Research Paper

Experimental investigation of R410A and R32 falling film evaporation on horizontal enhanced tubes

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HIGHLIGHTS

- Falling film evaporation with nucleating boiling are experimentally studied for R32 and R410A on four doubly enhanced tubes.
- HTCs exhibit two stages with decrease of film flow rate, a quasi-plateau stage and a sharp-decrease stage.
- HTCs increase with heat flux progressively before reaching the partial dryout regime.
- The integrated-fin tube performs best among the four enhanced tubes tested.

ABSTRACT

In this study, heat transfer performance on horizontal copper tubes of high pressure refrigerants R32 and R410A was investigated. An integrated fin (condensation enhanced) tube and three boiling enhanced tubes with 3-D different enhancement structures were tested. A plain tube was also tested for comparison. Effects of film flow rate, saturation temperature and heat flux on the falling film heat transfer coefficients were investigated. A tube bundle comprised of 6 enhanced boiling tubes was also tested to find the bundle effect. Experiments were carried out at saturation temperatures of 6, 10 and 16 °C, heat fluxes from 20 to 150 kW·m^{-2} and film flow rates from 0.01 kg·m^{-1}·s^{-1} to 0.14 kg·m^{-1}·s^{-1}. It is found that the effect of film Reynolds number on HTCs of enhanced tubes can be separated into two regimes, a quasi-plateau regime and a sharp decrease regime. HTCs increase with heat flux. Increase of saturation temperature has negligible effect on enhanced tubes as well as the tube bundle. The integrated fin tube performs best among the tubes tested. As a whole, R410A is inferior than R32. Tubes positioned below the top row possess the similar variation trend of HTCs with that of the first row but suffer an earlier dryout.

1. Introduction

Evaporator is an indispensable facility of a refrigeration and heat pump system. Several types of evaporators have been proposed and adopted, including flooded type (pool boiling), dried type and the falling-film type evaporator. Despite the fact that falling-film type evaporator was initially patented in 1888 [1], it was not a hot research topic until 1970s because of the oil crisis. From then on, it has been widely used in petrochemical industry, food processing, desalination industry and ocean thermal energy conversion system. With the environmental problems of ozone depletion and global warming becoming more and more serious, the usage of HCFCs in refrigeration and air-conditioning industry came into the eyes of the public all over the world. Montreal's Protocol and later revisions prompted the phasing

out of CFCs which in turn stimulated the usage of falling-film type evaporator in the field of refrigeration, air-conditioning and heat pump for its minuscule liquid inventory compared with the flooded type. Apart from that, falling film evaporator exhibits several other advantages over pool boiling evaporator, such as: higher heat transfer coefficients can be achieved, boiling temperature increase caused by hydrostatic head can be avoided, size of evaporator can be reduced and oil removal from the system is made easier. For these reasons, falling-film evaporator is a potential substitute for pool boiling evaporator. Fig. 1 shows its schematic diagram.

A large number of technical papers about the falling film evaporation have been published in the past several decades, for the simplicity of presentation, only three representative review papers are mentioned below. Ribatski and Jacobi [3] made a comprehensive and critical

Nomenclature

A	area, m^2		length, kg·m^{-1}·s^{-1}
c	enhanced factor	ϕ	heat transfer rate, W
D	diameter of tube, mm	μ	dynamic viscosity, kg·m^{-1}·s^{-1}
e	external fin height, mm		
fpi	external fins per inch	*Subscript*	
FFE	falling film evaporation		
h	heat transfer coefficient/HTC, W·m^{-2}·K^{-1}	c	condensation
k	overall heat transfer coefficient, W·m^{-2}·K^{-1}	e	evaporation
L	tested length of tube, mm	f	fouling
PB	pool boiling	Gni	Gnielinski equation
q	heat flux, kW·m^{-2}	l	liquid refrigerant
R	thermal resistance, m^2·K·W^{-1}	v	vapor refrigerant
r	latent heat, J·kg^{-1}	LMTD	logarithmic mean temperature difference
Re_Γ	film Reynolds number	i	inside of tube
T	temperature, °C	m	mean
t	inner fin height, mm	o	outside of tube
V	velocity of water, m·s^{-1}	p	pump
w	outlet width of the reentrant cavity, mm	r	reference
		sat	saturation
Greek		w	wall
		in	water inlet
Δ	variable differential	out	water outlet
Γ	liquid film flow rate on one side of the tube per unit	pre	prediction
		exp	experiment

Fig. 1. Diagram of a falling film evaporator [2].

review on the horizontal-tube falling film evaporation. They pointed out that further experimental work must be undertaken to broaden the current data base and resolve the contradictions in the extant data. In the years of 2014 and 2015, Fernández-Seara and Pardiñas [2] and Abed et al. [4], respectively, present their summary to reveal the state of art in the field of falling film evaporation research. Fernández-Seara and Pardiñas [2] demonstrated that experience in the application of falling film evaporator to heat pumps and refrigeration system was still limited. In both [2 and 4] the authors all stated that enhanced surface tubes have been appropriately applied in multiple applications with limited success, and further researches are highly needed. More references on the falling film evaporation on a horizontal single tube may be found in [5].

Because of the complicacy of two-phase flow in the falling film evaporator, there are many factors affecting its heat transfer performance which can be divided into two categories, designing factors and operating factors. Type of liquid distributor, structure of enhanced surfaces and bundle arrangement are the main designing factors which cannot be changed after installation of the system, while falling film flow rate, heat flux, saturation temperature and vapor flow are influencing operating factors which can be adjusted during operation [6–8]. The present paper will focus on the study of the enhanced tubes.

Structured surfaces with low fins or 3-D fins can improve the falling film boiling heat transfer by increasing the heat transfer area, creating cavities for bubbles' growing and enhancing the internal turbulence of the film. In the previous literature, several series of commercial enhanced tubes have been tested. A brief review of previous works on falling film evaporation with enhanced tubes is now conducted as follows.

