Condensation heat transfer and pressure drop characteristics of R134a, R1234ze(E), R245fa and R1233zd(E) in a plate heat exchanger

Zhang, Ji; Kærn, Martin Ryhl; Ommen, Torben; Elmegaard, Brian; Haglind, Fredrik

Published in:
International Journal of Heat and Mass Transfer

Link to article, DOI:
[10.1016/j.ijheatmasstransfer.2018.08.124](https://doi.org/10.1016/j.ijheatmasstransfer.2018.08.124)

Publication date:
2019

Document Version
Peer reviewed version

[Link back to DTU Orbit](#)

Citation (APA):
Zhang, J., Kærn, M. R., Ommen, T., Elmegaard, B., & Haglind, F. (2019). Condensation heat transfer and pressure drop characteristics of R134a, R1234ze(E), R245fa and R1233zd(E) in a plate heat exchanger. *International Journal of Heat and Mass Transfer*, 128, 136-149.
<https://doi.org/10.1016/j.ijheatmasstransfer.2018.08.124>

General rights

Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

- Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
- You may not further distribute the material or use it for any profit-making activity or commercial gain
- You may freely distribute the URL identifying the publication in the public portal

If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.

Condensation heat transfer and pressure drop characteristics of R134a, R1234ze(E), R245fa and R1233zd(E) in a plate heat exchanger

Ji Zhang*, Martin Ryhl Kærn, Torben Ommen, Brian Elmegaard, Fredrik Haglind

Department of Mechanical Engineering, Technical University of Denmark, Nils Koppels Allé,
Building 403, 2800 Kongens Lyngby, Denmark

Abstract

The fundamental understanding of the thermal-hydraulic performance of working fluids during condensation is important for the optimal design of the condenser in various thermodynamic cycles. This paper is aimed at obtaining flow condensation heat transfer and pressure drop characteristics in a plate heat exchanger during the working conditions of the condenser of either organic Rankine cycle power systems or heat pump units. The selected working fluids are two hydrofluorocarbons, R134a and R245fa, as well as their hydrofluoroolefin replacements, R1234ze(E) and R1233zd(E). Measurements of heat transfer coefficients and pressure drops were carried out with varying saturation temperature, mass flux, and liquid Reynolds number, ranging from 30 °C to 70 °C, 16 kg/m²s to 90 kg/m²s and 65 to 877, respectively. Based on commonly used existing correlations, new heat transfer and pressure drop correlations were developed, including the effect of the surface tension. The experimental data indicate that different heat transfer mechanisms occur at low liquid Reynolds number with the different working fluids. The results suggest higher heat transfer coefficients and pressure drops for R1234ze(E) and R1233zd(E) than for R134a and R245fa at the same working conditions. The new correlations enable significantly better prediction accuracies for the experimental results in this study than existing correlations, indicating that the surface tension is a suitable parameter to consider in mini and micro-scale condensation heat transfer.

* Corresponding author. Tel.: +45 45 25 13 87; fax: +45 45 25 19 61

E-mail address: jizhang@mek.dtu.dk (Ji Zhang)

1 **Keywords:** flow condensation, plate heat exchanger, organic Rankine cycle power systems, heat
2 pumps, HFO

3

Nomenclature			
Symbols		Subscripts	
A	heat transfer area, m ²	con	condensation
A_o	cross-sectional area on working fluid side of the PHE, m ²	cri	critical
b	amplitude of corrugation, m	dec	deceleration
Bo	Bond number	ele	elevation
c_p	specific heat capacity, J/kg K	eq	equivalent
D	diameter, m	exp	experimental
f	friction factor	fri	frictional
G	mass flux, kg/m ² s	h	hydraulic
h	heat transfer coefficient, W/m ² K	in	inlet
H	enthalpy, J/kg	l	liquid
K	coverage factor	lo	liquid only
k	thermal conductivity, W/m K	loc	local
L	length, m	m	mean
LMTD	log mean temperature difference, K	oil	oil
$\dot{m}$	mass flow rate, kg/s	out	outlet
N	number of channels	p	port
Nu	Nusselt number	pred	predicted
P	pressure, Pa	sat	saturation
Pr	Prandtl number	sub	subcooled
P_r	reduced pressure	sup	superheated
$\dot{Q}$	heat transfer rate, W	tot	total
Re	Reynolds number	v	vapor
t	thickness, m	w	water
T	temperature, °C	wall	wall
U	overall heat transfer coefficient, W/m ² K	wf	working fluid
$\dot{V}$	volume flow rate, L/min		
W	width, m		
We	Weber number		
x	vapor quality		
Greek Symbols		Abbreviations	
γ	dimensionless corrugation parameter	HC	hydrocarbon
β	chevron angle, °	HFC	hydrofluorocarbon
ϕ	enlargement factor of corrugation surface	HFO	hydrofluoroolefin
μ	dynamic viscosity, Pa•s	MAPD	mean absolute percentage deviation
ρ	mass density, kg/m ³	MBPD	mean bias percentage deviation
λ	corrugation pitch, m	ORC	organic Rankine cycle
Δ	difference	PHE	plate heat exchanger
ε	heat loss percentage	GWP	Global Warming Potential
δ	thickness of plate, m		
σ	surface tension, N/m		

4

5

1. Introduction

Plate heat exchangers (PHEs) are a type of compact heat exchanger widely used because of their high performance and compactness. Although the PHEs were originally developed for the single-phase heat transfer in the food industries, the use of PHEs as evaporators and condensers in industrial applications (such as refrigeration, air conditioning and power generation) has been introduced in the last 20 years [1,2]. Therefore, the fundamental understanding of the heat transfer and pressure drop characteristics of condensation in PHEs, as well as the corresponding prediction methods, are of high importance in order to design condensers for more efficient and economically feasible systems. However, the current research works in this field are quite limited, which is a major constraint in designing PHEs as condensers in industrial applications. In a recent review on condensation heat transfer in PHEs by Vakili-Farahani et al. [2], the authors stated that the use of PHEs in condensation has a relatively short history, the research on this topic is very limited so far, and theoretical predictions appear to be very difficult.

In addition, with respect to environmental issues, a new type of working fluids, hydrofluoroolefins (HFOs), has been developed as fourth-generation refrigerants. Compared with the conventional hydrofluorocarbons (HFCs), which have the largest share of the current refrigerant market [3], HFOs have much lower Global Warming Potential (GWP) [4]. Subsequently, the use of HFOs in various thermodynamic cycles has been increasingly studied, e.g., the use of R1234yf and R1234ze(E) in organic Rankine cycle (ORC) power systems [5] and the use of R1234ze(E) and R1234ze(Z) in heat pumps [6]. However, studies related to the condensation of HFOs in PHEs are indeed scarce in the open literature. Two relevant experimental investigations were carried out where R1234yf [7] and R1234ze(E) [8] were employed, respectively. Table 1 summarizes the existing studies on condensation heat transfer in PHEs using organic working fluids (HFCs, HFOs, Hydrocarbons (HCs)). As shown in the table, most of the experimental works consider HFCs as

1 working fluids with focus on condensation temperatures lower than 40 °C.

2 The objective of the present study is to investigate the condensation heat transfer of selected
3 HFCs and HFOs in a PHE at the working conditions prevailing in ORC units and heat pumps. The
4 two HFCs, R134a and R245fa, and their two HFO replacements, R1234ze(E) and R1233zd(E),
5 were evaluated for different mass flow rates and condensation temperatures. In terms of
6 condensation temperatures, the focus of the current research is 30 °C to 60 °C for the working fluids
7 R134a and R1234ze(E), and 40 °C to 70 °C for the working fluids R245fa and R1233zd(E).
8 Condensation temperatures of 40 °C to 70 °C are typical for the condenser in heat pumps, while
9 condensation temperatures of 30 °C to 40 °C are commonly used in the condenser of ORC power
10 systems [9,10], and condensation temperatures of 50 °C to 70 °C are typical for the condenser in
11 combined heat and power plants based on the ORC technology providing the heat for residential
12 applications (e.g., [11]). We used the experimental results to identify the prevailing heat transfer
13 mechanisms, and by comparison with the experimental data, we were able to evaluate the suitability
14 of the existing heat transfer and pressure drop correlations. Subsequently, we developed a new
15 prediction method for the thermal-hydraulic characteristics of condensation in PHEs by introducing
16 dimensionless numbers, taking account for the effects of surface tension and refitting existing
17 correlations using regression analysis of the experimental data presented in this paper. The intention
18 is for these new correlations to be used as a future reference for the design of condensers in ORC
19 systems and heat pumps.

