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Heat transfer characteristics of falling film evaporation on horizontal tube arrays

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A B S T R A C T

A falling film heat transfer test facility has been built for the measurement of falling film evaporation in a vacuum of about 1000 Pa. At this condition, only convective evaporation occurred in the liquid film. The Reynolds numbers of falling film over a range from 21.6 to 108.1 were tested on six-tube arrays made of enhanced or smooth tubes. Results show that the tubes with both enhanced outer and inner surfaces give high heat flux. Besides, as the Reynolds number increases, the heat transfer enhancement ratio of falling film evaporation decreases. A semi-analytical correlation is established to predict the heat transfer coefficients of falling film evaporation on smooth tube arrays, considering the contributions of partially dry-out and fully wet regimes, respectively. For enhanced tubes, the heat transfer enhancement ratios to the smooth tubes were also correlated.

1. Introduction

The industries of refrigeration, desalinization and petroleum refining all need evaporators with high efficiency to improve the performance of the systems. Because of the high heat transfer coefficients and low refrigerant charges in falling film evaporators, they appear as superior to the conventional evaporators [1,2]. Besides, falling-film type evaporators have lower pressure drops that they can be utilized in the poor conditions of small temperature difference and low heat flux. However, the falling film evaporation process involves two phase flow and heat transfer, and both evaporation and boiling occur in the liquid film. As a result, the falling film evaporation phenomena are complicated. Presently, the falling film evaporator industries still face the challenges of optimizing the design methods and operation strategies. Thus, many researchers are motivated to seek a better understanding of falling film evaporation.

Till now, different and contradictory heat transfer behaviors are derived from the experimental studies, as shown in Ribatski and Jacobi's review [2]. For smooth tubes at convection dominated conditions, as the Reynolds number of the falling film increases, the heat transfer coefficient usually decreases first, but increases after reaching a minimum value [3,4]; or the heat transfer coefficient increases with the film Reynolds numbers [5–7]; or the heat transfer coefficients increases to a peak value, and then decreases with increasing Reynolds number [8]. These different results might be attributed to different falling film modes. Under high heat flux condition, boiling may occur in the liquid film. When the boiling occurs, film Reynolds number will hardly have any effects on the evaporation heat transfer coefficients [3]. On the other hand, the heat flux will affect the heat transfer [2], as boiling is enhanced by increasing heat flux. However, the falling film evaporators always work under conditions of low heat flux, so nucleate sites might not always exist in the film.

Falling film evaporation on tube arrays is more complicated than on a single tube, because of the intertube evaporation and the turbulence caused by liquid falling from one tube onto the next [9]. Lorenz and Yung [9] identified the critical Reynolds number of 300, below which the falling film evaporation coefficients on tube arrays are less than those on a single tube. Additionally, when the Reynolds number is small, the lower tubes of the array will suffer more from partial dryout than those on higher layers. The liquid flow rate may become less at the lower tubes because of evaporation. Since the dry areas transfer the heat by natural convection only, a sudden drop of heat transfer coefficients is observed both on smooth tube arrays [4,7] and enhanced ones [10].

As to the predictions of heat transfer coefficients of falling film evaporation, many empirical correlations are derived, as shown in the review by Ribatski and Jacobi [2]. The dimensionless numbers, such as Reynolds number, Prandtl number and Archimedes number, as well as the pressure and temperature are used in these empirical correlations. On the other hand, there are also some analytically based models, which often divide the film on the tubes into different

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List of symbols

a	thermal diffusivity ($m^2\ s^{-1}$)
A	area (m^2)
Bo	Bond number $(\rho gHs)/\sigma$
c_p	specific heat ($J\ kg^{-1}\ K^{-1}$)
D	tube diameter (m)
g	gravitational acceleration ($m\ s^{-2}$)
H	fin height (m)
h	heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)
L	actual effective heated length of the test tubes (m)
L_d	the length of developing region (m)
L_h	wetted length of the enhanced tubes (m)
$M_{heating}$	mass flow rate of the heating water ($kg\ s^{-1}$)
Nu	Nusselt number, Nu_c is the Nusselt number for the fully develped convection region $(h/\lambda)(\nu^2/g)^{1/3}$
q	heat flux ($W\ m^{-2}$)
r	latent heat ($kJ\ kg^{-1}$)
R	radius (m)
Re	Reynolds number $4\Gamma/\mu$
s	pitch length or fins spacing (m)
T	temperature (°C)
U_o	overall heat transfer coefficient ($W\ m^{-2}\ K^{-1}$)

Greek symbols

Γ	liquid mass flow rate per unit length of tube (each side) ($kg\ m^{-1}\ s^{-1}$)
λ	thermal conductivity ($W\ m^{-1}\ K^{-1}$)
μ	dynamic viscosity ($kg\ m^{-1}\ s^{-1}$)
ν	kinematics viscosity ($m^2\ s^{-1}$)
ρ_V	density of the vapor ($kg\ m^{-3}$)
ρ_L	density of the liquid ($kg\ m^{-3}$)
σ	surface tension ($kg\ s^{-2}$)

Subscripts

cal	calculated values
e	enhanced tubes
exp	experimental values
i	inside of tubes
nuc	nucleation
o	outside of tubes
s	smooth tubes
sat	saturation

flow regions and then apply different physical models to them. Lorenz and Yung [11] treated the tube by unwrapping them to a vertical plate, and modeled the overall heat transfer coefficients by superposition of nucleate boiling and convective evaporation. Besides, they also divided the falling film flow into developing and developed film regions, and calculated the heat transfer coefficients for the two regions, respectively [11]. In addition Roques and Thome [10] defined a coefficient by comparing the heat transfer coefficients of falling film evaporation with nucleate boiling and those of pool boiling at the same heat flux. Both the models of Lorenz and Yung [11] and Roques and Thome [10] have taken considerations of pool boiling, because boiling is often observed in falling film evaporation especially on enhanced tubes. However, plugging pool boiling correlations into falling film evaporation models could make a larger error and become less applicable.

