

Entropy generation and thermodynamic analysis of pool boiling heat transfer on doubly enhanced tubes

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List of symbols

Nomenclature			
A	area, m ²	u	velocity, m/s
C	coefficient of Cooper equation	Subscripts	
d	diameter, m	ET	doubly enhanced tube
e	fin-depth, mm	ST	smooth tube
f	fraction factor	f	fluid
h	heat transfer coefficient, kW/(m ² ·k)	fou	fouling
i	the enthalpy, kJ/kg	gen	generation
k	the total heat transfer coefficient, kW/(m ² ·k)	i	inside of tube
L	the tube's length, m	o	outside of tube
m	mass flow rate, kg/s	l	liquid
P	pressure, Pa; perimeter, m	v	vapor
Pr	Prandtl number	r	refrigerant
q	heat flux, kW/m ²	s	saturation
Q	heat transfer rate, kW	w	wall
Re	Reynolds number	Greek symbols	
S	the entropy generation, W/(m·K)	γ	the latent heat, kJ/kg
T	temperature, K	λ	thermal conductivity, W/(m·K)
x	vapor quality	μ	dynamic viscosity, Pa·s
r	Thermal resistance, m ² /kW	ρ	density, kg/m ³
		σ	surface tension, N/m

ABSTRACT

To improve the efficiency of the heat exchanger tube, one smooth tube(ST) and four doubly enhanced tubes (EX1, EX2, EX3 and EX4) have been investigated for the pool boiling heat transfer experiments in this paper. The indicate that the pool boiling heat transfer coefficient of the doubly enhanced tubes increased visibly with the augmentation of heat flux through the bubble behavior. Heat transfer reinforcement effect of the doubly enhanced tubes are significantly better than that of the ST. Additionally, pool boiling heat transfer coefficient can be further enhanced by raising the saturate temperature. Entropy generation minimization analysis demonstrates that the heat transfer characteristics of like T-shaped tubes (EX1 and EX2) are superior to that of low fin tubes (EX3 and EX4). Particularly, tube EX1 exhibits higher pool boiling heat transfer efficiency. It is observed that a reasonable fin pitch is more advantageous for improving heat transfer characteristics. The utilization of entropy generation minimization analysis provides theoretical support for the design and optimization of doubly enhanced tubes.

Keywords: Enhanced structure; Pool boiling; Nucleation site; Entropy generation minimization

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1. Introduction

Pool boiling is a highly effective method for dissipating high heat at low temperature gradients through latent heat, making it the primary heat transfer mechanism in flooded evaporators. Enhanced surface technology, a widely used passive enhancement technique for pool boiling heat transfer, significantly increases nucleation sites on the superheated surface^[1-3]. Optimized surfaces play a crucial role in improving heat exchanger efficiency. The heat transfer evaluation criteria rely on the first law of thermodynamics. To further understand heat transfer mechanisms concerning the irreversibility caused by the heat transfer process, it is crucial to analyze based on the second law of thermodynamics.

Numerous studies have analyzed the heat transfer mechanism of enhanced tubes to achieve optimal heat transfer performance utilizing mechanical processing^[4-6]. Siddharth^[7] characterized the evolution of the contact angle as the bubble expands and the substrate contracts prior to the departure of the bubble. Regarding the bubble morphology during boiling in the off-nuclear state, simulation analysis indicates a transition in the shape of bubbles from hemispherical to bubbly. The theoretical boiling models depicted in Fig. 1, which visually illustrates the mechanism of pool boiling heat transfer. Transient conduction is one of the most essential mechanism, as superheated liquid transiently conducts in the main region. A microlayer forms on the superheated surface at the bottom of bubbles, with low thermal resistance. The fluid near the three-phase is evaporated significantly, which can promote the growth of bubbles. Under buoyancy, the bubbles detach and release heat. It generates a temperature gradient, significantly enhancing heat transfer by thermal boundary layer fluid. Additionally, part of the heat on the superheated surface is transferred to the fluid by natural convection, reducing heat conduction resistance and enhancing significantly heat transfer.