20 The main novel contributions of the paper are the following: i) condensation heat transfer
21 results in a PHE for two working fluids, namely, the HFC R245fa and the HFO R1233zd(E), for
22 which no studies have previously been reported in the open literature on this topic, ii) experimental
23 results for higher condensation temperatures (50 °C and 60 °C) for the working fluids R134a and
24 R1234ze(E) than presented previously for these working fluids [8,12–14], and iii) the effects of

1 surface tension are considered when deriving condensation heat transfer and pressure drop
2 correlations for a PHE.

3 The paper proceeds with a description of the method in Section 2, including the experimental
4 apparatus, Wilson plot test and data analysis. Test results and discussions are presented in Sections
5 3 and 4, respectively. Section 5 reports the conclusions of the study.

6 Table 1 Summary of research works of condensation heat transfer in PHEs.

Year	Authors	Working fluids	Working conditions	Process	Film-convective transition	Configuration of PHE
1999	Yan et al. [12]	R134a	$G = (60 \text{ to } 120) \text{ kg/m}^2\text{s}$; $P_{\text{con}} = (0.7 \text{ to } 0.9) \text{ MPa}$ ((26 to 35) °C)	Partial condensation; $x_m = 0.08 \text{ to } 0.86$	–	$\beta = 60^\circ$; $\lambda = 10 \text{ mm}$; $a = 3.3 \text{ mm}$; $L_p = 450 \text{ mm}$; $W_p = 70 \text{ mm}$; $N = 3$
2000	Palmer et al. [15]	R22; R290; R290/600a; R32/152a	$Re_l = 20 \text{ to } 250$; $T_{\text{con,in}} = 25 \text{ }^\circ\text{C}$	Complete condensation; $\Delta T_{\text{sup}} = 3.9 \text{ }^\circ\text{C}$	–	Brazed PHE; $N = 30$
2002	Thonon and Bontemps [16]	Propane; pentane; butane	$Re_{l,o} = 100 \text{ to } 2000$; $P_{\text{con,in}} = (1.5 \text{ to } 18) \text{ bar}$ (up to 80 °C)	Complete condensation	$Re_{\text{cri}} = 100 \text{ to } 1000$	$\beta = 45^\circ$; $L = 300 \text{ mm}$; $W = 300 \text{ mm}$
2003	Han et al. [17]	R410; R22	$G = (60 \text{ to } 120) \text{ kg/m}^2\text{s}$; $T_{\text{con}} = 20 \text{ }^\circ\text{C}$ and 30 °C	Partial condensation; $x = 0.15 \text{ to } 0.9$	–	$\beta = 45^\circ, 65^\circ$ and 70° ; $\lambda = 4.9 \text{ mm}$, 5.2 mm and 7.0 mm; $a = 2.55 \text{ mm}$; $L_p = 476 \text{ mm}$; $W_p = 69 \text{ mm}$; $N = 6$
2004	Würfel and Ostrowski [18]	Heptane	$Re_{l,\text{out}} = 350 \text{ to } 1650$; $Re_{v,\text{in}} = 9500 \text{ to } 45000$; $P_{\text{con}} = 1 \text{ bar}$ ($\approx 98 \text{ }^\circ\text{C}$)	Complete condensation	$Re_{\text{cri}} = 250$	$\beta = 30^\circ/30^\circ$, $30^\circ/60^\circ$ and $60^\circ/60^\circ$; $L = 560 \text{ mm}$; $W = 180 \text{ mm}$; $N = 3$
2005	Kuo et al. [19]	R410A	$G = (50 \text{ to } 150) \text{ kg/m}^2\text{s}$; $T_{\text{con,in}} = (20 \text{ to } 31.5) \text{ }^\circ\text{C}$	Partial condensation; $x_m = 0.10 \text{ to } 0.80$	–	$\beta = 60^\circ$; $\lambda = 10 \text{ mm}$; $a = 3.3 \text{ mm}$; $L_p = 450 \text{ mm}$; $W_p = 70 \text{ mm}$; $N = 3$
2008	Djordjević [13]	R134a	$G = (30 \text{ to } 65) \text{ kg/m}^2\text{s}$; $T_{\text{con,in}} = (26 \text{ to } 29) \text{ }^\circ\text{C}$	Complete condensation; $x_{\text{loc}} = 0.1 \text{ to } 0.9$	–	$\beta = 60^\circ$; $\lambda = 10 \text{ mm}$; $a = 3.3 \text{ mm}$; $L_p = 450 \text{ mm}$; $W_p = 70 \text{ mm}$; $N = 3$
2012	Grabenstein	R365mfc	$G = (20 \text{ to } 80)$	Complete	–	$\beta = 63^\circ$

	and Kabelac [20]		$\text{kg/m}^2\text{s}$; $P_{\text{con}} = (430 \text{ to } 640) \text{ kPa}$ (87 to 103) °C	condensation; $x_{\text{loc}} = 0.05 \text{ to } 1$		
2012 to 2013	Mancin et al. [21–23]	R407C; R410A; R32	$G = (15 \text{ to } 40) \text{ kg/m}^2\text{s}$; $T_{\text{con,in}} = 41.8 \text{ °C}$ and 36.5 °C	Partial condensation; $\Delta T_{\text{sup}} = 5 \text{ to } 25 \text{ K}$; $x_{\text{out}} = 0.0 \text{ to } 0.65$	$G_{\text{cri}} = 20 \text{ kg/m}^2\text{s}$	$\beta = 65^\circ$; $L = 325 \text{ to } 526 \text{ mm}$; $W = 94 \text{ to } 247 \text{ mm}$; $N = 4 \text{ to } 8$
2015 to 2016	Sarraf et al. [24,25]	Pentane	$G = (9 \text{ to } 30) \text{ kg/m}^2\text{s}$; $T_{\text{con,in}} = 41.8 \text{ °C}$ and 36.5 °C	Complete condensation; $\Delta T_{\text{sup}} = (5 \text{ to } 25) \text{ K}$	$G_{\text{cri}} = 15 \text{ kg/m}^2\text{s}$	$\beta = 55^\circ$; $\lambda = 6 \text{ mm}$; $a = 2.2 \text{ mm}$; $L = 521 \text{ mm}$; $W = 111.4 \text{ mm}$; $N = 3$
2008 to 2015	Longo et al. [7,8,14,26–29]	R134a; R410A; R600a; R290; R1270; R1234yf; R1234ze(E); R152a	$G = (5.3 \text{ to } 11.4) \text{ kg/m}^2\text{s}$; $\text{Re}_l = 120 \text{ to } 800$; $T_{\text{sat}} = (20 \text{ to } 40) \text{ °C}$	Partial condensation: $x_{\text{in}} = 0.92 \text{ to } 1$, $x_{\text{out}} = 0.0 \text{ to } 0.09$; complete condensation: $\Delta T_{\text{sup}} = (9.2 \text{ to } 11.2) \text{ K}$, $\Delta T_{\text{sub}} = (0 \text{ to } 4.9) \text{ K}$	$G_{\text{cri}} = (15 \text{ to } 20) \text{ kg/m}^2\text{s}$; $\text{Re}_{\text{cri}} = 200 \text{ to } 400$	$\beta = 65^\circ$; $\lambda = 8 \text{ mm}$; $a = 2 \text{ mm}$; $L_p = 278 \text{ mm}$; $W = 72 \text{ mm}$; $N = 8 \text{ and } 10$

2. Methods

2.1. Test facilities

Figure 1 Schematic of a chevron corrugation plate in the condenser [30].

A commercial brazed PHE was used as the condenser (test section) in the test rig. It has 16

plates in total, 8 cooling water passes and 7 working fluid passes. Figure 1 shows the schematic of a chevron corrugation plate, and Table 2 lists the main dimensions of the current stainless plate (which were measured).

Table 2 Geometrical data of the chevron plate.