Additionally, in most of the experiments carried out under atmospheric pressure, boiling and evaporation occur in the falling film at the same time. The combined boiling and evaporation effects make the study of falling film evaporation mechanism difficult. It is desirable to explore the individual effects, such as the convective evaporation heat transfer only. Besides, many literatures indicated that the heat transfer enhancement of enhanced tubes is mainly due to nucleate boiling [3,10], because the enhanced surfaces often form the nucleation sites and enhance boiling. However, few investigations on enhanced tubes have been done under the condition of film evaporation only. Therefore, the study of film evaporation, without boiling, on enhanced tube arrays will provide the important basic understanding to this complex process of combined boiling and evaporation.

In order to provide a better basic understanding and to guide the practical applications, experiments and predictions of heat transfer enhancement of enhanced tubes on the falling film evaporation are performed in the present work. We investigated the average heat transfer coefficients of water falling film on five types of enhanced tubes, with the smooth tubes for reference. Tests were conducted in a vacuum that few nucleation sites would exist on the tube surfaces. The effects of falling film Reynolds number from 21.6 to 108.1, and other parameters on falling film heat transfer are investigated. In addition, correlations considering the film's physical properties and the tubes' geometric dimensions are also established to predict the heat transfer coefficients of falling film on smooth tube arrays and the enhancement of the enhanced tubes arrays.

2. Experimental method

The schematic of the experimental setup is shown in Fig. 1(a). Since the experimental method and data reduction have been described in detail in a previous paper [8], only a brief description is given here. The working fluid of the liquid film is water, which evaporated at the absolute pressure of 1000 Pa. The liquid goes through a distributor, which is a perforated integral-fin tube with 0.8-mm-diameter holes on the top. Before entering the distributor, the liquid has a subcooling of about 0.5 K to prevent evaporation at the entrance of the vessel. But the subcooling was ignored in our data reduction process. The pitch of the tube arrays and the distance between the distributor and the top tube are both 25.4 mm. Besides, the heating water circuit provides the heat for film evaporation, while the condensing circuit helps to regulate and maintain the pressure in the test vessel by condensing the vapor.

Five types of enhanced tubes and the smooth tubes were tested in the experiments. The nominal outer diameters of the tested tubes are all 15.88 mm and their actual effective heated lengths are all 700 mm. Tubes A, B and C are the same type of tubes, but at the outer surfaces they have 40, 26 and 19 fins per inch, i.e. about 1575, 1024 and 748 fins per meter, respectively. However, tube A has enhanced inner surfaces while both Tube B and C having smooth inner surfaces. The fins of Tubes A, B and C are column-like protuberances on the outside surfaces, as shown in Fig. 1(b). Tube D has helically fins outside and internal ridges inside the tubes, as in Fig. 1(c). Tube E is made by corrugating copper alloy tubes, so both the outer and inner surfaces are corrugated, as shown in Fig. 1(d). The tubes' dimensions are listed in Table 1. The wetted length for the film flow outside the tubes can be calculated as follows:

