

Modelling of evaporative falling-film heat transfer over a horizontal tube

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Abstract

This article focuses on evaporative falling film heat transfer through numerical simulations. A two-dimensional symmetric liquid film flowing outside a single horizontal tube was simulated using the open-source CFD software OpenFOAM. The Tanasawa model was employed to consider mass transfer at the liquid-vapour interface, and the isoAdvector VOF method was used for interface tracking. The simulations explored the impact of film flow rate, tube surface heat flux, and tube diameter on the liquid film thickness and surface heat transfer coefficient. Additionally, the critical heat flux and operational limits were considered. The results demonstrate the accurate reproduction of evaporative falling film heat transfer performance by the present model. The film thickness exhibits a similar profile to adiabatic and sensible falling film heat transfer but provides smaller values. Similarly, the evaporative heat transfer coefficient exhibits significantly higher values and a gentle peripheral variation. The influence of heat flux on evaporative falling film heat transfer is contingent on the film flow rate; it can either enhance or weaken the process. Notably, a larger tube diameter negatively affects heat transfer, suppressing the impact of heat flux, and also increases the risk of liquid film dryout. For a given film flow rate, the average heat transfer coefficient increases initially, reaches a maximum value, and then decreases as the wall heat flux is increased. The peak heat flux, termed critical heat flux, increases with film flow rate but decreases with tube diameter. An operational limit exists where the heat flux is

lower than the critical heat flux and the film flow rate is higher than the critical film flow rate; under these conditions, evaporative falling film heat transfer operates in a fully wetting status.

Keywords: computational phase change; evaporation; evaporative heat transfer; falling film; numerical simulation

Nomenclature

D	diameter, m
F	source term due to surface tension
g	gravity acceleration, $\text{m}\cdot\text{s}^{-2}$
H	distributor height, m
p	pressure, Pa or bar
Re	film Reynolds number
r	latent heat, $\text{J}\cdot\text{kg}^{-1}$
S	source term due to phase change, $\text{kg}\cdot\text{m}^{-3}\cdot\text{s}^{-1}$
T	temperature, K or $^{\circ}\text{C}$
t	time, s

Greek

α	volume fraction
β	accommodation coefficient in Lee model, s^{-1}
Γ	film flow rate on one side of the tube per unit length, $\text{kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$

γ	accommodation coefficient in Tanasawa model
δ	film thickness, m
θ	peripheral angle from the upper stagnation point
μ	dynamic viscosity, $\text{kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$
ρ	density, $\text{kg}\cdot\text{m}^{-3}$
σ	surface tension coefficient, $\text{N}\cdot\text{m}^{-1}$

Subscripts

g	gas phase
l	liquid phase
sat	saturation state

1. Introduction

1.1. Falling film heat transfer

Falling liquid film heat transfer is ubiquitous in industrial applications, including condensation and evaporation processes such as distillation [1], refrigeration [2], and desalination [3], because of its outstanding behaviours in heat and mass transfer. When falling along a heated wall, the liquid film undergoes heating through conduction, convection, evaporation, or boiling, depending on the applied heat flux. By comprehensively understanding the underlying mechanisms and their interactions, engineers and researchers can develop strategies to enhance heat transfer efficiency, improve process performance, and ultimately drive innovation in key industries.

This technology has attracted considerable interest and research attention since its conceptualization, capturing the fascination of both numerical modelers and experimentalists alike [4, 5]. Tremendous investigations around crucial parameters such as film thickness, velocity, and profile, which serve as fundamental descriptors of the

liquid film's hydrodynamics and morphology were conducted under adiabatic conditions. In particular, stability and wave-like behaviors have garnered increased focus due to their significant impacts on heat transfer mechanisms [6-8]. In comparison, the knowledge of falling film phenomena under non-adiabatic scenarios seems to be more practical and useful for engineers and researchers, in which falling film heat transfer involving phase change, such as evaporation and boiling, is considerably more complex due to the influence of multiple factors [9, 10].

The process of falling film evaporation is governed by two distinct heat transfer regimes. Typically, when the tube surface heat flux is significantly lower than the threshold for the onset of nucleate boiling (ONB), the liquid film is initially heated through sensible heat. Subsequently, upon reaching a saturation state, evaporation occurs at the liquid film surface. Conversely, nucleate boiling prevails in the thin film if the heat flux surpasses the ONB threshold. Research on falling film heat transfer outside horizontal tubes has been extensive since the 1970s. Existing reviews [11-13] highlight that the majority of these studies rely on experimental approaches, with numerical investigations, particularly those addressing heat transfer with evaporation or condensation, being relatively scarce. However, owing to advancements in computational tools and methodologies, numerical investigations of non-adiabatic falling film flow have experienced unprecedented development.

Although the evaporative falling film heat transfer is focused on in the present study, the mass transfer model that governs the phase change intensity during falling film evaporation is briefly summarized in the following section, taking into account the model's crucial role in determining whether a computation is successful or not.

1.2. Review of mass transfer modelling

Phase change occurs when heat flux and liquid temperature reach specific thresholds, leading to the latent and sensible heat diffusions. Under these circumstances, a suitable mass transfer model is required to accurately calculate the heat transfer rate. Over the past decade, the literature has mostly used three mass transfer models that are commonly used.

The first of many models, extensively employed in published studies [14], was proposed by Lee in 1980 [15]. This model, most commonly utilized for addressing flow boiling [16-18] and condensation [19-21], determines mass transfer through:

$$S_g = -S_l = \beta \alpha_g \rho_g \frac{T - T_{\text{sat}}}{T_{\text{sat}}}, \quad (1)$$

for evaporation ($T > T_{\text{sat}}$), and:

$$S_l = -S_g = \beta \alpha_l \rho_l \frac{T_{\text{sat}} - T}{T_{\text{sat}}}, \quad (2)$$

for condensation ($T < T_{\text{sat}}$), where S_g and S_l are the volumetric mass transfer fluxes of the vapour and liquid, and β a constant factor representing mass transfer intensity [22] whose value can span a wide range depending on the flow geometry and regime, as well as the fluid property.

Numerical convergence is required to determine β , which is based on the assumption that the interfacial temperature, T_{int} , stays near the saturation temperature, T_{sat} [23]. A small β leads to a significant divergence of T_i from T_{sat} while large β induces convergence. For instance, β was set as 0.1 s^{-1} in Refs. [16, 17, 24] in the computation of flow boiling to ensure $T_{\text{int}} = T_{\text{sat}}$. Conversely, a larger value of β , such as 100 in Ref. [25], was used to make sure the interfacial temperature at $T_{\text{sat}} \pm 1 \text{ K}$. Notably, large β were applied in condensation simulations; β was set as $7.5 \times 10^5 - 1.0 \times 10^7 \text{ s}^{-1}$ to maintain $T_{\text{int}} = T_{\text{sat}} + (0.5 - 1.0) \text{ K}$ for the condensation simulation of R134a via a microchannel considering microgravity in Ref. [19]. Similarly, β was chosen as $5 \times 10^4 \text{ s}^{-1}$ to calculate water film condensation in laminar regime through a horizontal mini-tube in Ref. [26], while it was set as $1 \times 10^4 \text{ s}^{-1}$ for turbulent vertical downflow condensation of FC72 et al. [20]. In a helium liquefier simulation by Wei et al. [27], a value of $\beta = 1.75 \times 10^3 \text{ s}^{-1}$ was adopted for model validation, accessing liquefaction performance.

The second mass transfer model, referred to as the sharp interface model [28], assumes the interface to be saturated. Consequently, the interface converts any heat that passes through it into latent heat, which ultimately causes a phase change. When the interface stays at saturated status, the mass transfer intensity can be figured out by using the energy conservation. It is as follows for evaporation:

$$q = -\lambda_{\text{eff}} \nabla T_{\text{int}} \cdot \mathbf{n} = \dot{m} r, \quad (3)$$

in which q is the phase-change heat flux, $\mathbf{n}$ a unit vector of the interface, $\dot{m}$ the mass flux, and r the latent heat. And λ_{eff} , the effective thermal conductivity, is determined by $\alpha_g \lambda_g + \alpha_l \lambda_l$, with α_g and α_l the volume fractions of gas and liquid, respectively, and λ_g

and λ_l the thermal conductivities of gas and liquid, respectively. Accordingly, the mass transfer flux per unit volume is:

$$S_g = -S_l = \dot{m}|\alpha_g| = \frac{\lambda_{\text{eff}}(\nabla\alpha \cdot \nabla T)}{r}. \quad (4)$$

Eq. (4) shows that this model is only valid at the phase interface, as the multiplier $\nabla\alpha$ is non-zero there. When employing this model, it is mandatory to maintain the interface temperatures at T_{sat} [22]. This model is commonly utilized in published studies for addressing film boiling [28, 29] and evaporative cooling [30]. Lately, this model was simplified by assuming a linear distribution of liquid temperature near the interface [23]. Besides, the bubble growth was treated as independent of vapour conductivity considering negligible heat conduction of the vapour ($\lambda_g = 0$). Consequently:

$$S_g = -S_l = \frac{2\lambda_l(\nabla\alpha_l \cdot \nabla T)}{r}. \quad (5)$$

This model finds application in the evaporation simulation in heat pipes [31].