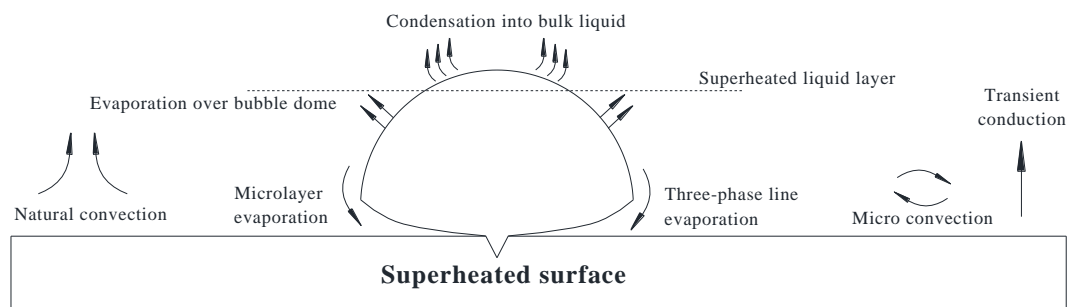


Fig.1 Theoretical pool boiling mechanism of heat transfer

The effects of several enhancement methods on pool boiling heat transfer have been investigated. Ribatski and Thome^[8] studied the HTC at a heat flux range of 20-70kW/m² on tubes GEWA-B, Turbo-CSL, and Turbo-BII. The three types of enhanced tubes had higher HTC, with heat transfer enhancement coefficient ratio of 2.4-5.2, 2.4-2.9 and 1.8-7.0, respectively. Ji et al.^[9,10] experimentally investigated the boiling heat transfer of three types of enhanced tubes. With the same structure, the ones with larger diameter produced more nucleate sites, which contributed to heat transfer efficiency. The reentrant cavity structure made it easier to enhance boiling heat transfer at specific heat flux. The heat transfer was not enhanced significantly when heat flux was further increased. Three types of fin structures of R410A pool boiling heat transfer have been explored by Li et al^[11] with mass flow ranging from 50 to 80kg/m². They found that the microporous tube HTC was higher than that of the integral fin tube, owing to the increased density

of the nucleation site. The small convex tube HTC was lower than that of the smooth tube owing to local dryness at high vapor quality.

Entropy generation analysis is an alternative method to assess irreversible losses during the heat transfer^[12–14]. Based on the second law of thermodynamics, Sahiti^[15] analyzed experimentally the entropy generation of heat exchangers with different flow lengths and pin lengths, In their study, the pin fin pipe was optimized based on entropy generation minimization. Dagtekin^[16] investigated the effect of three types of fins on entropy generation under the circular duct. The increase in the number and dimensionless length of thin and triangular fins had increased the entropy generation. Meanwhile, the increments in the angle of V-shaped and triangular fins had also increased significantly. Moghadasi et al^[17] developed a pool boiling model and validated its accuracy through experimental data. The analysis revealed the influence of parameters such as wall superheat, contact angle, and surface diameter on entropy generation. Ali^[18] conducted numerical simulations using CFX to analyze turbulent flow and heat transfer within an internally ribbed. The study investigated the influence of channel height, Prandtl number, diameter, and the arrangement of spot welds on entropy generation resulting from thermal effects and friction.

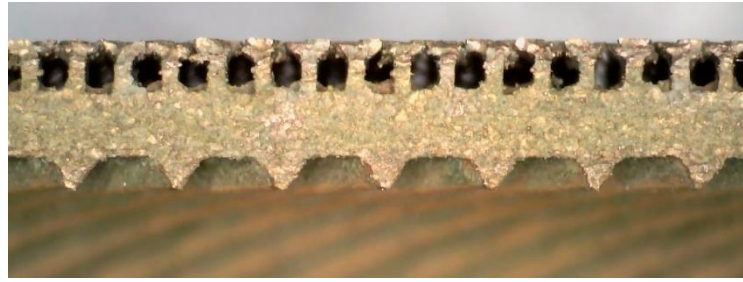
In summary, different enhanced structures have a huge impact on the pool boiling heat transfer performance. It is important to study the optimization of enhancement structures for tubes. However, there is limited research on the entropy generation analysis of fin structures under pool boiling conditions. Therefore, based on the experimental data of two typical doubly enhanced tubes with different fin pitches, this study employs the entropy generation minimization method to comprehensively analyze the effects of e/d_o (fin height to tube diameter ratio) and p/d_o (fin pitch to tube diameter ratio) on pool boiling heat transfer and entropy generation. Additionally, the optimum structure of doubly enhanced tubes are proposed. The utilization of entropy generation minimization analysis provides theoretical support for the design and optimization of doubly enhanced tubes. This experimental analysis could provide reference for the optimization of commercial enhanced tubes.