Parameters	Measured values
Length L	317 mm
Width W	76 mm
Port-to-port length L_p	278 mm
Port-to-port width W_p	40 mm
Diameter of inlet/outlet port D_p	18 mm
Chevron angle β	65 °
Corrugation pitch λ	7 mm
Amplitude of corrugation b	1 mm
Hydraulic diameter D_h (see the definition in Sec. 3.1)	3.4 mm

Figure 2 shows a schematic of the test facility. It consists of three fluid loops, one primary working fluid cycle in black and two auxiliary loops (thermal oil system and cooling water system) in red and blue, used to evaporate and condense the working fluids, respectively. In the main cycle, a variable speed volumetric pump was used to circulate the working fluid, as well as control the mass flow rate. Two PHEs, which were employed as pre-heater and evaporator by use of the thermal oil system, ensured that the subcooled working fluid at the inlet of the pre-heater was heated to superheated vapor at the outlet of the evaporator. Similarly, two PHEs function on the condensation side as the condenser (test section in this work) and subcooler, respectively. A cooling water system supplies the chilled water to the test section and subcooler. Four proportional valves were installed at the outlet of the four respective PHEs, in order to control the mass flow rates of the secondary fluids. A differential pressure transducer measures the pressure drop of the working fluid through the condenser. Moreover, the temperatures, mass/volume flow rates and pressures of the working fluids/thermal oil/chilled water in different locations were measured; see Figure 2. In addition, an additional PHE (termed PHE G), which has the same plate configuration as the

1 condenser, was installed between the two auxiliary loops. This PHE was used to obtain the water
 2 single-phase heat transfer coefficient in the current PHE based on the Wilson-plot method [31]; see
 3 Section 2.3.

Figure 2 Schematic of the test facility.

6 In heat pumps and ORC systems, the vapor at the condenser inlet is generally superheated [25].
 7 Therefore, we employed a slight superheating degree within 5 K for all the working fluids at the
 8 inlet of the condenser. The superheat was controlled through regulating the expansion valve as well
 9 as the heat input from the thermal oil. Moreover, at the outlet of the condenser, the working fluids
 10 were condensed to the saturated or slightly subcooled liquid within the subcooled degree of 5.5 K.
 11 Similarly, the subcooled degree of the working fluids was controlled by regulating the temperature

and flow rate of the chilled water. Table 3 summarizes the operating conditions for the experimental tests.

Table 3 Condenser test operating conditions.

$T_{\text{sat}} (^{\circ}\text{C})$	P_{sat} (bar)	G_{wf} (kg/m ² s)	q (kW/m ²)	Re_l	ΔT_{sup} (K)	ΔT_{sub} (K)	$\dot{V}_w$ (L/min)	$T_{w,\text{in}} (^{\circ}\text{C})$
29.7 to 71.0	2.9 to 16.3	16.0 to 90.0	4.0 to 57.4	65 to 877	1.6 to 4.9	0 to 5.3	14.9 to 15.6	13.8 to 65.2

A previous research work [30] has performed a validation of the experimental facility, based on the measurement of the single-phase heat transfer coefficient of R1234yf in the evaporator. In addition, the heat loss in the condenser based on the R1233zd(E) complete condensation process (from superheated vapor to subcooled liquid) was tested in this work. The heat loss percentage ε is defined as

$$\varepsilon = \frac{\dot{Q}_{w,\text{con}} - \dot{Q}_{\text{wf},\text{con}}}{\dot{Q}_{w,\text{con}}} 100 \%, \quad (1)$$

where the heat transfer rates of the working fluid and water in condenser $\dot{Q}_{\text{wf},\text{con}}$ and $\dot{Q}_{w,\text{con}}$ were calculated by

$$\dot{Q}_{\text{wf},\text{con}} = \dot{m}_{\text{wf}}(H_{\text{wf},\text{in}} - H_{\text{wf},\text{out}}), \quad (2)$$

and

$$\dot{Q}_{w,\text{con}} = \dot{m}_w c_{p,w}(T_{w,\text{out}} - T_{w,\text{in}}). \quad (3)$$

$T_{w,\text{in}}$ and $T_{w,\text{out}}$ are the condenser inlet and outlet temperatures of the water side, respectively, and $\dot{m}_w$ and $\dot{m}_{\text{wf}}$ are the mass flow rate of water and working fluid, respectively, $c_{p,w}$ is the specific heat of water, and $H_{\text{wf},\text{in}}$ and $H_{\text{wf},\text{out}}$ are the specific enthalpy of the working fluids at the inlet and outlet of the condenser. Figure 3 shows the test results of the heat loss rate, plotted as a function of mass flux G_{wf} (mass flow rate per unit cross-sectional area of flow), where G_{wf} was calculated based on the

cross-sectional area on working fluid side of the PHE A_o , defined as [32]

$$A_o = 2bWN_{wf}, \quad (4)$$

where N_{wf} is the number of channels (passages) on the working fluid side. As shown in the figure, the heat loss rate is less than 4 % with the mass flux ranging from 30 kg/m²s to 80 kg/m²s, suggesting that accurate measurements are obtained. Moreover, repeatability tests were conducted with R134a and R245fa for parts of the working conditions, indicating that the average deviation was 4.7 % and 3.8 %, respectively, for the heat transfer coefficient and pressure drop among the measurements.

Figure 3 Heat loss rate under the condensation heat transfer of R1233zd(E).

2.2. Data reduction

The whole heat transfer process consists of three regions: vapor desuperheating (region I), saturation condensation (region II), and liquid subcooling (region III); see Figure 4. In the analysis, we considered all three regions, though we did not treat the heat transfer in region I (desuperheating) separately; that is, we obtained the condensation heat transfer coefficients presented in this study by considering the heat transfer processes in regions I and II, and excluding that of region III. Not

1 treating the heat transfer in region I separately is justified by the fact that the heat transfer rate in
 2 this region accounts only for a small fraction (1.4 % to 4.6 %) of the heat transfer rate of the whole
 3 condensation process, having a small impact on the results. Besides, there are currently no
 4 appropriate heat transfer correlations available for the desuperheating region. A previous
 5 experimental study by Sarraf et al. [25] investigating the condensation heat transfer in a PHE,
 6 including a detailed analysis of the vapor desuperheating region, suggests that a thermal non-
 7 equilibrium process occurs in the desuperheating region including both liquid evaporation and
 8 vapor desuperheating, spatially and temporally distributed. This finding indicates that it is a non-
 9 trivial task to derive an accurate heat transfer correlation for this region. The details of the data
 10 reduction methods follow next.

Figure 4 Diagram of the heat transfer process in the condenser.

The geometric calculations of the chevron corrugation plates follow the definitions by Martin [33]. The hydraulic diameter of the working fluid channel between two chevron corrugation plates D_h is defined as

$$D_h = \frac{4b}{\phi}, \quad (5)$$

where the dimensionless parameter ϕ is the area enlargement factor caused by sinusoidal surface waviness and is calculated by

$$\phi = \frac{1}{6} (1 + \sqrt{1 + \gamma^2} + 4\sqrt{1 + \gamma^2/2}), \quad (6)$$

where γ is a dimensionless corrugation parameter, defined as

$$\gamma = \frac{2\pi b}{\lambda}. \quad (7)$$

The heat transfer rate of the working fluid in the subcooled heat transfer region $\dot{Q}_{wf,sub}$ is calculated as

$$\dot{Q}_{wf,sub} = \dot{m}_{wf}(H_l - H_{wf,out}), \quad (8)$$

where H_l is the saturated liquid specific enthalpy of the working fluids. According to the energy balance, the cooling water temperature at the inlet of region III, $T_{w,sub}$ is calculated by

$$T_{w,sub} = \frac{\dot{Q}_{wf,sub}}{c_{p,sub}\dot{m}_w} + T_{w,in}, \quad (9)$$

where the $\dot{m}_w$ is the mass flow rate of cooling water, $c_{p,sub}$ is the average specific heat of the water in region III, and $T_{w,in}$ is the cooling water temperature at the inlet of region I. The overall heat transfer coefficient in the subcooled region U_{sub} is determined by

$$\frac{1}{U_{sub}} = \frac{1}{h_{w,sub}} + \frac{1}{h_{wf,sub}} + \frac{t_{wall}}{k_{wall}}, \quad (10)$$

where the heat transfer coefficients of cooling water $h_{w,sub}$ and working fluids $h_{wf,sub}$ in region III are

calculated by the single-phase heat transfer correlation developed in the Wilson plot tests (see Section 2.3), and t_{wall} and k_{wall} are the thickness and thermal conductivity of the plate, respectively.

The heat transfer area of region III A_{sub} is calculated as

$$A_{\text{sub}} = \frac{\dot{Q}_{\text{sub}}}{U_{\text{sub}} \text{LMTD}_{\text{sub}}}, \quad (11)$$

where the log mean temperature difference in region III, LMTD_{sub} is defined as

$$\text{LMTD}_{\text{sub}} = \frac{(T_{\text{sat,out}} - T_{\text{w,sub}}) - (T_{\text{wf,out}} - T_{\text{w,in}})}{\ln \left(\frac{T_{\text{sat,out}} - T_{\text{w,sub}}}{T_{\text{wf,out}} - T_{\text{w,in}}} \right)}, \quad (12)$$

where $T_{\text{sat,out}}$ and $T_{\text{wf,out}}$ are the working fluid temperatures at the outlet of the regions II and III, respectively. The plate length for region III is calculated as

$$L_{\text{sub}} = \frac{A_{\text{sub}}}{A} L_{\text{p}}, \quad (13)$$

where A is the total heat transfer area of the condenser. The ratio A_{sub}/A ranges from 0.4 % to 2.0 % in the experiments.

