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

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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.

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

### Nomenclature

#### Symbols

- $A$: heat transfer area, m$^2$
- $A_o$: cross-sectional area on working fluid side of the PHE, m$^2$
- $b$: amplitude of corrugation, m
- $B_o$: Bond number
- $c_p$: specific heat capacity, J/kg K
- $D$: diameter, m
- $f$: friction factor
- $G$: mass flux, kg/m$^2$s
- $h$: heat transfer coefficient, W/m$^2$K
- $H$: enthalpy, J/kg
- $K$: coverage factor
- $k$: thermal conductivity, W/m K
- $L$: length, m
- $\text{LMTD}$: log mean temperature difference, K
- $\dot{m}$: mass flow rate, kg/s
- $N$: number of channels
- $Nu$: Nusselt number
- $P$: pressure, Pa
- $Pr$: Prandtl number
- $P_r$: reduced pressure
- $\dot{Q}$: heat transfer rate, W
- $Re$: Reynolds number
- $t$: thickness, m
- $T$: temperature, °C
- $\dot{U}$: overall heat transfer coefficient, W/m$^2$K
- $\dot{V}$: volume flow rate, L/min
- $W$: width, m
- $We$: Weber number
- $x$: vapor quality

#### Subscripts

- $\text{con}$: condensation
- $\text{cri}$: critical
- $\text{dec}$: deceleration
- $\text{ele}$: elevation
- $\text{eq}$: equivalent
- $\text{exp}$: experimental
- $\text{fri}$: frictional
- $\text{h}$: hydraulic
- $\text{in}$: inlet
- $\text{lo}$: liquid only
- $\text{loc}$: local
- $\text{m}$: mean
- $\text{oil}$: oil
- $\text{out}$: outlet
- $\text{p}$: port
- $\text{pred}$: predicted
- $\text{sat}$: saturation
- $\text{sub}$: subcooled
- $\text{sup}$: superheated
- $\text{tot}$: total
- $\text{v}$: vapor
- $\text{w}$: water
- $\text{wall}$: wall
- $\text{wf}$: working fluid

#### Greek Symbols

- $\gamma$: dimensionless corrugation parameter
- $\beta$: chevron angle, °
- $\phi$: enlargement factor of corrugation surface
- $\mu$: dynamic viscosity, Pa•s
- $\rho$: mass density, kg/m$^3$
- $\lambda$: corrugation pitch, m
- $\Delta$: difference
- $\varepsilon$: heat loss percentage
- $\delta$: thickness of plate, m
- $\sigma$: surface tension, N/m

#### Abbreviations

- HC: hydrocarbon
- HFC: hydrofluorocarbon
- HFO: hydrofluoroolefin
- MAPD: mean absolute percentage deviation
- MBPD: mean bias percentage deviation
- ORC: organic Rankine cycle
- PHE: plate heat exchanger
- GWP: Global Warming Potential
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
The objective of the present study is to investigate the condensation heat transfer of selected HFCs and HFOs in a PHE at the working conditions prevailing in ORC units and heat pumps. The two HFCs, R134a and R245fa, and their two HFO replacements, R1234ze(E) and R1233zd(E), were evaluated for different mass flow rates and condensation temperatures. In terms of condensation temperatures, the focus of the current research is 30 °C to 60 °C for the working fluids R134a and R1234ze(E), and 40 °C to 70 °C for the working fluids R245fa and R1233zd(E). Condensation temperatures of 40 °C to 70 °C are typical for the condenser in heat pumps, while condensation temperatures of 30 °C to 40 °C are commonly used in the condenser of ORC power systems [9,10], and condensation temperatures of 50 °C to 70 °C are typical for the condenser in combined heat and power plants based on the ORC technology providing the heat for residential applications (e.g., [11]). We used the experimental results to identify the prevailing heat transfer mechanisms, and by comparison with the experimental data, we were able to evaluate the suitability of the existing heat transfer and pressure drop correlations. Subsequently, we developed a new prediction method for the thermal-hydraulic characteristics of condensation in PHEs by introducing dimensionless numbers, taking account for the effects of surface tension and refitting existing correlations using regression analysis of the experimental data presented in this paper. The intention is for these new correlations to be used as a future reference for the design of condensers in ORC systems and heat pumps.