2. Experimental methods

2.1 Test section

In this experiment, a smooth tube(ST) and four doubly-enhanced tubes (EX1, EX2, EX3, and EX4) were employed for experimental research. ST ($d_o=19.05\text{mm}$) was used as the control group. Fig.2 shows the amplification sections of the four doubly enhanced tubes. The fins of EX1 and EX2 are 3D surfaces processed on the smooth tube with tiny opening recess, in which the recess diameter is larger than the opening size. The fin structures of EX1 and EX2 are roughly T-shaped, with the same e/d_o of 0.027 yet different p/d_o (the fin pitch to tube diameter ratio) of 0.029 and 0.026 respectively. The tubes EX3 and EX4 are typical integral fin tubes that, generally have the same low fin, with the same e/d_o of 0.041 but different p/d_o of 0.029 and 0.033 respectively. Four doubly enhanced tubes have triangular internal threads where fin height is about 0.35mm and fin pitch is approximately 1.18mm.

The test length is 2000mm. Table 1. lists the specific dimensional parameters of the five test tubes. Table2 illustrates the thermophysical properties of the refrigerant R410A under the test conditions. In this experiment, the working conditions are as follows: the Reynolds number inside the tube is kept constant, Saturate temperature are 5°C, 7°C and 10°C, respectively. The heat flux ranges from 10 to 60kW/m². Pool boiling heat transfer performance of the five test tubes are investigated.



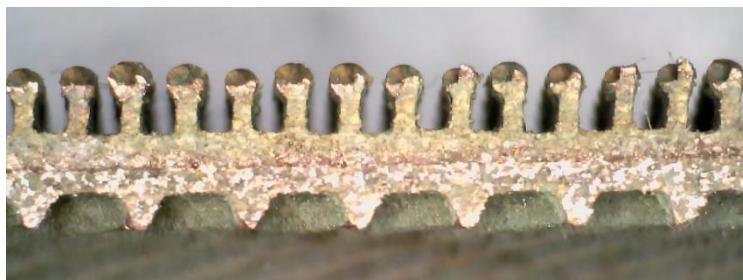
EX1



EX2



EX3



EX4

Fig2. Amplification section of four doubly-enhanced tubes

Table 1. Specific dimensional parameters of the test tubes

Tube	Outside					thickness δ (mm)	Inside		
	diameter d_o (mm)	fin-depth e (mm)	fin-pitch p (mm)	e/d_o	p/d_o		diameter d_i (mm)	fin-height (mm)	fin-pitch p_i (mm)
ST	19.05	/	/	/	/	0.752	18.298	/	/
EX1	19.05	0.52	0.54	0.037	0.029	0.565	17.964	0.347	1.175
EX2	19.05	0.52	0.50		0.026	0.599	17.928	0.363	1.176
EX3	19.05	0.78	0.55	0.041	0.029	0.792	17.479	0.325	1.187
EX4	19.05	0.78	0.63		0.033	0.774	17.494	0.405	1.188

Table 2. thermophysical properties of R410A

T_s (°C)	γ (kJ/kg)	λ_l (W/m·K)	C_{pl} (kJ/kg·K)	μ_v ($\mu\text{Pa}\cdot\text{s}$)	μ_l ($\mu\text{Pa}\cdot\text{s}$)	ρ_v (kg/m^3)	ρ_l (kg/m^3)	σ (N/m)
5	215.07	0.1002	1.547	12.454	152.14	1149.6	35.863	0.0082
7	212.48	0.0991	1.560	12.564	148.44	1141.2	38.187	0.0079
10	208.50	0.0974	1.579	12.731	143.02	1128.4	41.917	0.0074

2.2 Experimental apparatus

The schematic diagram of the horizontal single tube boiling heat transfer experimental system is shown in Fig. 2. The test bench consists of three circulating systems, refrigeration cycle system, water cycle system and ethylene glycol(EG) cycle system. After the liquid is sent to the preheater by the diaphragm pump, it enters the lower side of the experimental test section and evaporates into the vapor, which is then discharged from the upper side of the experimental test section. Water functions as a heat source for the test section. The inlet temperature of water is modulated by electrical heating power. The temperature of the thermostatic water tank in EG cycle is constant at -5°C to absorb the heat of the saturate refrigerant. Details of experimental equipment of the experimental system are shown in Table 3.

The saturation pressure and water mass flow rate are kept constant. The heat flux and vapor quality of refrigerant are changed during the test. The deviation of temperature is $\pm 0.1^\circ\text{C}$, and the deviation of pressure is within $\pm 0.1\%$ of the full range in the test section as the system balance can be recorded for the experiment data.

