

Impact of the geometric parameters on the thermal performance of a large-scale falling film evaporator for desalination

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ABSTRACT

Based on a validated distributed parameter model, the thermal performance of a large-scale falling film evaporator was simulated under different geometric parameters including the tube length and the row–column ratio of the tube bundle. On considering that the falling film evaporators applied in the multi-effect evaporation are operated under small temperature difference thus sensitive to the temperature variation inside the evaporator, the variation of the local temperature difference is presented in detail. The distributions of the local heat transfer coefficient and heat flux were also exhibited under different geometric parameters. Results show that both the tube length and tube bundle row–column ratio have significant impact on the distribution of thermodynamic parameters in the falling film evaporator. The maximum value of the average heat flux of the evaporator is obtained when the length of the tube is 7 m. The maximum heat flux of the evaporator is obtained when the tube bundle row–column ratio is 2.5.

Keywords: Falling film evaporator; Horizontal tube bundle; Distributed parameter model; Geometric parameters; Thermal performance

1. Introduction

Water shortage problems caused by the increasing of world population and rapid development of economy have heightened the need for technologies to obtain fresh water from seawater with high efficiency. Among various desalination technologies, the multi-effect evaporation (MEE) system has drawn extensive attention due to its advantages of enabling the use of low-grade energy, high heat transfer coefficient as well as the high quality of produced freshwater, minimum pressure drop, low investment, etc. The horizontal-tube falling film evaporator, featuring the advantages of high heat transfer coefficient, maximum use of available temperature difference, reliable liquid distribution, positive venting, etc., has been widely utilized in the MEE system [1].

The distribution of the heat transfer coefficient along the tube circumferential or axial directions was studied for a single horizontal tube in a falling film evaporator. Along the tube circumferential direction, the division of heat transfer regions has been the main subject of study with some of the earliest contribution coming from Chyu and Bergles [2] who analytically defined three regions including the jet impingement region, the thermal developing region, and the fully developed region. Recently, Mu [3] divided the liquid film outside the tube into three regions according to the variation of the heat transfer coefficient: the impinging zone, the thermal diffusion transfer zone and the wake detachment zone. For a single heat transfer tube, the heat transfer coefficient was found to have the highest value in the impinging zone on top of the tube [3]. As the liquid flows downward along the

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tube surface, the heat transfer coefficient exhibits a decreasing trend because of the weaker liquid film turbulence until the liquid approaches the bottom of the tube where the collision of liquid from both sides of the tubes contributes to a small increase of the local heat transfer coefficient; Along the tube axial direction, the distribution of the local heat transfer coefficient is mainly determined by the in-tube condensation process. Many scholars came to the same conclusion through analytical or experimental work [4,5] that the local condensation heat transfer inside a horizontal tube decreases along the axial direction. However, Shen et al. [6] found that the steady condensation film is hardly observed near the steam inlet of the inner tube which is responsible for the lower heat transfer coefficient near this area. For a horizontal tube bundle, the heat transfer coefficient shows non-uniform distribution along different directions. According to the analytical work of Kocamustafaogullari and Chen [7], the local heat transfer coefficient decreases with the increment of the tube row number until the thermal developed region is reached. Lorenz et al. [8], Moeykens et al. [9] and Yang et al. [10] observed the decreasing of the local heat transfer coefficient with the increase of the tube row number and attribute this phenomenon to the non-uniform distribution of the liquid films. Based on the distributed parameter model (DPM), Shen et al. [6] and Hou et al. [11] obtained the distribution of heat transfer coefficient in a large-scale falling film evaporator. It was concluded that the local heat transfer coefficient decreases with the increment of the tube row number mainly due to the weak in the seawater film turbulence outside the tubes. On consideration of the inter-tube vapor flow, the distribution of the heat transfer coefficient along the tube column direction of a falling film tube bundle were recently investigated by Gong et al. [12,13]. A smaller variation range of heat transfer coefficient was found along the tube column direction than along the tube row and length directions.

The low-temperature multi-effect evaporation (LT-MEE) system owns the characteristics of small temperature difference, low pressure drop, saturation states and high sensitivity [14]. The thermodynamic losses caused by the boiling point elevation (BPE) and flow resistance may cause remarkable inhibition of heat transfer performance of the system. For the analysis of the thermodynamic loss in the MEE system, the BPE of seawater or the flow resistance of vapor were considered as constant in earlier publications [15–17]. In later publications, Kamali et al. [18,19] and Kamali and Mohebinia [20] considered the flow two-phase resistance in the falling film evaporator and the change of physical properties of seawater with the change of the temperature and the salinity. El-Dessouky et al. [21,22], El-Dessouky and Ettouney [23] developed relative comprehensive models in which the BPE of seawater, the vapor flow resistance in the demister and the steam flow resistance inside the tube were considered.