The third model, derived from the Hertz–Knudsen equation, was developed by Schrage [32], and was generally called Schrage model. It assumes there is a temperature jump at the interface: $T_{\text{sat}}(P_l) = T_l \neq T_{\text{sat}}(P_g) = T_g$. In this context, the mass flux equals the mass transfer difference between evaporation (liquid to vapour) and condensation (vapour to liquid) for thermodynamic equilibrium cases:

$$\dot{m} = \frac{2}{2 - \gamma_c} \sqrt{\frac{M}{2\pi R}} \left(\gamma_c \frac{P_g}{\sqrt{T_g}} - \gamma_e \frac{P_l}{\sqrt{T_l}} \right), \quad (6)$$

where M represents the molecular weight of the fluid, R the universal gas constant ($8.314 \text{ J} \cdot \text{mol}^{-1} \cdot \text{K}^{-1}$), and the coefficient γ denotes the molecular percentage crossing the phase interface caused by phase change. The subscripts c and e indicate condensation and evaporation, respectively [33].

In numerous literature sources, $\gamma_c = \gamma_e = \gamma$ is often utilized in which the value of γ is typically designated by researchers. Based on a comprehensive review, Marek and Straub [33] concluded that γ can range from 0.001 to 1. Tanasawa [34] simplified the Schrage model [32] by introducing a saturated phase interface, based on which the mass flux being linear to the temperature jump is expressed as:

$$\dot{m} = \frac{2\gamma}{2-\gamma} \sqrt{\frac{M}{2\pi R}} \frac{\rho_g r(T-T_{\text{sat}})}{T_{\text{sat}}^{3/2}}, \quad (7)$$

Thus, the following relation is determined by the simplified Tanasawa model [34]:

$$S_g = -S_l = \dot{m} |\alpha_g|. \quad (8)$$

Similar to the second model, Tanasawa model exclusively applies to the phase interface. By using this model, $\gamma = 1$ was used for film boiling in Refs. [35, 36], while $\gamma = 0.1$ for the calculation of evaporative heat transfer with a turbulent liquid film falling vertically along a plate in Ref. [22]. Silvi et al. [37] adopted $\gamma = 0.1$ for annular flow boiling inside a vertical circular tube, and Municchi et al. [38] utilized $\gamma = 0.046$ in interface cells and $\gamma = 1$ elsewhere for flow boiling in multi-microchannels.

In comparison, Lee model and Tanasawa model are more efficient than sharp interface model [39]. Furthermore, Tanawasa model is more physical than Lee model in aspect of triggering mechanism in phase change [9], especially for liquid film evaporation, phase change occurring at the interface of liquid and vapor.

1.3. Aims and objectives

Although significant research has been conducted on falling film flow and heat transfer, there are still notable gaps, especially concerning the numerical descriptions of film thickness and heat transfer coefficients (HTC) under evaporative conditions. Most existing studies focus on sensible heat transfer [40], leaving a lack of comprehensive data and understanding of falling film evaporation from the perspective of numerical approaches. Besides, the evaporative heat transfer within the thin falling film is traditionally relied heavily on experimental methods, as acknowledged to us, however, the mechanisms of hydrodynamics and heat transfer often remain elusive due to the limited testing capabilities at such small scales. Fortunately, computational fluid dynamics (CFD) offers a promising alternative, particularly with the availability of open-source CFD platform, which supports a wealth of interfaces for user-defined functions, facilitating more comprehensive analyses of the phenomena at hand.

The primary focus of the present study is to explore evaporative heat transfer in a liquid film falling outside a horizontal tube. This is achieved through numerical simulations of liquid film evaporation in the saturation state. The study presents variations in film thickness, the local and average HTCs on the tube wall under various

film flow rates and heat fluxes. Phase change is confined to the free surface of the liquid film, and the heat flux is controlled at a specific value to avoid nucleate boiling. Moreover, a suitable mass transfer model is employed according to the characteristics of this problem. Another objective of this study is to achieve computational modelling of evaporative falling film on an engineering scale, based on which the profiles of local film thickness, as well as the local and average heat transfer coefficients under different levels of film flow rate and heat flux were illuminated. And eventually, the critical heat flux and the map of operational limits due to liquid film dryout were constructed, expecting to guide the practical design and operation of the falling film heat exchangers. Such as how to keep running in the range of operational limits in order to avoid dryout. What is more, the quantitative results of film thickness and HTC over the horizontal circular tube in this study, can be devoted to the construction of empirical correlations.

The calculations were conducted with the open-source CFD code OpenFOAM v1812, supplemented by the Volume-of-Fluid (VOF) approach to track the liquid-vapour interface. The mass transfer model developed by Tanasawa [34] was incorporated into the solver to tackle phase change during film flow under tube heating. To save computational cost and runtime, a 2-D model was built.

2. Numerical framework

2.1. Computational domain

Fig. 1 shows the computational region used in the current work, in which the inlet and outlet of the liquid are at the top and bottom of the region, respectively. The horizontal tube has a diameter of 25.4 mm. The diagram depicting liquid spreading is also illustrated schematically in Fig. 1 Both liquid spreading and heat transfer are ideally bilaterally symmetrical. Consequently, only half of the tube is considered in the computational domain. The liquid distributor is assumed to be a slit with a total width of 2.0 mm, resulting in a film flow rate (Γ) on this half.

2.2. Governing equations

To simplify the problem, the following assumptions are applied: the liquid is characterised by constant thermophysical properties according to the saturation liquid temperature, the liquid and vapour are incompressible, and the film flow is in the laminar regime.

The solving of the governing equations was fulfilled by the TwoPhaseFlow [41] solver embedded in the open-source code OpenFOAM. In order to track the phase interface, the isoAdvector [42] was used, a geometric VOF method for advection of sharp interfaces between two incompressible fluids. The Continuum Surface Force (CSF) [43] was adopted to consider surface tension, with plicRDF [44] utilized for reconstructing the phase interface between the phases.

Based on the assumption and the VOF method, the continuity equation considering the mass transfer during the liquid and vapour phases can be expressed as [45]:

$$\frac{\partial(\rho_l \alpha_l)}{\partial t} + \nabla \cdot (\rho_l \alpha_l \mathbf{v}_l) = S_l, \quad (9)$$

$$\frac{\partial(\rho_g \alpha_g)}{\partial t} + \nabla \cdot (\rho_g \alpha_g \mathbf{v}_g) = S_g, \quad (10)$$

where the S_l and S_g represent mass source terms accounting for phase change, which are determined from Eq. (8).

The momentum equation is:

$$\frac{\partial(\rho \mathbf{v})}{\partial t} + \nabla \cdot (\rho \mathbf{v} \otimes \mathbf{v}) = -\nabla P + \nabla(\mu \nabla \mathbf{v}) + \rho \mathbf{g} + \mathbf{F}, \quad (11)$$

where $\rho \mathbf{g}$ denotes gravity, and $\mathbf{F}$ the surface tension whose formulation will be described later.

The energy equation is:

$$\frac{\partial(\rho E)}{\partial t} + \nabla \cdot (\mathbf{v}(\rho E + P)) = \nabla \cdot (\lambda \nabla T) + S_q, \quad (12)$$

where S_q is the energy source caused by the mass transfer, defined as:

$$S_q = r S_l, \quad (13)$$

and E is the energy per unit mass with unit of $\text{J} \cdot \text{kg}^{-1}$, determined by [45]:

$$E = \frac{\alpha_l \rho_l E_l + \alpha_g \rho_g E_g}{\alpha_l \rho_l + \alpha_g \rho_g}. \quad (14)$$

In the above equation, the thermal physical parameters are determined as: $\rho = \rho_l \alpha_l + \rho_g \alpha_g$, $\mu = \mu_l \alpha_l + \mu_g \alpha_g$, $\lambda = \lambda_l \alpha_l + \lambda_g \alpha_g$.