Compared with the plain tube, enhanced tubes may delay the appearance of dry patches [9,10]. So far no specific enhanced tubes have been developed to work under falling film evaporation process, so it

1. Condenser (reservoir); 2. Evaporator; 3. Electromagnetic flowmeter; 4. RTDs; 5. Pressure gauge; 6. Condensate measuring container; 7. Exhausting valve; 8. Magnetic Pump; 9. Refrigerant charging valve; 10. Hot water pump; 11. Hot water tank; 12. Refrigerant outlet; 13. Cold water pump; 14. Cold water tank; 15. Liquid refrigerant distributor.

Fig. 2. Diagram of the experimental apparatus [25].

should be mentioned here that the so-called condensation-enhanced or boiling-enhanced tubes in this paper are referred to those enhanced tubes originally designed for enhancing condensation and pool boiling respectively.

Moeykens [11] tested plain tubes, two condensation-enhanced tubes, two boiling-enhanced tubes, and two finned tubes in a multi-tube falling film evaporation test facility. The results with R134a showed that heat transfer coefficients (HTCs) were slightly influenced by heat flux which indicated the coexistence of convective evaporation and nucleate boiling within the range tested (heat fluxes from 10 to $40 \, \mathrm{kW \cdot m^{-2}}$). Best performance was achieved with the condensation-enhanced tubes, whose HTCs were about 3 times as the plain tube under the same testing conditions, followed by the boiling-enhanced tubes and the finned tubes.

Thome [12–14], Roques [9,12], Habert [13,15] and Christians [14,16] extended the existing database of falling film evaporation HTCs with new refrigerants and structured surfaces. They tested plain tubes and several types of boiling-enhanced and condensation enhanced tubes in a tube bundle comprised of ten tubes with R134a and R236fa. The results showed that falling film HTCs changed slightly with film Reynolds number for large Reynolds number, until a certain value at which they decreased sharply due to dry patches' appearance. HTCs increased with heat flux for the plain and condensation enhanced tubes and decreased or remained unchanged for boiling enhanced tubes. In addition, R134a led to better heat transfer performance than R236fa. Whereas, there are still contradictions between their results and others'.

Li et al. [17] studied the heat transfer characteristics of falling film evaporation on horizontal tube arrays, and it was observed that when the Reynolds numbers were high, heat flux variation had little effects on the heat transfer coefficient of falling film evaporation on enhanced tube arrays.

Some researchers have proposed empirical correlations to predict the falling film HTCs on a single plain tube [5,18–20], among whom Zhao et al. [5] achieve considerable success in predicting not only their experimental results but also other researchers'. Whereas, the results of high pressure refrigerants are not included in their comparison.

With the purpose to decrease the negative effects of refrigeration and air-conditioning system on the climate, the refrigeration and air-conditioning industries have been devoting a lot to the conversion from HCFC (R12 and R22) to HFC(R123, R134a and R410A) and even a further step to HC (R290 and R600a) or HFO(R1234ze and R1234yf). Thereinto, R32 is recommended as one of the potential substitutes to R410A because its lower global warming potential. Based on the authors' knowledge, high pressure refrigerants (such as R410A and R32) can own higher heat transfer coefficients than lower pressure refrigerants and increase the circulation efficiency of a chiller system. R32 and R410A evaporation as well as condensation inside tube or microchannel have been studied by many researchers, say [21–24]. However, falling film evaporation heat transfer outside horizontal tubes with these two refrigerants has seldom been investigated. Also whether the pool boiling and condensation enhanced tubes can provide significant enhancement of the HTC still needs further study. In addition, due to the fact that enhanced tubes are often used in tube bundles, it is imperative to be aware of whether the data from tube bundles will significantly differ from its single tube counterparts.

Based on the above review and analysis, a plain tube, three boiling enhanced tubes and one condensation enhanced tube will be tested in this paper to compare their heat transfer performance. High pressure refrigerants R32 and R410A will be taken as the working fluids. Also, a tube bundle will be employed to evaluate the bundle effect. In this paper, the same enhanced tubes with Ref. [25] are tested for the

Table 1
Specifications of key measurement instruments.

Instruments	Model	Precision
Mass flow meter	SIEMENS MASS2100	0.1%
Volumetric flow meter	SIEMENS MAGFLO MAG5100W	0.1%
Pressure gauge	KELLER LEX1	0.05%
RTD	OMEGA Pt100 1/10 DIN	$\pm (0.03 + 0.0005|\mathrm{T}|)\mathrm{K}$
Data acquisition	Keithley digital voltmeter	$0.1 \, \mu\mathrm{V}$

Table 2
Specifications of tested tubes.

Tube number	Plain	1	2	3	4
D_o (mm)	19.06	19.04	19.03	18.89	19.05
D_i (mm)	17.19	16.99	17.00	16.61	17.17
fpi	–	50	51	45	48
e (mm)	–	0.5	0.71	0.57	0.56
w (mm)	–	0.063	0.280	0.119	0.074
t (mm)	–	0.3	0.34	0.32	0.46
L (mm)	535	535	535	500	515

simplicity of comparison between different refrigerants. Compared with R134a and R123 tested in [25], R32 owns lower GWP (Global Warming Potential). In addition, test will also be conducted for a tube bundle which was not investigated in Ref. [25].

In the following, the test facility will be introduced first, followed by data reduction and uncertainty analysis, then test results will be presented in details. Finally some conclusions will be drawn.

2. Experimental setup and tested tubes

The experimental system has three major circulation loops which are chilled (hot) water circuit, chilling (cold) water circuit and refrigerant circuit. Liquid refrigerant is stored at the bottom of the condenser, during operation, liquid refrigerant is pumped to the top of the

Tube No.1

Tube No.2

Tube No.3

Tube No.4

Fig. 3. Scanning pictures of enhanced surfaces [25].

evaporator and enters the liquid distributor. Then liquid refrigerant is distributed onto the tube's outer surface to absorb the heat of chilled water inside the tube and evaporates. Refrigerant vapor flows back to the condenser and condenses outside the outer surfaces of the tubes within which chilling water flows. The schematic diagram of the test facility is shown in Fig. 2.

