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Local flow boiling heat transfer characteristics in three-dimensional enhanced tubes

J. Chen [a,b], Wei Li [a,*]

[a] Department of Energy Engineering, Zhejiang University, Hangzhou 310027, China
[b] Co-Innovation Center for Advanced Aero-Engine, Department of Energy Engineering, Zhejiang University, 310027, China

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ABSTRACT

Experimental study on local flow boiling heat transfer characteristics in two three-dimensional dimple-grooved tubes (2EHT) and an equivalent smooth tube was performed. All the three test copper tubes have the same inner diameter of 11.07 mm and outer diameter of 12.7 mm. The working fluid was the near-azeotropic mixture, R410A. All test runs were conducted in a 2 m long horizontal tube-in-tube heat exchanger. Constant evaporation temperature was maintained at 10 °C when heat flux ranged from 32.6 kW/m^2 to 37 kW/m^2. Refrigerant quality varied from inlet 0.1 to outlet 0.9 and mass flux was controlled from 70 kg/(m^2 s) to 150 kg/(m^2 s). The enhanced surface areas of two 2EHT tubes are 1.02 and 1.03 times of the smooth tube, respectively. The local HTC results of saturate flow boiling were presented and compared with the existing correlations. Wall temperature was measured and critical heat flux effect on the tube-side evaporation was discussed. Local wall temperature results showed that the 2EHT tubes produce a better saturate evaporation at a relatively lower wall superheat. HTC increases with increasing heat flux when heat flux is smaller than CHF and decreases when heat flux is larger than CHF. Based on the local thermal properties, a new flow boiling heat transfer correlation for these 2EHT tubes is derived. The new correlation can predict all the experimental heat transfer coefficients within an error band of ±30% and 96.89% of test data within an error band of ±20%.

1. Introduction

Enhanced flow boiling systems are commonly used in aerospace application, refrigeration, air conditioning, micro and macro electronic devices and other thermal control systems to cool the heat source. Enhanced tubes such as dimple tubes, micro-fin tubes, surface coating tubes and other modified surface tubes are used to improve the heat transfer performance [1]. Li and Wu [1], Kim [2] and Jiang et al. [3] studied the flow boiling heat transfer characteristics in micro-fin tubes using different refrigerants. Li and Wu [1] proposed a semi-empirical heat transfer model which has predicted their experimental data within an error band of ±20%. Kim [2] and Jiang et al. [3] proposed a performance factor to evaluate the heat transfer enhancement for enhanced tubes. Li and Chen et al. [4] and Shafaee et al. [5] studied the modified surface tubes (dimpled, helically dimpled tubes) using mixture refrigerants. They found that dimpled tubes have a much larger evaporation heat transfer coefficient than smooth tubes. Yang et al. [6] studied the flow boiling heat transfer performance of nanowire-coated micro-channel and found that this device is more energy saving and more effective at low mass fluxes. Zhang et al. [7,8] studied the flow boiling heat transfer in internally grooved tubes and a new heat transfer model was proposed to predict the evaporation HTC. Goto et al. [9] experimentally studied the R410A and HCFC22 evaporation in two conventional spiral grooved tubes and they concluded that the groove structure has an important effect on liquid redistribution around the circumference to reduce the occurrence of local wall superheat. Earlier studies suggested that the dimpled and grooved structures are good choices to develop new types of enhanced tubes. However, recent researches related to flow boiling in dimple-grooved tubes are very limited because these new types of tubes (2EHT tubes) are newly developed.

A smooth tube with the same inner and outer diameter was used as a reference tube to evaluate the heat transfer performance of 2EHT tubes. Flow pattern map was used to predict the liquid and vapor phase distribution at a specified cross-section for smooth tube. Wojtan et al. [10] proposed a modified flow boiling map from the Kattan-Thome-Favrat flow pattern map [11]. They divided the stratified-wavy region in the Kattan-Thome-Favrat flow pattern map into slug, slug/stratified-wavy and stratified-wavy regions

* Corresponding author.
E-mail address: weili96@zju.edu.cn (W. Li).

