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Flow Visualization of Two-Phase R-245fa at Low Mass Flux in a Plate Heat Exchanger near the Micro-Macroscale Transition

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ABSTRACT

Two-phase R-245fa flow in a plate heat exchanger is experimentally investigated to understand the unique flow regimes found during adiabatic operation at low refrigerant mass flux. A transparent plate heat exchanger replica with 3.4 mm hydraulic diameter is 3D-printed for flow visualization using high-speed videography. Observed flow regimes support that the thermofluidic characteristics peculiar to plate heat exchanger (PHE) operation are due to the macro-microscale transitional two-phase flow from the coexistence of fluid inertial forces and surface tension effects, corresponding to the operating conditions. Maximum stable bubble diameter is bigger at low mass flux than at high mass flux, and the bubbles can become big enough to be fully confined in the millimeter-scale PHE channel to be deformed or elongated. This represents the main thermo-physical characteristics of two-phase flow in mini- and microchannels, which is different from turbulent mixing flow easily found at high-mass-flux operation or in channels of conventional macroscale. Flow morphology involving complex bubble coalescence and breakup dynamics is captured and analyzed in relation to the fluid properties and geometric obstructions provided by the plate heat exchanger channel. While there exist previous studies, and even heat transfer coefficient correlations, suggesting the potential microscale flow regimes in PHEs, this is the first time evidence is presented via flow visualization.

1. INTRODUCTION

Plate heat exchangers (PHEs) have a wide range of applications due to their compactness (and resulting low-charge operation), thermal effectiveness, design flexibility, and cost competitiveness. Due to the turbulent-like conditions achieved inside small hydraulic diameter channels (on the order of a few millimeters) at Reynolds numbers as low as 150, the single-phase convective heat transfer coefficient in PHEs can be eight times of that observed in a circular pipe at the same Reynolds number (Yan & Lin, 1999). It has been claimed that the complex millimeter-scale corrugation of typical plate heat exchanger channels is intended to induce turbulence for active convection and mixing. Design of heat exchangers have been based on the manufacturers' proprietary knowledge or empirical models assuming turbulent flow. However, the flow regime prevailing at low mass flux operation, with potential bubble confinement effects and dryout conditions, is at low Reynolds numbers and might not be turbulent.

With the advancement of heating, ventilation, and air conditioning (HVAC) technology and enhanced insulation technology, research on the operation of PHEs with low refrigerant mass fluxes, to meet the lowered thermal load, is now more important. Recent technologies such as low-temperature lift heat pumps (which employ PHEs, at low mass flux) have further necessitated research for PHEs operating at mass fluxes below $50 \text{ kgm}^{-2}\text{s}^{-1}$ (Lee *et al.*, 2013). Miniaturization of evaporators, resulting in flow diameter reduction, targets to lower materials and manufacturing cost. It also leads to refrigerant charge reduction, but one of the side effects includes the higher pressure drop. Operations at low mass flux can directly help lower pressure drop and there has been industry research on this matter with different types of evaporators (Fenko *et al.*, 2016).

The two-phase flow and associated heat transfer mechanism is complex in the intricate, millimeter-scale PHE passages, exacerbating efforts to understand and predict the heat exchanger performance. Many of the available correlations badly over- or underestimate some published heat transfer and pressure drop data, especially those not used in

developing the correlation in question (Amalfi *et al.*, 2016a, 2016b). Many correlations are also often limited by the range of conditions they cover, such as mass flux and Reynolds number (Kandlikar, 1991). Explanations of behavior based on our understanding of flows in conventional-size (“macro”) channels do not always hold for PHEs passages, or other small channels in micro- or miniscale, due to differences in the bubble dynamics and flow regimes (Cheng & Xia, 2017). Furthermore, many of the existing correlations assume turbulent flows, but BPHEs can operate at relatively low Reynolds numbers. Dryout conditions have also been often overlooked due to a limited number of data sets covering the regime for correlation development (Lee *et al.*, 2014).