According to the literature review above, despite numerous and comprehensive studies on the distribution of the heat transfer coefficient in a horizontal tube falling film evaporator, investigation on the variation of the distribution of the local heat transfer coefficient of a large-falling film evaporator of difference geometrical dimension has rarely been carried out. Furthermore, considering the high sensitivity of the thermal performance of the falling film evaporator

to the variation of the temperature difference, the concept of temperature difference loss is defined in this paper. The analysis of the temperature difference loss within a single falling film evaporator could benefit to the more accurate thermal design of the LT-MEE system. The present study, therefore, provided the impact of the geometrical dimension on the heat transfer performance of the evaporator. The thermodynamic parameters distribution is presented. The temperature difference loss caused by the BPE and the inter-tube vapor flow resistance are quantified under different geometrical dimensions. The information is expected to benefit the geometrical design of a large-scale falling film evaporator.

2. Experimental apparatus and mathematical

Three experiments were carried out to study the heat transfer performance of falling film evaporation outside a horizontal tube, the condensation heat transfer performance as well as the flow characteristics inside a circular tube and the vapor pressure drop across a tube bundle. Several validated correlations were obtained from the set of experiments and a comprehensive DPM was developed to simulate the heat transfer performance in a large-scale horizontal-tube falling film evaporator. In previous work, the flow charts for the three experiments and the related correlations as well as the algorithm were described in detail [13]. The model and the operating conditions are briefly summarized as follows: the physical model is a large-scale horizontal-tube falling film evaporator containing a tube bundle that has 160 rows and 80 columns of tubes. The configuration of the evaporator is shown in Fig. 1a. On the shell side of the evaporator, the seawater is uniformly distributed on the top tubes and it flows downward by gravity. The seawater continuously evaporates when it is heated by the steam inside the tubes. On the tube side, the steam is condensed into liquid by the seawater on the shell side. The tube bundle is divided into a three-dimensional network as shown in Fig. 1b. For each grid, the seawater module on the shell side, the tube-side steam module and the inter-tube vapor module are calculated, respectively. Considering the symmetric distribution of thermal parameters along the tube column direction, half of the tube bundle is selected as the calculation area with column 1 representing the boundary of the tube bundle and column 40 the center. The validation of the present model is shown in Fig. 2. The average vapor mass flow rates per square meter are compared between the operating results and the simulation results using present model. It can be seen from Fig. 2 that the prediction values are coincident with the operating data with the relative deviation less than 4.2%. The simulation results are also compared with the model developed by Hou et al. [11] using DPM. It is indicated that the DPM is an effective method when it is applied in simulating large-scale falling film evaporators.

3. Results and discussions

3.1. Effect of the total tube length on the distribution of thermodynamic parameters

This section considers the impact of the tube length on the thermal performance of the evaporator under the

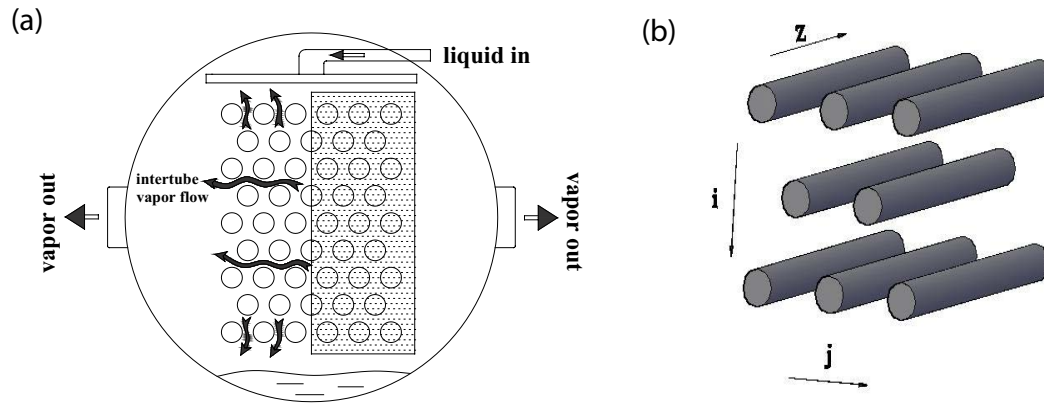


Fig. 1. Schematic of falling film evaporator and grid generation. (a) Cross-section of the evaporator and (b) three-dimensional grid generation.