$$L_h = L + 2H\frac{L}{s}. \quad (1)$$

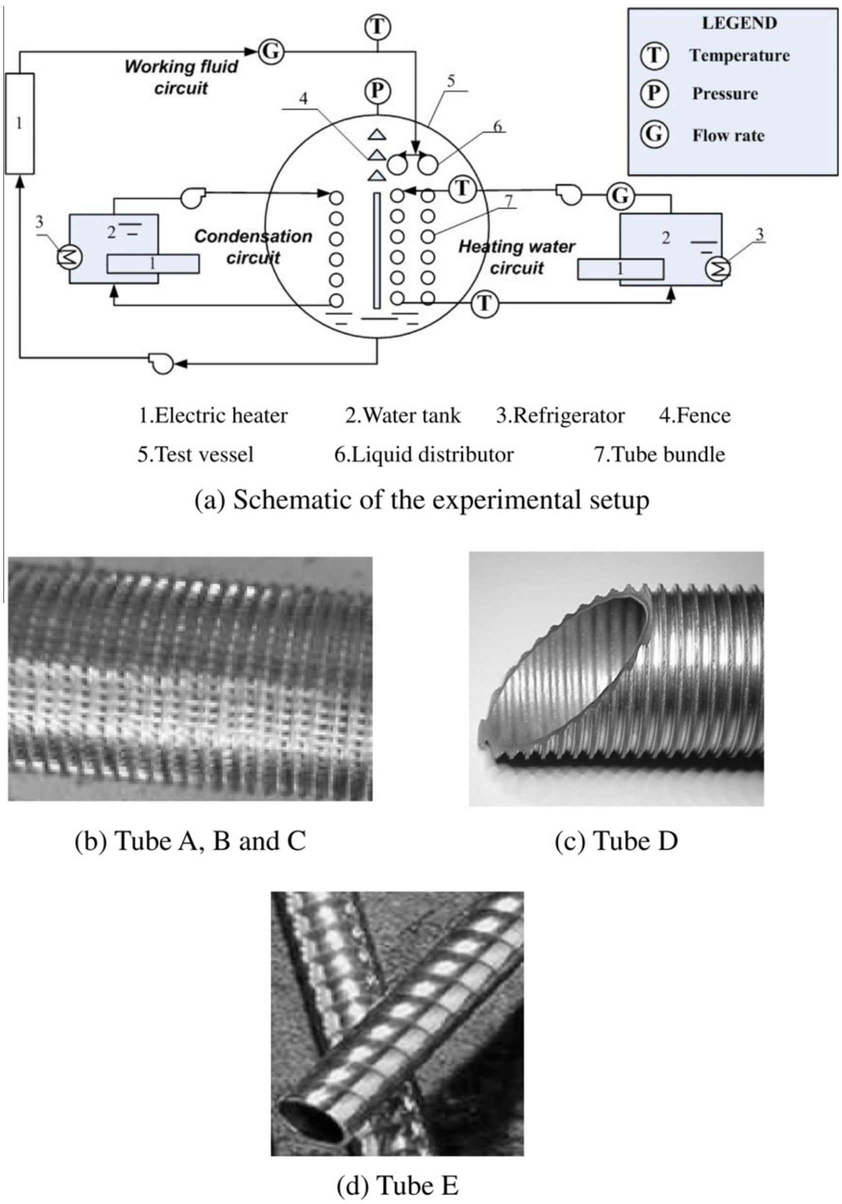

Fig. 1. Experimental setup and the enhanced tubes, (a) schematic of the experimental setup, (b) Tube A, B and C, (c) Tube D and (d) Tube E.

Table 1
Dimensions of the tubes in the experiments.

Tube	Type	A	B	C	D	E	Smooth tubes
L	(mm)	700	700	700	700	700	700
Nominal D_o	(mm)	15.88	15.88	15.88	15.88	15.88	15.88
nominal D_i	(mm)	13.89	13.6	13.6	14.02	14.45	14.45
H	(mm)	0.345	0.355	0.355	1.092	0.356	/
s	(mm)	0.635	0.9769	1.337	2.309	7.19	/
L_h	(mm)	1461	1209	1072	1362	769	/
STC_i		0.032	0.027	0.027	0.036	0.104	0.027
Fins per inch		40	26	19	11	Corrugated	Smooth

Calculations have been done to identify the existence of the nucleation sites. The relation between the nucleation superheat and the nucleation radius follows [12],

$$\Delta T_{nuc} = \frac{2\sigma}{R_{nuc}(dp/dT)_{sat}} = \frac{2\sigma T_{sat}}{R_{nuc} r}\left(\frac{1}{\rho_v} - \frac{1}{\rho_L}\right). \quad (2)$$

The water film evaporated at a vacuum of 1000 Pa in our tests, and the highest temperature differences were less than 10 K. Thus, from Eq. (2), the nucleation radius for superheat of 10 K should be 0.22 mm. In order to generate bubbles, water film should be thicker than 0.44 mm. In falling film evaporation, we did not observe any appropriable large nucleation sites on the enhanced surface, so boiling will be unlikely to occur. Besides, the absolute working pressure of our tests was about 1000 Pa; and in this vacuum condition, homogeneous boiling would not be generated at low superheat. In fact, from the glass window on the test vessel, we could not observe any boiling phenomenon during the experiments. Thus, the falling film in our experiments evaporated without boiling, and only the convective evaporation is studied in this paper.

The thermal resistance method was used to calculate the heat transfer coefficients of falling film evaporation. As in our experiments the falling film flows outside the six-tube arrays, the average heat transfer coefficients of the entire arrays were calculated and analyzed. The inlet and outlet temperature of the heating water were measured by RTD Pt100, thus the overall heat flux of the system can be calculated. The heat flux was evaluated based on the outer surface areas of the whole tube array using the nominal outer diameter of the tubes. Then, the overall heat transfer coefficients of the tube arrays are calculated as follows:

$$U_o = \frac{q}{\Delta T_{LMTD}}, \tag{3}$$

where ΔT_{LMTD} is the LMTD between the heating water and the saturated falling film.

The inner heat transfer coefficients of heating water flowing inside the tubes was calculated by the Sieder and Tate's relation [13]. The STC_i coefficients for the tubes are derived by Wilson-plot method, and are listed in Table 1.

The wall resistance is ignored, because for the enhanced tubes in most of the conditions, $D_o \ln(D_o/D_i)/2\lambda$ is much smaller than $1/h_o$. Thus, the average heat transfer coefficient of the falling film evaporation in the tube arrays can be calculated as follows:

$$h_o = 1 \Big/ \left(\frac{1}{U_o} - \frac{1}{h_i} \cdot \frac{D_o}{D_i} \right). \tag{4}$$

The uncertainty analysis was also described in details in the previous paper [8]. Table 2 shows the accuracy of the instruments. Only the uncertainties of temperature, pressure and flow rate were considered, because the measurements of other parameters, such as length, are of minor uncertainties. A propagation of uncertainty analysis was performed as follows:

$$\frac{\Delta h_o}{h_o} = h_o \left[\left(\frac{\Delta U_o}{U_o} \cdot \frac{\Delta U_o}{U_o} \right)^2 + \left(\frac{D_o}{D_i} \cdot \frac{\Delta h_i}{h_i} \cdot \frac{1}{h_i} \right)^2 \right]^{1/2}. \tag{5}$$

The average relative errors for h_o are 9.73%, 12.4%, 10.7%, 8.51%, 4.83%, 8.85% for Tubes A, B, C, D, E and smooth tube, respectively.

3. Heat transfer characteristics and discussion

3.1. Heat transfer characteristics

When the Reynolds number is less than 200, falling film flows at laminar region [14]. In our experiments, the evaporator worked at the maximum Reynolds numbers at about 108.1. Thus, the falling film in our experiments was at laminar flow. In the laminar region, the thick boundary layer will be a barrier for heat transfer; and in the fully developed region, the boundary layer is the thickest. So the developing region has a better heat transfer capability than the developed region. Besides, at the laminar flow, the thickness of the film will also become a barrier to heat transfer from the tube wall to the film surface. Thus, the boundary layer and the film thickness are two major resistances for the heat transfer of laminar falling film evaporation. Several film flow modes are defined by Hu and Jacobi [15]. In the general case of our experiments on enhanced tubes, the droplet-jet mode was observed when film Reynolds number exceeded 43.2, and droplet mode was observed in film Reynolds smaller than 32.4. Additionally, we compared the average values of the Nusselt numbers of falling film evaporation with the correlation of Hu and Jacobi [16] in a previous paper [8], as shown in Fig. 2. Our data fits the jet mode correlation from Hu and Jacobi better than the droplet mode correlation. But their correlations do not show the peak values around the transition point.