For this problem, the continuity equation is expressed as follows based on the VOF method:

$$\partial \alpha_g / \partial t + \mathbf{v} \cdot \nabla (\alpha_g) = 0. \quad (15)$$

According to the CSF model [43], the surface tension is incorporated into the source term in the momentum equations, as following expression:

$$F = \sigma \kappa \mathbf{n} \frac{\rho}{(\rho_l + \rho_g)/2}, \quad \mathbf{n} = \nabla \alpha, \quad \kappa = -\nabla \cdot \left(\frac{\mathbf{n}}{|\mathbf{n}|} \right), \quad (16)$$

where σ denotes the coefficient of the surface tension, and κ the curvature of the interface.

The liquid and tube were assumed respectively to be copper and water. Consequently, the contact angle of water on the copper tube was set at 10° according to Ref. [46] in all cases to account for the contact angle effect.

To optimize computational resources, the grid system used in our work consists of non-uniform quadrilaterals generated using the ICEM program (see Fig. 1). Refined grids near the solid boundary were arranged. Specifically, the size of the grids varies from 0.1 mm near the tube wall (liquid inlet with a width of 1.0 mm) to 0.72 mm at the right boundary for all tube diameters. Fig. 1 determines all boundary conditions in this study.

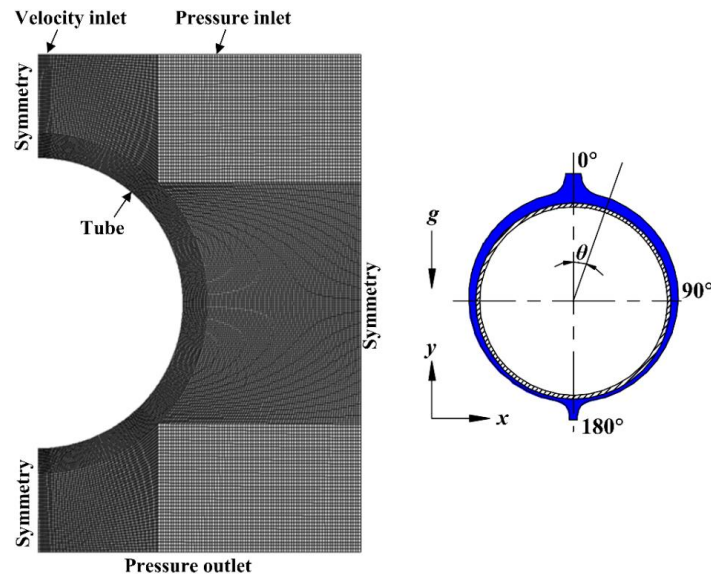


Fig. 1. Schematic of the calculation domain and the grid system.

2.3. Data reduction

For a given condition, the computation is continued until the film around the tube becomes steady, whence time-averaged local film thickness (δ_θ) and HTC (h_θ) are calculated:

$$\delta_\theta = \frac{1}{\Delta t} \int_0^{\Delta t} \delta(\theta, t) dt, \quad (17)$$

$$h_\theta = \frac{1}{\Delta t} \int_0^{\Delta t} \frac{q}{T_w(\theta, t) - T_i} dt, \quad (18)$$

in which, $\delta(\theta, t)$ and $T_w(\theta, t)$ depict respectively the transient local film thickness and tube wall temperature, q the heat flux imposed on the tube wall, and T_i the inlet water temperature.

Finally, the average HTC (h_{ave}) is obtained by integrating along the semi-periphery:

$$h_{ave} = \frac{1}{\pi} \int_0^\pi h_\theta d\theta. \quad (19)$$

3. Results and discussion

3.1. Grid independence verification

Grid independence verification was carried out with calculations for such grid densities as 30,875, 50,225, 73,367, and 99,377. The following parameters were involved: $\Gamma = 0.168 \text{ kg} \cdot \text{m}^{-1} \cdot \text{s}^{-1}$, $D = 25.4 \text{ mm}$, and $H = 6.3 \text{ mm}$. Fig. 2 compares the peripheral HTCs (h_θ) of the four cases. Obviously, h_θ varies minimally as the grid number increases above 73367. As a result, the grid configuration of 73,367 was used in all cases. Additionally, calculations based on various time step settings were conducted. The local HTC are compared in Fig. 3, from which it can be observed that the time steps of $5\text{e-}6 \text{ s}$ and $1\text{e-}5 \text{ s}$ have negligible effect on the results. To save calculation costs, an adapted time-step was adopted through maintaining the Courant number lower than 0.3.

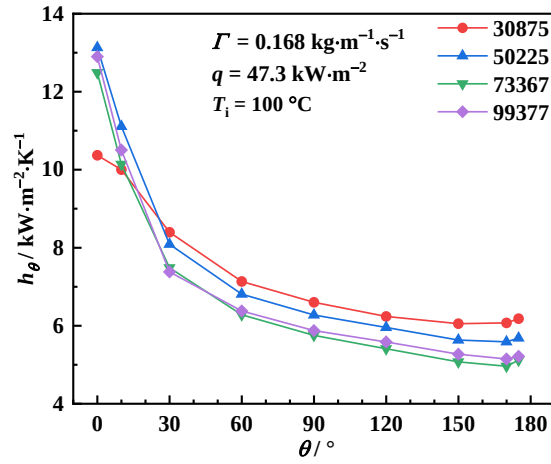


Fig. 2. Grid independence test for different grid numbers.

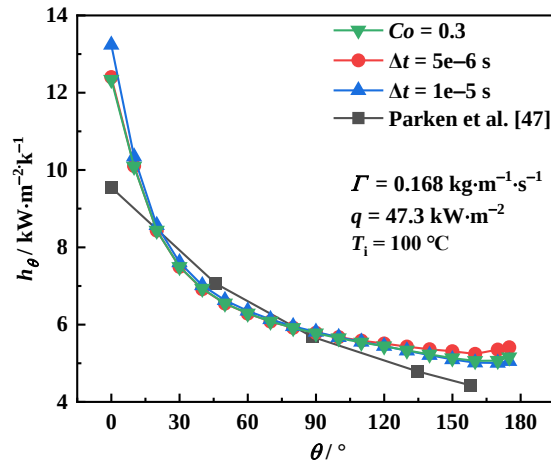


Fig. 3. Time step independence test for different time step settings.

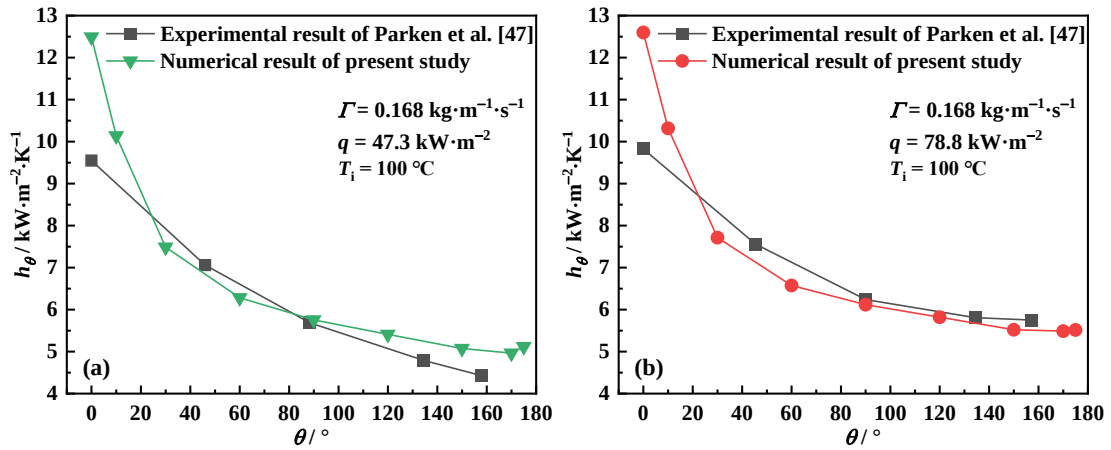


Fig. 4. Comparison of HTC and predicted results by Parken et al. [47] for heat fluxes: (a) $q = 47.3 \text{ kW}\cdot\text{m}^{-2}$, (b) $q = 78.8 \text{ kW}\cdot\text{m}^{-2}$.

3.2. Model validation

Two cases with $\Gamma = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, $q = 47.3$ and $78.8 \text{ kW}\cdot\text{m}^{-2}$, and $T_{\text{in}} = 100 \text{ }^{\circ}\text{C}$, were calculated to validate the reliability of our model. Here, the value of γ in the

Tanasawa model used in this study was 0.1. Fig. 4 compares the current numerical data and the experimental results of Parken et al. [47]. Notably, the maximum and average deviations at the stagnation point are 21% and 6%, respectively, for $\Gamma = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ and $q = 47.3 \text{ kW}\cdot\text{m}^{-2}$, which we consider adequate for the purpose of the present study. Besides, to clear the designation of the value of γ , the peripheral HTC obtained based on different γ are plotted in Fig. 5. It can be seen that the value of γ influences little on the local HTC except the region of θ from 120° to 180° . Accordingly, the $\gamma = 0.1$ was designated in all calculations.