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

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

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

<table>
<thead>
<tr>
<th>Year</th>
<th>Authors</th>
<th>Working fluids</th>
<th>Working conditions</th>
<th>Processlovakia</th>
<th>Film-convective transition</th>
<th>Configuration of PHE</th>
</tr>
</thead>
<tbody>
<tr>
<td>1999</td>
<td>Yan et al. [12]</td>
<td>R134a</td>
<td>$G = (60$ to $120) \text{ kg/m}^2\text{s}; P_{\text{con}} = (0.7$ to $0.9) \text{ MPa}$ ($26$ to $35 \degree\text{C}$)</td>
<td>Partial condensation; $x_m = 0.08$ to $0.86$</td>
<td>–</td>
<td>$\beta = 60\degree; \lambda = 10 \text{ mm}; a = 3.3 \text{ mm}; L_p = 450 \text{ mm}; W_p = 70 \text{ mm}; N = 3$</td>
</tr>
<tr>
<td>2000</td>
<td>Palmer et al. [15]</td>
<td>R22; R290; R290/600a; R32/152a</td>
<td>$Re = 20$ to $250; T_{\text{con},in} = 25 \degree\text{C}$</td>
<td>Complete condensation; $\Delta T_{\text{sup}} = 3.9 \degree\text{C}$</td>
<td>–</td>
<td>Brazed PHE; $N = 30$</td>
</tr>
<tr>
<td>2002</td>
<td>Thonon and Bontemps [16]</td>
<td>Propane; pentane; butane</td>
<td>$Re_{\text{lo}} = 100$ to $2000; P_{\text{con},in} = (1.5$ to $18) \text{ bar}$ (up to $80 \degree\text{C}$)</td>
<td>Complete condensation</td>
<td>$Re_{\text{el}} = 100$ to $1000$</td>
<td>$\beta = 45\degree; L = 300 \text{ mm}; W = 300 \text{ mm}$</td>
</tr>
<tr>
<td>2003</td>
<td>Han et al. [17]</td>
<td>R410; R22</td>
<td>$G = (60$ to $120) \text{ kg/m}^2\text{s}; T_{\text{con},in} = 20 \degree\text{C}$ and $30 \degree\text{C}$</td>
<td>Partial condensation; $x = 0.15$ to $0.9$</td>
<td>–</td>
<td>$\beta = 45\degree, 65\degree$ and $70\degree; \lambda = 4.9 \text{ mm}, 5.2 \text{ mm and 7.0 mm}; a = 2.55 \text{ mm}; L_p = 476 \text{ mm}; W_p = 69 \text{ mm}; N = 6$</td>
</tr>
<tr>
<td>2004</td>
<td>Würfel and Ostrowski [18]</td>
<td>Heptane</td>
<td>$Re_{\text{out}} = 350$ to $1650; Re_{\text{v},in} = 9500$ to $45000; P_{\text{con}} = 1 \text{ bar}$ ($\approx 98 \degree \text{C}$)</td>
<td>Complete condensation</td>
<td>$Re_{\text{el}} = 250$</td>
<td>$\beta = 30\degree/30\degree, 30\degree/60\degree$ and $60\degree/60\degree; L = 560 \text{ mm}; W = 180 \text{ mm}; N = 3$</td>
</tr>
<tr>
<td>2005</td>
<td>Kuo et al. [19]</td>
<td>R410A</td>