The heat transfer rate and heat transfer area for the sum of regions I and II, $\dot{Q}_{\text{sat+sup}}$ and $A_{\text{sat+sup}}$ are given by

$$\dot{Q}_{\text{sat+sup}} = \dot{Q} - \dot{Q}_{\text{sub}}, \quad (14)$$

$$A_{\text{sat+sup}} = A - A_{\text{sub}}, \quad (15)$$

where the total heat transfer rate $\dot{Q}$ is calculated as

$$\dot{Q} = \dot{m}_{\text{wf}}(H_{\text{wf,in}} - H_{\text{wf,out}}). \quad (16)$$

The condensation heat transfer coefficient h_{wf} for the sum of regions I and II is calculated as

$$\frac{1}{h_{wf}} = \frac{1}{U_{sat+sup}} - \frac{1}{h_{w,sat+sup}} - \frac{t_{wall}}{k_{wall}}, \quad (17)$$

where the heat transfer coefficient of water for the sum of regions I and II, $h_{w,sat+sup}$ is calculated by the single-phase heat transfer correlation developed in the Wilson plot tests (see Section 2.3) and the overall heat transfer coefficient for the sum of regions I and II, $U_{sat+sup}$ is defined as

$$U_{sat+sup} = \frac{\dot{Q}_{sat+sup}}{A_{sat+sup} LMTD_{sat+sup}}, \quad (18)$$

where the log mean temperature difference for the sum of regions I and II, $LMTD_{sat+sup}$ is calculated as [34]

$$LMTD_{sat+sup} = \frac{(T_{sat} - T_{w,out}) - (T_{sat} - T_{w,sub})}{\ln \left(\frac{T_{sat} - T_{w,out}}{T_{sat} - T_{w,sub}} \right)}, \quad (19)$$

where T_{sat} is the saturation temperature in the condenser, which is an arithmetic average value of $T_{sat,out}$ and $T_{sat,in}$.

The frictional pressure drop for the sum of regions I and II is determined from

$$\Delta P_{fri} = \Delta P_{tot} - \Delta P_p + \Delta P_{ele} + \Delta P_{dec} - \Delta P_{sub}, \quad (20)$$

where the ΔP_{tot} is the total pressure drop measure by the differential pressure transducer. The ΔP_p is caused by the manifold and port, defined as

$$\Delta P_p = 0.75 G_p^2 \left(\frac{1}{2\rho_{v,sup}} + \frac{1}{2\rho_{l,out}} \right), \quad (21)$$

where the mass flux at port G_p is calculated based on the cross-sectional area of the plate port, $\rho_{v,sup}$ is the density of superheated vapor at inlet and $\rho_{l,out}$ is the density of liquid at the outlet. The deceleration and elevation pressure drops ΔP_{dec} and ΔP_{ele} were estimated from the homogenous model of the two-phase flow [35]:

$$\Delta P_{\text{dec}} = G_{\text{wf}}^2 \Delta x \left(\frac{1}{\rho_v} - \frac{1}{\rho_l} \right), \quad (22)$$

$$\Delta P_{\text{ele}} = g \rho_m (L_p - L_{\text{sub}}), \quad (23)$$

where ρ_l and ρ_v are the densities of liquid-phase and vapor-phase, respectively and Δx is the vapor quality difference between the inlet and outlet of the condenser, which is equal to 1 for complete condensation. The average two-phase density between the inlet and outlet of the condenser ρ_m is calculated at the average vapor quality between the inlet and outlet x_m , obtained by

$$\frac{1}{\rho_m} = \frac{x_m}{\rho_v} + \frac{1-x_m}{\rho_l}, \quad (24)$$

where x_m is 0.5 for complete condensation. The single-phase pressure drop at region III ΔP_{sub} was calculated by

$$\Delta P_{\text{sub}} = 2 f_{\text{sub}} \frac{L_{\text{sub}}}{D_h} \frac{G_{\text{wf}}^2}{\rho_l}, \quad (25)$$

where the single-phase friction factor was calculated based on the correlation suggested by Martin [33].

2.3. Wilson plot test

In order to calculate the single-phase heat transfer coefficients in Eqs. (10) and (17), the liquid-liquid heat transfer between water and oil was examined using PHE G. The analysis follows the Wilson plot technique [31] to determine a correlation for the water heat transfer coefficient. Adriano et al. [36] present a detailed description on how to derive the thermal oil single-phase heat transfer correlation using the modified Briggs and Young method [31]. In this work, we employed a similar method. In order to minimize the uncertainties of the Wilson plot results, we conducted three tests with different oil inlet temperatures, following the guideline suggested by Sherbini et al. [37]. According to the modified Briggs and Young method, the heat transfer correlations of the

1 water and oil sides are defined as

$$2 \quad h_w = C_1 \text{Re}_w^a \text{Pr}_w^{1/3} k_w / D_h (\mu_w / \mu_{w,\text{wall}})^{0.14}, \quad (26)$$

$$3 \quad h_{\text{oil}} = C_2 \text{Re}_{\text{oil}}^{0.8} \text{Pr}_{\text{oil}}^{1/3} k_{\text{oil}} / D_h (\mu_{\text{oil}} / \mu_{\text{oil},\text{wall}})^{0.14}, \quad (27)$$

4 where the $\text{Re}_{\text{oil}}/\text{Re}_w$ and $\text{Pr}_{\text{oil}}/\text{Pr}_w$ are the Reynolds numbers and Prandtl number of the oil/water,
 5 respectively, and μ_{oil}/μ_w and $\mu_{\text{oil},\text{wall}}/\mu_{w,\text{wall}}$ are the dynamic viscosities based on mean oil/water
 6 temperature and wall temperature, respectively.

7 Table 4 summarizes the operating conditions, coefficients C_1 and exponents of the water
 8 Reynolds number for the three tests. The maximum deviation among three coefficients is 3.5 % and
 9 among exponents is 1.4 %, which indicate consistency among the tests and demonstrate that the
 10 method is reliable. By averaging the results for C_1 and a , the following single-phase heat transfer
 11 correlation for water was obtained:

$$12 \quad \text{Nu} = 0.4225 \text{Re}^{0.733} \text{Pr}^{1/3} (\mu / \mu_{\text{wall}})^{0.14}, \quad 400 < \text{Re} < 1100, 2.8 < \text{Pr} < 4.5. \quad (28)$$

13 Figure 5 shows the heat transfer coefficients calculated by Eq. (28) for the working conditions of
 14 Test 1, together with the prediction results by several existing heat transfer correlations (Focke et al.
 15 [38], Hayes et al. [39], Chisholm and Wanniarachchi [40], Gasche [15], Martin [33] and Dović et al.
 16 [41]). The results of these correlations are comparable with those obtained in this study,
 17 demonstrating that Eq. (28) provides reasonable results. For example, the differences in heat
 18 transfer coefficient calculated by Eq. (28) and the correlations by Focke et al. [38] and Hayes et al.
 19 [39] are within 5 % and 10 %, respectively.

20 Table 4 Condenser operating test conditions.

	C_1	a	Re_w	Pr_w	Re_{oil}	Pr_{oil}	R^2
Test 1	0.4139	0.7391	700 to 1100	2.8 to 4.4	450	54	0.994
Test 2	0.4282	0.7287	580 to 940	2.9 to 4.5	360	65	0.981
Test 3	0.4254	0.7312	400 to 700	3.0 to 4.4	275	82	0.997

1

2

3 Figure 5 Comparison of Eq. (28) with different water single-phase heat transfer correlations in the
 4 open literature for the working conditions of test 1.

5

6 The difference between the true mean temperature difference and LMTD in the Wilson plot
 7 test, caused integrately by many factors including the number of plates, effects of end plates, flow
 8 arrangement and so on [42], was also evaluated by calculating the correction factor F_t for each test
 9 point (in total 22 points) following the method presented by Mancin et al. [22]. The results suggest
 10 that the figure for the correction factor F_t ranges from 0.935 to 0.986 with an average value of 0.973.
 11 Moreover, the differences in the condensation heat transfer coefficient of the working fluids when
 12 using the calculated value of the correction factor F_t and assuming F_t equal to unity range from 0.3 %
 13 to 2.0 %, indicating that for the conditions presented in this paper the correction to LMTD has a
 negligible effect of the PHE performance.