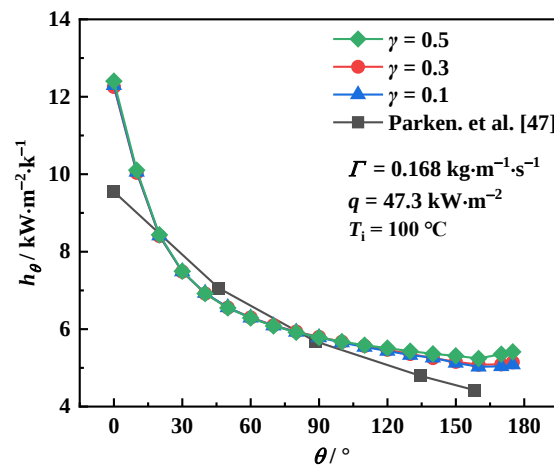


Fig. 5. The peripheral HTC obtained based on various coefficient γ .

3.3. Time history analysis of liquid film

Fig. 6 illustrates the liquid film evolution on a horizontal tube with $D = 25.4 \text{ mm}$ and $H = 6.3 \text{ mm}$ under an evaporative condition: $\Gamma = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, $T_i = 100 \text{ }^\circ\text{C}$, and $q = 47.3 \text{ kW}\cdot\text{m}^{-2}$. As can be seen, the liquid film exhibits behaviour similar to the case without phase change throughout the entire duration. For more details, refer to Zhao et al. [48]. The main difference is that the film thickness with phase change is slightly smaller. However, the liquid film dryout does not occur mainly due to the lower heat flux.

To gain a comprehensive understanding of the time history of evaporative falling film on a horizontal tube, Fig. 7 displays similar liquid film evolution under condition similar to those in Fig. 6 but with a larger heat flux of $160.0 \text{ kW}\cdot\text{m}^{-2}$ was imposed. As seen in Fig. 7 (a-i), the liquid film undergoes similar evolutions to the case with heat flux of $47.3 \text{ kW}\cdot\text{m}^{-2}$ (see Fig. 6 (a-i)). At $t = 3.165 \text{ s}$, a notable phenomenon occurs, particularly in the region adjacent to the tube bottom, where liquid film breakout is

observed attributed to the liquid film dryout. Additionally, as seen in Fig. 7 (j), besides, liquid film accumulations appear near the three-phase contact points, possibly because of resistance from the vapor, and droplets are ejected under the action of vapor shear and centrifugal forces. As seen in Fig. 7 (k), liquid films detach from the tube, forming liquid tongues, under further action of vapor shear and centrifugal forces. As time progresses, the affected area of liquid film dryout increases, as seen in Fig. 7 (l), in where ejected droplets, liquid tongues, and liquid film film dryout coexist.

It is worth noting that, the liquid film distribution described above are just snapshots at individual time points. Therefore, it is reasonable to speculate that these dryout regions may be either permanent or temporary depending on the levels of film flow rate and heat flux. Consequently, the thresholds of film flow rate and heat flux are significant for for predicting liquid film dryout. Overall, during evaporative falling film heat transfer on a horizontal tube, the liquid film dryout basically first appears at the lower part of the tube.

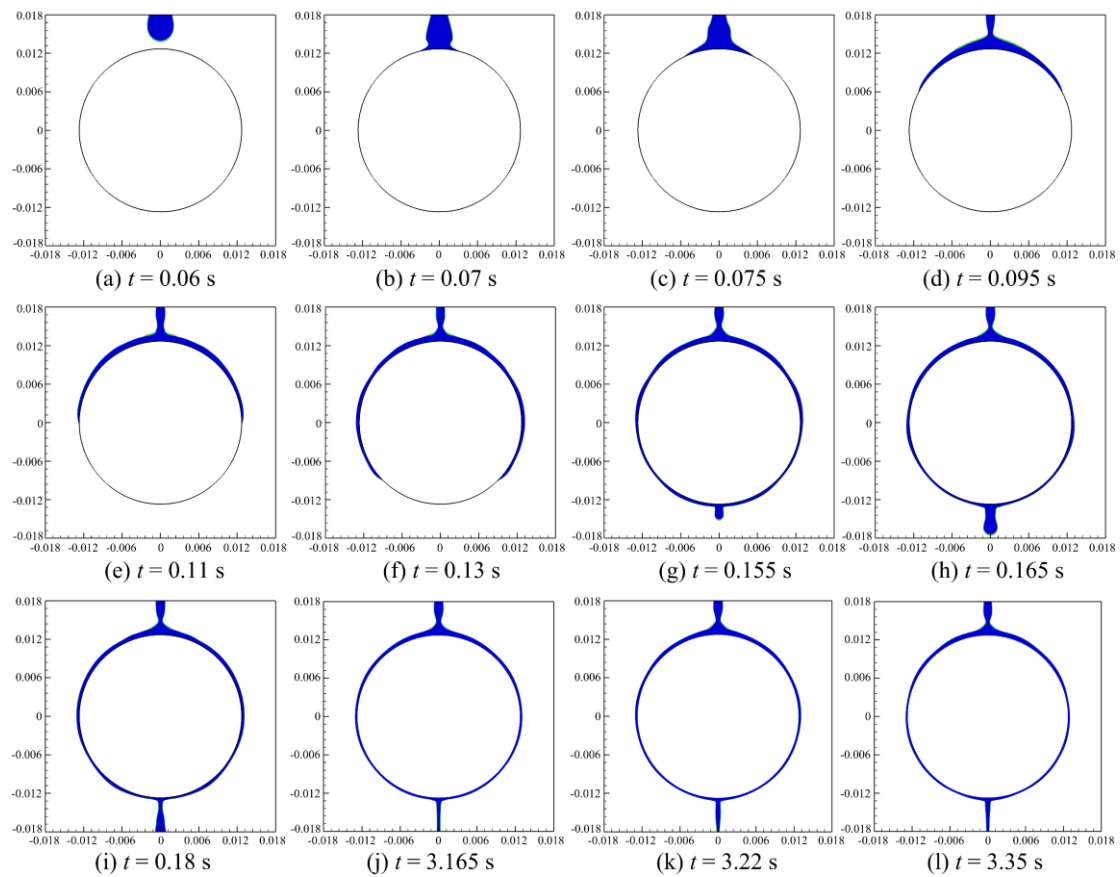


Fig. 6. Liquid film distributions at different times, $D = 25.4$ mm, $H = 6.3$ mm, $\Gamma = 0.168$ kg·m⁻¹·s⁻¹, $T_i = 100$ °C, and $q = 47.3$ kW·m⁻².