Even though the small hydraulic diameter typical to PHEs has been acknowledged as one of the most significant features of the heat exchanger (Amalfi *et al.*, 2016a, 2016b, Vakili-Farahani *et al.*, 2014a, 2014b), size effects have only recently been considered. What is important in considering size effects is the realization that an increasing heat transfer coefficient with heat flux does not necessarily mean nucleate boiling prevalence in small channels. Such behavior can be due to elongated or large deformed bubbles confined inside the channels, i.e., confined bubble flow. Thus, what is interpreted at the macroscale as nucleate boiling can be convection-dominated evaporation through the thin liquid film surrounding confined bubbles (Jacobi & Thome, 2002). There is currently no well-established quantitative criterion to determine when such bubble dynamics and flow regime changes occur, especially with low mass flux flows in the narrow and complex channels of a PHE (Ong & Thome, 2011a, 2011b).

The possibility of microscale boiling from evaporating thin films surrounding confined bubbles in BPHE at low mass flux was shown by Kim *et al.* (2018). While there exist correlations based on confinement number to determine microscale or macroscale flow morphology in PHEs (Amalfi *et al.*, 2016a, 2016b), the microscale flow regimes are not clearly shown. It was suggested that bubble confinement flow morphology and effect is important when looking at two-phase flow in a narrow space between two vertical surfaces (Bonjour & Lallemand, 1998), but such flow regime was never visualized in a PHE. Previous results were focused on the turbulent flows to induce mixing, as presented by Grabenstein *et al.* (2017), Tribbe & Müller-Steinhagen (2001), and Yan & Lin (1999).

This current work is focused on flow visualization of two-phase R-245fa inside the complex passages formed in chevron-patterned plate heat exchangers (PHEs) operating at low mass flux conditions. It elucidates important aspects associated with the flow regimes involving bubble confinement and elongation effects, which have received little attention in previous studies of two-phase flows in PHEs. It has significance scientifically, because of the added importance of viscous and surface tension forces to flow morphology. An experimental investigation was conducted for a transparent 3D-printed PHE replica (with the hydraulic diameter of 3.4 mm) operating with two-phase R-245fa under adiabatic conditions. The experimental design and measurement procedures are described next. A characterization of the flow regimes found from high-speed visualization in the PHEs is then presented and discussed, followed by main conclusions and recommendations.

2. EXPERIMENTAL SET-UP

A 3D-printed transparent replica of a PHE with 3.4 mm hydraulic diameter is used as the test section. The PHE is shown in Figure 1, and its detailed geometric parameters are given in Table 1. This test section is 3D printed with optically clear resin by stereolithography apparatus (SLA). A CAD model was created based on the geometric specification of a sample PHE. Direction of 3D printing was carefully selected to prevent internal support structures from forming due to overhang features of the PHE corrugations. High-speed flow visualization captures details of the prevailing flow regimes in the PHE. A 100 W DC floodlight was used for illumination for high-speed videography. Optical distortion from the corrugated heat exchanger plate is shown in Figure 2. Features can appear bigger or smaller in the test section.

The experimental apparatus consisted of four fluid loops as shown schematically in Figure 3 and the details are thoroughly described by Kim *et al.* (2018). The refrigerant loop, the water loop, and all water-ethylene glycol (EG) loops were equipped with pumps with variable frequency drive (VFD), valves, reservoirs, and instruments as shown in the schematic, allowing temperature and flow rate control. The setup included various control devices and measurement instruments with a computerized data acquisition system.

The refrigerant loop was charged with R-245fa. Refrigerant entered the heat exchanger from the bottom right port and exited through the top right port shown in Figure 1(b). Therefore, the refrigerant stream flows upward. The refrigerant temperature was measured at inlets and outlets of the heat exchanger using a resistance temperature detector (RTD)

Figure 1 3D-printed PHE section for flow visualization

Figure 2 Optical distortion test using (a) square grid and (b) a PHE plate over the same grid

at each port. Pressure was measured at the refrigerant stream inlet and outlet of the test section using absolute pressure transducers of stainless steel diaphragm type with a silicon strain gage. Refrigerant mass flow rate was measured using a Coriolis-effect flow meter.