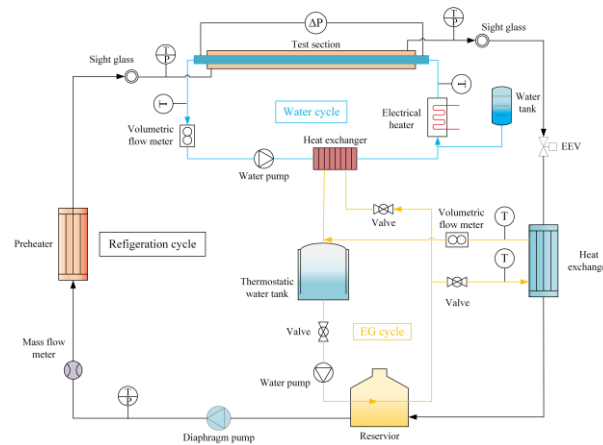


Fig. 3 Schematic diagram of horizontal single tube experimental system.

Table3. Details of experimental equipment

Equipment	Model	Parameters
Diaphragm pump	SJ3-M-200/2.8	Flow range: 0-200L/h; Precision: $\pm 2\%$.
Mass flow meter	DMF-1-2-A	Flow range: 0-150kg/($\text{m}^2\cdot\text{s}$); Precision: $\pm 0.1\%$.
Volumetric flow meter	HMLBG-DN103	Flow range: 0.2-4.2 m^3/h ; Precision: $\pm 0.5\%$.
Electronic expansion valve(EEV)	E3V55SSR10d	Step range: 0-480; Pressure range:0-30bar.
Preheater	BL14-36D	Rated heat exchange:15kW
Heat exchanger	BL14-34D /28D	Rated heat exchange:15kW
Water pump	CHL2-20/40	Rated flow: 2m/h; Rated head: 15m or 29m

3. Performance evaluation

3.1 Heat transfer rate and heat balance

Heat transfer rate, determined by release by heating water and absorption by chilling refrigerant, is expressed as:

$$Q_r = m_r(i_o - i_i) \quad (1)$$

$$Q_w = m_w c_p (T_i - T_o) \quad (2)$$

Where T_i and T_o are the water temperatures of inlet and outlet respectively, i_i and i_o are the refrigerant enthalpy of inlet and outlet, respectively, and m_r and m_w are the mass flow rates of chilling refrigerant and heating water respectively.

The total heat transfer coefficient(HTC) can be calculated in Eq(3)

$$k = \frac{q}{\Delta T_m} \quad (3)$$

Where q is the heat flux defined in Eq. (4), and ΔT_m is the logarithmic mean temperature difference in tested section tubes by refrigerantsaturate temperature, as defined in Eq. (5):

$$q = \frac{Q_r + Q_w}{2A_o} \quad (4)$$

$$\Delta T_m = \frac{|T_i - T_o|}{\ln|(T_i - T_s)/(T_o - T_s)|} \quad (5)$$

Where A_o is the external surface area of tested tube, and T_s is the saturate temperature.

The heat transfer inside the test tube is defined by Gnielinski Equation^[19]:

$$Nu = \frac{(f/8)(Re-1000)Pr_f}{1+12.7\sqrt{f/8}(Pr_f^{2/3}-1)} \left[1 + \left(\frac{d_i}{L}\right)^{2/3} \right] \frac{Pr_f^{0.11}}{Pr_w} \quad (7)$$

$$Re = \frac{\rho u d}{\nu} \quad (8)$$

$$f = \frac{2\Delta P d_i}{L\rho u^2} \quad (9)$$

$$h_i = \frac{Nu\lambda}{d_i} \quad (10)$$

Wilson plot method is a suitable technique to investigate the pool boiling HTC, which is based on the thermal resistance separation method in the heat transfer of tested tube. The pool boiling HTC of doubly enhanced tube is defined as:

$$\frac{1}{h_o} = \frac{1}{k} - \frac{A_o}{A_i} \frac{1}{h_i} - r_w - r_{fou} \quad (11)$$

Where h_o and h_i are the HTC of external and internal tested tube, respectively, A_i is the internal surface area of tested tube, k is the total HTC, r_w is the thermal resistance of tube wall, and r_{fou} is the thermal resistance for fouling.

