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Heat transfer characteristics in horizontal tube bundles for falling film evaporation in multi-effect desalination system

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HIGHLIGHTS

- 2 dimensional modelling of falling film evaporation
- Prediction of film coefficient and evaporation for different tube sizes
- Experimental studies on falling film evaporation
- Comparison of in-line staggered tube layouts in tube bundle
- Comparison of CFD, experimental data and published literature

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ABSTRACT

Horizontal tube falling film evaporation finds wider applications in multi-effect distillation system in recent years. The latent heat released inside the tube due to condensation of vapor is transferred to the falling film on the outer surface of the tube resulting in convective evaporation of water film. The evaporator consists of multiple rows and columns of horizontal tubes. Heat transfer from condensing film inside the tube bundle is more or less uniform while there is a large variation in heat transfer outside the tube bundle. This paper focuses on Computational Fluid Dynamics (CFD) analysis of falling film evaporation of seawater on a single tube and bundle of tubes using ANSYS Fluent 13.0®. CFD results are validated with published data available in the literature and also with experimental studies carried out. The effect of feed rate, tube diameter, wall temperature, etc. on the heat transfer is studied. Local film coefficient around the tubes of 19.05, 25.4 and 50.8 mm ∅ for different film Reynolds numbers is discussed. It is observed that convective evaporation heat transfer performance increases with feed rate, but decreases with tube diameter. In-line tube configuration is found to be better compared to staggered tube configuration.

1. Introduction

Falling film evaporators find more applications in refrigeration and desalination industries due to less liquid inventory, high heat transfer coefficient, low temperature drop, low pressure drop, etc. Falling film evaporation takes place outside the tube geometry with two-phase heat transfer. Convective evaporation as well as nucleate boiling occurs in the film as it flows over the tube depending on the heat flux conditions. There are three modes of flowing film which are drop mode, column mode and sheet mode which is related to the fluid flow rate. Heat transfer coefficient around tube surface is influenced by the film Reynolds number and the mode of the film. Film mode in turn depends on film Reynolds number. Heat transfer in tube bundles compared to a single tube is significant as evaporators and condensers made of in-line or staggered tubes of considerable numbers. Most of the theoretical and experimental research on falling film evaporation is based on a single tube. In multi-effect desalination (MED) system, seawater is sprayed over the tube bundle and flows as a thin film over the tubes due to gravity under vacuum conditions. Liquid maldistribution and local dry out are common problems in such systems. Normally upper tubes experience excess flow rate and film thickness while the lower strata in tube bundle have dry out conditions. Experimental investigations conducted in the past show different trends in heat and mass transfer due to the simultaneous occurrence of nucleate boiling and convective evaporation. At lower heat flux only convective evaporation takes place and normally film coefficient increases with film Reynolds number. At high heat flux conditions nucleate boiling takes place as observed in experimental studies and the film Reynolds number has no significant influence. Falling film evaporators normally work under low heat flux conditions with convective evaporation. This paper discusses the Computational Fluid

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Dynamics (CFD) studies as well as experimental investigation on horizontal tube bundle for heat and mass transfer under various operating conditions.

A review of investigations on tube bundle was published by Ribatski and Jacobi [1]. Working fluids used in most of the investigations are refrigerants such as R 134a and ammonia. Fujita and Tsutsui [2] carried out experimental studies on 5 vertical tubes with refrigerant R-112 and the highest film coefficient was observed at the topmost tube. Zeng et al. [3] carried out experimental studies on in-line and staggered tubes with ammonia. It was reported that rectangular pitch configuration gave better results. Experimental studies of Moeykens et al. [4] carried out with R-134a and R-123 reported that staggered pitch gave better results compared with in-line tubes at high heat flux conditions. Roques et al. [5] carried out experimental studies with 10 tubes in vertical array using R-134a and observed that film coefficient reaches a stabilized value at $Re_f = 500$. It is also reported that the lower tubes showed a higher film coefficient. Habert M. [6] carried out experimental with R134a and R236fa to study the effects different working fluids, type of tubes, feed rates and heat flux conditions. It was reported that tubes present in the 4^{th} to 6^{th} row gave better film coefficients.

Sharma R. and Mitra S.K. [7] conducted studies on tube bundles with 120 rows of 22 mm tube size adopting triangular tube configuration for brackish water at atmospheric conditions. Tube drying and reduction in heat transfer coefficient was observed below $Re_f = 400$. Lower tubes in the bundle experienced reduction in film coefficient and liquid evaporation. Theoretical and experimental investigations on tube or tube bundles were presented by Yang and Wang [8], We Li et al. [9,10], Mu et al. [11], Jani et al. [12] and Shen et al. [13] in recent times. In certain cases the film coefficient increased with feed rate, stabilized or decreased beyond a limit. There observed a gradual reduction in heat transfer coefficient with bundle depth and film dry out occurred in low feed rates. Present study, using CFD and experimental methods, was due to the lack of information on tube bundles compared to single tube and to study the reduction in performance in tubes in the vertical direction in the bundle. A comparison of in-line and staggered tubes were carried out with sea water and fresh water which has applications in seawater desalination.