Before entering the test section, the subcooled refrigerant passed through a pre-evaporator comprising of a brazed plate heat exchanger (BPHE) which was used to set the vapor quality supplied to the test evaporator. The hot water stream supplied to the pre-evaporator heated the refrigerant to a saturated two-phase flow with the desired vapor quality. Sufficient insulation between the pre-evaporator outlet and the test evaporator inlet preserved that condition. The heating element is rated at 4.5 kW. T-type thermocouples were used for temperature measurements at the inlets and outlets of the heat exchanger.

After leaving the test section, the refrigerant entered a BPHE condenser where heat was rejected to a water-ethylene glycol mixture (65% ethylene glycol by volume) in a loop cooled by a 20 kW chiller. Thermocouples were used to measure temperatures at the inlets and outlets of the condenser BPHE. The refrigerant then entered a receiver, and flowed through an after-condenser (using the same chiller loop) to ensure sub-cooling adequate to avoid cavitation (10 $^\circ\text{C}$ or more) in the refrigerant pump. This pump supplied refrigerant to the pre-evaporator, completing the refrigerant loop.

Ultimately, this apparatus allowed the adiabatic test section to be supplied with a refrigerant flow of prescribed mass flux, vapor quality, and saturation temperature. The experimental uncertainties and ranges of the measurement devices are provided in Table 2. Data were collected and analyzed when steady state operation was achieved, which was decided to prevail when temperature fluctuations were less than 0.4 K for at least 20 minutes. Average temperatures during steady state were used to evaluate thermal physical properties of the working fluids, such as density, thermal conductivity, and viscosity. Properties of the refrigerant were obtained from the Engineering Equation Solver (EES) software, while the water-ethylene glycol properties were calculated using the manufacturer's datasheet for DowthermTM SR-1 fluid.

Table 2 Specification of the measurement devices and experimental uncertainties

Parameter	Type	Uncertainty	Range
Temperature	T-type thermocouple	± 0.10 °C	-15 to 70 °C
	RTD sensor	± 0.11 °C	0 to 100 °C
Pressure	Strain-gage	$\pm 0.26\%$	0 to 1 MPa
	Strain-gage	$\pm 0.56\%$	0 to 3 MPa
Flow rate	Coriolis effect	$\pm 0.06\%$	0 to 272 kg h ⁻¹
	Coriolis effect	$\pm 0.10\%$	0 to 517 kg h ⁻¹
	Coriolis effect	$\pm 0.09\%$	0 to 2722 kg h ⁻¹

Figure 3 Schematic of the experimental setup

3. RESULTS AND DISCUSSION

High-speed videography is used to study flow morphology inside a PHE. There is an apparent difference between two-phase refrigerant flow morphology at high mass flux and low mass flux, as shown in Figure 4 and Figure 5. This comparison is made at the same fluid saturation temperature, $T_{sat} = 31.98$ °C, to ensure the same physical properties which can alter the flow morphology, such as surface tension, viscosity, and density, with values shown in Table 3. Vapor quality also plays a significant role in determining flow regime. Vapor quality at the test section, x , is calculated based on the enthalpy values at the inlet, saturated liquid, and saturated vapor, h_i , h_l , and h_v respectively, such that

$$x = \frac{h_i - h_l}{h_v - h_l} \quad (1)$$

h_i is evaluated using the enthalpy of single-phase liquid entering the pre-evaporator, $h_{pe,i}$, which is determined by the measured pressure and temperature, heating provided at the pre-evaporator, Q_{pe} , and mass flow rate of refrigerant, m_r :

Table 3 R-245fa properties at $p_{sat} = 190$ kPa

p_{sat}	T_{sat}	σ	μ_l	μ_v	ρ_l	ρ_v
190 kPa	31.98 °C	0.01274 N m ⁻¹	0.0003689 kg m ⁻¹ s ⁻¹	0.00001043 kg m ⁻¹ s ⁻¹	1319 kg m ⁻³	10.83 kg m ⁻³