<td>$G = (50$ to $150) \text{ kg/m}^2\text{s}; T_{\text{con},in} = (20$ to $31.5) \degree\text{C}$</td>
<td>Partial condensation; $x_m = 0.10$ to $0.80$</td>
<td>–</td>
<td>$\beta = 60\degree; \lambda = 10 \text{ mm}; a = 3.3 \text{ mm}; L_p = 450 \text{ mm}; W_p = 70 \text{ mm}; N = 3$</td>
</tr>
<tr>
<td>2008</td>
<td>Djordjević [13]</td>
<td>R134a</td>
<td>$G = (30$ to $65) \text{ kg/m}^2\text{s}; T_{\text{con},in} = (26$ to $29) \degree\text{C}$</td>
<td>Complete condensation; $x_{\text{loc}} = 0.1$ to $0.9$</td>
<td>–</td>
<td>$\beta = 60\degree; \lambda = 10 \text{ mm}; a = 3.3 \text{ mm}; L_p = 450 \text{ mm}; W_p = 70 \text{ mm}; N = 3$</td>
</tr>
<tr>
<td>2012</td>
<td>Grabenstein</td>
<td>R365mfc</td>
<td>$G = (20$ to $80)$</td>
<td>Complete</td>
<td>–</td>
<td>$\beta = 63\degree$</td>
</tr>
<tr>
<td>and Kabelac [20]</td>
<td>kg/m²s; $P_{\text{con}} = (430$ to $640)$ kPa (87 to 103) °C</td>
<td>condensation; $x_{\text{loc}} = 0.05$ to 1</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------</td>
<td>-----------------</td>
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<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2012 to 2013</td>
<td>Mancin et al. [21–23]</td>
<td>R407C; R410A; R32</td>
<td>$G = (15$ to $40)$ kg/m²s; $T_{\text{con,in}} = 41.8$ °C and 36.5 °C</td>
<td>Partial condensation; $\Delta T_{\text{sup}} = 5$ to $25$ K; $x_{\text{out}} = 0.0$ to 0.65</td>
<td>$G_{\text{cri}} = 20$ kg/m²s</td>
<td></td>
</tr>
<tr>
<td>2015 to 2016</td>
<td>Sarraf et al. [24,25]</td>
<td>Pentane</td>
<td>$G = (9$ to $30)$ kg/m²s; $T_{\text{con,in}} = 41.8$ °C and 36.5 °C</td>
<td>Complete condensation; $\Delta T_{\text{sup}} = (5$ to $25)$ K</td>
<td>$G_{\text{cri}} = 15$ kg/m²s</td>
<td></td>
</tr>
<tr>
<td>2008 to 2015</td>
<td>Longo et al. [7,8,14,26–29]</td>
<td>R134a; R410A; R600a; R290; R1270; R1234yf; R1234ze(E); R152a</td>
<td>$G = (5.3$ to $11.4)$ kg/m²s; $Re_{l} = 120$ to $800$; $T_{\text{sat}} = (20$ to $40)$ °C</td>
<td>Partial condensation; $x_{\text{in}} = 0.92$ to 1, $x_{\text{out}} = 0.0$ to 0.09; complete condensation: $\Delta T_{\text{sup}} = (9.2$ to $11.2)$ K, $\Delta T_{\text{sub}} = (0$ to $4.9)$ K</td>
<td>$G_{\text{cri}} = (15$ to $20)$ kg/m²s; $Re_{\text{cri}} = 200$ to 400</td>
<td></td>
</tr>
</tbody>
</table>