Film Reynolds number can affect the length of developing region and the film thickness. As the Reynolds number increases, the developing region becomes longer. However, the film thickness will grow with Reynolds number. In Fig. 3, we can see the relationship between Reynolds number and heat transfer coefficient. As the Reynolds number increases, the heat transfer coefficients will increase to a peak value and then gradually decrease. This trend is already discussed before by Li et al. [8] and not to be mentioned in details here. The same trend was observed for the heat flux. The highest heat flux is defined as $q_{plateau}$ by Ribatski and Thome [7]. But in their results, the heat transfer coefficients will reach the highest value at Reynolds number about 800 and remain at that value when Re is larger than 800; before that, the heat transfer coefficients increase linearly with Reynolds number [7]. In our research, the heat transfer coefficients become lower after the peak at Reynolds number of about 60; the same with the heat flux. The plateau of heat flux can be observed in Fig. 3 as well. Tube E, which has both corrugated inner and outer tube surfaces, has $q_{plateau}$ as high as 24.5 kW m^{-2}, and the values of the $q_{plateau}$ of Tube B and C are a little lower. As the heat flux is calculated based on the nominal outer surfaces, which are the same for the tubes, higher heat flux means more liquid will be evaporated. So the tubes with both enhanced inner and outer surface are appropriate for falling film evaporators.

From Fig. 3, we see that the heat transfer coefficients of falling film evaporation on enhanced tubes are higher than those on

Table 2
Experimental uncertainties.

Position (system)	Items	Instruments' accuracy
Tank	Temperature Pressure	±0.1 K ±0.15%
Refrigerant circuit	Temperature Flow rate	±0.1 K ±6.67 × 10^{-4} kg/s
Heating water circuit	Flow rate Temperature	±3.33 × 10^{-3} kg/s ±0.1 K
Condensing water circuit	Flow rate Temperature	±3.33 × 10^{-3} kg/s ±0.2 K

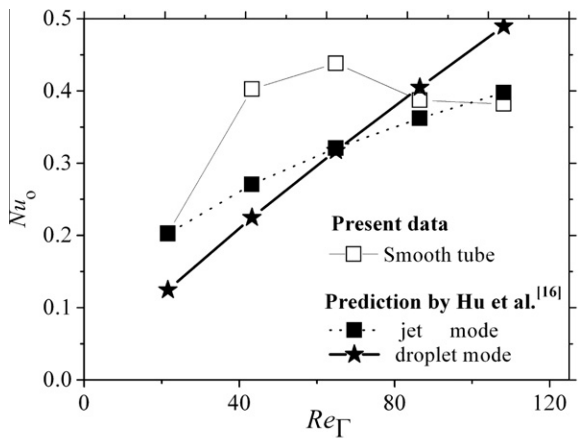

Fig. 2. Comparisons between the present data and the correlations by Hu and Jacobi [16] in a previous paper [8].