Figure 4 Flow visualization of refrigerant at high mass flux under adiabatic conditions: $G = 100 \text{ kg m}^{-2}\text{s}^{-1}$, $x = 0.1$, $\text{Re}_l = 830$, $\text{Re}_v = 3260$, and $p_{sat} = 190 \text{ kPa}$. Image was enhanced by subtracting background and adjusting contrast

$$h_i = h_{pe,i} + \frac{Q_{pe}}{m_r} \quad (2)$$

At high mass flux operation, as shown in Figure 4, active mixing of liquid and vapor phases is observed in the whole PHE channel. Some previous work involving flow visualization results also showed similar actively mixing flows, such as turbulent mist flow (Yan & Lin, 1999) and a torrent perpendicular to the mean flow (Grabenstein *et al.*, 2017). Small bubbles travel with liquid in a highly chaotic motion in the heat exchanger, while they were also showing apparent flow recirculation zones near the left edge and bottom left corner of the PHE acting as traceable illuminated fluid particles, as illustrated in Figure 4. Maximum stable bubble diameter (Kocamustafaogullari & Ishii, 1995) in this experimental condition with high mass flux is observed to be smaller than that of low mass flux operation. Bubble breakup is highly prevalent due to the momentum flux caused by high inertial forces of the fluid, while the vapor and liquid eddies collide at high frequency. Another major reason for bubble breakup at higher Reynolds number is turbulent pressure fluctuations along the liquid – vapor interfaces (Liao & Lucas, 2009). Reynolds numbers of liquid phase (Re_l) and vapor phase (Re_v) are defined respectively below and used for comparative analysis of flow morphology:

$$\text{Re}_l = \frac{G(1-x)D_h}{\mu_l} \quad (3)$$

$$\text{Re}_v = \frac{GxD_h}{\mu_v} \quad (4)$$

The chevron pattern of the plate heat exchanger is intended to promote flow mixing as described above to enhance heat transfer performance. At low mass flux, however, larger bubbles and vapor slugs, separated from the liquid phase by stable liquid – gas interfaces as presented in Figure 5, were instead observed far from the inlet and outlet ports. Such flow morphology is more prevalent when surface tension can have dominance over inertial force or gravity, as often found in two-phase flows in micro- and minichannels. Similar PHE flow visualization results were found in a previous study using water (Shiomi *et al.*, 2004), which has higher surface tension than refrigerants. Near the PHE inlet and outlet ports where inertial force has significance, irregular bubbly flow was captured when $x = 0.1$ and partial film flow was observed at $x = 0.4$, as illustrated in Figure 6.

Figure 5 Flow visualization at low mass flux operations: R-245fa flow at $G = 10 \text{ kg m}^{-2}\text{s}^{-1}$, $x = 0.2$, $Re_l = 74$, $Re_v = 652$, and $p_{sat} = 190 \text{ kPa}$ (enhanced images)

Figure 6 Two-phase flow morphology at low mass flux, $G = 10 \text{ kg m}^{-2}\text{s}^{-1}$, and $p_{sat} = 190 \text{ kPa}$: (a) $x = 0.1$ and (b) $x = 0.4$ (enhanced images)

Surface tension dominance over inertial forces can be quantified by confinement number (Co) or Bond number (Bd). It has been reported that Co should be greater than approximately 0.3 for bubble confinement to be effective, which is also an indication of microscale flow prevalence (Ong & Thome, 2011a, 2011b):

$$Co = \sqrt{\frac{1}{Bd}} = \frac{1}{D_h} \left(\frac{\sigma}{g(\rho_l - \rho_v)} \right)^{0.5} \gtrsim 0.3 \quad (5)$$

This is revised from the previously widely accepted threshold of 0.5, realizing that fluid properties such as surface tension, phase densities, and viscosity were found to be highly important. In the current experimental set-up, Co is found to be ≈ 0.29 and elongated or deformed large bubbles confined inside the PHE channels provide visual evidence.