Volume flow rate of chilled and chilling water and mass flow rate of liquid refrigerant are measured with electromagnetic flow meter. A digital pressure gauge is employed to measure the pressure in the evaporator. RTDs are installed at the inlet and outlet of the water to measure the temperature differences. Specifications of measurement instruments are listed in Table 1.

The liquid distributor consists of a preliminary and a secondary distributors. They are two rectangular stainless steel boxes but the top plate of the second one is open-ended. The liquid level in the second box varies with film flow rate which makes the liquid refrigerant flow under gravity. For both boxes, orifices with diameter of 2.0 mm and spacing of 20.0 mm are drilled at the bottom surfaces.

Four enhanced tubes with different enhanced structures are tested on the experimental system. They all have 3-D fins on the outer surface and spiral ribs inside the tube to enhance water side heat transfer. Specifications of enhanced tubes are listed in Table 2 and geometries and scanning pictures of the 3-D fins are shown in Fig. 3. A plain tube is also tested for comparison.

Configuration of the tube bundle is demonstrated in Fig. 4, where the four semi-dummy tubes and one dummy tube is arranged in the tube bundle to simulate the triangular-pitch tube bundle. During the tests, there was no water inside dummy tubes which meant no heat transfer for these tubes. The bottom of the liquid distributor is equipped by two half tubes right above tube No. 5 and No. 9. The two half tubes have a continuous slit along the axial direction to make the liquid refrigerant in the distributor flow through them to tubes No. 5 and No. 9. Tubes in the bundle have the same enhanced structure with tube No. 3. The vertical and horizontal tube pitches are 22.5 mm and 19.5 mm respectively.

High pressure refrigerants, R32 and R410A, are used as the working fluids. Experimental measurements are carried out for each single tube individually. The ranges of working conditions are listed in Table 3. During the test, the pressures both at the top and the bottom of the evaporator are measured with two digital pressure gauges, and their difference is within the precision of the pressure gauge which indicates that the pressure drop of falling film evaporation within the evaporator is negligible. In addition, the corresponding saturated temperature corresponds well with the liquid refrigerant temperature in the condenser/reservoir. The difference between them (pressure-corresponding saturation temperature and liquid temperature in reservoir) is within 0.1 °C.
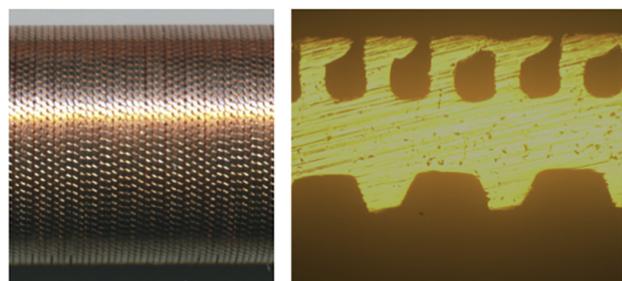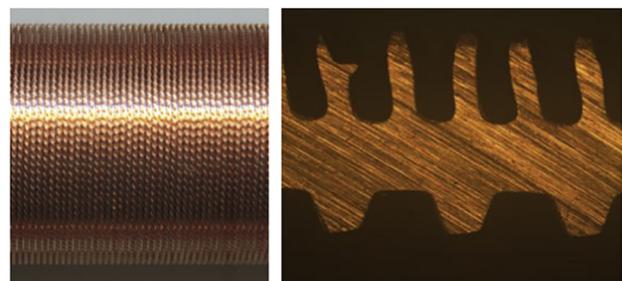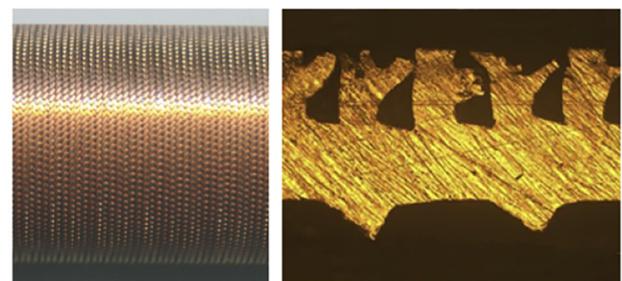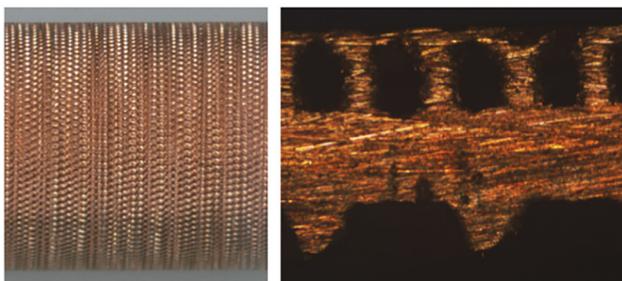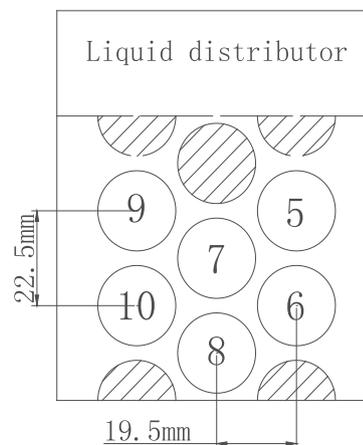
Fig. 4. Tube bundle layout.

Table 3
Experimental conditions for Wilson plot and falling film evaporation.

	Wilson plot	Falling film evaporation
T_{sat} (°C)	6	6, 10, 16
q (kW·m^{-2})	40	20–150
V_{water} (m·s^{-1})	0.7–3.3	2
Γ (kg·m^{-1}·s^{-1})	0.08	0.01–0.14

Table 4
Uncertainties of outside heat transfer coefficients.

