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Flow boiling in horizontal flattened tubes: Part II – Flow boiling heat transfer results and model

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ABSTRACT

Experiments of flow boiling heat transfer were conducted in four horizontal flattened smooth copper tubes of two different heights of 2 and 3 mm. The equivalent diameters of the flattened tubes are 8.6, 7.17, 6.25, and 5.3 mm. The working fluids were R22 and R410A. The test conditions were: mass velocities from 150 to 500 kg/m^2 s, heat fluxes from 6 to 40 kW/m^2 and saturation temperature of 5 °C. The experimental heat transfer results are presented and the effects of mass flux, heat flux, and tube diameter on heat transfer are analyzed. Furthermore, the flow pattern based flow boiling heat transfer model of Wojtan et al. [L. Wojtan, T. Ursenbacher, J.R. Thome, Investigation of flow boiling in horizontal tubes: Part I – A new diabatic two-phase flow pattern map, Int. J. Heat Mass Transfer 48 (2005) 2955–2969; L. Wojtan, T. Ursenbacker, J.R. Thome, Investigation of flow boiling in horizontal tubes: Part II – Development of a new heat transfer model for stratified-wavy, dryout and mist flow regimes, Int. J. Heat Mass Transfer 48 (2005) 2970–2985], using the equivalent diameters, were compared to the experimental data. The model predicts 71% of the entire database of R22 and R410A ±30% overall. The model predicts well the flattened tube heat transfer coefficients for R22 while it does not predicts well those for R410A. Based on several physical considerations, a modified flow boiling heat transfer model was proposed for the flattened tubes on the basis of the Wojtan et al. model and it predicts the flattened tube heat transfer database of R22 and R410A by 85.8% within ±30%. The modified model is applied to the reduced pressures up to 0.19.

1. Introduction

Compared to a circular tube, a flattened tube has a higher internal surface area-to-cross-sectional flow area ratio, which can potentially be used to enhance the heat transfer rate and increase the compactness of the heat exchanger. Furthermore, flattened heat transfer tubes can greatly reduce the refrigerant charge in direct-expansion evaporators and condensers. Additionally, external potential advantages of flattened tube profiles are reduced air-side pressure drop and increase air-side heat transfer. So far, there are very limited studies on two-phase flow and heat transfer in flattened tubes in the literature. Wilson et al. [1] investigated refrigerant charge, two-phase pressure drop and heat transfer during condensation of two refrigerants R134a and R410A in several flattened tubes. Their results show significant reduction in refrigerant mass as a tube is flattened. They also indicate enhancement of condensation heat transfer and increase of pressure drop in the flattened tubes. Krishnaswamy et al. [2] investigated condensation heat transfer of steam–air mixtures in a horizontal flattened tube. They also proposed a simple heat transfer model. Their model predicts their data satisfactorily. Koyama et al. [3] conducted experiments on two-phase pressure drop and heat transfer of condensation of refrigerant R134a in multi-port extruded flattened tubes with hydraulic diameters of 1.114 and 0.807 mm. They concluded that to establish a prediction method of the pressure drop and heat transfer characteristics of pure refrigerant condensing in a small diameter tube, more experimental data for small diameter tubes should be investigated by considering the following terms: (1) flow patterns, (2) the effect of tube diameter, and (3) the interaction effect among the vapor shear stress and the gravitational acceleration and the surface tension. As for flow boiling in flattened tubes, however, there is no study available in the literature. In order to design a flattened tube evaporator, it is important to understand and predict the two-phase flow and flow boiling heat transfer characteristics inside horizontal tubes. In particular, in the case of a flattened tube having a very small height, the confinement of such a tube greatly affects two-phase and flow boiling heat transfer characteristics [4–7]. In Part II, experimental results of flow boiling heat transfer are presented and analyzed.

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Nomenclature

A_L	cross-sectional area occupied by liquid phase (m^2)
c_p	specific heat at constant pressure (J/kg K)
D	internal tube diameter (m)
F	nucleate boiling correction factor
G	total vapor and liquid two-phase mass flux (kg/m^2 s)
h	heat transfer coefficient (W/m^2 K)
k	thermal conductivity (W/m K)
M	molecular weight (kg/kmol)
Pr	Prandtl number ($=c_p\mu/k$)
p	pressure (Pa)
p_r	reduced pressure ($=p/p_{crit}$)
q	heat flux (W/m^2)
Re_V	vapor phase Reynolds number ($=GxD_e\,/\mu_V\varepsilon$)
Re_δ	liquid film Reynolds number ($=4G(1-x)\delta/\mu_L(1-\varepsilon)$)
T	temperature (°C)
t	tube wall thickness (m)
x	vapor quality
Greeks	
δ	liquid film thickness (m)
ε	cross-sectional vapor void fraction
μ	dynamic viscosity (N s/m^2)
θ	angle of tube perimeter (rad)
ρ	density (kg/m^3)
σ	standard deviation (%)
ξ	relative error (%)
$\bar{\xi}$	average error (%)
$\lvert\bar{\xi}\rvert$	mean error (%)
Subscripts	
cb	convective boiling
$crit$	critical
dry	dry
e	equivalent
IA	intermittent to annular flow
L	liquid
nb	nucleate boiling
sat	saturation
tp	two-phase flow
V	vapor
w	wall
wet	wetted
wi	inside wall
wo	outside wall