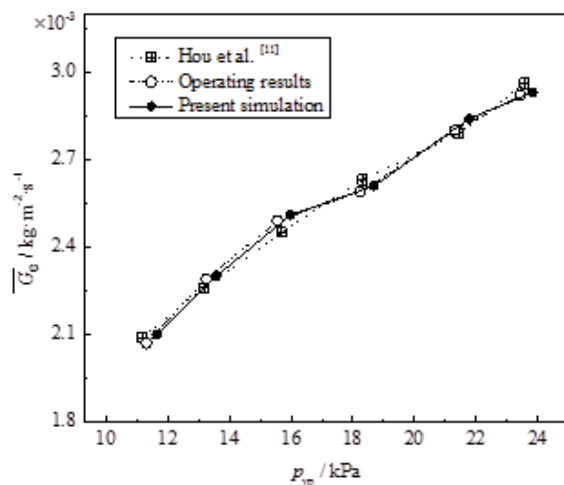


Fig. 2. Validation of the present model.

premise that the row–column ratio a constant and the total heat transfer area of the evaporator almost invariant. The geometric parameters and the operating conditions are listed in Tables 1 and 2.

3.1.1. Influence of the tube length on temperature difference loss and the distribution of temperature difference

On consideration of the high sensitivity of the thermal performance of the falling film evaporator to the variation of the temperature difference, the concept of “local temperature difference loss” is defined as the difference between the apparent temperature ΔT_{app} and the local temperature difference ΔT .

Fig. 3 shows the variation of the maximum temperature difference loss under different tube lengths. It can be seen from Fig. 3 that the tube length has less effect on the maximum temperature difference loss caused by the variation of steam pressure and salinity (δT_{st} and δT_s), but has greater effect on the maximum temperature difference loss caused by the variation of vapor pressure outside the tube (δT_{vp}). With the increase of L , δT_{st} remains almost unchanged. The local temperature difference loss caused by the change of

Table 1
Geometric parameters of the evaporator

Tube length/m	Row number	Column number	Row–column ratio	Total heat transfer area/m ²
5	204	102	2	8,302.02
6	184	92	2	8,104.76
7	172	86	2	8,262.44
8	160	80	2	8,171.15

Table 2
List of operating conditions

Parameter	Value
Steam inlet temperature/°C	63
Apparent temperature difference/°C	3
Average steam velocity/m s ⁻¹	40
Seawater inlet salinity/g kg ⁻¹	30
Seawater inlet spray density/kg m ⁻¹ s ⁻¹	0.06

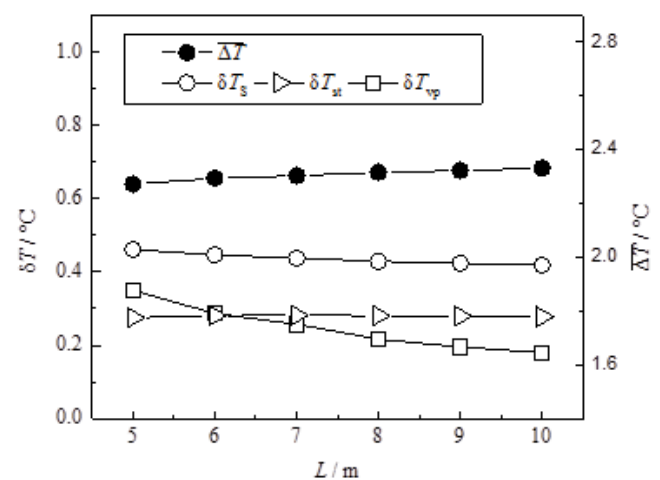


Fig. 3. Temperature difference loss of the evaporator under different tube lengths.

steam pressure in the tube decreases exponentially in the tube length direction. So the local temperature difference loss has larger values near the steam inlet. As the steam gradually approaches the tube outlet, the steam velocity decreases and the local temperature difference loss per unit length decreases rapidly. The simulation results in this section show that under the operating conditions of the evaporator selected in this paper, much of the total steam pressure drop Δp_{st} in the tube is generated within the first 5 m along the tube length direction. Therefore, when L is increased from 5 to 10 m, the maximum loss of temperature difference caused by the change of steam pressure in the tube is less than 2.5%.

With the increase of L , the maximum temperature difference loss caused by the increase of BPE of seawater, δT_s , slightly decreases. This is because that under the premise of the constant total heat transfer area, when L is increased, the row number of the tube bundle decreases and the concentration ratio of the sea water CR is gradually reduced which leads to the decrease in the increasing rate of BPE from top to bottom of the tube bundle.

With the increment of L , the maximum temperature difference loss caused by the vapor pressure drop on the shell side, δT_{vp} , decreases significantly. When L is increased from 5 to 10 m, the reduction of δT_{vp} reaches to 5.67% of ΔT_{app} . This is because when the heat transfer area of the evaporator remains unchanged, the number of rows and rows of the tube bundle decreases with the increase of L , and the flow resistance of the inter-tube vapor continuously decreases. From Fig. 3, it can also be seen that the average temperature difference of the evaporator increases slightly with the increase of L mainly due to the decrease of δT_{vp} and δT_s .