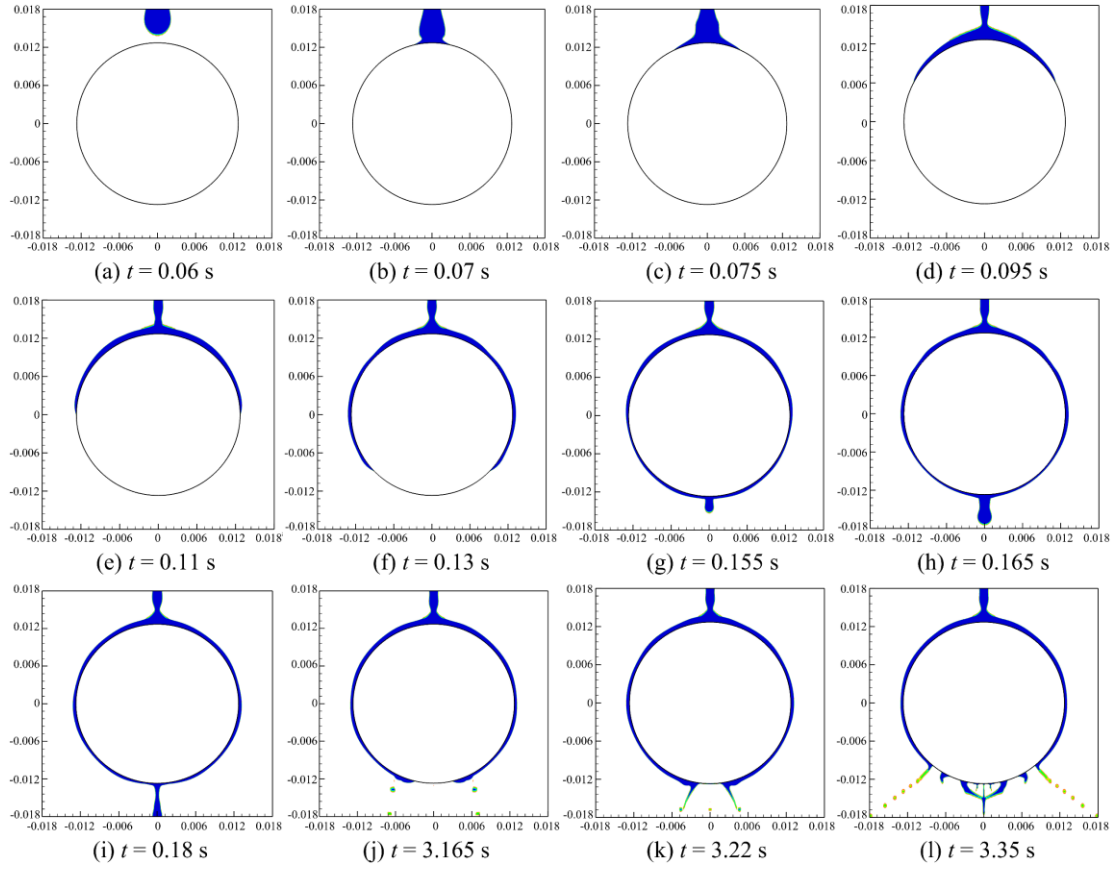


Fig. 7. Liquid film distributions at different times, $D = 25.4$ mm, $H = 6.3$ mm, $\Gamma = 0.168$ kg·m⁻¹·s⁻¹, $T_i = 100$ °C, and $q = 160.0$ kW·m⁻².

3.4. Effect of film flow rate

It is consistently revealed that film flow rate overwhelmingly dominates falling film heat transfer through influencing liquid film characteristics including thickness, velocity and profile, even dry patch status [9]. With this in mind, the fluctuations of peripheral film thickness and HTC with film flow rate are examined.

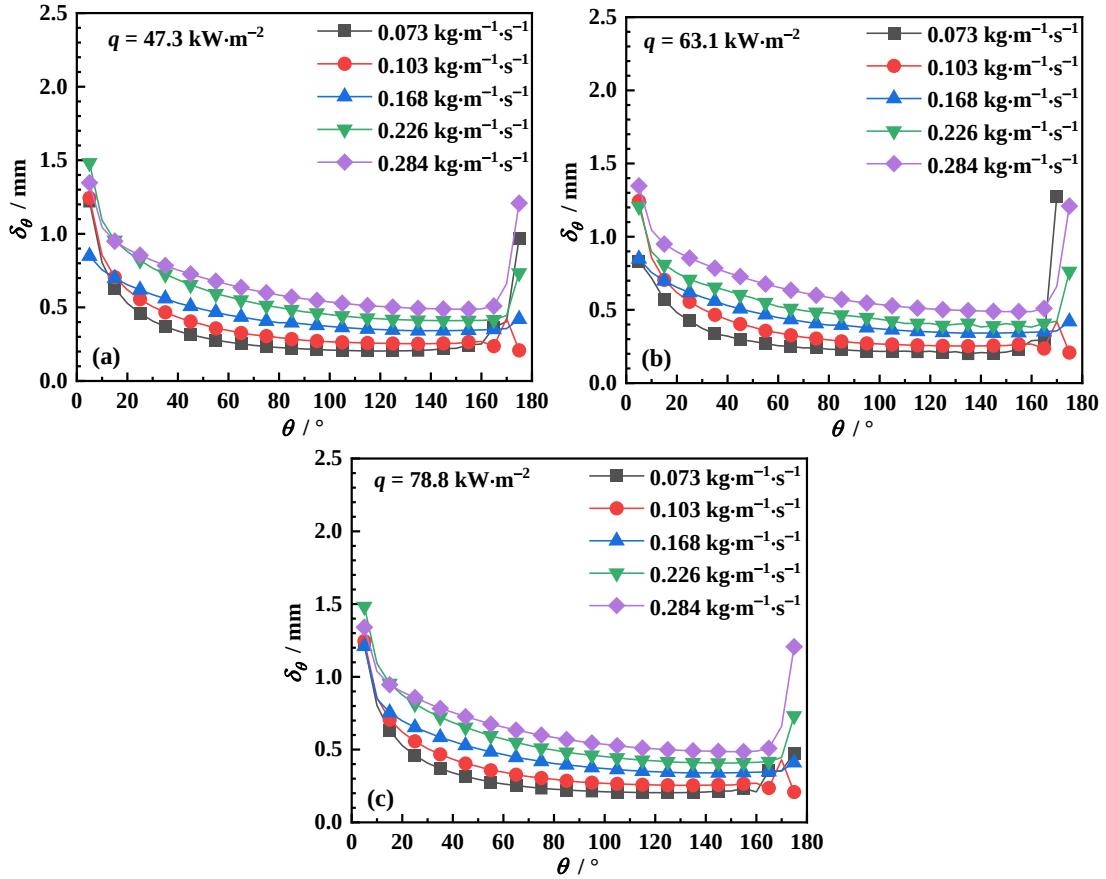


Fig. 8. Peripheral film thickness variations at film flow rates: (a) $q = 47.3 \text{ kW} \cdot \text{m}^{-2}$, (b) $q = 63.1 \text{ kW} \cdot \text{m}^{-2}$, (c) $q = 78.8 \text{ kW} \cdot \text{m}^{-2}$.

Fig. 8 displays the fluctuations of the local film thickness, δ_θ , under three heat fluxes. As plotted in Fig. 4 (a), at $q = 47.3 \text{ kW} \cdot \text{m}^{-2}$ for example, δ_θ first decreases sharply, then decreases gently, and then increases suddenly. With an ascending film flow rate, δ_θ notably increases within θ from 13° to 165° . This trend is less pronounced in the impingement and detachment zones. Similar profiles are observed at $q = 63.1$ and $78.8 \text{ kW} \cdot \text{m}^{-2}$ in Fig. 8 (a) and (b), respectively. In sum, δ_θ increases with the film flow rate for the laminar regime and without liquid film dryout condition. Furthermore, the trends of film thickness profiles are highly similar to the ones in adiabatic [49] and sensible heat transfer [50, 51] conditions, but the values are smaller due to evaporation.