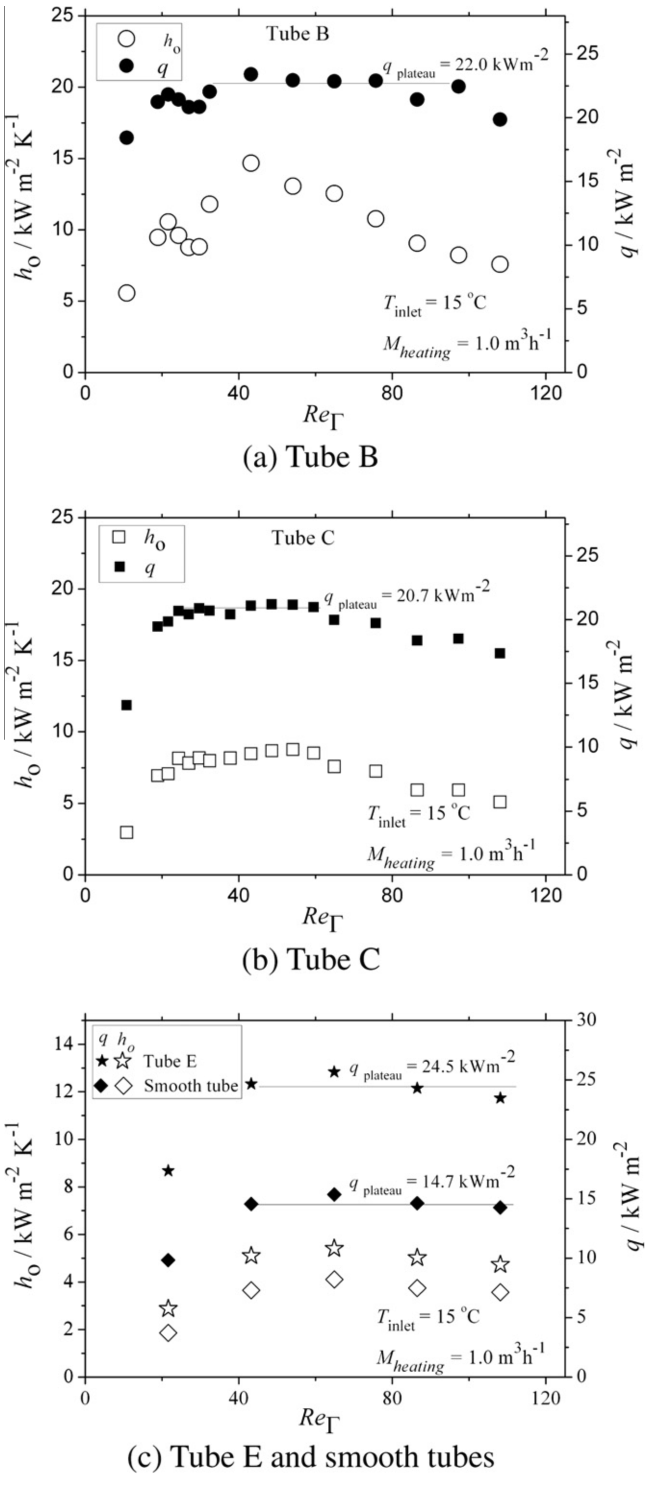

Fig. 3. Effect of Reynolds number on heat transfer coefficients and q_{plateau} when T_{inlet} = 15 °C.

smooth tubes. The fins or protuberances on the enhanced tube will make the boundary layer thinner, and disturb the film flow. So the heat transfer coefficients will be larger on these enhanced surfaces. We compared the heat transfer enhancement capabilities more carefully in Fig. 4 for enhanced tubes versus smooth ones, and define $h_{o,e}/h_{o,s}$ as the heat transfer enhancement ratio. The enhancement ratio decreases with Reynolds number for most of the enhanced tubes. As shown by Li et al. [8], the enhanced tubes have smaller transition Reynolds numbers from fully wet to partially dry than smooth tubes. In the present experiments, the enhanced tube arrays are still wet when the Reynolds number is low. So the enhancement ratio is large in the transition region. When the tubes are fully wet, the liquid film could cover the whole fins on the tubes. From the observation through the glass window, when the Reynolds number exceeds 64.9, the liquid film was thick enough to cover the fins around the impinging point of the liquid droplet and jet. As the fins or protuberances are immersed in the liquid film, they could not enhance evaporative heat transfer very much. So the enhancement ratio decreases gradually in the fully wet regime. In this way, we know that the enhanced tubes will work the best when the liquid flow rate is low.

Besides, the enhancement ratio of Tube B drops most quickly as Reynolds number increases. This means that in this case the liquid film will induce a great problem if it is thick enough to cover the whole fin. As the fin is covered by the liquid film, the fins could not help the film distribute or disturb the film flow to enhance the heat transfer. Though Tube A has more fins than Tube B, the heat transfer enhancement ratios of the two tubes are almost the same. Thus, it means that there is a fin number when the tubes could have the maximum heat transfer performance between 26 and 40 fpi.

3.2. Effects of temperature difference

Generally, temperature differences may affect the falling film evaporation when boiling occurs, because ΔT will enhance bubbly boiling and make the heat flux increase. But in convective evaporation, situations will be different. In our experiments, the film evaporation occurred in a tubular heat exchanger, so the logarithmic mean temperature is used here to study the effects of temperature differences on falling film evaporation on tube arrays. As shown in Fig. 5, we observed that the heat flux increases almost linearly with temperature differences. This means that the heat transfer coefficients will not change with the increasing ΔT. Besides, the heat transfer coefficients of falling film evaporation remain the same under most of the fully wet conditions when the Reynolds numbers are high. But in the case when Reynolds number is 21.6, higher heat flux causes more liquid evaporate; many dry patches appear on the tubes. Severe dryout will make the heat transfer worse, so heat transfer coefficients decreases with increasing heat flux, and when the Reynolds number is 21.6, the film flow will not be steady in the mode of droplet. This makes the data fluctuate more than other cases.

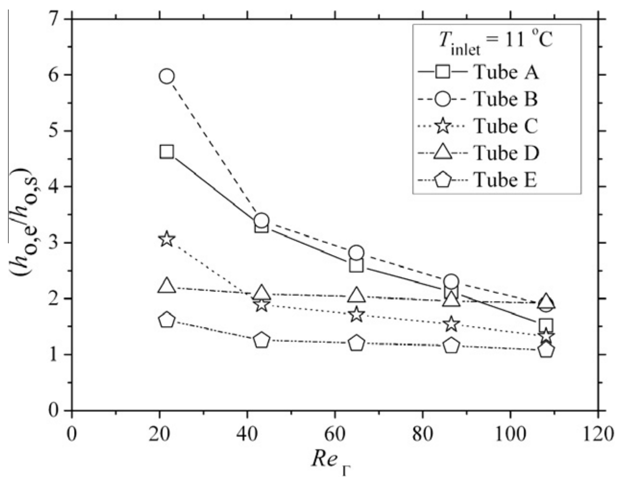

Fig. 4. Heat transfer enhancement ratios over Reynolds numbers for different types of tubes.