14 2.4. Uncertainties analysis

15 In this study, the uncertainty of temperature measurement was ± 0.19 K. The errors associated
 16 with the mass flow rate and volume flow rate measurements were ± 0.015 % and ± 0.5 %, respectively.
 17 The uncertainty of pressure and pressure difference uncertainties were ± 0.45 % FS
 18 (Full Span of 5 MPa) and ± 0.046 %, respectively. We performed the uncertainty analyses of the

main parameters in this study in accordance with the Kline and McClintock method [43], and Table 5 presents the corresponding results with the coverage factor $K = 2$.

Table 5 Uncertainty of main parameters.

Parameters	Uncertainty ($K = 2$)
Hydraulic diameter, D_h	2.8 %
Heat transfer rate, $\dot{Q}$	1.7 % to 4.1 %
Water heat transfer coefficient, h_w	2.4 % to 5.0 %
Working fluid heat transfer coefficient, h_{wf}	3.8 % to 12.1 %
Frictional pressure drop, ΔP_{fri}	3.2 % to 15.2 %

3. Results

In this section, we present the experimental heat transfer coefficients and pressure drops of the four working fluids and indicate their thermal and hydraulic characteristics in the PHE. Through comparison of the experimental results with those of existing heat transfer and pressure drop correlations, we evaluated the suitability of the existing correlations and subsequently were able to develop a new prediction method for the thermal-hydraulic characteristics of condensation in PHEs.

3.1. Heat transfer

Figure 6 depicts the heat transfer coefficient variation of the four working fluids as a function of the liquid Reynolds number with different saturation temperatures/reduced pressures. The liquid Reynolds number Re_l is defined as

$$Re_l = \frac{G_{wf}(1-x_m)D_h}{\mu_l}, \quad (28)$$

where the x_m is the mean vapor quality of the inlet and outlet. As shown in Figure 6, the heat transfer coefficients of each working fluid at the same saturation temperature are strongly dependent on the liquid Reynolds number. For the majority of the test conditions, the heat transfer coefficients increase with increasing liquid Reynolds number, which suggests a shear-controlled heat transfer process. In contrast, for several test conditions of R134a and R1234ze(E) at low liquid

1 Reynolds number, the results were independent of the liquid Reynolds number, indicating that the
2 processes are characterized by film condensation. The transition between the film and convective
3 condensation regions occurs at $Re_l \approx 200$. However, for R245fa and R1233zd(E), the shear-
4 controlled region dominates throughout the whole range of the liquid Reynolds number, and no film
5 condensation was found. The results suggest that the heat transfer coefficients increase when the
6 saturation temperatures decrease.

(a)

(b)

(c)

(d)

Figure 6 Heat transfer coefficients of the four working fluids as a function of liquid Reynolds number with different saturation temperatures.

Moreover, the results shown in Figure 6 indicate that the HFOs, R1234ze(E) and R1233zd(E), have higher heat transfer coefficients than their HFC alternatives, R134a and R245fa. Based on a common range of liquid Reynolds numbers for each saturation temperature, Table 6 provides quantifications of the average changes of h_{wf} and ΔP_{fri} . The largest average increase in h_{wf} between R1234ze(E) and R134a is 35% at $Re_l = 80$ to 480 and $T_{sat} = 30$ °C, while the corresponding increase between R1233zd(E) and R245fa is 31 % at $Re_l = 140$ to 450 and $T_{sat} = 70$ °C.

Table 6 Average changes in h_{wf} and ΔP_{fri} of R1234ze(E)/R1233zd(E) compared with R134a/R245fa.

		30 °C	40 °C	50 °C	60 °C	70 °C
R1234ze(E)/R134a	h_{wf}	35 %	25 %	14 %	12 %	—
	ΔP_{fri}	31 %	17 %	23 %	24 %	—
R1233zd(E)/R245fa	h_{wf}	—	11 %	17 %	24 %	31 %
	ΔP_{fri}	—	8 %	7 %	21 %	26 %

3.2. Pressure drop

(a)

(b)

(c)

(d)

Figure 7 Frictional pressure drops of the four working fluids as a function of liquid Reynolds number with different saturation temperatures.

Figure 7 shows the frictional pressure drop variation of the four working fluids as a function of the liquid Reynolds number at various saturation temperatures. The results suggest that the frictional pressure drop increases when the liquid Reynolds number (mass flux) increases and the saturation temperatures decreases. The temperature differences between $T_{\text{sat,out}}$ and $T_{\text{sat,in}}$ caused by the pressure drop across the PHE were 0.05 K to 0.49 K for R134a, 0.12 K to 1.14 K for R1234ze, 0.22 K to 3.19 K for R245fa and 0.31 K to 3.32 K for R1233zd(E).

Generally, the increase of pressure drop caused by the decrease of the saturation temperature is more significant at the high liquid Reynolds number. Moreover, similar to the heat transfer results, R1234ze(E) and R1233zd(E) show higher frictional pressure drops than R134a and R245fa. Table 6 presents a comparison of the pressure drop results. The largest average increase in frictional pressure drop of R1234ze(E) and R1233zd(E) compared with R134a and R245fa is 31 % at $T_{\text{sat}} = 30\text{ }^{\circ}\text{C}$ and 26 % at $T_{\text{sat}} = 70\text{ }^{\circ}\text{C}$, respectively.

3.3. Existing correlations

In order to quantify deviations between the experimental results and correlations, we employed

two parameters, the mean absolute percentage deviation (MAPD) and the mean bias percentage deviation (MBPD):

$$\text{MAPD} = \frac{1}{n} \sum_{i=1}^n \left| \frac{\text{data}_{i,\text{pred}} - \text{data}_{i,\text{exp}}}{\text{data}_{i,\text{exp}}} \right| \times 100 \%, \quad (30)$$

and

$$\text{MBPD} = \frac{1}{n} \sum_{i=1}^n \left(\frac{\text{data}_{i,\text{pred}} - \text{data}_{i,\text{exp}}}{\text{data}_{i,\text{exp}}} \right) \times 100 \%, \quad (31)$$

where the $\text{data}_{i,\text{pred}}$ and $\text{data}_{i,\text{exp}}$ are the predicted values calculated by the correlations and experimental values, respectively. The MAPD indicates the average deviation between the predicted and experimental values, while the MBPD by using actual deviations rather than absolute deviations, indicates whether the predicted values on average represent an overestimation (positive value) or underestimation (negative value) of the experimental values.

We selected four correlations for the heat transfer coefficient comparison. One of them is an early and widely used correlation developed by Akers et al. [44] for annular flow in a horizontal tube, including a definition of an equivalent Reynolds number Re_{eq} , accounting for the vapor core and the interfacial shear stress:

$$\text{Re}_{\text{eq}} = \frac{G_{\text{eq}} D_h}{\mu_l}, \quad (32)$$

where the equivalent mass flux G_{eq} is defined as

$$G_{\text{eq}} = G_{\text{wf}} \left[1 - x_m + x_m \left(\frac{\rho_l}{\rho_v} \right)^{0.5} \right]. \quad (33)$$

Yan et al. [12] and Würfel and Ostrowski [18] developed another two correlations considered in this work. Based on an evaluation of a databank collected from different condensation heat transfer studies, Vakili-Farahani et al. [2] found that these correlations provide the best accuracy. Longo et

al. [45] developed the fourth correlation considered in this work. Their correlation depends on two heat transfer regions divided by the equivalent Reynolds number and includes the effect of the presence of superheated vapor in the condensation process based on the Webb model [46]. Among the four correlations, Yan et al. [12] and Würfel and Ostrowski [18] correlations were developed based on the mean values of test data along the channel, while Akers et al. [44] and Longo et al. [45] correlations were developed aiming to predict the local heat transfer coefficient. Therefore, we applied the average values of experimental data for the heat transfer process to calculate the predicted values by the former two correlations and applied numerical integration along the channel to compute the average heat transfer coefficient by the latter two correlations. The general equation for numerical integration is defined as:

$$h_{wf,m} = \left(\frac{1}{A}\right) \int_0^A h \, dA. \quad (34)$$

Figure 8 depicts a comparison of the experimental and predicted values using the four correlations for heat transfer coefficients. The MAPD and MBPD of each correlation with respect to all four working fluids are also presented in the figure to indicate their prediction accuracy. Compared with the other three correlations, which generally underestimate the experimental results, the correlation by Yan et al. [12] provides a better prediction accuracy, resulting in a MAPD and MBPD of 11.8 % and -3.0 %, respectively. However, Yan et al. [12] developed their correlation only based on the experimental results of R134a, resulting in more accurate predictions for R134a and its replacement R1234ze(E), compared with those of R245fa and R1233zd(E). Especially for R1233zd(E), the MAPD of 21.2 % and MBPD of -21.2 % are much higher than the corresponding average values based on all the working fluids.