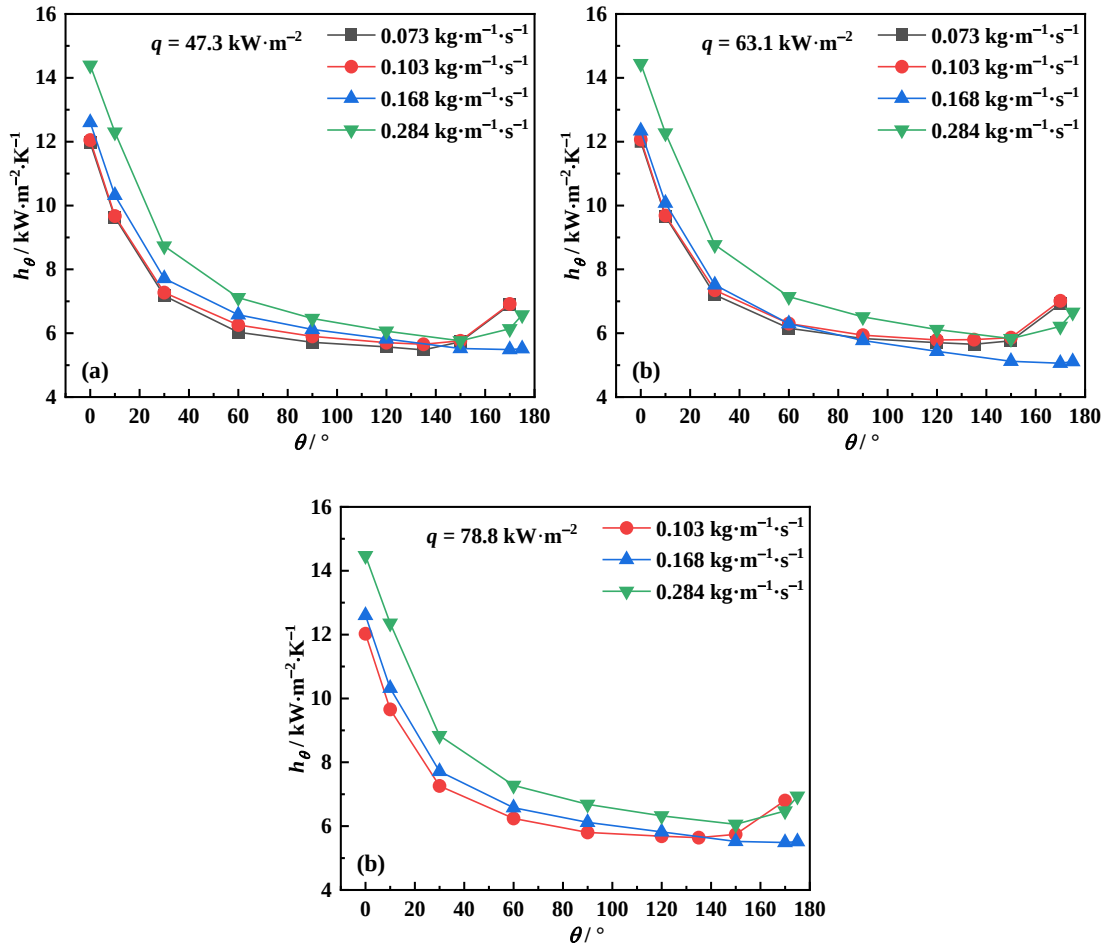


Fig. 9. Peripheral HTC variations at film flow rates: (a) $q = 47.3 \text{ kW}\cdot\text{m}^{-2}$, (b) $q = 63.1 \text{ kW}\cdot\text{m}^{-2}$, (c) $q = 78.8 \text{ kW}\cdot\text{m}^{-2}$.

Fig. 9 shows local HTC, h_θ , profiles for three heat fluxes. At $q = 47.3 \text{ kW}\cdot\text{m}^{-2}$, h_θ generally increases with the film flow rate, as displayed in Fig. 9 (a). In detail, the cases with $\Gamma = 0.073$ and $0.103 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ provide similarly worse heat transfer performance due to dry patches. With increasing flow rate, up to 0.168 or even $0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, heat transfer is noticeably improved. Interestingly, the HTC shows a slightly increase after around 130° for $\Gamma = 0.103$ and $0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, while the HTC keeps decreasing for $\Gamma = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$. The different trends are related to the different velocity fields under different film flow rates. It was found that there are noticeable vortices and waves in the region of after around 130° at $\Gamma = 0.103$ and $0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, which are beneficial to heat transfer, however, the liquid film flows quietly at $\Gamma = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$. At $q = 63.1 \text{ kW}\cdot\text{m}^{-2}$ (Fig. 9 (b)), the flow rate continues to facilitate heat transfer. The difference is that h_θ at $q = 0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ drops substantially, especially on the lower tube's half (θ from 90° to 180°). Further, at $q = 0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, h_θ exhibits a similar profile and slight superiority to that of $0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$. E.g., at θ of

135°, it is $5.828 \text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ at $0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ vs. $5.76 \text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ at $0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$. As heat flux increases to $78.8 \text{ kW}\cdot\text{m}^{-2}$ (see Fig. 9 (c)), h_θ of flow rates from 0.103 and $0.168 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ remains constant or is marginally enhanced, while that of a flow rate at $0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ is significantly enhanced. It is worth noting that the flow rate of $0.073 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ was not plotted in Fig. 9 (c) because such low values are caused by film dryout. Summarily, h_θ generally increases with the film flow rate but the increase extent differs with the heat flux.

As discussed above, the film thickness increases at higher flow rates, indicating a corresponding increase in thermal conduction resistance through the film. The heat transfer performance seems to improve with increasing thermal conduction resistance, which is reasonable since a larger film flow rate would be expected to enhance liquid film liquid disturbances, and to increase the wetting area. Both of them are advantageous to heat transfer. In addition, the effect of film flow rate is dependent on the level of heat flux in phase change scenarios. At lower heat flux levels, convective heat transfer contributes more significantly than evaporation. In this case, variations in film flow rate greatly affect heat transfer performance by altering film thickness and velocity. At higher heat flux levels, evaporation becomes the dominant mechanism in falling film heat transfer. Here, the film flow rate mainly determines the wetting ratio of the tube surface due to the higher risk of liquid film dryout. Consequently, heat transfer performance increases with the film flow rate up to a critical value but remains constant beyond this point.

In comparison to the sensible heat transfer conditions studied in [48, 51], the profiles of evaporative falling film HTC drop more gently within the impingement zone (θ from 0° to 10°) and present much higher levels. For instance, at θ of 90° it is $6.46 \text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ for evaporative falling film heat transfer vs. $4.03 \text{ kW}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$ for sensible falling film heat transfer. Additionally, the curves of the local HTC are more scattered. This trend is because a higher film flow rate promotes heat transfer, increases the liquid film temperature, and ultimately enhances evaporation.

3.5. Effect of heat flux

Evaporation differs significantly from sensible heat transfer. It has been clearly shown that the sensible falling film heat transfer does not rely on the heat flux [9, 52].

This section investigates the impact of heat flux on the evaporative falling film heat transfer.

The fluctuations in the local HTC at various heat fluxes are displayed in Fig. 10. Generally, the impact of heat flux on HTC varies substantially depending on the film flow rate. Herein, results based on three film flow rates are compared. At $\Gamma = 0.103 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, as displayed in Fig. 10 (a), h_θ increases slightly as the heat flux rises from 47.3 to 63.1 $\text{kW}\cdot\text{m}^{-2}$, while decreases conversely as the heat flux further ascends to 78.8 $\text{kW}\cdot\text{m}^{-2}$, throughout the entire tube periphery. At $\Gamma = 0.226 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, as shown in Fig. 6 (b), h_θ gradually grows as the heat flux grows from 47.3 to 94.6 $\text{kW}\cdot\text{m}^{-2}$ but decreases suddenly to a level even lower than that for the heat flux of 47.3 $\text{kW}\cdot\text{m}^{-2}$ as the heat flux climbs to 126.0 $\text{kW}\cdot\text{m}^{-2}$, throughout the entire tube periphery. More universal, a larger heat flux provides a lower h_θ at a smaller film flow rate but a higher h_θ at a larger film flow rate.

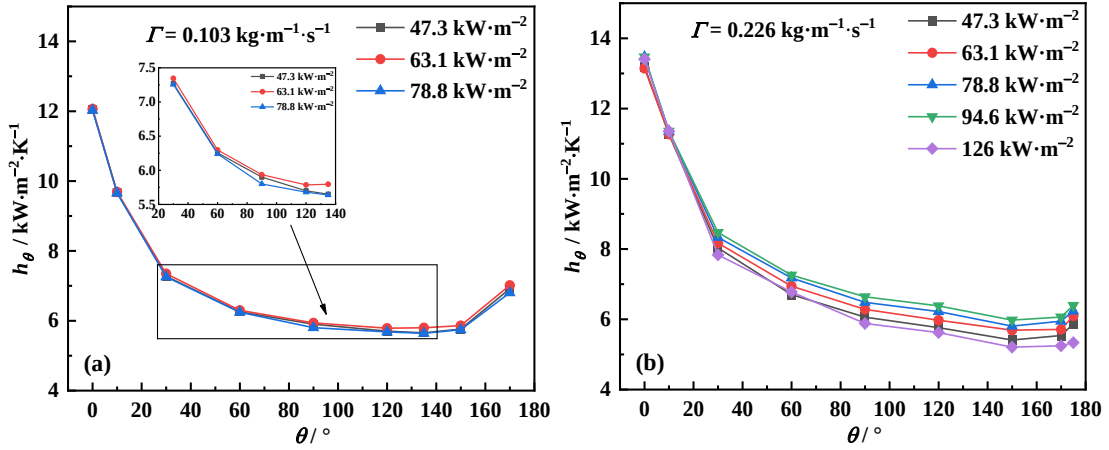


Fig. 10. Peripheral HTC variations at heat fluxes: (a) $\Gamma = 0.103 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, (b) $\Gamma = 0.226 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$.

As described, evaporative heat transfer is affected by the heat flux either positively or negatively, relying on the film flow rate and heat flux. Specifically, at a lower level of film flow rate, a higher heat flux facilitates heat transfer due to the resulting larger superheat temperature difference ($T - T_{\text{sat}}$ in Eq. (7)). This agrees well with the findings of Parken et al. [47]. However, as the heat flux increases to a higher level, which is a threshold to some extent, a higher heat flux deteriorates evaporation due to the liquid film dryout. In contrast, at a higher film flow rate, the threshold of heat flux grows as the tolerance of the liquid film dryout is strengthened by sufficient liquid feedings. As a result, the evaporative falling film performance is weakened at $q = 78.8$, and 126.0 $\text{kW}\cdot\text{m}^{-2}$ for $\Gamma = 0.103$ and 0.226 $\text{kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, respectively. Here, the heat flux and film

flow rate are combined to determine the evaporative falling film heat transfer, as discussed above.