2. Methods

2.1. Test facilities

![Figure 1 Schematic of a chevron corrugation plate in the condenser [30].](image)

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.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Measured values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length $L$</td>
<td>317 mm</td>
</tr>
<tr>
<td>Width $W$</td>
<td>76 mm</td>
</tr>
<tr>
<td>Port-to-port length $L_p$</td>
<td>278 mm</td>
</tr>
<tr>
<td>Port-to-port width $W_p$</td>
<td>40 mm</td>
</tr>
<tr>
<td>Diameter of inlet/outlet port $D_p$</td>
<td>18 mm</td>
</tr>
<tr>
<td>Chevron angle $\beta$</td>
<td>65 $^\circ$</td>
</tr>
<tr>
<td>Corrugation pitch $\lambda$</td>
<td>7 mm</td>
</tr>
<tr>
<td>Amplitude of corrugation $b$</td>
<td>1 mm</td>
</tr>
<tr>
<td>Hydraulic diameter $D_h$ (see the definition in Sec. 3.1)</td>
<td>3.4 mm</td>
</tr>
</tbody>
</table>

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

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

Figure 2 Schematic of the test facility.
and flow rate of the chilled water. Table 3 summarizes the operating conditions for the experimental tests.

Table 3 Condenser test operating conditions.

<table>
<thead>
<tr>
<th>$T_{\text{sat}}$ (°C)</th>
<th>$P_{\text{sat}}$ (bar)</th>
<th>$G_{\text{wf}}$ (kg/m$^2$s)</th>
<th>$q$ (kW/m$^2$)</th>
<th>Re</th>
<th>$\Delta T_{\text{sup}}$ (K)</th>
<th>$\Delta T_{\text{sub}}$ (K)</th>
<th>$V_w$ (L/min)</th>
<th>$T_{w,\text{in}}$ (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>29.7 to 71.0</td>
<td>2.9 to 16.3</td>
<td>16.0 to 90.0</td>
<td>4.0 to 57.4</td>
<td>65 to 877</td>
<td>1.6 to 4.9</td>
<td>0 to 5.3</td>
<td>14.9 to 15.6</td>
<td>13.8 to 65.2</td>
</tr>
</tbody>
</table>

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 $\varepsilon$ is defined as

$$\varepsilon = \frac{\dot{Q}_{w,\text{con}} - \dot{Q}_{\text{wf,con}}}{\dot{Q}_{w,\text{con}}} \times 100\%,$$

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

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

and

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

$T_{w,\text{in}}$ 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 $H_{\text{wf,in}}$ and $H_{\text{wf,out}}$ 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 $G_{\text{wf}}$ (mass flow rate per unit cross-sectional area of flow), where $G_{\text{wf}}$ was calculated based on the
cross-sectional area on working fluid side of the PHE $A_o$, defined as [32]

$$A_o = 2bWN_{wf},$$  \hspace{1cm} (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/m$^2$s to 80 kg/m$^2$s, 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).](image)

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

$$D_h = \frac{4b}{\varphi},$$

(5)

where the dimensionless parameter $\varphi$ is the area enlargement factor caused by sinusoidal surface waviness and is calculated by

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

(6)

where $\gamma$ is a dimensionless corrugation parameter, defined as

$$\gamma = \frac{2\pi b}{\lambda}.$$

(7)

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

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

(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

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

(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

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

(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 $t_{\text{wall}}$ and $k_{\text{wall}}$ are the thickness and thermal conductivity of the plate, respectively.

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

$$A_{\text{sub}} = \frac{\hat{Q}_{\text{sub}}}{U_{\text{sub}} \text{LMTD}_{\text{sub}}},$$

(11)

where the log mean temperature difference in region III, $\text{LMTD}_{\text{sub}}$ is defined as

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

(12)

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

$$L_{\text{sub}} = \frac{A_{\text{sub}}}{A} L_p,$$

(13)

where $A$ is the total heat transfer area of the condenser. The ratio $A_{\text{sub}}/A$ 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, $\hat{Q}_{\text{sat+sup}}$ and $A_{\text{sat+sup}}$ are given by

$$\hat{Q}_{\text{sat+sup}} = \hat{Q} - \hat{Q}_{\text{sub}},$$

(14)

$$A_{\text{sat+sup}} = A - A_{\text{sub}},$$

(15)

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

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

(16)

The condensation heat transfer coefficient $h_{\text{wf}}$ for the sum of regions I and II is calculated as
\[
\frac{1}{h_{w,f}} = \frac{1}{U_{sat + sup}} - \frac{1}{h_{w,sat + sup}} - \frac{t_{wall}}{k_{wall}},
\]  
(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

\[
U_{sat + sup} = \frac{Q_{sat + sup}}{A_{sat + sup} LMTD_{sat + sup}},
\]  
(18)

where the log mean temperature difference for the sum of regions I and II, \(LMTD_{sat + sup}\) is calculated

as [34]