Tube number	Uncertainty of h_o q		
	20 kW·m^{-2}	40 kW·m^{-2}	60 kW·m^{-2}
Plain	15.4%	19.6%	24.0%
No. 1	13.0%	13.6%	14.3%
No. 2	19.3%	18.4%	20.2%
No. 3	14.2%	16.2%	15.4%
No. 4	15.1%	15.1%	14.2%

Table 5
Enhanced factors of water side heat transfer coefficients.

Tube number	1	2	3	4	–	–
c_i	3.011	2.872	3.122	3.423	–	–
Tube number	5	6	7	8	9	10
c_i	3.122	3.22	3.0	3.22	3.39	3.407

Fig. 6. Variation of falling film evaporation HTCs with film Reynolds number for single plain tube.

(a) single tube
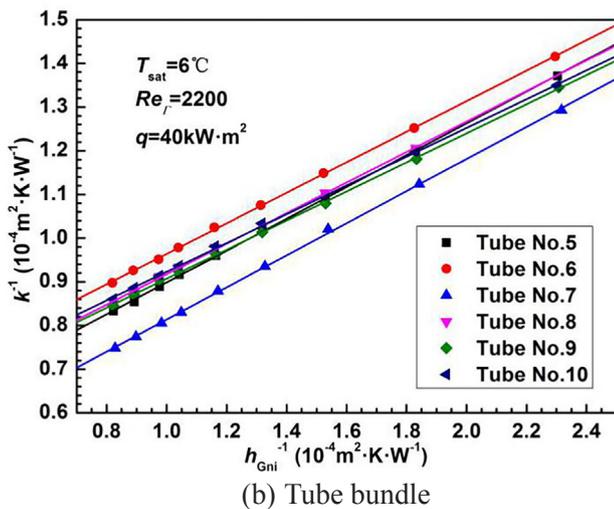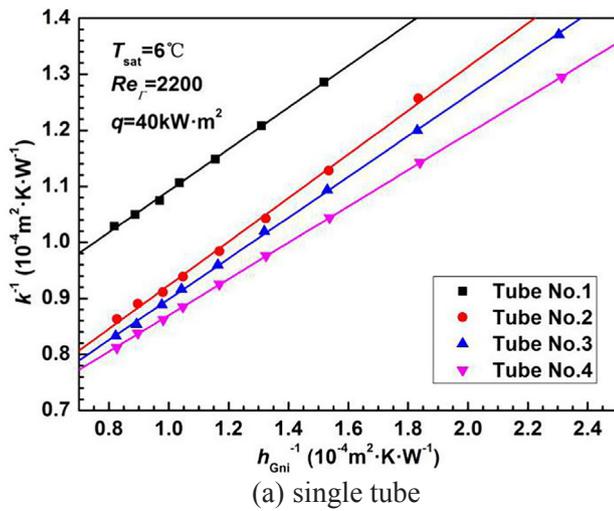
(b) Tube bundle

Fig. 5. Variation of k^{-1} with h_{Gni}^{-1} for single tube and the tube bundle.

3. Data reduction and uncertainty analysis

3.1. Heat transfer rate and heat balance

Heat dissipated by the chilled water and absorbed by the chilling water is calculated by Eqs. (1) and (2) respectively:

$$\phi_e = \dot{m}_e c_p (T_{e,in} - T_{e,out}) \tag{1}$$

$$\phi_c = \dot{m}_c c_p (T_{c,out} - T_{c,in}) \tag{2}$$

where $\dot{m}_e$ and $\dot{m}_c$ are mass flow rate of chilled water and chilling water respectively, c_p is the specific heat capacity of water inside the tube.

Following heat balance requirement of the system should be satisfied:

$$(\phi_e + \phi_p - \phi_c) + /\phi_r \leq 5\% \tag{3}$$

where ϕ_p is the power of the canned motor pump which is immersed in the bulk of liquid refrigerant, 1.5 kW; ϕ_r is the reference heat transfer rate, defined by Eq. (4):

$$\phi_r = (\phi_e + \phi_c - \phi_p)/2 \tag{4}$$

For all the test data presented in this paper, heat balance deviation is less than 5% which ensures the reliability of determining the overall heat transfer coefficients in next part.
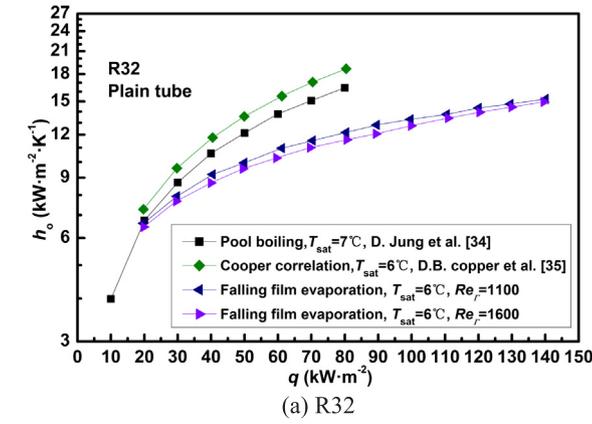
(a) R32
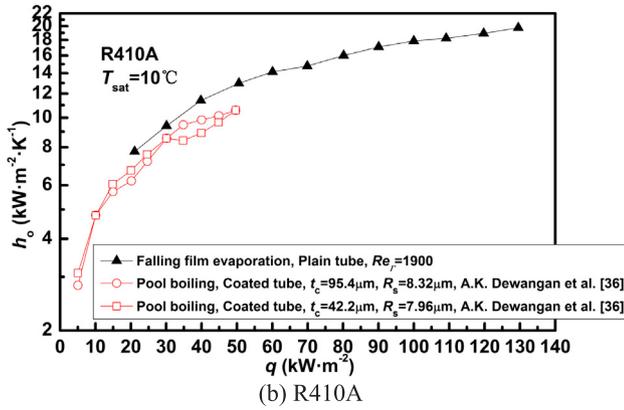
(b) R410A

Fig. 7. HTCs comparison between FFE and PB on plain surfaces.