Figure 7 Deformed bubbles confined in narrow space between two corrugated vertical surfaces (microscale flow morphology). $G = 10 \text{ kg m}^{-2}\text{s}^{-1}$, $x = 0.1$, $Re_l = 83$, $Re_v = 326$ and $p_{sat} = 190 \text{ kPa}$ (enhanced images)

Figure 8 Schematic of the bubble deformation and breakup in PHE

When the two-phase refrigerant flow is supplied to the heat exchanger from the inlet at the bottom right, the vapor phase can travel diagonal upward direction towards the top left corner in a slug form along the valley of the corrugated channel, as described in Figure 5. Vapor phase has tendency to move directly upward due to buoyancy. As slugs travel upward, they face geometric structural obstacles, which are heat exchanger joints, and recirculating vortices that break them up into multiple bubbles. These bubbles can further break up to form more daughter bubbles or coalesce with neighboring bubbles, based on the balance among various forces such as buoyancy, shear stress, surface tension, as well as inertial force. The dynamics of bubbles at low mass flux with channel geometry and obstruction are considered highly different from what is observed at turbulent processes found at high mass flux (Link *et al.*, 2004; Protière *et al.*, 2010).

A confined large deformed bubble is captured and shown in Figure 7. As the large bubble travels upward in the PHE channel, it becomes elongated, deformed, or broken down to daughter bubbles based on its interaction with the internal geometry of the heat exchanger and various forces. The joint where two plate corrugations meet can be considered as a column-like obstacle (Jin & Hrnjak, 2017b). While the eccentric collision of a large bubble and a heat exchanger joint can lead to deformation of the bubble, symmetric head-to-head contact can cause a bubble breakup at the prevailing Bond number condition (Jin & Hrnjak, 2017a; Li *et al.*, 2014), as schematically shown in Figure 8. During the deformation or breakup process, formation of several very small daughter bubbles is also observed. These small bubbles can be shearing off from a larger bubble due to the large viscosity difference between liquid and vapor phases. Also, bubbles can attach to the contact points of the plates (Grabenstein *et al.*, 2017) and some large bubbles leave small bubbles on the PHE joint surface as they pass around the obstruction.

As large bubbles travel upwards due to their buoyancy while changing shapes per confinement effect provided by the narrow complex channel of PHE, bottom left corner of the heat exchanger tends to be left with small bubbles recirculating with liquid vortices, as shown in Figure 9. Due to this recirculation, the upward velocity of the vapor phase tends to be slower than the mean upward flow velocity. During recirculation, some bubbles are entrained in the wake of a preceding bubble and they eventually coalesce to form a bigger bubble. Once a large enough coalesced bubble is formed near a plate contact point, it can then travel upward more rapidly, experiencing further dynamic changes involving deformation and breakup.

Figure 9 Small bubbles in the flow recirculation zone, $G = 10 \text{ kg m}^{-2}\text{s}^{-1}$, $x = 0.2$, $Re_l = 74$, $Re_v = 652$; (a) overall PHE and (b) zoomed-in section (enhanced images)

4. CONCLUSIONS

Unique micro- and macroscale flow morphology in narrow spaces between corrugated vertical surfaces of a PHE has been captured, validating some previously published findings that suggest potential microscale boiling mechanisms in PHEs. During low refrigerant mass flux, surface tension dominates inertial forces. Vapor slug and large confined bubble elongation, deformation, and breakup occur in the PHE and the associated flow morphologies were significantly different from those encountered during turbulent mixing at high mass flux. Their behavior is explained in terms of Reynolds number and various flow properties, such as surface tension and viscosity. Research on the effect of micro- and macroscale two-phase flow morphology can further contribute to understanding the highly complex heat transfer and pressure drop processes in PHE, where there is not yet a successful universal correlation.

NOMENCLATURE

Bd	Bond number	(–)
Co	confinement number	(–)
D	diameter	(m)
G	mass flux, $G = m/(Wb)$	($\text{kg m}^{-2}\text{s}^{-1}$)
g	acceleration due to gravity	(m s^{-2})
h	enthalpy	(J kg^{-1})
m	mass flow rate	(kg s^{-1})
p	pressure	(kPa)
Re	Reynolds number	(–)
T	temperature	(°C)
x	vapor quality	(–)

Greek Symbols

μ	viscosity	($\text{kg m}^{-1}\text{s}^{-1}$)
ρ	density	(kg m^{-3})
σ	surface tension	(N m^{-1})

Subscript

h	hydraulic
i	inlet
l	liquid
sat	saturation
v	vapor

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