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

\[
\Delta P_{fr} = \Delta P_{tot} - \Delta P_p + \Delta P_{ele} + \Delta P_{dec} - \Delta P_{sub},
\]  
(20)

where the \(\Delta P_{tot}\) is the total pressure drop measure by the differential pressure transducer. The \(\Delta P_p\) is

caused by the manifold and port, defined as

\[
\Delta P_p = 0.75 G_p^2 \left( \frac{1}{2 \rho_{v, sup}} + \frac{1}{2 \rho_{l, out}} \right),
\]  
(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 \(\Delta P_{dec}\) and \(\Delta P_{ele}\) were estimated from the homogenous

model of the two-phase flow [35]:
\[
\Delta P_{\text{dec}} = G_{\text{wf}}^2 \Delta x \left( \frac{1}{\rho_v} - \frac{1}{\rho_l} \right),
\]
\[
\Delta P_{\text{ele}} = g \rho_m (L_p - L_{\text{sub}}),
\]
where \( \rho_l \) and \( \rho_v \) are the densities of liquid-phase and vapor-phase, respectively and \( \Delta x \) 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 \( \rho_m \) is calculated at the average vapor quality between the inlet and outlet \( x_m \), obtained by
\[
\frac{1}{\rho_m} = \frac{x_m}{\rho_v} + \frac{1-x_m}{\rho_l},
\]
where \( x_m \) is 0.5 for complete condensation. The single-phase pressure drop at region III \( \Delta P_{\text{sub}} \) was calculated by
\[
\Delta P_{\text{sub}} = 2f_{\text{sub}} \frac{L_{\text{sub}} G_{\text{wf}}^2}{D_h \rho_l},
\]
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
water and oil sides are defined as

\[ h_w = C_1 \text{Re}_w^{0.733} \text{Pr}_w^{0.14} \frac{k_w}{D_h (\mu_w/\mu_{w,\text{wall}})} \]  \hspace{1cm} (26)

\[ h_{\text{oil}} = C_2 \text{Re}_{\text{oil}}^{0.8} \text{Pr}_{\text{oil}}^{0.14} \frac{k_{\text{oil}}}{D_h (\mu_{\text{oil}}/\mu_{\text{oil,wall}})} \]  \hspace{1cm} (27)

where the \( \text{Re}_{\text{oil}}/\text{Re}_w \) and \( \text{Pr}_{\text{oil}}/\text{Pr}_w \) are the Reynolds number and Prandtl number of the oil/water, respectively, and \( \mu_{\text{oil}}/\mu_w \) and \( \mu_{\text{oil,wall}}/\mu_{w,\text{wall}} \) are the dynamic viscosities based on mean oil/water temperature and wall temperature, respectively.

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

\[ \text{Nu} = 0.4225 \text{Re}^{0.733} \text{Pr}^{0.14} (\mu/\mu_{\text{wall}}), \hspace{1cm} 400 < \text{Re} < 1100, 2.8 < \text{Pr} < 4.5. \]  \hspace{1cm} (28)

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

Table 4 Condenser operating test conditions.

<table>
<thead>
<tr>
<th>Test</th>
<th>( C_1 )</th>
<th>( a )</th>
<th>( \text{Re}_w )</th>
<th>( \text{Pr}_w )</th>
<th>( \text{Re}_{\text{oil}} )</th>
<th>( \text{Pr}_{\text{oil}} )</th>
<th>( R^2 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>Test 1</td>
<td>0.4139</td>
<td>0.7391</td>
<td>700 to 1100</td>
<td>2.8 to 4.4</td>
<td>450</td>
<td>54</td>
<td>0.994</td>
</tr>
<tr>
<td>Test 2</td>
<td>0.4282</td>
<td>0.7287</td>
<td>580 to 940</td>
<td>2.9 to 4.5</td>
<td>360</td>
<td>65</td>
<td>0.981</td>
</tr>
<tr>
<td>Test 3</td>
<td>0.4254</td>
<td>0.7312</td>
<td>400 to 700</td>
<td>3.0 to 4.4</td>
<td>275</td>
<td>82</td>
<td>0.997</td>
</tr>
</tbody>
</table>
Figure 5 Comparison of Eq. (28) with different water single-phase heat transfer correlations in the open literature for the working conditions of test 1.

