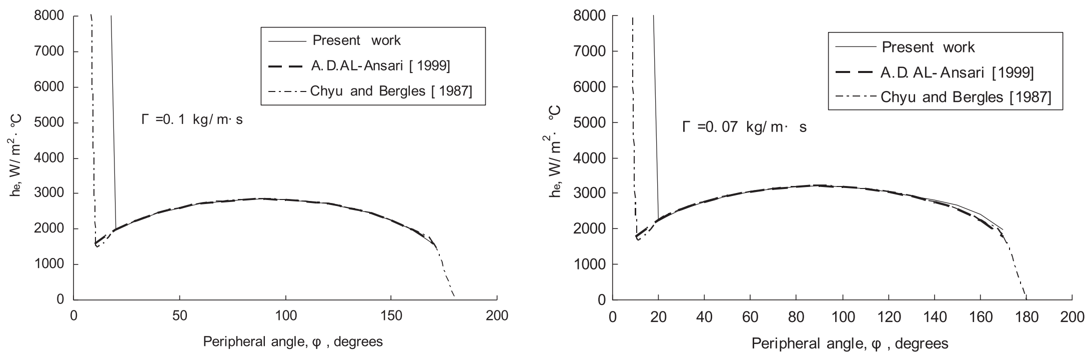

Fig. 2. Comparison of local evaporation heat transfer coefficients between the calculated and experimental results.

evaporating heat transfer coefficient increases as flow density decreases. A possible reason for this is that the thinner film thickness corresponds to less conduction resistance in the film. Thus, the low flow density contributes to a high evaporation rate, given that there is no dry patch on the tube surface.

The influence of evaporation temperature on average evaporation heat transfer coefficient with different flow density is shown in Fig. 6. It can be seen that the average evaporation heat transfer coefficient increases with an increase in evaporation temperature. It can be

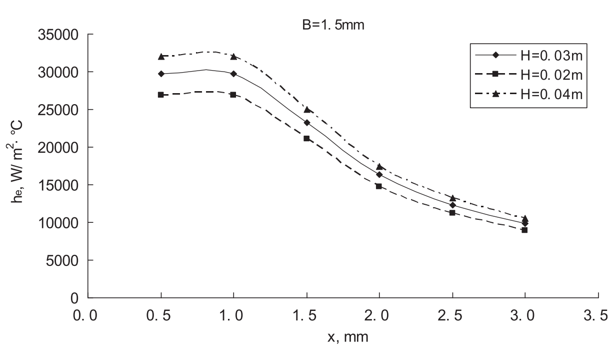

Fig. 3. Local evaporation heat transfer coefficients at different H along x.

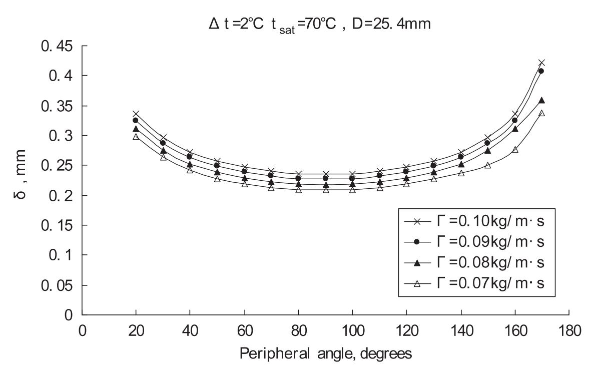

Fig. 4. Local film thickness along the perimeter.

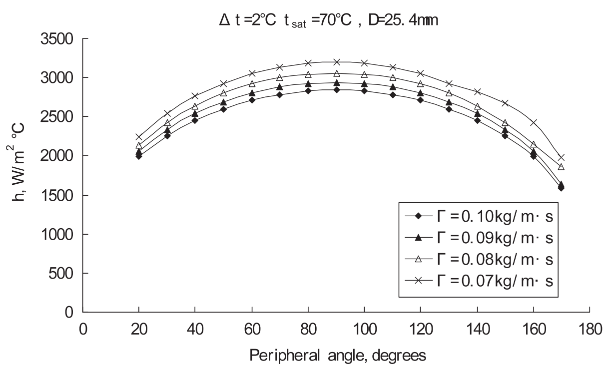

Fig. 5. Local evaporation heat transfer coefficients along the perimeter.

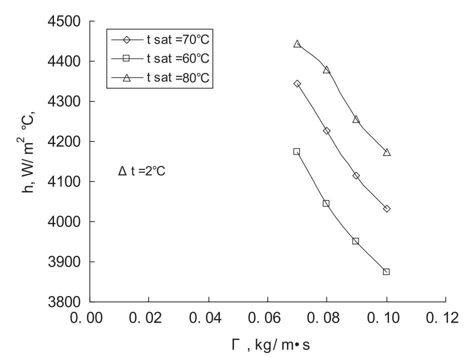

Fig. 6. Influence of evaporation temperature on average evaporation heat transfer coefficient.

explained by the fact that the liquid film thickness decreases due to the reduction of liquid viscosity with an increase in evaporation temperature.

4. Conclusions

Based on the given mathematical models, various parameters affecting the thermodynamic characteristics are presented and analysed in this paper.

- Satisfactory agreement of the evaporation heat transfer coefficients is found between the calculated by the present models and those calculated by the previous mathematical models, which proves that the developed models were reliable.
- A decrease in flow density contributes to an increase in local evaporation heat transfer coefficient and a decrease in local film thickness. It is at the peripheral angle of 90° that the maximum local evaporation heat transfer coefficient and the minimum local film thickness are attained.
- An increase in evaporation temperature causes an increase in evaporation heat transfer coefficient due to the reduction of liquid film thickness created by the decrease in liquid viscosity.

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Nomenclature

B	jet nozzle width, m
D	outer diameter of tube, m
G	acceleration of gravity of tube, m/s^2
H	liquid feed height, m
h	heat transfer coefficient, W/m^2 ℃
$\bar{h}$	average heat transfer coefficient, W/m^2 ℃
h_{fg}	latent heat of vaporization, W/m^2 ℃
k	thermal conductivity, W/m K
Pr	Prandtl number
R	outer radius of tube, m
Re	Reynolds number
t	temperature,℃
Δt	wall superheat $= t_w - t_s$
u	velocity, m/s
x	distance along the heating surface, m
y	distance in the direction normal to the heating surface, m
α	thermal diffusivity, m^2/s
Γ	mass flow rate of film per unit length on one side, kg/m·s
δ	film thickness, m
μ	dynamic viscosity, Pa·s
υ	kinematic viscosity, m^2/s
ρ	density, kg/m^3
φ	angular position, degree

Subscripts

d	developing region
f	liquid; film
w	wall
s	saturation; stagnation
fd	fully developed region
g	gas; vapor
i	impingement
j	jet

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