Fig. 4 shows the distribution of local temperature difference, ΔT , under different L . As described above, on

the premise that the total area of the tube bundle remains unchanged, with the increase of L , the concentration ratio of sea water outside the tube decreases. Thus the decreasing rate of ΔT from top to bottom of the tube bundle gradually decreases with the increase of L , as shown in Fig. 4. With the increase of L , because that the row-column ratio remains unchanged, the heat transfer areas of the three zones (top, middle and bottom zones divided according to the secondary steam flow direction outside the tube) has little variation (the variation rate is less than 0.3%). It indicates that for any cross section of the tube bundle, the distribution of the steam flow lines outside the tube remains almost unchanged with the increase of L . However, due to the decrease of the row number RN and the column number CN, the total flow resistance of the secondary steam outside the tube decreases. Fig. 4 shows that along the vapor flow directions, the variation gradient of the ΔT gradually decreases with the increase of L .

3.1.2. Impact of the tube length on the distribution of heat transfer coefficient and heat flux

Figs. 5 and 6 show the distributions of local heat transfer coefficient K and heat flux q under different L . It is indicated that L has a significant effect on the local heat transfer coefficient K and the heat flux q along the tube length direction. When L is increased, the in-tube steam has higher condensation rate, and at the steam outlet, the steam has lower velocity. Lower steam velocity leads to smaller disturbance force on the steam-liquid interface. Thus the convective heat transfer between the steam and liquid phases as well as the heat transfer in the condensate layer is weaker near the steam outlet. On the other hand, when L is increased, more condensate is accumulated at the bottom of the tube. Both the

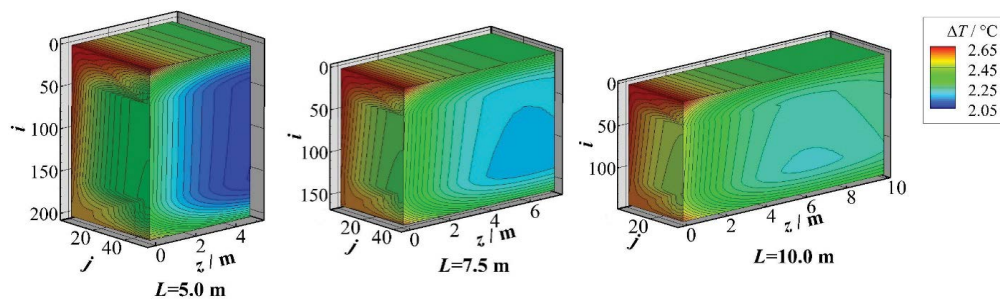


Fig. 4. Impact of the tube length on the distribution of ΔT .

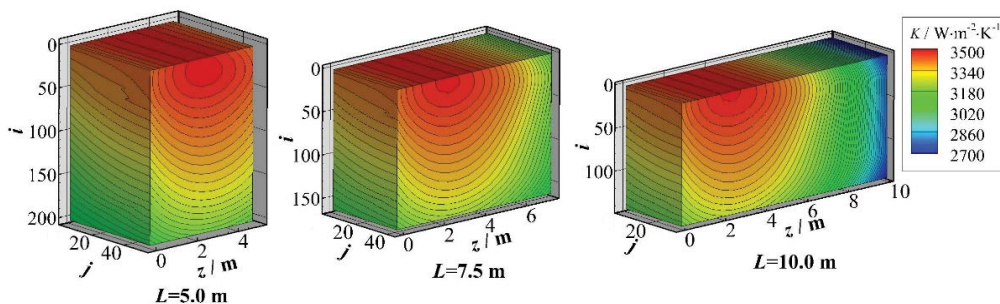


Fig. 5. Impact of the tube length on the distribution of K .

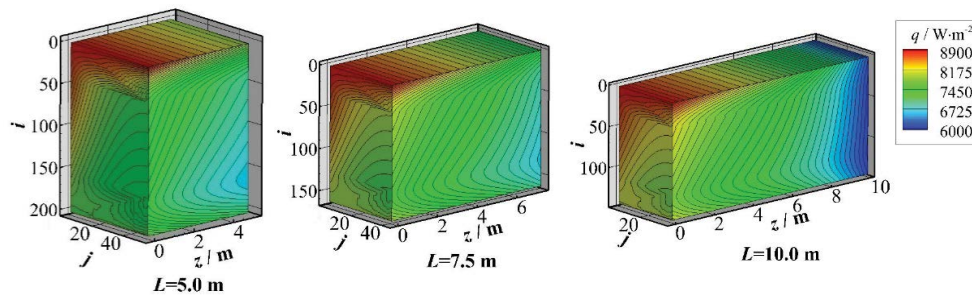


Fig. 6. Impact of the tube length on the distribution of q .

above phenomena are unfavorable to the condensation heat transfer performance in the tube. Therefore, both the local heat transfer coefficient K and the heat flux q decrease more significantly near the steam outlet when the tube bundle has longer tube length L , as shown in Figs. 5 and 6.