(a)

(b)

(c)

(d)

Figure 8 Predicted heat transfer coefficients using different correlations versus the experimental results.

For the frictional pressure drop (see Figure 9), we compared experimental results with predictions by two existing pressure drop correlations by Hsieh and Lin [47] and Khan et al. [48]. Vakili-Farahani et al. [2] recommended these two correlations due to their good predictions. The results indicate that the correlation by Hsieh and Lin [47] predicts the frictional pressure drop for R134a with a MAPD of 12.7 % and a MBPD of -0.7 %, and for R1234ze(E) the MAPD and MPBD are 15.7 % and -0.4 %, respectively. The MAPD and the MBPD for the prediction of the pressure drop of R1234ze(E) using the correlation by Khan et al. [48] are 14.4 % and 11.3 %, respectively.

1 However, both correlations overestimate most of the experimental data. Based on the comparison
 2 results shown in Figures 8 and 9, one may conclude that more accurate correlations need to be
 3 developed for predicting the heat transfer and pressure drop data in this study.

Figure 9 Predicted pressure drops using different correlations versus experimental results.

3.4. New correlations

As the channel size decreases, the surface tension becomes dominant over the gravitational force in micro and mini-scale condensation heat transfer, while its effects are generally negligible in macro-channels [47,48]. By influencing the behaviour of the liquid phase in the flow, the surface tension has significant effects on flow patterns and hence affects the heat transfer and pressure drop characteristics. According to Ref. [49], $D_h < 1$ mm refers to micro-channels, and mini-channels refers to $1 \leq D_h \leq 5$ mm. According to this definition, the hydraulic diameter of the PHE in this study, $D_h = 3.4$ mm, belongs to mini-channels. However, to the best of the authors' knowledge, none of the existing correlations with respect to condensation in PHEs incorporate the effects of surface tension. In this work, we developed a new heat transfer correlation through a modification to the correlation by Yan et al. [12] (see Table 6), which has provided reasonable predictions for all the working fluids except for R1233zd(E); see Figure 8(b). A dimensionless number, the Bond

number (Bo), involving the working fluid's physical parameter surface tension σ , was introduced in the new heat transfer correlation. We used two equations including the Bond number to regress the experimental data:

$$\text{Nu} = C \text{Re}_{\text{eq}}^{0.4} \text{Pr}_l^{1/3} \text{Bo}^b, \quad (35)$$

$$\text{Nu} = C \text{Re}_{\text{eq}}^a \text{Pr}_l^{1/3} \text{Bo}^b, \quad (36)$$

where the Bond number is defined as

$$\text{Bo} = \frac{g(\rho_l - \rho_v)D_h^2}{\sigma}. \quad (37)$$

Equation (35) was proposed to a large degree by following the correlation by Yan et al. [12], i.e., keeping the same exponents of Re_{eq} and Pr_l and only fitting the exponent of Bo and the first constant, while Eq. (36) was used to fit further the exponent of Re_{eq} based on Eq. (35), aiming to improve the regression. Figure 10 shows a comparison between the measured and predicted values for the heat transfer coefficient. In addition, Table 7 presents all the heat transfer correlations as well as their prediction results. The results suggest that through the introduction of the Bond number and further fitting of exponent a , the MAPD and MBPD of Eqs. (35) and (36) are reduced compared with those of the correlation by Yan et al. [12], especially for the working fluid R1233zd(E), indicating a significantly improved prediction accuracy.

Similar to the approach for developing the heat transfer correlation, we developed a new pressure drop correlation by using regression analysis in the form

$$f = C \text{Re}_{\text{eq}}^a \text{We}^b, \quad (38)$$

where the friction factor is considered to be a function of the equivalent Reynolds number, following the same format as the correlations by Hsieh and Lin [47] and Khan et al. [48]. In

addition, it includes the dimension number, Weber number (We), defined as

$$We = \frac{G_{wf}^2 D_h}{\rho_m \sigma}. \quad (39)$$

The Weber number, expressing the ratio of inertia forces to surface tension forces, indicates the relative importance of the fluid's kinetic energy compared with its surface tension. The final pressure drop correlation reads as follows:

$$f = 0.0146 Re_{eq}^{0.9814} We^{-1.0064}. \quad (40)$$

Figure 11 shows a comparison of experimental results with the predicted values of the friction factor calculated by Eq. (40), with 94 % of the experimental data predicted within ± 25 % with a MAPD of 9.2 % and a MBPD of 0.7 %.

Figure 10 Comparison of predictions of the proposed correlations (Eqs. (35) and (36)) with experimental data for heat transfer coefficients.

Table 7 Summary of heat transfer correlations as well as their prediction accuracies of experimental results.

	Equation	All fluids		R1233zd(E)	
		MAPD	MBPD	MAPD	MBPD
Yan et al. [12]	$Nu = 4.118 Re_{eq}^{0.4} Pr_l^{1/3}$	11.8 %	-3.0 %	21.2 %	-21.2 %

Eq. (35)	$Nu = 12.36Re_{eq}^{0.4}Pr_l^{1/3}Bo^{-0.3715}$	8.6 %	1.2 %	6.9 %	-4.1 %
Eq. (36)	$Nu = 4.3375Re_{eq}^{0.5383}Pr_l^{1/3}Bo^{-0.3872}$	6.4 %	0.3 %	5.7 %	-5.1 %

Figure 11 Comparison of predictions of the proposed correlation (Eq. (40)) and experimental data for friction factors.

Table 8 Range of dimensionless numbers in Eqs. (35), (36) and (40).

Re_{eq}	Pr_l	Bo	We
1207 – 4827	3.1 – 6.5	11.7 – 28.3	1.8 – 68.2

4. Discussion

As may be noted in Figs. 6 and 7, both the heat transfer coefficient and friction factor increase with increasing mass flow rate and decreasing saturation temperature. This finding agrees with that concluded in Ref. [2] by reviewing the relevant studies in this area. As for the convective condensation (shear-controlled heat transfer region), the increase of the mass flow rate and the decrease of saturation temperature induce higher vapor velocity and more turbulence, thereby enhancing the heat transfer. In addition, at the lower saturation temperatures, working fluids have the larger vapor-liquid density differences causing larger shear stresses, in turn resulting in more

1 intensified convection at the vapor-liquid interface.

2 A clear transition between film and convective condensation occurs at $Re_l \approx 200$ for R134a
3 and R1234ze(E). This value agrees in line with $Re_l = 250$ for heptane obtained by Würfel and
4 Ostrowski [18] and $Re_l \approx 200$ to 300 for R134a obtained by Longo [14]. The corresponding
5 transition for mass flux is 20 kg/m²s to 25 kg/m²s in this study, which agrees reasonably well with
6 results obtained by Mancin et al. [23] and Longo et al. [8], namely, $G = 20$ kg/m²s for R410A and
7 R407C and $G = 20$ kg/m²s for R1234ze(E), respectively. Therefore, although the research works
8 with respect to film-convective condensation transition are very limited, most of the results suggest
9 that the transition generally occurs at $Re_l \approx 200$ to 300 or $G = 20$ kg/m²s to 25 kg/m²s. We
10 recommend for future work more studies aimed at finding the accurate definition of the transition
11 for different working conditions. Moreover, it should be noted from the results obtained in this
12 paper that the whole heat transfer process of R245fa and R1233zd(E), even at $Re_l < 200$, is
13 characterized by convective condensation without the appearance of film condensation. This may
14 be attributed to the difference in properties of the working fluids. Within the working conditions in
15 this study, R245fa and R1233zd(E) have on average 2.6 times smaller vapor density and 2.9 times
16 larger liquid-vapor density ratios than those of R134a and R1234ze(E). The former leads to a higher
17 vapor velocity and thereby causes a stronger disturbance at the vapor-liquid interface, while the
18 latter results in a larger shear stress between the vapor and liquid phases. Both these factors promote
19 a heat transfer process dominated by convective condensation.