(a) R32

(b) R410A

Fig. 8. Comparison of HTCs between experimental and prediction results [5]

3.2. Overall heat transfer coefficients

The overall heat transfer coefficients of the tested tubes can be expressed by Eq. (5):

$$k = \frac{\phi_e}{A_o \Delta T_{LMTD}} \tag{5}$$

where ΔT_{LMTD} is the logarithm mean temperature difference between water and refrigerant saturation temperature.

ΔT_{LMTD} is defined by Eq. (6).

$$\Delta T_{LMTD} = \frac{|T_{e,in} - T_{e,out}|}{\ln\left[(T_{e,in} - T_{sat})/(T_{e,out} - T_{sat})\right]} \tag{6}$$

where T_{sat} is the saturation temperature of the refrigerant.

3.3. Water side heat transfer coefficients

By the thermal resistance analysis, the overall thermal resistance can be expressed as Eq. (7):

$$\frac{1}{k} = \frac{1}{h_o} + R_w + R_f + \frac{1}{h_i}\frac{D_o}{D_i} \tag{7}$$

where D_i and D_o are the inner and outer diameter of the test tubes, R_w is the thermal resistance of the tube wall, h_i is the water side convective heat transfer coefficient and R_f is the fouling thermal resistance. Since the tested tubes have been cleaned before installation, fouling thermal resistance is neglected in this study.

For the tubes with internal enhancement structures, convective heat transfer coefficients between water and internal surfaces can be expressed by Eq. (8):

$$h_i = c_i h_{Gni} \tag{8}$$

where c_i is enhanced factor of the internal surface, obtained by Wilson plot method, which is described in detail by Yang and Tao [26]. Gnielinski correlation [27,28] is adopted to calculate the heat transfer coefficients of the plain internal surface, h_{Gni}.

3.4. Shell side heat transfer coefficients

Hence, shell-side heat transfer coefficient can be determined by Eq. (9):

$$h_o = \left[\frac{1}{k} - \frac{1}{h_i}\frac{D_o}{D_i} - R_w\right]^{-1} \tag{9}$$

3.5. Heat flux, film Reynolds number and average HTCs of single tube and the tube bundle

Heat flux in this paper is the area-averaged heat flux on the outer surface of the tube tested:

$$q = \frac{\phi_e}{A_o} \tag{10}$$

where ϕ_e is heat transfer rate of the tested tube, A_o is the outer surface area of the tested tube. For the tube bundle, the heat flux of tube No. 9 is taken as the heat flux of the tube bundle.

Film Reynolds number is defined by Eq. (11):
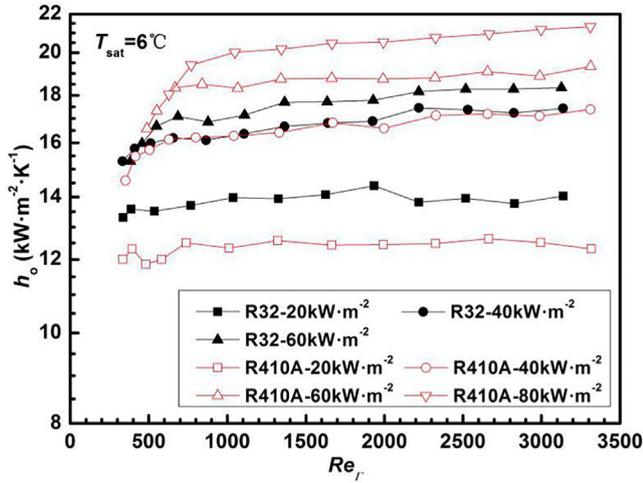
(a) Tube No.1
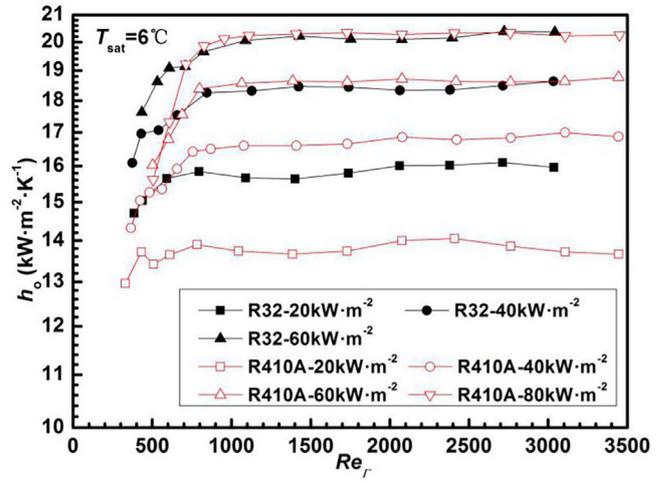
(c) Tube No.3
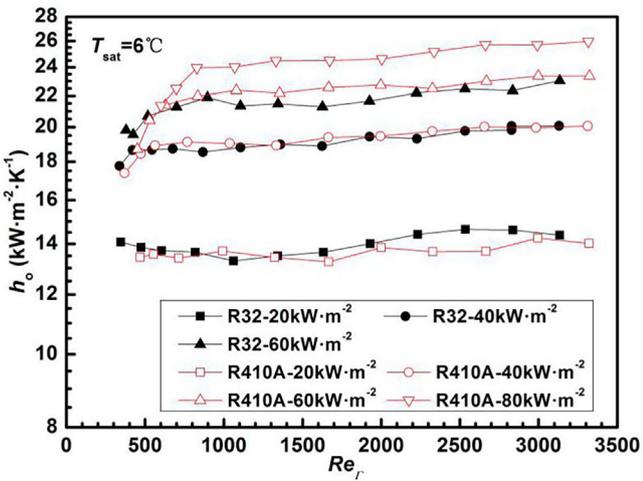
(b) Tube No.2
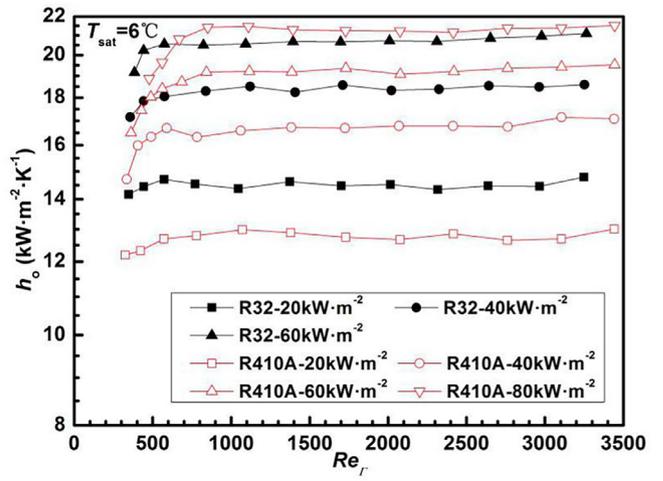
(d) Tube No.4