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Nomenclature

A_i	inner surface area of test tube, m^2
A_{LD}	dimensionless parameter, $A_{LD} = A(1-\varepsilon)/d_i^2$
A_{VD}	dimensionless parameter, $A_{LD} = A\varepsilon/d_i^2$
A	cross-sectional area, m^2
c_p	specific heat, J kg^{-1} K^{-1}
D	hydraulic diameter, m
d_i	inner diameter, m
d_o	outer diameter, m
E	enhanced factor
Fr	Froude number, $Fr = G^2/(\rho^2 gD)$
G	mass flux, kg m^{-2} s^{-1}
g	gravity acceleration, m s^{-2}
h	heat transfer coefficient, W/m^2 K^{-1}
h_{lv}	latent heat of vaporization, J kg^{-1}
L	tube length, m
q	heat flux, W/m^2
Re_l	Reynolds number of liquid phase Re_l=G(1 − x)D/μ_l
Q	heat transfer rate, W
S	suppress factor
S_{par}	projected surface area, mm^2
S_{dar}	developed surface area, mm^2
S_a	arithmetical mean height, $Sa = \frac{1}{A}\iint_A \lvert z(x,y)\rvert dxdy$
S_q	root mean square height, $Sq = \sqrt{\frac{1}{A}\iint_A z^2(x,y)dxdy}$
S_p	maximum peak height from mean plane, µm
S_v	maximum pit height from mean plane, µm
S_z	maximum height, $Sz = Sp + Sv$
S_{sk}	Skewness of the height distribution
T	temperature, K
x	vapor quality
V	voltage, V
I	electric current, A
W	mass flow rate, kg s^{-1}
We	Weber number, $We = G^2D/(\rho\sigma)$

Greek symbols

λ	thermal conductivity of copper, W m^{-1} K^{-1}
σ	surface tension, N m^{-1}
ρ	density, kg m^{-3}
ε	void fraction
μ	viscosity, Pa s
θ	dry-out angle

Subscripts

CHF	critical eat flux
eV	evaporation
in	inlet
IA	Transition of intermittent to annular
out	outlet
l	liquid phase
v	vapor phase
ref	refrigerant
lv	liquid to vapor phase change
sat	saturation
pre	preheating section
o	outer wall of test tube
i	inner wall of test tube
water	water
local	local parameters

and added annular-to-dryout, dryout-to-mist boundary curves into that map. Thome and Hajal [12] also redefined the Kattan-Thome-Favrat flow pattern map according to their experimental observations and test data. The Wojtan map [10] was used to predict flow patterns in the smooth tube (shown in Fig. 3), and then the same mass flux was tested with 2EHT tubes.

Fundamental issues of flow boiling heat transfer also have been widely investigated by many researchers. Kandlikar [13] studied the flow boiling heat transfer characteristics of mini-/micro-channels, and they concluded that isolated bubble and confined bubble or slug flow and annular flow would appear in these channels at low mass flux. Moreover, the surface tension plays a significant role in liquid distribution. Cheng [14] reviewed a large variety of data on critical heat flux effect on flow boiling in micro channel and confined space. They found that the existing critical heat flux (CHF) results show a great discrepancy, and no generalized correlations can correctly predict CHF in micro-channels and confined spaces. Thus, more CHF studies on flow boiling should be conducted, especially for enhanced tubes. Barraza et al. [15] experimentally studied the flow boiling of azeotropic mixtures in small channels so as to supplement the lack of data, and indicated that models suitable for pure refrigerant cannot predict the HTC of azeotropic mixtures boiling. Cheng and Xia [16] reported that the channel size effect on flow boiling was not completely understood and suggested that the flow pattern map was a promising method to understand the heat transfer behavior. The critical heat flux phenomenon is a crisis signal of heat transfer deterioration when the mist flow or dry-out flow occurs as explained by Wu and Li [17]. They proposed a new correlation to predict the saturated CHF which can predict almost 97% of their data within an error band of ±30%. Cheng [18] reviewed and summarized the CHF studies on flow boiling in micro-channels and confined spaces, which helped to understand the CHF effect on heat transfer coefficient as presented in Figs. 7 and 8. The literature review shows that most of the previous studies focused on smooth tubes and micro-fin tubes, and the heat transfer characteristics of these new 2EHT tubes are not available.

The objective of this paper is to study the flow boiling heat transfer characteristics of two types of three-dimensional dimple-grooved tubes. The heat transfer mechanisms related to the mass flux, local wall temperature superheat, critical heat flux and flow pattern maps are located. The HTC is calculated from the direct measurement parameters. The HTC results are then compared with an equivalent smooth tube. The experimental data has provided a comprehensive understanding on local flow boiling heat transfer near the nominal critical heat flux for two tested 2EHT tubes and the smooth tube. Another purpose is to develop a heat transfer model for the 2EHT tubes to reduce the large error band predicted by existing flow boiling heat transfer models [19–21].

2. Experimental facilities

Fig. 1 shows the cycle diagram of this experimental system and the detailed distribution of all T-type thermocouples that are used to measure the temperatures of tube wall and water. As is presented in Fig. 1(a), the test system includes three loops: the refrigerant circulation loop, the water circuit which is used to control the outlet quality of the test section by changing the water inlet temperature, and the condensation section which is used to subcool the refrigerant coming from the outlet of the test section, and is simplified as a condenser in Fig. 1(a). The inlet quality is adjusted by changing the voltage and current supply of the preheating section. The refrigerant driven by a digital gear pump enters the