Furthermore, a modified flow pattern based heat transfer model for these flattened tubes was proposed for R410A and R22 flow boiling in the flattened tubes.

Flow patterns are very important in understanding the very complex two-phase flow phenomena and heat transfer trends in flow boiling [8]. Flow patterns at diabatic conditions are intrinsically related to the corresponding flow boiling heat transfer characteristics. The flow patterns can be used to explain physically the heat transfer mechanisms and characteristics. Over the past years, a number of flow pattern based flow boiling heat transfer models for various fluids were developed in LTCM. One of the earliest such kind of models is the Kattan–Thome–Favrat flow boiling heat transfer model [9–11] which was developed based on experimental data of R134a, R123, R502, R402A, and R404A in horizontal smooth tubes. The model predicts local heat transfer coefficients based on the local flow patterns and has methods for predicting heat transfer coefficients in the annular, intermittent, stratified-wavy and stratified flow regimes. Later on, Wojtan et al. [12,13] extended the Kattan–Thome–Favrat [9–11] flow map to include a new dryout region, new mist flow regime transition and subdivided the stratified-wavy regime into three sub-regimes (slug, stratified-wavy and slug, and stratified-wavy flows) based on their observations and dynamic void fraction measurements for R22 and R410A. They also developed the corresponding heat transfer methods for these flow regimes. Cheng et al. [14,15] developed a new flow map and a new flow boiling heat transfer model for CO_2 evaporation using the model of Wojtan et al. [12,13] as their starting point. Furthermore, new data allowed an updated flow map and heat transfer model to be proposed for CO_2 [16,17]. These models generally predict their respective flow boiling heat transfer data well. Recently, da Silva Lima et al. [18] compared the model of Wojtan et al. to their new flow boiling heat transfer data for R134a at three saturation temperatures, mass fluxes and two heat fluxes and found the model predicted their new data well. Furthermore, da Silva Lima et al. [19] compared their flow pattern observation with ammonia for a wide range of conditions the flow pattern map of Wojtan et al. and found that the map worked well for ammonia. Therefore, in the present study, the flow map and heat transfer model of Wojtan et al. were used to identify the flow regimes and to compare to the experimental heat transfer data in the flattened tubes.

2. Test setup and conditions

The test facility, test tubes, test section, measurement system and uncertainties are described in Part I. Flow boiling heat transfer coefficients of R22 and R410A were measured in the flattened tubes for the range of mass velocities from 150 to 500 kg/m^2 s, heat fluxes from 6 to 40 kW/m^2 and saturation temperature of 5 °C. This corresponds to reduced pressures p_r of 0.12 for R22 and 0.19 for 410A, respectively. Table 1 shows the test flattened tube number and dimensions (refer to Fig. 3 in part I). For each test run, these parameters were manually controlled to achieve desired test conditions. The measurements obtained with a National Instruments data acquisition system were monitored through a Personal Computer. Each experimental point resulted from the average of 10 acquisition cycles. Each acquisition cycle corresponds to an average from 100 acquisitions made in approximately 0.02 s. All measurements were made at steady state conditions.

3. Data reduction and test procedures

In the present study, the physical properties were obtained from REFPROP of NIST Version 6.01 [20]. The measurement and calculation of the local values of outside wall temperature T_{wo}, refrigerant saturation temperature T_{sat}, the local heat flux at the local measurement position q and vapor quality x are given in Part I of this study. With these measured parameters, the local flow boiling heat transfer coefficient can now be determined as

$$h_{tp} = \frac{q}{\overline{T}_{wi} - T_{sat}}, \tag{1}$$

Table 1
Flattened tube dimensions (mm).

Tube	H	W	t	D_e	D_h
Flattened tube No. 1	2	18.6	1.02	7.17	3.71
Flattened tube No. 2	3	17	0.76	8.6	5.35
Flattened tube No. 3	2	9.44	1.02	5.3	3.5
Flattened tube No. 4	3	7.87	0.86	6.25	4.88