It is worth emphasizing that the influence in heat flux on evaporative falling film heat transfer is the most significant difference from sensible heat transfer performance. Moreover, the effect of heat flux on evaporative falling film heat transfer is dependent in turn on the film flow rate. Specifically, the heat transfer coefficient increases with heat flux when the film flow rate is sufficient to fully wet the surface. However, it decreases sharply when the film flow rate is insufficient, causing liquid film dryout.

3.6. Effect of tube diameter

Tube diameter is a crucial design parameter for falling film heat transfer. Zhao et al. [48] concluded that changes in tube diameter alter the heat transfer profile along the tube periphery by influencing the thermal boundary layer length and the impingement area on the tube's periphery. Besides, the tube diameter also influences the film thickness, as presented in Zhao et al. [50], where a larger tube diameter decreases film thickness. This changes the thermal conduction resistance and affects the tolerance of liquid film dryout.

Fig. 11 shows the variations in the local HTC with peripheral angle. Here, the results based on various heat fluxes, film flow rate for tubes, and tube diameters are respectively illustrated. At $\Gamma = 0.103 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, as shown in Fig. 11 (a) and (b), it can be observed that the tube diameter has a marginal effect on the profiles of h_θ but leads to a slight reduction in value. At $\Gamma = 0.284 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$, as seen in Fig. 11 (c) and (d), it is evident that h_θ for the tube with a larger diameter decreases materially, and the profiles of h_θ become less scattered. More generally, a larger tube diameter dose harm to evaporative falling film heat transfer, and supress the influence of heat flux.

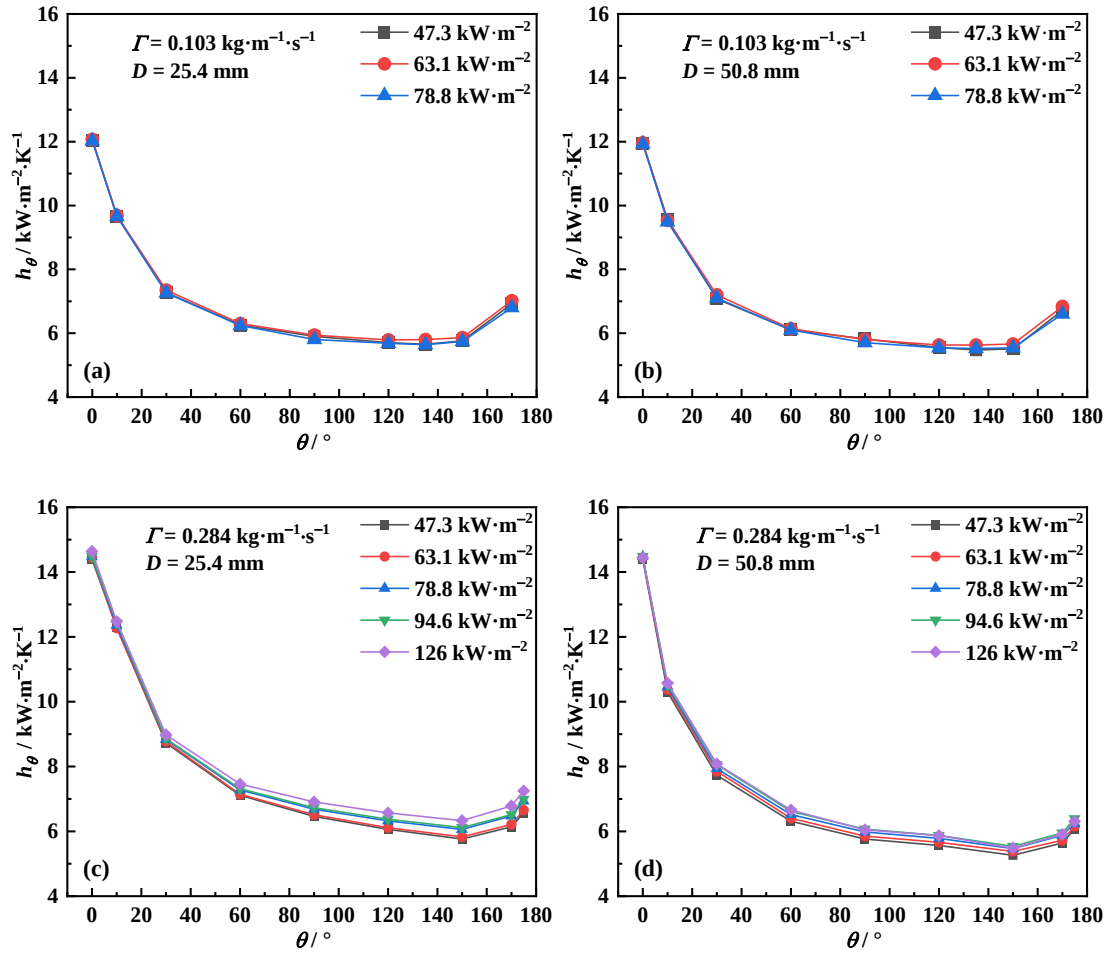


Fig. 11. Peripheral HTC variations for different tube diameters.

3.7. Critical heat flux

As highlighted by Zhao et al. [9], there is a threshold of heat flux at a fixed film flow rate during evaporative falling film heat transfer. This threshold, also known as the critical heat flux, signifies the point beyond which the HTC experiences a rapid decline due to liquid film dryout. Unlike its counterpart in nucleate pool boiling, where it represents the end of nucleate boiling because of bubble accumulation on the heating wall, the critical heat flux in evaporative falling film heat transfer is primarily linked to liquid film dryout derived from the liquid shortage. This section examines variations in the average HTC as a function of heat flux.

The average HTC for a range of film flow rates and tube diameters is plotted against heat flux in Fig. 12. For $D = 25.4$ mm, as shown in Fig. 12 (a), a larger film flow rate contributes to a higher h_{ave} at the same heat flux. Moreover, with increasing heat flux, h_{ave} initially increases to a peak and then decreases for all film flow rates. A similar trend can be observed for $D = 50.8$ mm, as illustrated in Fig. 12 (b).

To have a thorough grasp of the impact of heat flux, the critical heat flux is investigated. Here, the critical heat flux is defined as the heat flux at which the HTC's reach peak values before declining rapidly to a lower value under liquid film dryout. As displayed in Fig. 12, the critical heat flux is denoted by an arrow. Beyond the critical heat flux, evaporative falling film heat transfer shrinks due to liquid film dryout. It can be evident that the critical heat flux generally grows with increasing film flow rate, but declines with increasing tube diameter.

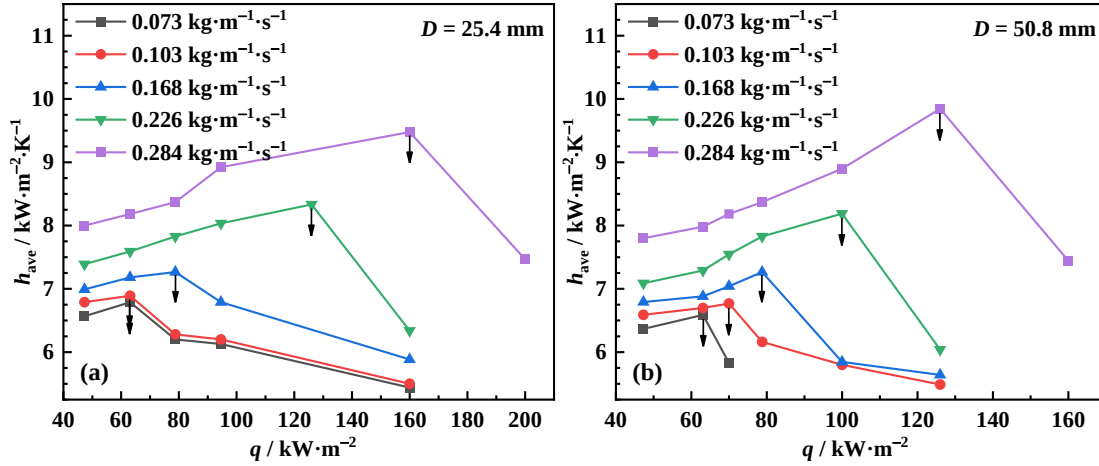


Fig. 12. Average HTC variations with heat flux at different film flow rates.