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

### 2.4. Uncertainties analysis

In this study, the uncertainty of temperature measurement was ±0.19 K. The errors associated with the mass flow rate and volume flow rate measurements were ±0.015 % and ±0.5 %, respectively. The uncertainty of pressure and pressure difference uncertainties were ±0.45 % FS (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.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainty (K = 2)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulic diameter, $D_h$</td>
<td>2.8 %</td>
</tr>
<tr>
<td>Heat transfer rate, $\dot{Q}$</td>
<td>1.7 % to 4.1 %</td>
</tr>
<tr>
<td>Water heat transfer coefficient, $h_w$</td>
<td>2.4 % to 5.0 %</td>
</tr>
<tr>
<td>Working fluid heat transfer coefficient, $h_{wf}$</td>
<td>3.8 % to 12.1 %</td>
</tr>
<tr>
<td>Frictional pressure drop, $\Delta P_{fr}$</td>
<td>3.2 % to 15.2 %</td>
</tr>
</tbody>
</table>

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

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

where $x_m$ 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
Reynolds number, the results were independent of the liquid Reynolds number, indicating that the processes are characterized by film condensation. The transition between the film and convective condensation regions occurs at $Re_l \approx 200$. However, for R245fa and R1233zd(E), the shear-controlled region dominates throughout the whole range of the liquid Reynolds number, and no film condensation was found. The results suggest that the heat transfer coefficients increase when the saturation temperatures decrease.
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 \( \Delta P_{fri} \). 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>
<thead>
<tr>
<th></th>
<th>30 °C</th>
<th>40 °C</th>
<th>50 °C</th>
<th>60 °C</th>
<th>70 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>R1234ze(E)/R134a</td>
<td>( h_{wf} )</td>
<td>35 %</td>
<td>25 %</td>
<td>14 %</td>
<td>12 %</td>
</tr>
<tr>
<td></td>
<td>( \Delta P_{fri} )</td>
<td>31 %</td>
<td>17 %</td>
<td>23 %</td>
<td>24 %</td>
</tr>
<tr>
<td>R1233zd(E)/R245fa</td>
<td>( h_{wf} )</td>
<td>–</td>
<td>11 %</td>
<td>17 %</td>
<td>24 %</td>
</tr>
<tr>
<td></td>
<td>( \Delta P_{fri} )</td>
<td>–</td>
<td>8 %</td>
<td>7 %</td>
<td>21 %</td>
</tr>
</tbody>
</table>

3.2. Pressure drop

![Diagram](image)
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 decrease. 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 $T_{\text{sat}} = 30 \, ^{\circ}\text{C}$ and 26 % at $T_{\text{sat}} = 70 \, ^{\circ}\text{C}$, 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):

\[
\text{MAPD} = \frac{1}{n} \sum_{i=1}^{n} \left| \frac{\text{data}_{i,\text{pred}} - \text{data}_{i,\exp}}{\text{data}_{i,\exp}} \right| \times 100 \%, 
\]

(30)

and

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

(31)

where the \( \text{data}_{i,\text{pred}} \) and \( \text{data}_{i,\exp} \) 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 \( \text{Re}_{\text{eq}} \), accounting for the vapor core and the interfacial shear stress:

\[
\text{Re}_{\text{eq}} = \frac{G_{\text{eq}} D_h}{\mu_l}, 
\]

(32)

where the equivalent mass flux \( G_{\text{eq}} \) is defined as

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

(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:

\[ h_{\text{wf}, m} = \left( \frac{1}{A} \right) \int_0^A h \, dA. \] \hspace{1cm} (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.
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 MBPD 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.
However, both correlations overestimate most of the experimental data. Based on the comparison results shown in Figures 8 and 9, one may conclude that more accurate correlations need to be developed for predicting the heat transfer and pressure drop data in this study.