As mentioned above, with the total heat transfer area of evaporator almost unchanged and the row–column ratio remains constant, the tube row number RN and the tube column number CN decrease proportionally with the increase of L . The variation amplitude of seawater spray density, Γ , seawater salinity, S , and temperature difference, ΔT , along the row and column directions decreases. Therefore, the variation gradient of K and q along the tube row and column directions decreases which means both K and q are more uniformly distributed along these two directions, as shown in Figs. 5 and 6.

3.1.3. Optimal tube length

Based on the above analysis, it can be seen that the spatial distribution of the thermodynamic parameters in the horizontal tube falling film evaporator varies with the change of tube length. The difference in the distribution of thermal parameters further affects the overall heat transfer performance of the evaporator.

Fig. 7 shows the variation of the average heat transfer coefficient $\bar{K}$ and the average heat flux $\bar{q}$ of the evaporator with the change of L . On the one hand, it is shown in

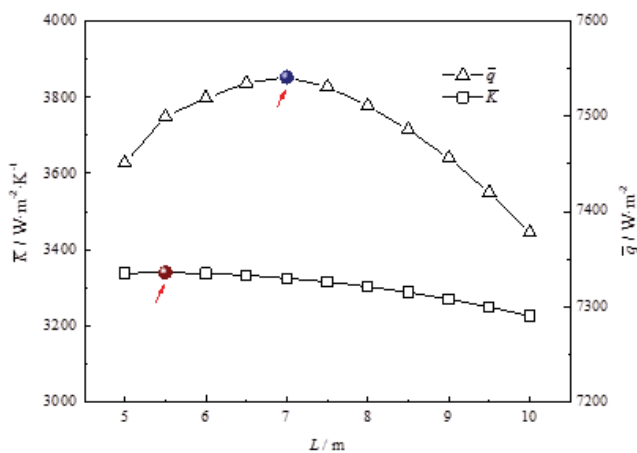


Fig. 7. Average heat flux and heat transfer coefficient under different tube lengths.

Fig. 5 that the local heat transfer coefficient K of the evaporator increases first and then decreases along the tube length direction. However, K exhibits greater decreasing range near the steam outlet than the rising range near the steam inlet. This makes the average heat transfer coefficient $\bar{K}$ exhibits a slight increasing trend followed by a remarkable decreasing trend when the total tube length L is increased from 5 to 10 m. The maximum value of $\bar{K}$ is obtained when the length of the tube L is 5.5 m, as shown in Fig. 7. On the other hand, it is shown in Fig. 3 that with the increase of L , the temperature difference loss of evaporator decreases and the average temperature difference increases $\Delta \bar{T}$ slightly. It is indicated in Fig. 7 that the highest average heat transfer coefficient of the evaporator does not lead to the maximum heat transfer. Under the combined influence of the average heat transfer coefficient $\bar{K}$ and the average temperature difference $\Delta \bar{T}$, the average heat flux of the evaporator $\bar{q}$ increases first and then decreases with the increase of L . $\bar{q}$ reaches its maximum value when L is 7 m. That is, under the selected operating conditions and the geometric parameters of the evaporator in this paper, the optimum tube length of the evaporator is 7 m taking the maximum average heat flux as the criterion.

3.2. Impact of the row–column ratio on the distribution of thermodynamic parameters

In this section, the influence of the row–column ratio on the spatial distribution of thermodynamic parameters of the evaporator is discussed on the premise that the total heat transfer area of evaporator remains almost unchanged and the total tube length remains a constant. The geometric parameters of the evaporator are shown in Table 3. The setting of evaporator operating conditions is of the same as listed in Table 2.

3.2.1. Influence of the row–column ratio on the temperature difference loss and the distribution of temperature difference

Fig. 8 shows the variation of temperature difference at different row to column ratios, RN/CN . On the premise that the total heat transfer area of the evaporator basically unchanged and the length of the heat transfer tube a fixed value, the cross-sectional area of the tube bundle is almost unchanged with the change of RN/CN . Thus the number of tube column decreases with the increase of RN/CN . For the regular triangular pitch bundle, because that the horizontal inter-tube vapor flow is more inclined to generate smaller

flow resistance than the vertical inter-tube vapor flow, the area of the horizontal inter-tube vapor flow increases with the increment of RN/CN. As a result, the ratio of the maximum temperature difference loss caused by the vapor pressure to the apparent temperature difference decreases from 12.78% to 3.73%. On the premise that the total tube length and the seawater spray density on the top row remain unchanged, the concentration ratio of seawater has higher values under larger RN/CN. Therefore, the ratio of the maximum temperature difference loss caused by the variation of salinity of seawater outside the tube, δT_{st} , to the apparent temperature difference increases from 13.42% to 15.79%. Due to that the total tube length remains unvaried, the maximum temperature difference loss caused by the variation of in-tube steam

pressure, δT_{st} , has no significant change with the increment of RN/CN. Based on the above analysis, the maximum temperature difference loss caused by the change of inter-tube vapor pressure outside the tube, δT_{vp} , and the maximum temperature difference loss caused by the change of seawater salinity outside the tube, δT_s , have relative more significant variation with the change of RN/CN, but exhibit opposite variation trends. Fig. 8 shows that with the change of RN/CN, δT_{vp} plays a dominant role on the variation of the average temperature difference $\Delta \bar{T}$. With the increase of RN/CN, it is due to that δT_{vp} has remarkable decrease, $\Delta \bar{T}$ shows a slight increasing trend.