Fig. 9. Falling film evaporation HTCs of R410A and R32 on enhanced tubes.

$$Re_{\Gamma} = \frac{4\Gamma}{\mu_1} \tag{11}$$

where Γ (kg·m^{-1}·s^{-1}) is the film flow rate on one side of the tested tube per unit length, μ_l (kg·m^{-1}·s^{-1}) is the dynamic viscosity of liquid refrigerant. For the tube bundle, film Reynolds numbers of the tubes positioned in the first row are taken as the film Reynolds numbers for each single column.

Average HTCs of the tube bundle is the area-weighted average HTCs of the outer surface of the individual tested tube in the tube bundle.

3.6. Uncertainty analysis

Uncertainty of calculating Re_{Γ} depends on the precision of the mass flow meter (see Table 1), uncertainty of μ_l as well as the pressure drop between the condenser/reservoir and the evaporator. As indicated above, the difference between the two temperatures (pressure-corresponding saturation temperature and liquid temperature in reservoir) is within 0.1 °C which suggests that the boiling process is saturation boiling. Since the measurement uncertainty of mass flow rate is negligible, the uncertainty of Re_{Γ} is subjected to the uncertainty of dynamic viscosity of liquid refrigerant which is also quite small (usually less than 2%).

Uncertainty of h_o cannot be estimated directly because the outside thermal resistance was separated from the overall thermal resistance. So the uncertainties of h_o is estimated using the method suggested in

[29–32]. The estimated uncertainties of k are less than 3.5% for all test conditions. The accuracy for determining heat transfer coefficient on the tube side is quoted to be within 10% [33]. For all experimental data, the percentage of water side thermal resistance varied from 36% to 58%. Results of uncertainty analysis for h_o are shown in Table 4, and the maximum uncertainty of h_o is less than 25%.

4. Results and discussion

4.1. Water side enhanced factors

Enhanced factors of the internal surfaces were obtained by Wilson Plot Method in which the saturation temperature and heat flux are fixed and the internal water velocity is changed from 0.7 to 3.3 m·s^{-1}. In this study, the heat flux and saturation temperature were kept almost constant ($q = 40 \pm 0.5$ kW·m^{-2} and $T_{sat} = 6 \pm 0.02$ °C). For the enhanced tubes, k^{-1} vs. h_{Gni}^{-1}, is plotted and their relationship can be well described by a straight line, as seen in Fig. 5. The slope of a line is equal to $D_o/(c_i D_i)$, from which the enhanced factor c_i can be obtained.

Results of the single tube and the tube bundle are shown in Table 5. Since the 6 enhanced tubes in the tube bundle (tube No. 5–10) have the same enhanced structure on both sides of the tubes, the average value of their enhanced factors 3.23 is taken as the enhanced factor.
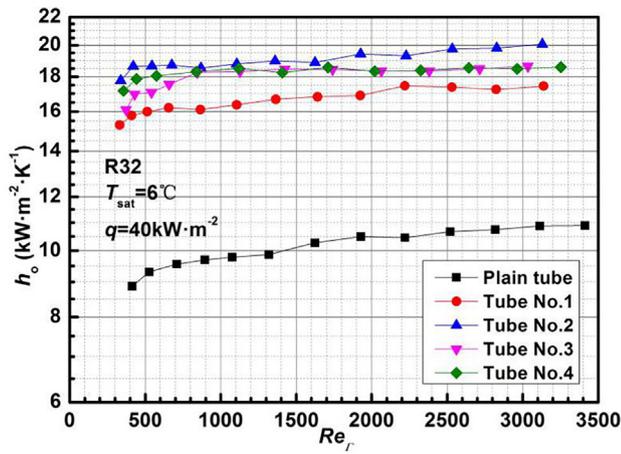
(a) R32
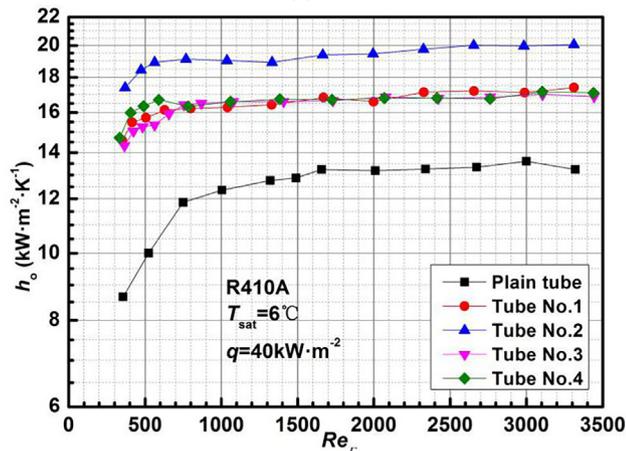
(b) R410A

Fig. 10. HTCs comparison among different tubes.