3.2 Entropy generation analysis

The heat transfer evaluation criteria above are based on the firstlaw of thermodynamics. To further understand the mechanism of heat transfer enhancement concerning the irreversibility caused by the process of heat transfer, it is necessary to perform analyses based on the second law of thermodynamics. The entropy generation per unit length S'_{gen} in a finite control volume of length dz is developed by Revellin et al^[21] as follow:

$$S'_{gen} dz = m[s_v dx + x ds_v + (1-x)s_1] - Q/T_w \quad (12)$$

The values of di_v and dh_l can be determined in Eq.(14) and Eq.(15)

$$di_v = T_v ds_v + v_v dp_v \quad (13)$$

$$di_1 = T_1 ds_1 + v_1 dp_1 \quad (14)$$

By assuming $T_s=T_v=T_l$ and $dP=dP_v=dP_l$ in saturated boiling flow, the value of s_{lv} is derived in Eq.(15)

$$s_{lv} = i_{lv}/T_s \quad (15)$$

By simplifying expression Eq.(12) :

$$S'_{gen} dz = \frac{m}{T_s} (i_{lv} dx + x di_v + (1-x) di_l) - \frac{Q}{T_w} - \frac{m(xv_v + (1-x)v_l)}{T_s} dp \quad (16)$$

Therefore, the equation for determining the entropy generation per unit length by energy transfer can be calculated below:

$$S'_{gen} = q^2 P / h T_w T_s \quad (17)$$

Moreover, the entropy generation number(N_s) is expressed as the ratio of entropy generation for doubly enhanced tubes to entropy generation for the smooth tube.

$$N_s = S'_{gen,ET} / S'_{gen,ST} \quad (18)$$

Table 3 shows that the uncertainty analysis of experimental calculation parameters is performed to simplify the results. According to previous studies^[22,23], the uncertainties of h_o , h_i and K were about 14.91%, 14.13% and 5.98%, respectively. The estimated uncertainty of entropy generation is $\pm 2.54\%$ to $\pm 5.72\%$.

Table 3 Uncertainty of parameters

Parameter	uncertainty
Diameter	± 0.1 mm
Length	± 0.5 cm
Temperature	± 0.1 K
Pressure	$\pm 0.1\%$
Heat flux	$\pm 0.5\%$
HTC	$\pm 5.98\%$ to $\pm 14.91\%$
Entropy generation	$\pm 2.54\%$ to $\pm 5.72\%$

4. Result and discussion

4.1 Validation of the experimental system

To test the accuracy of this experimental system, the pool HTC of the Cooper equation(19) was compared with current data and the results from other studies^{[9][23]} as shown in Fig.4. The authors are grateful to the reviewer for the valuable suggestions. In this paper, the experimental data of Ji^[9] and Kedzierski^[23] are added, and the specific experimental conditions are shown in Table 4. The deviation of the experimental data of Ji^[9] and Kedzierski^[23] from Cooper's formula is about 15%, which verified the reliability of Cooper's equation. The relative deviation of $\pm 20\%$ between experimental and theoretical values indicates that the present experimental results are reliable. The Cooper equation^[24] is expressed as:

$$h_{op} = Cq^{0.67} M_r^{-0.5} p_r^m (-\log p_r)^{-0.55} \quad (19)$$

$$C = 90W^{0.33} / (m^{0.66} \cdot K)$$

$$m = 0.12 - 0.21 \log \{ R_p \} \mu m$$

Where h_{op} is heat transfer coefficient of refrigerant side outside the tube, kW/(m² · K); C is empirical coefficients for surface-liquid; M_r is relative molecular mass of liquids; p_r is the ratio of the pressure of a liquid to the critical pressure of that liquid; m is empirical coefficient; R_p is

average roughness of the surface.

Table 4. The specific experimental conditions

Authors	Saturate temperature	Refrigerant	Test section
Ji ^[9]	6 °C	R134a	Smooth tubes with an outer diameter of 19.06 mm
Kedzierski ^[23]	5 °C	R1336mzz(Z)	Turbo-ESP surface