Fig. 9 shows the distribution of the temperature difference with the change of RN/CN. It is shown that when RN/CN has smaller values, the tube bundle is shorter but wider. As described above, for the regular triangular pitch tube bundle, horizontal inter-tube vapor flow is more inclined to generate smaller flow resistance, thus the total inter-tube vapor flow resistance from the inside to the boundary of the tube bundle increases with the decrement of RN/CN due to the increases of the tube size along the horizontal direction. In consequence, with the decrement of RN/CN, the area composed of the vertical inter-tube vapor flow gradually increases. That is, the ratio of the top zone and the bottom zone to the volume of the evaporator increases significantly. Accordingly, the center zone composed of the horizontal inter-tube vapor flow gradually decreases. As can be seen from Fig. 9, the variation gradient of ΔT from the inner to the boundary of the tube bundle is larger under smaller RN/CN. In addition, comparing Fig. 4 with Fig. 9, it is indicated that the maximum of local temperature difference loss normally occurs at the boundary of the center zone and the middle zone. When RN/CN increases, the minimum value of the heat transfer temperature difference gradually moves towards the bottom row of the tube bundle due to the increase of the center zone and the decrease of the bottom zone. When RN/CN increases, the boiling point of seawater at the bottom row increases more obviously because of the increase of the seawater concentration ratio. In Fig. 9, comparing the local heat transfer temperature difference between the top row and the bottom row, it can be seen that under larger RN/CN, the variation range of ΔT along the row direction is more remarkable. Along the tube length direction, with the increase of RN/CN, the total temperature drop of the steam between the steam inlet and outlet remains almost unchanged. In consequence, it is indicated that the average temperature difference drop between the steam inlet and outlet has little change with the variation of RN/CN.

Table 3
Geometric parameters of the evaporator

Row number	Column number	Row-column ratio	Total tube length/m	Total heat transfer area/m ²
120	106	1.13	8	8,120.08
140	92	1.52	8	8,222.22
160	80	2.00	8	8,171.15
180	72	2.50	8	8,273.29
200	64	3.13	8	8,171.15
220	58	3.79	8	8,145.62

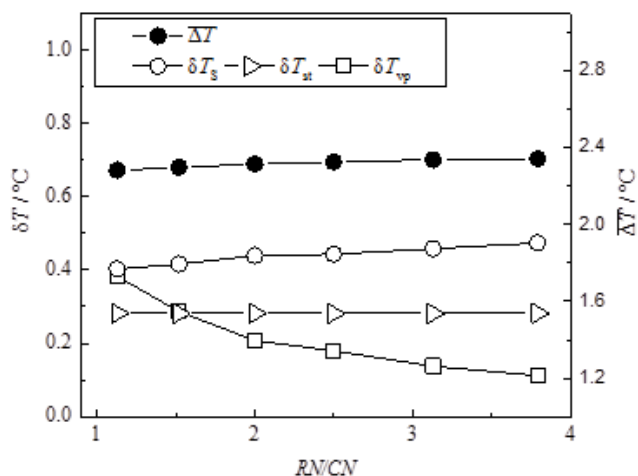


Fig. 8. Impact of the row-column ratio on the distribution of the local heat transfer coefficient.

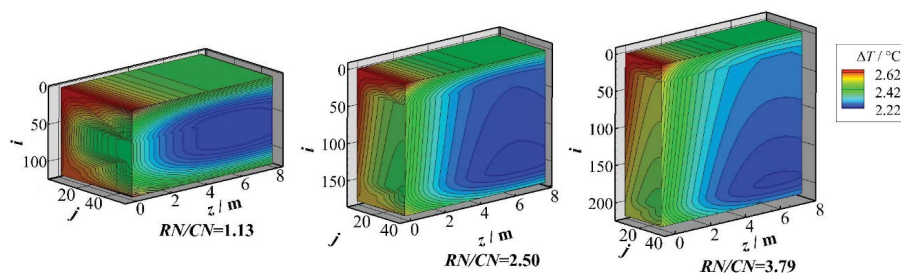


Fig. 9. Impact of the row-column ratio on the distribution of ΔT .