20 Although involving surface tension to develop the correlation enables the better prediction for
21 the experimental data obtained in this study, the physics on how surface tension affects the
22 condensation flow in a plate heat exchanger are not clear due to the limited number of studies
23 available in the field; especially visualization studies are lacking. Nevertheless, an effort is made

1 here briefly to explain some of the physical phenomena. In noncircular micro and mini-channels,
2 the surface tension generates a transverse pressure gradient in the condensate film, which leads to a
3 condensate flow towards the corners and thins the film along the flat sides of the channel, lowering
4 the thermal resistances. This is in accordance with the essence of surface tension, i.e., minimizing
5 the surface area of the fluid. The effects of surface tension on flow condensation in noncircular
6 micro and mini-channels have been experimentally proved and theoretically analysed by numerous
7 research works (e.g., Refs. [50,51]). In terms of brazed PHEs, the contact points where two plates
8 are brazed together play a similar role as those of the corners in noncircular micro and mini-
9 channels, i.e., the condensate film is pulled towards the contact points leading to a thinner liquid
10 film on the remaining parts of the channel. The inference for the distribution of the condensate film
11 in the flow passage of PHEs needs to be verified, preferably by experimental work.

12 It needs to be added that it would be desirable to use more experimental data for the
13 development and validation of the correlations presented in this paper. However, at present no other
14 measurements of complete condensation representing the conditions prevailing in the condenser of
15 ORC units and heat pump systems are available in the open literature.

16 **5. Conclusions**

17 We conducted an experimental investigation to study the condensation heat transfer and
18 pressure drop performances of R134a, R245fa, R1234ze(E) and R1233zd(E) in a plate heat
19 exchanger, covering a wide temperature range from 30 °C to 70 °C. Based on correlations by Yan et
20 al. [12] and Hsieh and Lin [47], we developed new heat transfer and pressure drop correlations,
21 both taking into account the effect of surface tension on mini-channel condensation. The new
22 correlations were developed for a larger number of working fluids and for a wider range of working
23 conditions than those of Yan et al. [12] and Hsieh and Lin [47]. We have demonstrated significant
24 improvements in terms of the prediction accuracy of heat transfer and pressure drop.

The experimental results indicate that the heat transfer coefficient and pressure drop increase with increasing liquid Reynolds number and decreasing saturation temperature. Due to the difference in physical properties, R134a and R1234ze(E) have a filmwise-convective condensation transition at $Re_l = 200$, while R245fa and R1233zd(E) present convective condensation in the whole heat transfer process. Generally, the HFOs R1234ze(E) and R1233zd(E) have higher heat transfer coefficients and pressure drops than their HFC counterparts R134a and R245fa. The new heat transfer and pressure drop correlations have mean absolute percentage deviations of 6.4 % and 9.3 % and mean bias percentage deviations of 0.3 % and 0.7 %, respectively.

Acknowledgement

The research presented in this paper has received funding from the People Programme (Marie Curie Actions) of the European Union's Seventh Framework Programme (FP7/2007-2013) under the REA grant agreement n° 609405 (COFUNDPostdocDTU), and Innovation Fund Denmark with the THERMCYC project (www.thermcyc.mek.dtu.dk, project ID: 1305-00036B). The financial support is gratefully acknowledged.

1 **References**

- 2 [1] L. Wang, B. Sundén, R.M. Manglik, Plate Heat Exchangers: Design, Applications and
3 Performance, WIT Press, Southampton, UK, 2007.
- 4 [2] J.R. Thome, R.L. Amalfi, F. Vakili-farahani, Two-phase Heat Transfer and Pressure Drop
5 within Plate Heat Exchangers, in: *Encycl. Two-Phase Heat Transf. Flow II*, 2015: pp. 145–
6 215.
- 7 [3] W. Liu, D. Meinel, C. Wieland, H. Spliethoff, Investigation of hydrofluoroolefins as
8 potential working fluids in organic Rankine cycle for geothermal power generation, *Energy*.
9 67 (2014) 106–116. doi:10.1016/j.energy.2013.11.081.
- 10 [4] J.S. Brown, HFOs new, low global warming potential refrigerants, *ASHRAE J.* 51 (2009)
11 22–29.
- 12 [5] C.M. Invernizzi, P. Iora, M. Preßinger, G. Manzolini, HFOs as substitute for R-134a as
13 working fluids in ORC power plants: A thermodynamic assessment and thermal stability
14 analysis, *Appl. Therm. Eng.* 103 (2016) 790–797. doi:10.1016/j.applthermaleng.2016.04.101.
- 15 [6] C. Kondou, S. Koyama, Thermodynamic assessment of high-temperature heat pumps using
16 low-GWP HFO refrigerants for heat recovery, *Int. J. Refrig.* 53 (2015) 126–141.
17 doi:10.1016/j.ijrefrig.2014.09.018.
- 18 [7] G.A. Longo, C. Zilio, Condensation of the lowGWPrefrigerant HFC1234yf inside a brazed
19 plate heat exchanger, *Int. J. Refrig.* 36 (2013) 612–621. doi:10.1016/j.ijrefrig.2013.08.013.
- 20 [8] G.A. Longo, C. Zilio, G. Righetti, J.S. Brown, Condensation of the low GWP refrigerant
21 HFO1234ze(E) inside a Brazed Plate Heat Exchanger, *Int. J. Refrig.* 38 (2014) 250–259.
22 doi:10.1016/j.ijrefrig.2013.08.013.
- 23 [9] J. Bao, L. Zhao, A review of working fluid and expander selections for organic Rankine
24 cycle, *Renew. Sustain. Energy Rev.* 24 (2013) 325–342. doi:10.1016/j.rser.2013.03.040.
- 25 [10] S. Quoilin, M. Van Den Broek, S. Declaye, P. Dewallef, V. Lemort, Techno-economic
26 survey of organic rankine cycle (ORC) systems, *Renew. Sustain. Energy Rev.* 22 (2013)
27 168–186. doi:10.1016/j.rser.2013.01.028.
- 28 [11] C. Wieland, D. Meinel, S. Eyerer, H. Spliethoff, Innovative CHP concept for ORC and its
29 benefit compared to conventional concepts, *Appl. Energy*. 183 (2016) 478–490.
30 doi:10.1016/j.apenergy.2016.08.193.
- 31 [12] Y.-Y. Yan, H.-C. Lio, T.-F. Lin, Condensation heat transfer and pressure drop of refrigerant
32 R-134a in a plate heat exchanger, *Int. J. Heat Mass Transf.* 42 (1999) 993–1006.
33 doi:10.1016/S0017-9310(98)00217-8.
- 34 [13] E. Djordjević, S. Kabelac, S. Šerbanović, Heat transfer coefficient and pressure drop during
35 refrigerant R-134a condensation in a plate heat exchanger, *Chem. Pap.* 62 (2008) 78–85.
36 doi:10.2478/s11696-007-0082-8.
- 37 [14] G.A. Longo, Refrigerant R134a condensation heat transfer and pressure drop inside a small
38 brazed plate heat exchanger, *Int. J. Refrig.* 31 (2008) 780–789.
39 doi:10.1016/j.ijrefrig.2007.11.017.