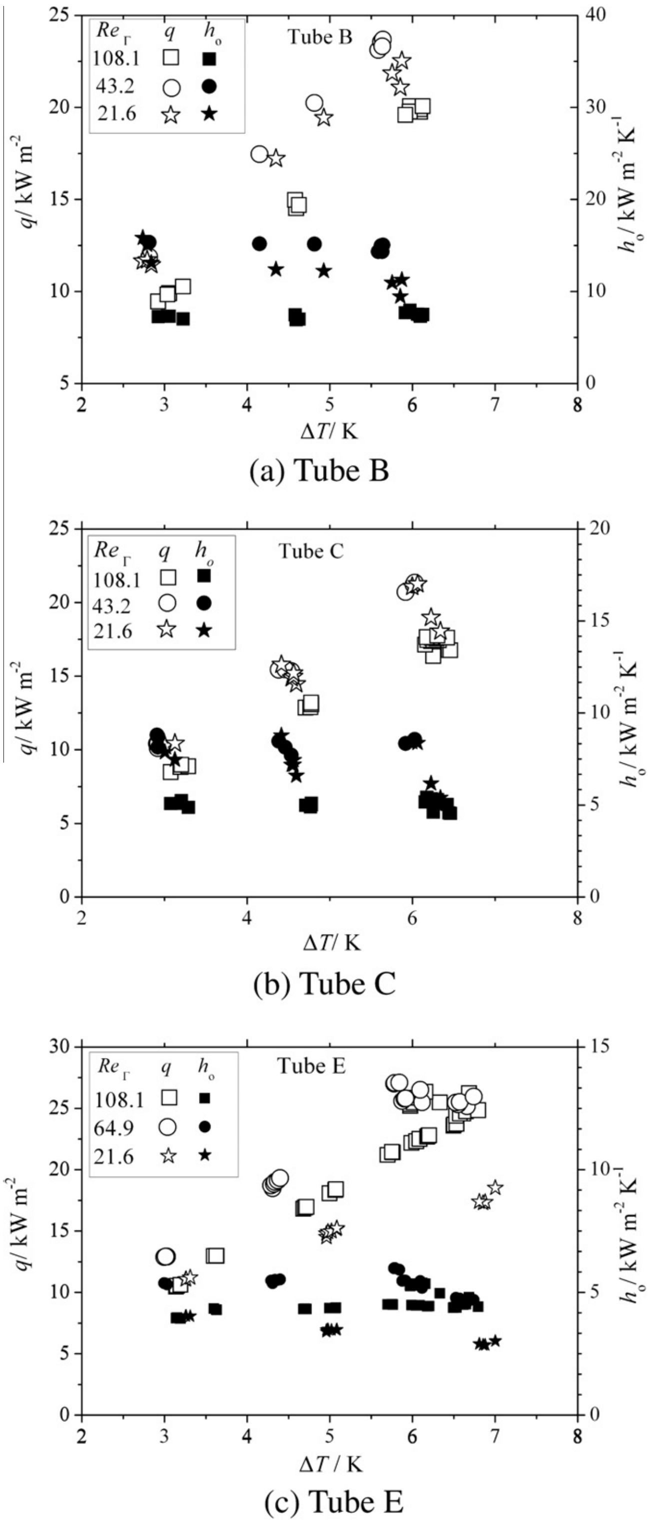

Fig. 5. The effects of temperature differences on heat flux and heat transfer coefficients of falling film evaporation on enhanced tubes, (a) Tube B, (b) Tube C, and (c) Tube E.

Furthermore, we notice that there is a turning point of heat flux in Tube E as shown in Fig. 5(c). This means that at large temperature difference, dryout occurred on the tubes, the heat flux transferred through the film could not increase. As much liquid evaporated by the high heat flux, heat transfer coefficients decreased. But since falling film evaporators often work at small temperature differences, the dryout caused by high ΔT may not occur easily. Additionally, as the turning points did not show up in Fig. 5(a) and (b) when ΔT is about 6 K, it means that Tube B or C might have larger temperature differences for dryout to occur. Because there are column-like protuberances on Tubes B and C, the protuberances indeed help the liquid to distribute along the tubes, and dryout may not occur easily on the tubes with the similar enhanced surfaces.

Thus, we know that the heat transfer coefficients can still be kept high at low heat flux and low film Reynolds number on modified finned tubes B or C. This is quite important for falling film evaporator design, because this type of evaporators will work well at poor conditions like low heat flux and small temperature differences. Thus, the enhancement of convective evaporation will be of great importance.

4. Analytical models

4.1. Model for smooth tube array

Typically, falling film evaporation model will be analyzed by dividing into different flow regions, i.e. free fall, jet impingement, thermal developing and fully developed regions [2]. The free fall and jet impingement regions cover only a very small portion of the tube surface, and could be ignored. So Lorenz and Yung [11] developed a model for falling film evaporation on horizontal smooth tubes, combining both convective evaporation and nucleate boiling effects. The heat transfer coefficient of falling film evaporation is calculated as follows [11]:

$$h_o = h_b + h_d \frac{L_d}{L} + h_c\left(1 - \frac{L_d}{L}\right), \tag{6}$$

where h_b, h_d and h_c are the heat transfer coefficients of nucleate boiling, of film evaporation in the developing and of developed region, respectively; L_d is the length of developing region and L is the "unwrapped" length of the tube, namely $L = \pi D_o/2$.

In the developing region, Lorenz and Yung [11] calculated the average heat transfer coefficient from energy balance, giving

$$h_d = \frac{3}{8} c_p \frac{\Gamma}{L_d}. \tag{7}$$

Based on the uniform film thickness given by Nusselt theory and ignoring the bubbles in the film, the length of developing region is estimated as [11]

$$L_d = \frac{\Gamma^{4/3}}{4\pi\rho a}\sqrt[3]{\frac{3\mu}{g\rho^2}}. \tag{8}$$

Additionally, the other two heat transfer coefficients in Eq. (6) are calculated empirically. The heat transfer coefficients of nucleate boiling, h_b, can be calculated by the empirical correlations given by the tube manufacturers; and the film evaporation in the developed region, h_c, is calculated by the experimental correlation on the tubes. So the overall heat transfer coefficients could be calculated by adding these three components in Eq. (6).

For our experiments of falling film evaporation in a vacuum, the nucleate boiling term can be eliminated. In addition, as the falling film evaporation goes through fully wet to partially dry on tubes, the heat transfer coefficients of falling film evaporation should be modeled in these two regimes separately. Therefore, we make the following assumptions for the new model:

1. The impingement effects and the liquid loss from splashing when the liquid film falls from upper tube to lower tube are all ignored.
2. The heat loss from the end of the tubes is ignored.