3.2.2. Influence of the row–column ratio on the distribution of the heat transfer coefficient and the heat flux

Fig. 10 shows the distribution of the local heat transfer coefficient K under different RN/CN. It is indicated that RN/CN has more significant effects on the distribution of K along the tube row direction. As analyzed above, the concentration ratio of seawater increases with the increase of RN/CN under the premise that the spray density of seawater in the top row remains unchanged. As a result, when the evaporator has larger value of RN/CN, the local spray density of seawater near the bottom row is smaller and the local salinity is higher which lead to a more significant decrease of K from top to bottom row as shown in Fig. 10. The simulation results show that when RN/CN is increased from 1.13 to 3.79, the decreasing rate K from top to bottom row increases from 4.15% to 8.64%. Along the tube length direction, with the change of RN/CN, for the top row, the distribution of K along the tube length direction remains unchanged due to the unvaried thermodynamic states of fluid for both the inside and outside of tubes under the same operating conditions. For the rest rows, with the increase of RN/CN, it is due to the increase of the seawater concentration ratio, the temperature difference shows larger decreasing range from top to bottom row, and the heat flux exhibits a more significant decreasing trend. For the tubes that are closer to the bottom row, the in-tube steam has smaller condensation ratio. Thus the inhibiting effect of condensate on the heat transfer coefficient is weaker and the K shows smaller variation range along the tube length direction with the increase of RN/CN. When RN/CN = 1.13, the maximum value of K along the length direction is 10.13% higher than the minimum value on average, while when RN/CN = 3.79, the maximum value of K is 6.92% higher than the minimum value

on average. The main parameter that affects the distribution of K along the tube length direction is the local temperature difference. As the increase of RN/CN, the variation range of heat transfer temperature difference along this direction decreases gradually as shown in Fig. 9. It leads to the decrease of the variation range of K along the tube row direction and thus a more uniform distribution of K as shown in Fig. 10.

Fig. 11 shows the distribution of heat flux q in the evaporator under different RN/CN. It can be seen from the above analysis that both the heat transfer coefficient and temperature difference show larger decreasing range from top to bottom along the row direction with the increase of RN/CN which lead to more non-uniform distribution of heat flux along this direction. The simulation results show that when RN/CN is increased from 1.13 to 3.79, the average decreasing rate of q from top to bottom row increases from 6.05% to 12.61%. Comparing Figs. 9 and 11, it is indicated that with the increase of RN/CN, because of the decrease of the variation range of the gradient of ΔT from the center to the boundary of the tube bundle, the heat flux is more uniformly distributed within the tube bundle. In addition, in the three zones divided by the flow directions of the inter-tube vapor, the area of the central zone increases gradually with the increase of RN/CN, so the minimum value of the heat flux gradually moves towards the bottom of the tube bundle. Under the condition of changing RN/CN but keeping L unchanged, as mentioned above, the average decrease of ΔT along the tube length direction remains almost unchanged, while the variation range of K gradually decreases. It results in that when RN/CN is increased from 1.13 to 3.79, the decreasing rate of heat flux along the length direction decreases from 20.30% to 15.81%.

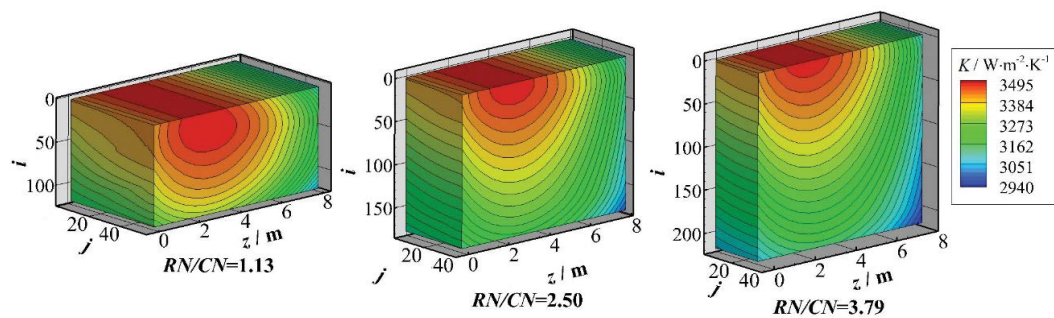


Fig. 10. Impact of the tube length on the distribution of K .

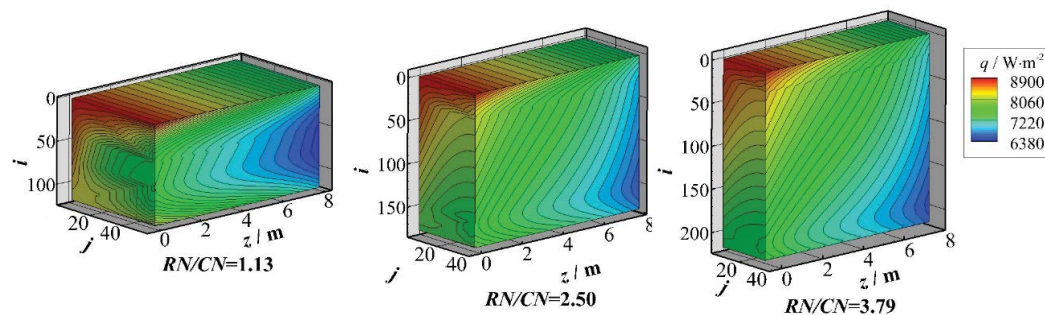


Fig. 11. Impact of the tube length on the distribution of q .