- 1 [15] S.C. Palmer, W.V. Payne, P. a Domanski, Evaporation and Condensation Heat Transfer
2 Performance of Flammable Refrigerants in a Brazed Plate Heat Exchanger, NIST
3 Interagency/Internal Rep. (2000) 6541.
- 4 [16] B. Thonon, A. Bontemps, Condensation of Pure and Mixture of Hydrocarbons in a Compact
5 Heat Exchanger: Experiments and Modelling, Heat Transf. Eng. 23 (2002) 3–17.
6 doi:10.1080/01457630290098718.
- 7 [17] D.H. Han, K.J. Lee, Y.H. Kim, The characteristics of condensation brazed plate heat
8 exchangers with different chevron angles, J. Korean Phys. Soc. 43 (2003) 66–73.
9 doi:10.1016/S1359-4311(03)00061-9.
- 10 [18] R. Würfel, N. Ostrowski, Experimental investigations of heat transfer and pressure drop
11 during the condensation process within plate heat exchangers of the herringbone-type, Int. J.
12 Therm. Sci. 43 (2004) 59–68. doi:10.1016/S1290-0729(03)00099-1.
- 13 [19] W.S. Kuo, Y.M. Lie, Y.Y. Hsieh, T.F. Lin, Condensation heat transfer and pressure drop of
14 refrigerant R-410A flow in a vertical plate heat exchanger, Int. J. Heat Mass Transf. 48 (2005)
15 5205–5220. doi:10.1016/j.ijheatmasstransfer.2005.07.023.
- 16 [20] V. Grabenstein, S. Kabelac, Experimental and theoretical analysis of the local condensation
17 heat transfer in a plate heat exchanger, J. Phys. Conf. Ser. 395 (2012) 012169.
18 doi:10.1088/1742-6596/395/1/012169.
- 19 [21] S. Mancin, D. Del Col, L. Rossetto, R32 partial condensation inside a brazed plate heat
20 exchanger, Int. J. Refrig. 36 (2013) 601–611. doi:10.1016/j.ijrefrig.2012.10.019.
- 21 [22] S. Mancin, D. Del Col, L. Rossetto, Partial condensation of R407C and R410A refrigerants
22 inside a plate heat exchanger, Exp. Therm. Fluid Sci. 36 (2012) 149–157.
23 doi:10.1016/j.expthermflusci.2011.09.007.
- 24 [23] S. Mancin, D. Del Col, L. Rossetto, Condensation of superheated vapour of R410A and
25 R407C inside plate heat exchangers: Experimental results and simulation procedure, Int. J.
26 Refrig. 35 (2012) 2003–2013. doi:10.1016/j.ijrefrig.2012.06.001.
- 27 [24] K. Sarraf, S. Launay, G. El Achkar, L. Tadrist, Local vs global heat transfer and flow
28 analysis of hydrocarbon complete condensation in plate heat exchanger based on infrared
29 thermography, Int. J. Heat Mass Transf. 90 (2015) 878–893.
30 doi:10.1016/j.ijheatmasstransfer.2015.07.036.
- 31 [25] K. Sarraf, S. Launay, L. Tadrist, Analysis of enhanced vapor desuperheating during
32 condensation inside a plate heat exchanger, Int. J. Therm. Sci. 105 (2016) 96–108.
33 doi:10.1016/j.ijthermalsci.2016.03.001.
- 34 [26] G.A. Longo, R410A condensation inside a commercial brazed plate heat exchanger, Exp.
35 Therm. Fluid Sci. 33 (2009) 284–291. doi:10.1016/j.expthermflusci.2008.09.004.
- 36 [27] G.A. Longo, Heat transfer and pressure drop during hydrocarbon refrigerant condensation
37 inside a brazed plate heat exchanger, Int. J. Refrig. 33 (2010) 944–953.
38 doi:10.1016/j.expthermflusci.2011.01.018.
- 39 [28] G.A. Longo, The effect of vapour super-heating on hydrocarbon refrigerant condensation
40 inside a brazed plate heat exchanger, Exp. Therm. Fluid Sci. 35 (2011) 978–985.
41 doi:10.1016/j.expthermflusci.2011.01.018.

- 1 [29] G.A. Longo, C. Zilio, G. Righetti, J.S. Brown, Condensation of the low GWP refrigerant
2 HFC152a inside a Brazed Plate Heat Exchanger, *Exp. Therm. Fluid Sci.* 68 (2015) 509–515.
3 doi:10.1016/j.jrefrig.2013.08.013.
- 4 [30] J. Zhang, A. Desideri, M.R. Kærn, T.S. Ommen, J. Wronski, F. Haglind, Flow boiling heat
5 transfer and pressure drop characteristics of R134a, R1234yf and R1234ze in a plate heat
6 exchanger for organic Rankine cycle units, *Int. J. Heat Mass Transf.* 108 (2017) 1787–1801.
7 doi:10.1016/j.jheatmasstransfer.2017.01.026.
- 8 [31] R.K. Shah, Assessment of modified wilson plot techniques for obtaining heat exchanger
9 design data, in: *Ninth Int. Heat Transf. Conf.*, Jerusalem, Israel, 1990.
- 10 [32] R.S. Shah, D. Sekulić, Heat Exchanger Surface Geometrical Characteristics, in: *Fundam.*
11 *Heat Exch. Des.*, John Wiley & Sons, Inc., Hoboken, New Jersey, 2003.
- 12 [33] H. Martin, A theoretical approach to predict the performance of chevron-type plate heat
13 exchangers, *Chem. Eng. Process.* 35 (1996) 301–310.
- 14 [34] E. Granryd, I. Ekroth, P. Lundqvist, Å. Menlinder, B. Palm, P. Rohlin, *Refrigerating*
15 *Engineering*, Royal Institute of Technology, KTH, Stockholm, 2005.
- 16 [35] J.G. Collier, J.R. Thome, *Convective Boiling and Condensation*, third Edit, Oxford
17 University Press, New York, United States, 1994.
- 18 [36] A. Desideri, J. Zhang, M.R. Kærn, T.S. Ommen, J. Wronski, V. Lemort, F. Haglind, An
19 experimental analysis of flow boiling and pressure drop in a brazed plate heat exchanger for
20 organic Rankine cycle power systems, *Int. J. Heat Mass Transf.* 113 (2017) 6–21.
21 doi:10.1016/j.jheatmasstransfer.2017.05.063.
- 22 [37] A. Sherbini, A. Joardar, A. Jacobi, Modified Wilson-Plot Technique for Heat Exchanger
23 Performance: Strategies for Minimizing Uncertainty in Data Reduction, in: *Int. Refrig. Air*
24 *Cond. Conf.*, Purdu, Indiana, United States, 2004.
25 <http://docs.lib.purdue.edu/cgi/viewcontent.cgi?article=1626&context=iracc>.
- 26 [38] W.W. Focke, J. Zachariades, I. Olivier, The effect of the corrugation inclination angle on the
27 thermohydraulic performance of plate heat exchangers, *Int. J. Heat Mass Transf.* 28 (1985)
28 1469–1479. doi:10.1016/0017-9310(85)90249-2.
- 29 [39] N. Hayes, A. Jokar, Z.H. Ayub, Study of carbon dioxide condensation in chevron plate
30 exchangers; Heat transfer analysis, *Int. J. Heat Mass Transf.* 54 (2011) 1121–1131.
31 doi:10.1016/j.jheatmasstransfer.2010.11.010.
- 32 [40] Z.H. Ayub, Plate heat exchanger literature survey and new heat transfer and pressure drop
33 correlations for refrigerant evaporators, *Heat Transf. Eng.* 24 (2003) 3–16.
34 doi:10.1080/01457630304056.
- 35 [41] D. Dović, B. Palm, S. Švaić, Generalized correlations for predicting heat transfer and
36 pressure drop in plate heat exchanger channels of arbitrary geometry, *Int. J. Heat Mass*
37 *Transf.* 52 (2009) 4553–4563. doi:10.1016/j.jheatmasstransfer.2009.03.074.
- 38 [42] R.K. Shah, W.W. Focke, *Plate Heat Exchanger and Their Design Theory*, in: *Heat Transf.*
39 *Equip. Des.*, Hemisphere Publishing, Washington, 1988.
- 40 [43] S.J. Kline, F.A. McClintock, Describing uncertainties in single-sample experiments, *Mech.*

Eng. 75 (1953) 3–8.

[44] W.W. Akers, H.A. Deans, O.K. Crosser, Condensation heat transfer within horizontal tubes, Chem. Eng. Prog. Symp. Ser. 55 (1959) 171–176.

[45] G.A. Longo, G. Righetti, C. Zilio, A new computational procedure for refrigerant condensation inside herringbone-type Brazed Plate Heat Exchangers, Int. J. Heat Mass Transf. 82 (2015) 530–536. doi:10.1016/j.ijheatmasstransfer.2014.11.032.

[46] R.L. Webb, Convective Condensation of Superheated Vapor, J. Heat Transfer. 120 (1998) 418–421. doi:10.1115/1.2824266.

[47] Y.Y. Hsieh, T.F. Lin, Evaporation Heat Transfer and Pressure Drop of Refrigerant R-410A Flow in a Vertical Plate Heat Exchanger, J. Heat Transfer. 125 (2003) 852. doi:10.1115/1.1518498.

[48] T.S. Khan, M.S. Khan, M.C. Chyu, Z.H. Ayub, Experimental investigation of evaporation heat transfer and pressure drop of ammonia in a 60° chevron plate heat exchanger, in: Int. J. Refrig., Elsevier, 2012: pp. 336–348. doi:10.1016/j.ijrefrig.2011.10.018.

[49] J.R. Thome, Mini- and Microchannel Condensation, in: Encycl. Two-Phase Heat Transf. Flow I, 2015: pp. 231–284. doi:10.1142/9789814623216_0015.

[50] D. Del Col, S. Bortolin, A. Cavallini, M. Matkovic, Effect of cross sectional shape during condensation in a single square minichannel, Int. J. Heat Mass Transf. 54 (2011) 3909–3920. doi:10.1016/j.ijheatmasstransfer.2011.04.035.

[51] H.S. Wang, J.W. Rose, A Theory of Film Condensation in Horizontal Noncircular Section Microchannels, J. Heat Transfer. 127 (2005) 1096. doi:10.1115/1.2033905.