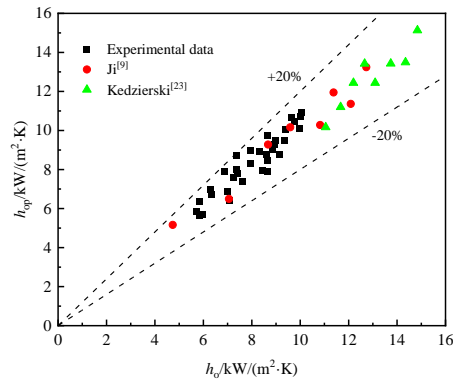


Fig.4 Comparison of experimental HTC in ST with Cooper equation

4.2 Effect of work condition on pool boiling heat transfer coefficient

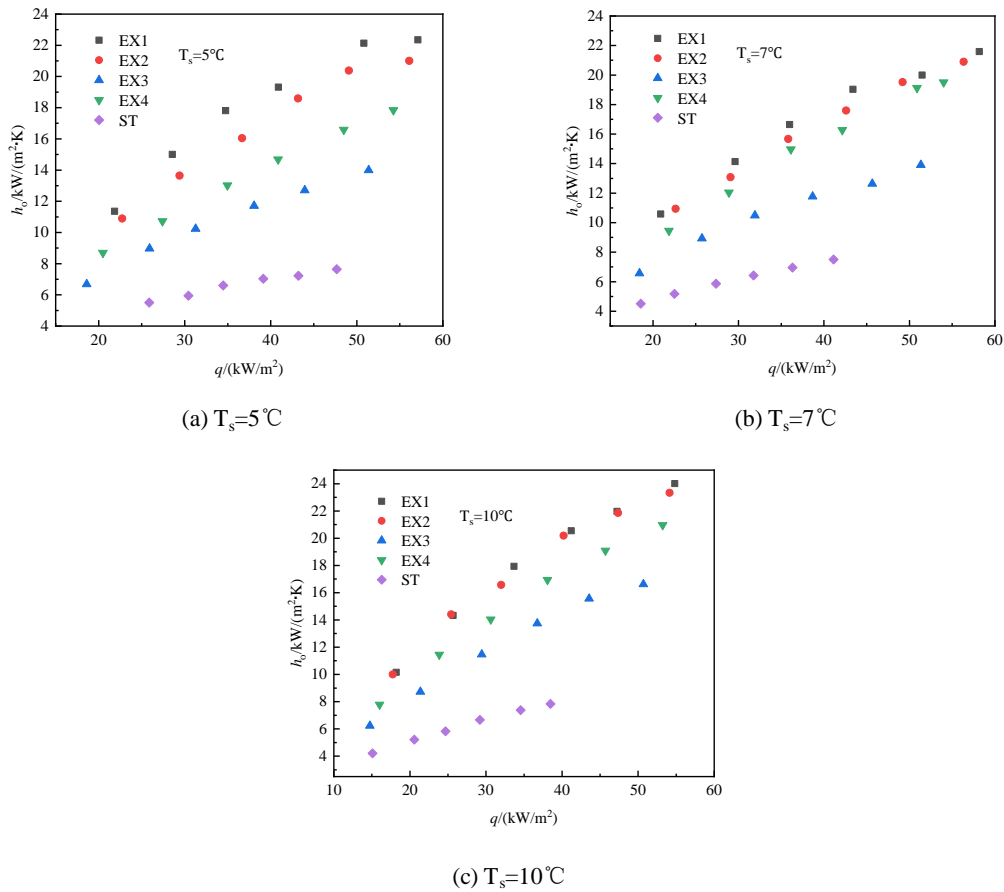


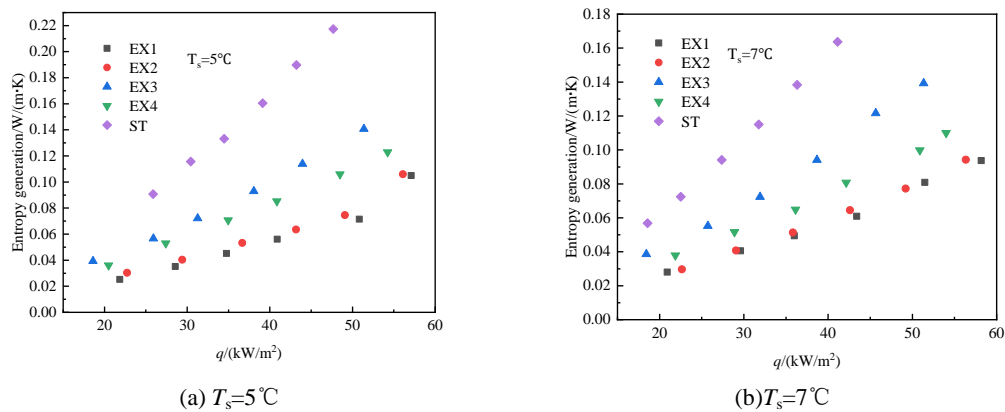
Fig.5 Pool boiling HTC versus heat flux for tubes at saturate temperatures of 5 °C 、 7°C and 10°C