Fig. 12. HTCs of 410A on tube No. 9, 10.
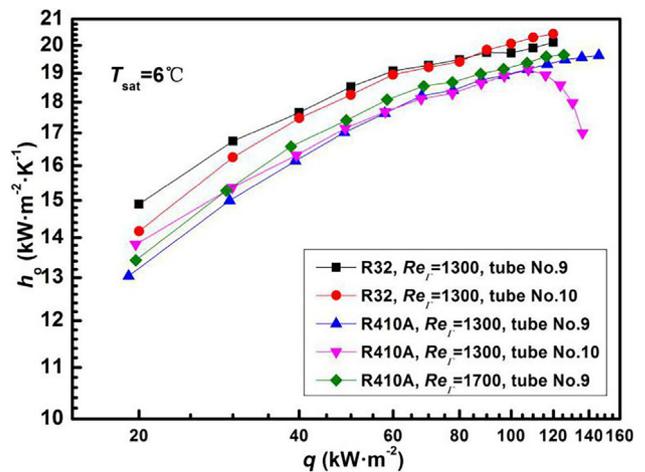
Fig. 13. Variation of HTCs with heat flux of tube No. 9, 10.

Fig. 11. HTCs of R32 on tube No. 9, 10.

4.2. Falling film evaporation HTCs of the plain tube

Fig. 6 shows the results for the plain tube of two refrigerants R32 and R410A. Experiments are carried out at two saturation temperatures and three heat fluxes. For each curve, there are two obvious stages, a quasi-plateau stage at higher film Reynolds number and a sharp decrease stage when film Reynolds number is lower than a certain value. The threshold film Reynolds numbers are located from 500 to 800 for

the cases studied. In addition, HTCs increase with heat flux monotonously for both refrigerants when $T_s = 6°C$. HTCs of R410A are about 20% higher than those of R32 for each working condition. At a typical working condition of an evaporator ($q = 40\,kW·m^{-2}$), HTCs at saturation temperature of 10 °C are inferior to those at 6 °C. Comparison with the experimental results of Zhao et al. [5] shows that HTCs of R32 and R410A are all larger than those of R134a. This phenomenon can be explained by larger liquid thermal conductivity, less viscosity and larger latent heat of these two refrigerants compared with R134a.

Comparison of HTCs between falling film evaporation and pool boiling is shown in Fig. 7. For R32, results of the present study are compared with the work of Jung et al. [34] as well as Cooper's correlation [35], and we can see from the figure that HTFs of pool boiling (PB) are higher than those of falling film evaporation (FFE), and the advantage of PB increases with increase of heat flux. However, the variation trends of PB HTCs with heat flux are similar with FFE which suggests that FFE is dominated by nucleate boiling for the cases studied. Film Reynolds number has negligible influence on HTCs of falling film evaporation within the heat fluxes tested. For R410A, the present results are compared with the experimental results of Dewangan et al. [36], and similar variation trends of HTCs with heat flux are also observed, while HTCs of FFE are higher than those of PB even the pool boiling experiments were carried out on tubes with coated surfaces. It is authors' consideration that PB is very sensitive to surface condition, and the test results shown above are taken from different authors groups, hence further researches at more comparable test conditions are needed
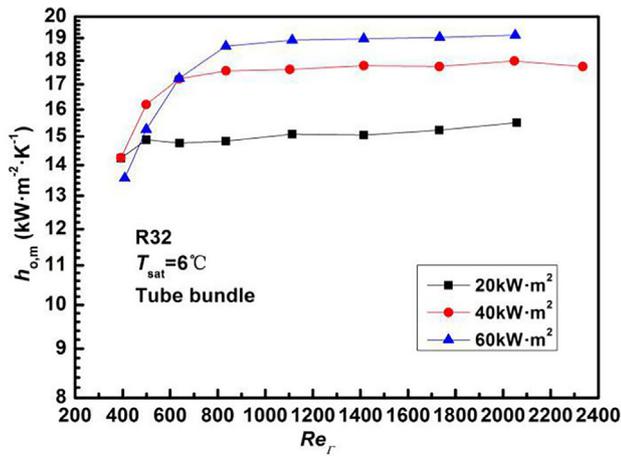
(a) R32
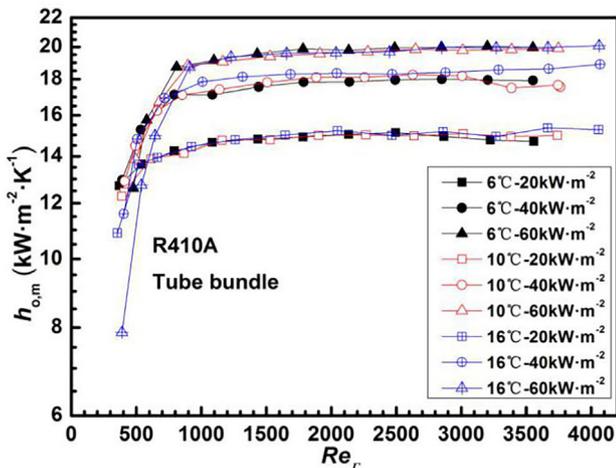
(b) R410A

Fig. 14. Average HTCs of the tube bundle.

to obtain more convincible comparison results.

Zhao et al. [5] have developed two correlations considering pretty influencing factors, and achieved success in predicting HTCs with several refrigerants. Two heat transfer regimes, full wetting regime and partial dryout regime as shown in Fig. 6, are identified in their paper to predict HTCs separately. Comparison between the predictive results by Zhao's correlation of full wetting regime and the present experimental data is shown in Fig. 8. Since the criteria of $Re_{threshold}$ in their paper cannot be applied to the present study, only the HTCs of apparent full wetting regime are compared. What should be pointed out here is that the adopted latent heat of vapor of R410A is the average value of its components R32 and R125.