Fig. 13 shows transient distributions of the liquid phase over the tube corresponding to conditions with critical heat fluxes in Fig. 12. As shown, all dryout points appear in the lower half of the tube, close to the lower stagnation point. These phenomena are reasonable because dryout is generally caused by evaporation. In the lower half of the tube, the liquid film absorbs sufficient heat, accelerating its evaporation and thinning. Comparing Fig. 13 (a) and (b) reveals that for a constant heat flux, the dryout point is postponed by a higher film flow rate because a larger film flow rate makes up for the liquid loss due to evaporation. Similarly, when comparing Fig. 13 (b) and (c), the dryout point expands as the heat flux increases from 63.1 to 126.0 kW·m⁻², despite the film flow rate almost doubling from 0.103 to 0.226 kg·m⁻¹·s⁻¹. It is easy to be understood that a higher heat flux promotes evaporation, amplifying the dryout risk. Further increasing the film flow rate to 0.284 kg·m⁻¹·s⁻¹ prevents dryout, even with a 27% increase in heat flux. For the tube with a larger diameter of 50.8 mm, dryout also occurs early due to film thickness reduction that increases film temperature and promotes evaporation, as evident in the comparison between Fig. 13 (a) and (e), (b) and (f), as well as (c) and (g). Moreover, when the film flow rate increases to 0.284 kg·m⁻¹·s⁻¹,

dryout is still absent (see Fig. 13 (h)), but the risk of dryout is higher compared to a tube with a smaller diameter (see Fig. 13 (a)).

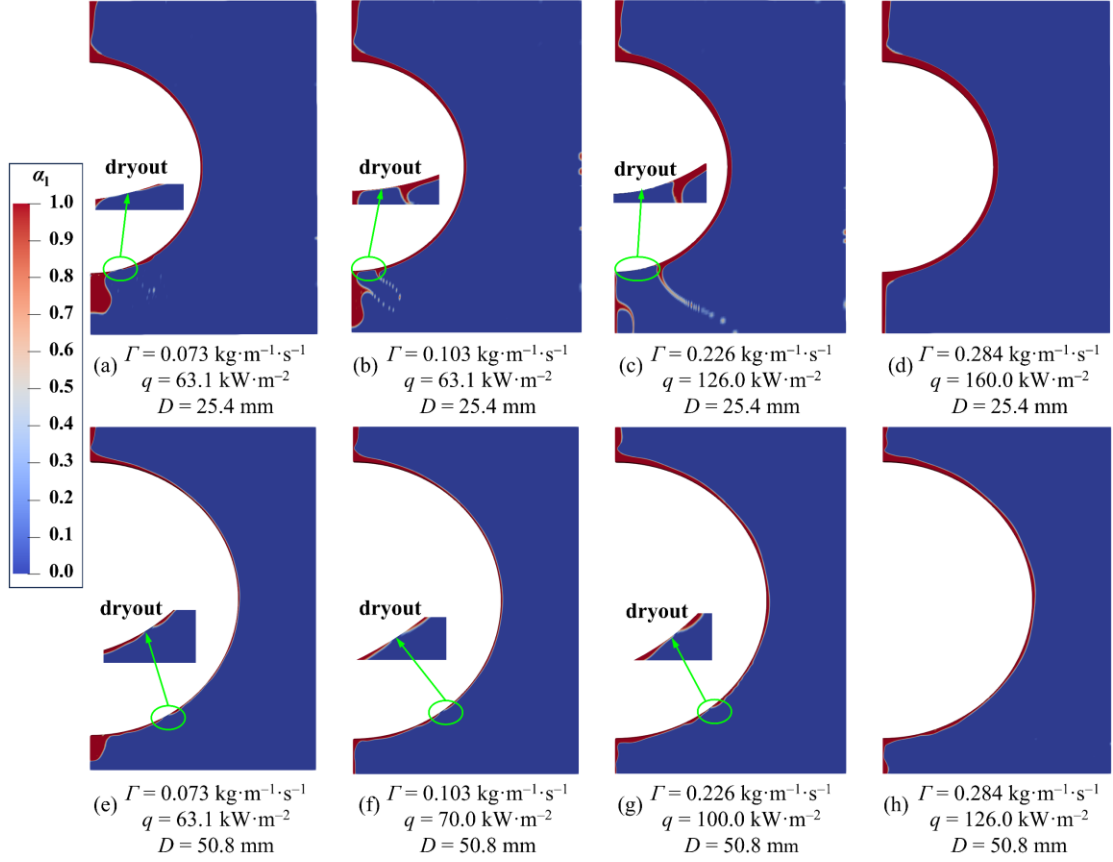


Fig. 13. Transient liquid film contours at conditions of critical heat flux over the tube.

It can be speculated that the occurrence of falling film dryout is influenced by both liquid film flow and evaporation. The former determines the stability or disturbance of the liquid film, while the latter dominates the thickness of the remaining (unevaporated) liquid film. Therefore, the liquid film dryout results from the combined impacts of film flow rate and heat flux. In different scenarios, the occurrence of liquid film dryout may be determined either by reducing the film flow rate or by expanding the heat flux.

3.8. Operational limits

As discussed earlier, the occurrence of liquid film dryout is determined by both film flow rate and heat flux. Consequently, a liquid film dryout map can be constructed to illustrate the operational limits of falling film evaporation.

Higher film flow rates suppress liquid film dryout or rewet dry patches, thereby delaying the inception of the critical heat flux. Specifically, a lower (higher) film flow rate results in a lower (higher) critical heat flux, at which liquid film dryout occurs. Fig.

14 presents a liquid film dryout map, composed of coordinates representing film flow rate and heat flux, based on the current cases under consideration. From this map, for a tube with any diameter in an area with a lower heat flux than the critical heat flux and a larger film flow rate than the critical film flow rate (the film flow rate corresponds to the critical heat flux), evaporative falling film heat transfer operates in a fully wetting status. This condition is recommended for a fixed heat flux or film flow rate. For example, the heat flux should not exceed $100 \text{ kW}\cdot\text{m}^{-2}$ when the film flow rate is $0.226 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ for a tube with a diameter of 50.8 mm , or similarly, the film flow rate should exceed $0.226 \text{ kg}\cdot\text{m}^{-1}\cdot\text{s}^{-1}$ when the heat flux is $100 \text{ kW}\cdot\text{m}^{-2}$.

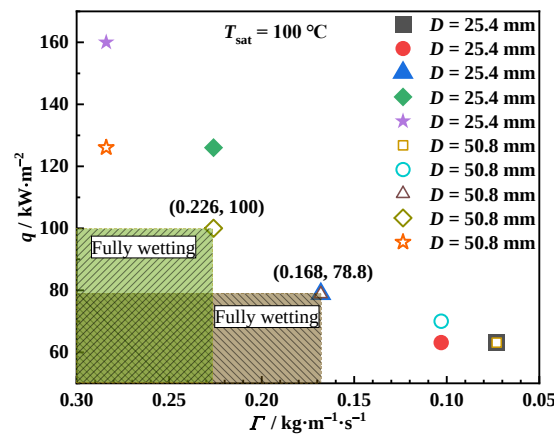


Fig. 14. Critical heat flux as a function of film flow rate.

4. Conclusions

This paper presents a two-dimensional numerical model, implemented using the open-source CFD code OpenFOAM v1812, to study and elucidate the flow and heat transfer characteristics of evaporative falling films on a single horizontal tube. The study explored the impacts of the film flow rate, heat flux, and tube diameter, including discussions on critical heat flux and operational limits.

The key findings from the results of the study are summarized below:

- (1) The present model can accurately reproduce evaporative falling film heat transfer performance.
- (2) The film thickness exhibits a profile similar to that in adiabatic and sensible falling film heat transfer but with smaller values. The peripheral evaporative heat transfer coefficient varies gently and attains significantly higher levels.
- (3) The impact of heat flux on evaporative falling film heat transfer depends on the

- film flow rate level, resulting in either enhancement or weakening.
- (4) Larger tube diameters negatively affect heat transfer, suppressing the influence of heat flux, while also increasing the risk of liquid film dryout.
 - (5) For a fixed film flow rate, the average heat transfer coefficient initially increases, reaches a peak, and then decreases with an increase in heat flux.
 - (6) The critical heat flux grows with the flow rate and declines with the tube diameter.
 - (7) An operational limit has been identified; when the heat flux is lower than the critical heat flux, and the film flow rate is higher than the critical film flow rate, evaporative falling film heat transfer operates in a fully wetting status.

Acknowledgments

This research has been funded by the National Natural Science Foundation of China (No. 51976144; 51908444). The authors also gratefully acknowledge the support of the Youth Innovation Team of Shaanxi Universities, the International Joint Research Center for Building Service Science and the Underground Space Environment, Shaanxi (China). The work was also supported by the UK Engineering and Physical Sciences Research Council (EPSRC) [grant number EP/T03338X/1]. For the purpose of Open Access, the authors have applied a CC BY public copyright licence to any Author Accepted Manuscript version arising from this submission.

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