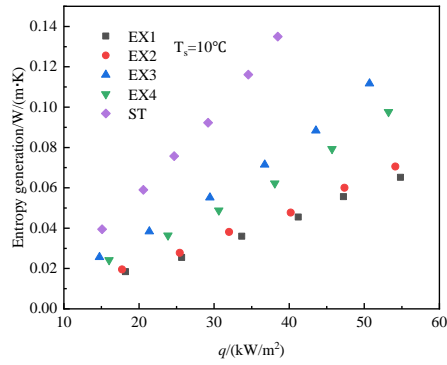
Fig. 5 illustrates the pool boiling HTC against heat flux at saturated temperatures of 5 °C, 7°C,

and 10°C, respectively. From Fig.5(a), the difference in pool boiling HTC of the doubly enhanced tube and the ST is small at low heat fluxes, due to the fact that few bubbles are produced in each tunnel. The reinforcement ratio of the pool boiling HTC is about 1-1.5 times, and thereby, the pool boiling heat transfer of doubly enhanced tubes outperforms that of the tube ST. Heat is efficiently conducted through the microlayer, leading to fluid vaporization. Pool boiling is primarily enhanced at the liquid-vapor interface with the increment of heat flux. Heat flux and pool boiling heat transfer coefficient are positively correlated. The nucleation rate of bubbles demonstrates a substantial increase. Influenced by inertia force, nucleation sites on the upper surface of the horizontal tube are larger than those on the bottom surface. Consequently, a large amount of bubbles coalesce and cover the superheated surface. However, there is limited heat transfer from the fluid to the superheated surface, which leads to an insignificant improvement in the pool boiling HTC with heat fluxes greater than 40kW/m². In short, the nucleate boiling effect is obviously enhanced to a certain extent with the increase of heat flux. Regarding the enhancement of the nucleate pool boiling, the doubly enhanced tubes with reentrant cavity and tunnels produce much more bubbles than the ST owing to the potential ability to trap more nucleation sites. The enhanced tubes are formed by upgrading the surface structure of the smooth tube with an increase in the heat transfer surface area and the bubble dynamics to improve heat transfer performance.

As seen in Fig. 5(a)-(c), the pool boiling HTC at saturate temperature 10°C is higher than that of 5°C. The growth and separation of bubbles are influenced not only by enhanced surface also by physical properties, such as surface tension, viscosity and latent heat. As shown in Table 3, the latent heat is reduced slightly with the increment of saturate temperature, bubbles are produced more quickly. Lower latent heat promotes transient conduction, thereby augmenting boundary layer perturbation. A decrease in surface tension requires a smaller minimum size for bubble generation and leads to an increase in the number of nucleate site. In addition, when liquid viscosity is reduced, bubbles are more easily separated from the surface.

4.3 Effect of geometric structure on entropy generation





(c) $T_s=10^\circ\text{C}$

Fig.6 Entropy generation versus heat flux for tubes at saturate temperatures of 5°C 、 7°C and 10°C

The changes in entropy generation versus heat flux for the tubes at saturate temperatures of 5°C , 7°C and 10°C are presented in Fig. 6(a)-(c). The doubly enhanced tubes produce the formation of bubbles within the tunnel at a specified heat flux. Importantly, the doubly enhanced tubes exhibit substantially lower irreversible losses in comparison to the ST. The reduction ratio of the entropy generation ranges from 0.9 to 1.9 times. Additionally, Fig.7 shows the heat flux versus superheat for the doubly enhanced tubes at saturate temperature of 10°C . With increasing superheat, there is a significant rise in heat flux, accompanied by a noticeable increase in irreversible losses within the heat transfer process, leading to a visible growth in entropy generation.

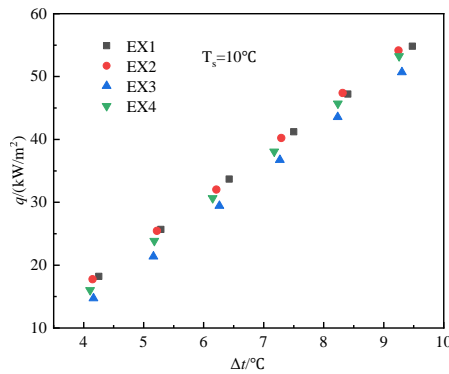


Fig.7 Heat flux versus superheat for doubly enhanced tubes at saturate temperature of 10°C