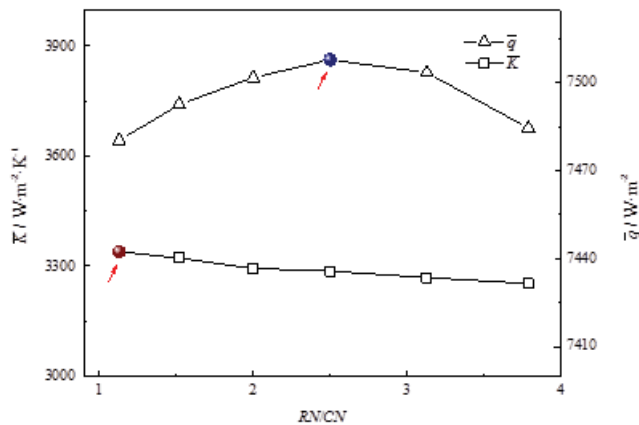


Fig. 12. Average heat flux and heat transfer coefficient under different row–column ratios.

3.2.3. Optimal tube bundle row–column ratio

Fig. 12 shows the variation of the average heat transfer coefficient and the average heat flux of the evaporator with the change of tube bundle row–column ratio RN/CN. With the increase of RN/CN, the average heat transfer coefficient of evaporator decreases continuously mainly because that the local heat transfer coefficient K decreases more significantly along the row direction, as shown in Fig. 12. It is shown in Fig. 8 that with the increase of RN/CN, the temperature difference loss of evaporator decreases and the average heat transfer temperature difference $\Delta \bar{T}$ increases. Under the combined influence of the average heat transfer coefficient and the average temperature difference, the average heat flux first increases and then decreases with the increase of RN/CN. The maximum value is obtained when t RN/CN is 2.5. That is, the optimal row–column ratio of the evaporator is 2.5 with the average heat flux as the criterion under the selected operating conditions and the geometric size of the evaporator in this paper.

4. Concluding remarks

This study presents the thermal performance of a large horizontal-tube falling film evaporator based on the DPM. Considering the sensitivity of the heat transfer performance of the falling evaporator to the change of the temperature difference, the variation of the temperature difference loss and its effects on the thermal performance of the falling evaporator is investigated under different geometric parameters of the tube bundle.

Under the selected operating conditions and geometric parameters, the maximum temperature difference loss caused by the change of the inter-tube vapor pressure and the BPE of seawater in the evaporator decreases with the increase of the total tube length. The maximum temperature difference loss caused by the change of in-tube steam pressure remains almost unchanged. With the increment of tube length, both the heat transfer coefficient and heat flux are more non-uniformly distributed in the tube length direction but more uniformly distributed along the tube column direction. The maximum value of the average heat flux of the evaporator is obtained when the length of the tube is 7 m.

The maximum temperature difference loss caused by the increase of BPE of seawater in evaporator increases slightly with the increase of row–column ratio of the tube bundle. The maximum temperature difference loss caused by the change of in-tube steam pressure remains almost unchanged, but the maximum temperature difference loss caused by the change of inter-tube vapor pressure is reduced remarkably. With the increment of the row–column ratio, both the heat transfer coefficient and heat flux are more non-uniformly distributed along the tube column direction but more uniformly distributed along the tube length direction. The maximum heat flux of the evaporator is obtained when the tube bundle row–column ratio is 2.5.

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Symbols

BPE	—	Boiling point elevation of seawater, °C
CN	—	Column number
CR	—	Concentration ratio of seawater
K	—	Local heat transfer coefficient, $W m^{-2} K^{-1}$
L	—	Total tube length, m
q	—	Heat flux, $W m^{-2}$
RN	—	Row number of the tube bundle
S	—	Local seawater salinity, $g kg^{-1}$
Δp_{st}	—	Local steam pressure drop, Pa
ΔT	—	Local temperature difference, °C
ΔT_{app}	—	Apparent temperature difference of the evaporator, °C
δT_{vp}	—	Maximum temperature difference loss caused by the inter-tube vapor flow resistance, °C
δT_s	—	Maximum temperature difference loss caused by the variation of seawater salinity, °C
δT_{st}	—	Maximum temperature difference loss caused by the in-tube steam flow resistance, °C

Greeks

Γ	—	Local seawater flow rate, $kg m^{-1} s^{-1}$
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Subscripts

S	—	Salinity
st	—	In-tube steam
vp	—	Inter-tube vapor

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