2. Factors influencing film heat transfer

In falling film evaporation, seawater is sprayed on the top of the tube and allowed to flow along the curved tube surface. According to Chyu and Bergles [14] flow region can be divided into four. Liquid jet is impinging on top of the tube where stagnation region is formed. At the impingement zone, the flow takes a sharp turn and because of this the film coefficient is high due to the high velocity gradient. This is followed by the developing region, where the heat transfer coefficient decreases with increase in thermal boundary layer thickness. Heat transfer coefficient further reduces as the thermal layer is fully developed. At the bottom of the tube the thickness of film increases and flow is detached from the tube bottom surface and the schematic details of falling film with four regions is reported earlier by Raju Abraham and Mani [15].

Falling film evaporation is governed by conduction heat transfer across the thin film and convective evaporation from liquid to vapor at the interface. If the heat flux is sufficiently high nucleate boiling takes place at the tube surface in addition to the convective evaporation and bubbles start to form in the film. This further enhances the heat transfer coefficient. As the flow proceeds around the tube, it is accelerated by the gravity force and decelerated by the viscous force. A viscous boundary layer develops until it is extended to the free surface of the film.

Chyu and Bergles [14] developed models for heat transfer prediction in the distinct flow regions along the tube surface. This model was later used by Sharma and Mitra [7] for heat transfer prediction for film evaporation of sea water. According to Chyu and Bergles, local heat transfer coefficient at the tube surface at the stagnation flow zone can be estimated as

$$h_s = 1.03\,Pr^{\frac{1}{3}}k\left[\frac{d\left(\frac{u_{max}}{u_j}\right)}{d\left(\frac{x}{w}\right)}\frac{u_j}{v\cdot w}\right]^{0.5}. \tag{1}$$

Velocity gradient is constant in stagnation zone and the angle for the region of stagnation is given by $\theta_s = 0.6(\frac{w}{R})$. Similarly the local surface heat transfer coefficient at the impingement zone is given by

$$Nu_i = \frac{h_i x}{k} = 0.73\,Pr^{\frac{1}{3}}Re_x{}^{0.5} \tag{2}$$

for laminar boundary layer and

$$Nu_i = \frac{h_i x}{k} = 0.037\,Pr^{\frac{1}{3}}Re_x{}^{0.8} \tag{3}$$

for turbulent boundary layer.

Impingement angle is estimated by

$$\theta_i = 2.0\left(\frac{W}{R}\right). \tag{4}$$

Film Reynolds number, the ratio of inertia force to viscous force within film, is estimated based on half liquid mass flow rate per meter length of tube. In the thermal developing region, film thermal gradient is developed and major part of heat is utilized for heating of film. The film heat transfer coefficient is estimated based on the heat fluxes at impingement and developing region.

The angle for film development is given by

$$\theta_d = \frac{1}{\pi\alpha R}\left(\frac{3\mu\Gamma^4}{g\rho^5}\right)^{\frac{1}{3}} \tag{5}$$

$$h_{d,(\theta_i-\theta_d)} = \frac{q_{d,(0-\theta_d)}\theta_d - q_{d,(0-\theta i)}\theta_i}{(\theta_d-\theta_i)(T_w-T_s)}. \tag{6}$$

Heat flux is related with film thickness as proposed by Nusselt's theory which is given by

$$\delta_{(\theta)} = \left[\frac{3\mu\Gamma}{g\rho_f(\rho_f-\rho_g)\sin\theta}\right]^{\frac{1}{3}}. \tag{7}$$

Liquid film thickness gradually reduces from the top of the tube and reaches a constant near the middle and increases then as the film flows toward the bottom of the tube. Theoretical and experimental studies with better measurement techniques in the recent past have proved wide variations in predictions. Mustafa and Negeed [16] proposed correlation for film thickness with tube diameter as a parameter influencing the film thickness. Correlation for film thickness developed by Hou et al. [17], based on experimental studies, incorporates the influence of tube diameter and inter-tube spacing.

$$\delta_{(\theta)} = C\left[\frac{3\mu\Gamma}{\rho_f\cdot(\rho_f-\rho_g)g\cdot\sin\theta}\right]^{\frac{1}{3}}\left(\frac{s}{d}\right)^n \tag{8}$$

The constants C and n are depending on the angular position of film. In the fully developed thermal region, the thermal gradient is stabilized and heat is transferred to the film surface for convective evaporation. The following correlations are suggested by Chyu and Bergles [14].