To gain a deeper understanding of pool boiling heat transfer performance on doubly enhanced tubes, fin structure parameters such as depth and pitch were examined based on entropy generation minimization analysis. EX1 and EX2 tubes have the same external fin structure resembling a T-shape, with the same e/d_o of 0.027 but different p/d_o of 0.029 and 0.026, respectively. EX3 and EX4 tubes feature generally the same external low fin, in which e/d_o are both 0.041 but p/d_o are different, 0.029 and 0.033 respectively. Variances in nucleate site stems from distinct enhanced surfaces, thus affecting the boiling heat transfer performance. The cavity opening mouth of the doubly enhanced tube is smaller than the reentrant cavity, which increases the nucleate site and promotes bubble aggregation between the fins. The fin pitch of EX1 tube is smaller than that of EX2 tube, while the fin pitch of EX3 tube is larger than that of EX2 tube. From Fig. 6(a)-(c), the entropy generation of EX1 tube is lower than that of EX2 tube, which indicate that the heat transfer efficiency of EX1 tube is more adequate. The entropy generation of EX4 tube is lower than that of EX3 tube, which suggests that the heat transfer efficiency of EX4 tube is more sufficient. Excessively thin or dense fin pitch may limit the formation, growth, and

detachment of bubbles, thus further reducing boiling heat transfer. It also indicates that the augmentation of fin pitch has certain restrictions on pool boiling heat transfer enhancement. EX1 and EX3 tubes have different cavity shapes while the fin pitch ($p/d_o=0.029$) is the same. EX1 tube has a smaller cavity opening diameter and a relatively larger heat transfer surface area, resulting in entropy generation in EX1 lower than EX3 tube. The heat transfer performance of EX1 is higher than that of EX3 at constant heat flux. Reasonable fin pitch is more conducive to reducing the irreversible losses of the heat transfer process based on the principle of entropy generation minimization.

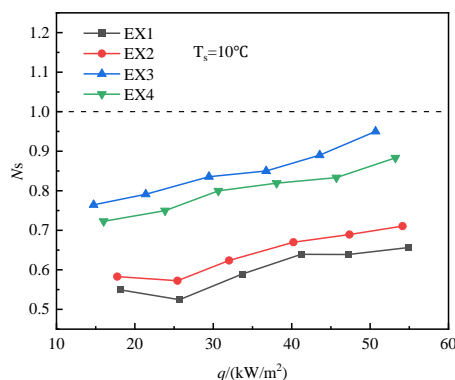


Fig.8 Entropy generation numbers versus heat flux for doubly enhanced tubes at saturate temperature of 10°C.

Fig.8 presents the entropy generation numbers versus heat flux for the enhanced tubes at saturate temperature of 10°C. The total entropy generation in the doubly enhanced tubes is smaller than that of the ST. The total entropy generation in doubly enhanced tubes is smaller than that of the smooth tube when $N_s < 1$, which makes the utilization of doubly enhanced tubes desirable in the condition. Alternatively, the total entropy generation of doubly enhanced tubes is larger than that of the smooth tube when $N_s \geq 1$, and thus using the smooth tube is more reasonable in this case. As seen in Fig. 8, $N_s < 1$ indicates that heat transfer efficiency of the four doubly enhanced tubes are higher than that of the ST. Heat transfer characteristic of T-shaped tubes (EX1 and EX2) are higher than that of low fin tubes (EX3 and EX4) at constant heat flux. With the T-shaped tube being enhanced by notching on the ST, the pool boiling heat transfer performance in T-shaped tube is better than that of the low fin tube. In the case of a similar structure surface, regarding the effect of longitudinal fin pitch, the nucleate site of tube EX1 ($p/d_o=0.029$) with sparser fins is more than that of tube EX2 ($p/d_o=0.026$), while the nucleate site of tube EX4 ($p/d_o=0.033$) with denser fins is less than that of tube EX3 ($p/d_o=0.029$) per unit area. Due to the excessively small fin pitch of the tube EX3, the fin overcrowding restricts heat transfer on the local superheated surface, potentially resulting in dry spots and deteriorating boiling heat transfer performance. The pool boiling heat transfer of EX1 is superior to that of other three doubly enhanced tubes. Meanwhile, suitable fin pitch is beneficial to the improvement of boiling heat transfer performance.

Conclusions

By analyzing the change trend of pool boiling HTC and the entropy generation of test tubes at different heat flux, the pool boiling heat transfer characteristics of different enhanced structures on horizontal tubes were studied. The following conclusions are reached:

(1) The heat flux and saturate temperature have a positive correlation with the pool boiling heat transfer characteristics within a certain range. Moreover, bubbles are more prone to detachment

with lower surface tension and dynamic viscosity.

(2) Entropy generation minimization analysis demonstrates that heat transfer efficiency of the four doubly enhanced tubes are higher than that of the ST. Pool boiling heat transfer of EX1 is superior to that of other doubly enhanced tubes.

(3) The heat transfer of T-shaped tubes (EX1 and EX2) are higher than that of low fin tubes (EX3 and EX4) at constant heat flux. Furthermore, a reasonable fin pitch is more beneficial to improving the heat transfer performance.

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