3. Dry patches on the tube surfaces will affect the heat transfer coefficients of the developed regions only that the changes of flow modes will be ignored.
4. The effects of dry patches on heat transfer coefficients are mainly due to the decrease of Reynolds numbers.

Regard the assumption 3, when the liquid film flow downwards, dry patches may appear on the lower tubes first. Because of the evaporation, the dry patches on the lower part of the tube will damage the developed regions. Decreasing film flow rate will have two effects on falling film evaporation. On one hand, the thinner film caused by the decreasing flow rate will enhance heat transfer because the tip of the fins are over the liquid film. On the other hand, the decreasing flow rate will cause dry patches on the tube surface, so the heat transfer will be deteriorated. Because the developing regions are located on the top of the tubes, the effects of partial dry out on the developing region will not be as strong as those on the developed regions. Thus, when dry patches appear on the tube, the heat transfer coefficients on the developed regions will become lower. In this sense, when we correlate the heat transfer coefficients, only the heat transfer coefficients of the developed regions will be affected in the partially dry regime of tube.

Accordingly, we could model the falling film evaporation heat transfer for tubes partially dry and fully wet regimes separately. We can see from the discussion above that the Reynolds number will affect the heat transfer capability greatly. This is because on one hand, the increasing of Reynolds numbers will make the liquid film thicker. On the other hand, when the inlet temperature of heating water varies, the heat transfer coefficients of falling film evaporation will not show much change. Additionally, the surface tension does not affect the heat transfer of falling film evaporation on smooth tubes significantly. So we will only use the Reynolds number to correlate the heat transfer coefficients in the developed region. Thus, using the regression for multiple variables, we get the following correlation for $Nu_{c,wet,}$

$$Nu_{c,wet} = 182.1Re^{-1.56}. \tag{9}$$

When the tube is partially dry, the heat transfer coefficient of falling film evaporation is governed by the film Reynolds number. When the Reynolds number is low, the ratio of dry area to the whole tube surface area will be large. So the heat transfer coefficients will be lower. Thus, another dimensionless Reynolds number, Re/Re_{tran}, is used to correlate the data. The Re_{tran} is the transition Reynolds number, where the heat transfer coefficient reaches the peak value. In our experiments of falling film evaporation, the transition Reynolds number of smooth tubes is about 54.1. Then, the correlation becomes,

$$Nu_{c,dry} = Nu_{c,wet}\left(\frac{Re}{Re_{tran}}\right)^{2.67}. \tag{10}$$

The heat transfer coefficients of developed falling film evaporation can be derived from the Nusselt number values in Eqs. (9) and (10). The heat transfer coefficients in the developing region can be calculated by Eq. (7). Thus, the heat transfer coefficients of falling film evaporation can be calculated as Eq. (6) with elimination of the boiling component h_b. Fig. 6 shows comparison between the experimental and analytical results of the overall heat transfer coefficients of the film evaporation on smooth tubes. It indicates that the errors are larger when the Reynolds numbers are below the transition value. When dry patches appears on the tube surfaces, as a result of the change of flow modes and the impacting points, the heat transfer coefficients varies. Also, the errors would be larger in the dry out regime.

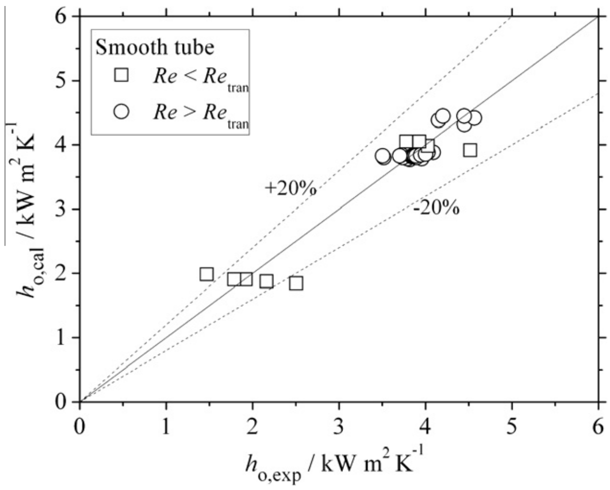

Fig. 6. Comparisons between the calculated and the experimental heat transfer coefficients of falling film evaporation on smooth tube arrays.

4.2. Model for enhancement ratio

From Fig. 3, we see that the enhancement ratios of different enhanced tubes have different decreasing rates with increasing Reynolds number. Thus, we use the Bond number, which relates the fin geometry and the effect of surface tension, to correlate the enhancement ratio. Consider the liquid film flows outside the finned tubes, the Bond number could be modified as,

$$Bo = \frac{\rho g D_h^2}{\sigma} = \frac{\rho g H s}{\sigma}. \tag{11}$$

This is because when the liquid film flows through the fins, the cross section area between two fins is calculated by multiplying the fin height and fin pitch length, i.e. the product of H and s. In another view, the present Bond number for the finned tubes describes the ratio of gravity over surface tension.