It can be seen from Fig. 8 that for R32, the empirical correlation over predicts the HTCs more than 50% even up to 180%. While for R410A, the correlation can reasonably well predict the HTCs with more than 80% cases' deviation within ± 30%. The largest deviation happens at $T_{sat} = 16$ °C and heat flux of 20 kW·m^{-2}, and it may be resulted from relatively higher experimental uncertainty under this working condition. Consequently, the correlation must be studied further to extend its applicable range, including the high pressure refrigerants.

4.3. Falling film evaporation HTCs on different enhanced surfaces

Fig. 9 shows the variation of HTCs of the two refrigerants with film Reynolds number and heat flux on the four types of enhanced surface. As a whole, similar with the plain tube, there are two obvious stages for each curve, especially at higher heat fluxes.

By careful expecting (inspecting) the test data, following features may be noted. First, for all enhanced tubes tested, HTCs of the refrigerants increase with heat flux, exhibiting an appreciable feature of PB; Second, at low heat flux (20 kW·m^{-2}) almost all test data of the four enhanced tubes and two refrigerants are positioned in the quasi-plateau stage, indicating that the enhanced structures are helpful to delay the occurring of partial dryout; Third, at higher heat flux for all the four enhanced tubes R410A exhibits an earlier occur of the turning point from the quasi-plateau stage to the sharp decrease stage; Fourth, as far as the values of HTCs are concerned, tube No. 2 behaves the best. Its HTCs at heat flux of 80 W·m^{-2} for 410A and at 60 W·m^{-2} for R32 are almost equal to 24 W·m^{-2} K^{-1}and at 22 W·m^{-2}K^{-1}, respectively, being the highest HTCs in the test range of this paper.

It is interesting to compare the FFE HTCs of the smooth tube and the enhanced tubes. Fig. 10 shows the comparison of HTCs of the four enhanced tubes under a typical heat flux of 40 kW·m^{-2}. We can see that tube No. 2 exhibits the highest HTCs for both R32 and R410A. While tube No. 1 behaves the worst, even though for 410A, tubes No. 2 and 3 have similar heat transfer coefficients with Tube No. 1. What should be emphasized here is that tube No. 2 is designed for enhancing condensation heat transfer while performs best in the falling film evaporation process with nucleate boiling.

4.4. Falling film evaporation heat transfer characteristics in a tube bundle

Experiments are also carried out for a tube bundle in staggered triangular arrangement comprised of 6 enhanced tubes as shown in Fig. 4. In the following, the HTCs of a single row (tubes No. 9, 10) are presented first and then the HTCs of the whole bundle at three heat fluxes are provided.

Fig. 11 shows HTCs of R32 for tube No. 9 and No. 10. Each curve exhibits two stages as discussed above. Under the same saturation temperature and the same nominal heat flux, HTCs of tube No. 9 are slightly higher than tube No. 10 in the full wetting regime, while in the partial dryout regime, HTCs of the lower tube decrease more sharply. The film Reynolds number of the turning points of the upper tube is a bit larger than that of the lower tube. It can also be found from Fig. 11 that the effect of saturation temperature on heat transfer is negligible for enhanced tubes. Experimental data of R410A are shown in Fig. 12. In the quasi-plateau regime the HTCs of the upper and lower tubes are almost the same, while in the partial dryout regime the lower tube exhibits an earlier turning point and a sharper decrease in HTCs.

HTC variations with heat flux for tube No. 9 and No. 10 are shown in Fig. 13. It can be found that the HTCs increase with heat flux implying nucleate boiling dominated heat transfer mode. For each tube R32 performs appreciably better than R410A, while for each refrigerant, the HTCs of the upper tube is a bit larger than those of the lower one. It worth noting that for R410A boiling on tube No. 10 at film Reynolds number of 1300 dryout occurs when heat flux is larger than 110 kW m^{-2}. This is because at the same test condition, the actual film flow rate on tube No. 10 is less than that of tube No. 9, leading to an earlier occurrence of the partial dryout regime.

Tube bundle averaged HTCs of R32 and R410A are shown in Fig. 14. Two significant features may be noted. First with the increase in heat flux, both the averaged HTCs and the film Reynolds number of the turning point increase; Second the saturation temperature has negligible influence on FFE HTCs.

5. Conclusions

Heat transfer characteristics of falling film evaporation with nucleating boiling are experimentally studied in this paper for high pressure refrigerant R32 and R410A on four doubly enhanced tubes. The outside surface of one tube is of integrated fin while all the other three tubes have 3-D micro-fin structure. Following conclusions can be drawn:

1. HTCs of falling film evaporation exhibit two fundamental stages with decrease of film flow rate, a quasi-plateau stage where film Reynolds has very mild influence on heat transfer coefficient and a sharp-decrease stage where dryout occurs because of the insufficient liquid supply.
2. For both refrigerants, HTCs increase with heat flux progressively before reaching the partial dryout regime. The effect of saturation temperature on the enhanced tubes as well as the tube bundle is negligible. While for smooth tube HTCs decrease with the increase of saturation temperature.
3. The integrated-fin tube performs best among the four enhanced tubes tested for both refrigerants at the typical working condition of 6 °C and 40 kW·m^{-2}. As a whole, the HTCs of R32 are larger than those of 410A, while for the smooth tube the result is the opposite.
4. Tubes positioned in the second row possess the similar variation trend of HTCs with that of the first row when no dry patches occur, however they suffer an earlier dryout because less flow rate supply compared with the upper ones.
5. Correlation of Zhao et al. [5] for smooth tube can predict the experimental data of R410A of smooth tube quite reasonably with 81.3% of 91 points' deviation within ± 30%, while it appreciably over-predicts the HTCs of R32.

Acknowledgment

Appendix A. Supplementary material

Supplementary data associated with this article can be found, in the online version, at http://dx.doi.org/10.1016/j.applthermaleng.2018.03.060.

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