![Figure 9](image)

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 \( \sigma \), was introduced in the new heat transfer correlation. We used two equations including the Bond number to regress the experimental data:

\[ \text{Nu} = C \text{Re}_{eq}^{0.4} \text{Pr}_{l}^{1/3} \text{Bo}^{b}. \]  
(35)

\[ \text{Nu} = C \text{Re}_{eq}^{a} \text{Pr}_{l}^{1/3} \text{Bo}^{b}. \]  
(36)

where the Bond number is defined as

\[ \text{Bo} = \frac{g(\rho_{l} - \rho_{v})D_{h}^{2}}{\sigma}. \]  
(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 \( \text{Re}_{eq} \) and \( \text{Pr}_{l} \) and only fitting the exponent of Bo and the first constant, while Eq. (36) was used to fit further the exponent of \( \text{Re}_{eq} \) 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

\[ f = C \text{Re}_{eq}^{a} \text{We}^{b}. \]  
(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].
addition, it includes the dimension number, Weber number (We), defined as

\[ \text{We} = \frac{G_{wf}^2 D h}{\rho m \sigma} \]  

(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:

\[ f = 0.0146 \text{Re}_{eq}^{0.9814} \text{We}^{-1.0064}. \]  

(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.

<table>
<thead>
<tr>
<th>Equation</th>
<th>All fluids</th>
<th>R1233zd(E)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>MAPD</td>
<td>MBPD</td>
</tr>
<tr>
<td>Yan et al. [12]</td>
<td>[ \text{Nu} = 4.118 \text{Re}<em>{eq}^{0.4} \text{Pr}</em>{l}^{1/3} ]</td>
<td>11.8 %</td>
</tr>
</tbody>
</table>
Eq. (35) \[ \text{Nu} = 12.36 \text{Re}_{\text{eq}}^{0.4} \text{Pr}_l^{1/3} \text{Bo}^{-0.3715} \]

<table>
<thead>
<tr>
<th>Eq. (36)</th>
<th>[ \text{Nu} = 4.3375 \text{Re}_{\text{eq}}^{0.5383} \text{Pr}_l^{1/3} \text{Bo}^{-0.3872} ]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nu</td>
<td>12.36 \text{Re}_{\text{eq}}^{0.4} \text{Pr}_l^{1/3} \text{Bo}^{-0.3715}</td>
</tr>
<tr>
<td></td>
<td>4.3375 \text{Re}_{\text{eq}}^{0.5383} \text{Pr}_l^{1/3} \text{Bo}^{-0.3872}</td>
</tr>
</tbody>
</table>

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

<table>
<thead>
<tr>
<th>Re_{eq}</th>
<th>Pr_l</th>
<th>Bo</th>
<th>We</th>
</tr>
</thead>
<tbody>
<tr>
<td>1207 – 4827</td>
<td>3.1 – 6.5</td>
<td>11.7 – 28.3</td>
<td>1.8 – 68.2</td>
</tr>
</tbody>
</table>

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
intensified convection at the vapor-liquid interface.

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

Although involving surface tension to develop the correlation enables the better prediction for the experimental data obtained in this study, the physics on how surface tension affects the condensation flow in a plate heat exchanger are not clear due to the limited number of studies available in the field; especially visualization studies are lacking. Nevertheless, an effort is made...
here briefly to explain some of the physical phenomena. In noncircular micro and mini-channels, the surface tension generates a transverse pressure gradient in the condensate film, which leads to a condensate flow towards the corners and thins the film along the flat sides of the channel, lowering the thermal resistances. This is in accordance with the essence of surface tension, i.e., minimizing the surface area of the fluid. The effects of surface tension on flow condensation in noncircular micro and mini-channels have been experimentally proved and theoretically analysed by numerous research works (e.g., Refs. [50,51]). In terms of brazed PHEs, the contact points where two plates are brazed together play a similar role as those of the corners in noncircular micro and mini-channels, i.e., the condensate film is pulled towards the contact points leading to a thinner liquid film on the remaining parts of the channel. The inference for the distribution of the condensate film in the flow passage of PHEs needs to be verified, preferably by experimental work.

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

5. Conclusions

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

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