When the liquid flows through the fins, surface tension may have two important effects on the heat transfer coefficients [17]. On the one hand, the surface tension will help the liquid disperse longitudinally, so the film will be thinner and the heat transfer coefficients of falling film evaporation will be higher. On the other hand, if the fin is very high, the surface tension will help the reten-

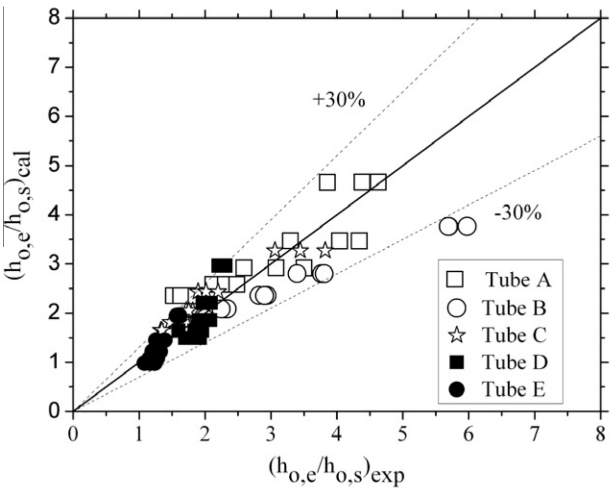

Fig. 7. Comparisons between the calculated and the experimental enhancement ratios of falling film evaporation on enhanced tube arrays.

tion of the liquid at the bottom of the fins, so the film thickness will be larger near the bottom of the fin. The heat transfer could be deteriorated by the thick liquid film.

So we correlate the enhancement ratio in this form,

$$h_{o,e}/h_{o,s} = 5.635Bo^{-0.164}Re^{-0.426}(L_h/L)^{0.732}. \quad (12)$$

As shown in Fig. 7, 90.1% of the data fall within the error of ±30%. Thus, the correlation could predict the heat transfer enhancement ratio satisfactorily.

5. Conclusions

An experimental study has been carried out to investigate the characteristics of evaporative heat transfer of falling film on horizontal tube arrays on smooth and enhances tubes. The following conclusions are derived:

1. The tubes that have both corrugated outer and inner surfaces have higher heat fluxes values than those enhanced on the outer surfaces only.
2. As the falling film Reynolds number increase, the heat transfer enhancement ratio of falling film evaporation decreases exponentially. The enhanced tubes, which have column-like protuberances, have better heat transfer performances at low Reynolds numbers than the helical finned or corrugated tubes.
3. Increasing the temperature difference will make the heat flux higher, and when the Reynolds numbers are high, heat flux variation has little effects on the heat transfer coefficient of falling film evaporation on enhanced tube arrays. But when the film Reynolds numbers are low, high heat flux causes large dry areas that the heat transfer coefficients decreases.
4. A semi-analytical model was established for the heat transfer coefficients of falling film evaporation on smooth tube array when the boiling is suppressed. The model considers the heat transfer characteristics of partially dryout and fully wet regimes separately, and the heat transfer coefficients of the enhanced tube arrays are correlated using the enhancement ratios.

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References

[1] J.R. Thome, Falling film evaporation: state-of-the-art review of recent work, J. Enhanc. Heat Transfer 6 (2–4) (1999) 263–277.
[2] G. Ribatski, A.M. Jacobi, Falling-film evaporation on horizontal tubes – a critical review, Int. J. Refrig. 28 (5) (2005) 635–653.
[3] Z.H. Liu, J. Yi, Falling film evaporation heat transfer of water/salt mixtures from roll-worked enhanced tubes and tube bundle, Appl. Therm. Eng. 22 (1) (2002) 83–95.
[4] Y. Fujita, M. Tsutsui, Experimental investigation of falling film evaporation on horizontal tubes, Heat Transfer Jpn. 27 (1998) 609–618.
[5] L.P. Yang, S.Q. Shen, Experimental study of falling film evaporation heat transfer outside horizontal tubes, in: Conference on Desalination and the Environment, Halkidiki, GREECE, 2007.
[6] W.H. Parken et al., Heat-transfer through falling film evaporation and boiling on horizontal tubes, J. Heat Transfer – Trans. ASME 112 (3) (1990) 744–750.
[7] G. Ribatski, J.R. Thome, Experimental study on the onset of local dryout in an evaporating falling film on horizontal plain tubes, Exp. Thermal Fluid Sci. 31 (6) (2007) 483–493.
[8] W. Li, X.Y. Wu, Z. Luo. Falling film evaporation of water on horizontal configured tube bundles, in: The 14th International Heat Transfer Conference, Washington, DC, 2010.
[9] J.J. Lorenz, D. Yung, Film breakdown and bundle-depth effects in horizontal-tube, falling-film evaporators, J. Heat Transfer – Trans. ASME 104 (3) (1982) 569–571.
[10] J.F. Roques, J.R. Thome, Falling films on arrays of horizontal tubes with R-134a. Part II: Flow visualization, onset of dryout, and heat transfer predictions, Heat Transfer Eng. 28 (5) (2007) 415–434.
[11] J.J. Lorenz, D. Yung, Note on combined boiling and evaporation of liquid-films on horizontal tubes, J. Heat Transfer – Trans. ASME 101 (1) (1979) 178–180.
[12] A. Bejan, A.D. Kraus, Heat Transfer Handbook, John Wiley & Sons, Inc., Hoboken, New Jersey, 2003. pp. 640–643.
[13] E.N. Sieder, G.E. Tate, Heat transfer and pressure drop of liquids in tubes, Ind. Eng. Chem. Res. 28 (1936) 1429–1435.
[14] J.J. Lorenz, D.T. Yung, Combined boiling and evaporation of liquid films on horizontal tube, in: Proceedings of the Fifth Ocean Thermal Energy Conversion (OTEC), Miami Beach, 1978.
[15] X. Hu, A.M. Jacobi, The intertube falling film. 1. Flow characteristics, mode transitions, and hysteresis, J. Heat Transfer – Trans. ASME 118 (3) (1996) 616–625.
[16] X. Hu, A.M. Jacobi, The intertube falling film. 2. Mode effects on sensible heat transfer to a falling liquid film, J. Heat Transfer – Trans. ASME 118 (3) (1996) 626–633.
[17] R.K. Shah, S.Q. Zhou, K.A. Tagavi, The role of surface tension in film condensation in extended surface passages, J. Enhanc. Heat Transfer 6 (2–4) (1999) 179–216.