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Condensation heat transfer and pressure drop characteristics of zeotropic mixtures of R134a/R245fa in plate heat exchangers

Ji Zhang*, Brian Elmegaard, Fredrik Haglind

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

Abstract

Zeotropic mixtures have shown great potential to improve the system efficiency compared with the use of pure fluids as the working fluids. However, there is a lack of analysis of the heat transfer during the phase change of zeotropic mixtures in plate heat exchangers, which is a common heat transfer device used as the evaporator and condenser in these systems. This paper presents an experimental analysis of the flow condensation heat transfer and pressure drop characteristics of zeotropic mixtures in a plate heat exchanger. The objectives are to obtain prediction methods for the heat transfer coefficient and frictional pressure drop and to quantify the degradation of the heat transfer coefficient by using mixtures. Zeotropic mixtures of R134a/R245fa with five different quantitative compositions were experimentally tested with the bubble point temperatures of 30 °C to 50 °C and various mass fluxes. The experimental results indicate that condensation of the mixtures is governed by a shear-controlled process, similar to what happens with pure fluids. Moreover, the heat transfer degradation of the zeotropic mixture caused by the mixture effects increases with decreasing condensation temperature and mass flux. There is a 48 % maximum decrease in the heat transfer coefficient for the mixture of R134a/R245fa with 0.431/0.569 in mass fraction. A modified Silver-Bell-Ghaly method provides a good prediction for the heat transfer data with a mean absolute percentage deviation of 12.2%. A pure-fluid correlation developed in a previous work by the authors enables a prediction with an 8.6 % mean absolute percentage deviation for the pressure drop data.

Keyword: zeotropic mixture, condensation, plate heat exchanger, prediction method

* Corresponding author. Tel.: +45 45 25 13 87; fax: +45 45 25 19 61
E-mail address: jizhang@mek.dtu.dk (Ji Zhang)
## Nomenclature

### Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>heat transfer area, m$^2$</td>
</tr>
<tr>
<td>$Bd$</td>
<td>Bond number, $-$</td>
</tr>
<tr>
<td>$c_p$</td>
<td>specific heat capacity, J/kg K</td>
</tr>
<tr>
<td>$D$</td>
<td>diameter, m</td>
</tr>
<tr>
<td>$F$</td>
<td>friction factor, $-$</td>
</tr>
<tr>
<td>$f_i$</td>
<td>interfacial roughness correction factor</td>
</tr>
<tr>
<td>$G$</td>
<td>mass flux, kg/m$^2$s</td>
</tr>
<tr>
<td>$H$</td>
<td>enthalpy, J/kg</td>
</tr>
<tr>
<td>$h_{lg}$</td>
<td>latent heat of condensation, J/kg</td>
</tr>
<tr>
<td>$L$</td>
<td>length, m</td>
</tr>
<tr>
<td>LMTD</td>
<td>log mean temperature difference, K</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass flow rate, kg/s</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure, Pa</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number, $-$</td>
</tr>
<tr>
<td>$Q$</td>
<td>heat transfer rate, W</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number, $-$</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature, °C</td>
</tr>
<tr>
<td>$U$</td>
<td>overall heat transfer coefficient, W/m$^2$K</td>
</tr>
<tr>
<td>$u_{dr}$</td>
<td>drift velocity between the two phases, m/s</td>
</tr>
<tr>
<td>$X$</td>
<td>vapor quality, $-$</td>
</tr>
<tr>
<td>$Z$</td>
<td>ratio between the sensible heat transfer of vapor phase and the total heat transfer, $-$</td>
</tr>
</tbody>
</table>

### Subscripts

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>b</td>
<td>bubble point</td>
</tr>
<tr>
<td>con</td>
<td>condensation</td>
</tr>
<tr>
<td>d</td>
<td>dew point</td>
</tr>
<tr>
<td>dec</td>
<td>deceleration</td>
</tr>
<tr>
<td>ele</td>
<td>elevation</td>
</tr>
<tr>
<td>eq</td>
<td>equivalent</td>
</tr>
<tr>
<td>exp</td>
<td>experimental</td>
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<tr>
<td>fri</td>
<td>frictional</td>
</tr>
<tr>
<td>film</td>
<td>liquid film</td>
</tr>
<tr>
<td>g</td>
<td>glide</td>
</tr>
<tr>
<td>h</td>
<td>hydraulic</td>
</tr>
<tr>
<td>in</td>
<td>inlet</td>
</tr>
<tr>
<td>l</td>
<td>liquid</td>
</tr>
<tr>
<td>m</td>
<td>mean</td>
</tr>
<tr>
<td>mix</td>
<td>zeotropic mixture</td>
</tr>
<tr>
<td>out</td>
<td>outlet</td>
</tr>
<tr>
<td>p</td>
<td>port</td>
</tr>
<tr>
<td>pred</td>
<td>predicted</td>
</tr>
<tr>
<td>pure</td>
<td>pure working fluid</td>
</tr>
<tr>
<td>s</td>
<td>superficial</td>
</tr>
<tr>
<td>tot</td>
<td>total</td>
</tr>
<tr>
<td>v</td>
<td>vapor</td>
</tr>
<tr>
<td>w</td>
<td>water</td>
</tr>
<tr>
<td>wall</td>
<td>wall</td>
</tr>
<tr>
<td>wf</td>
<td>working fluid</td>
</tr>
</tbody>
</table>

### Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$M$</td>
<td>dynamic viscosity, Pa$\cdot$s</td>
</tr>
<tr>
<td>$\rho$</td>
<td>mass density, kg/m$^3$</td>
</tr>
<tr>
<td>$\rho^*$</td>
<td>density ratio of liquid phase to vapor phase, $-$</td>
</tr>
<tr>
<td>$\Delta$</td>
<td>difference</td>
</tr>
<tr>
<td>$\Delta$</td>
<td>thickness, m</td>
</tr>
<tr>
<td>$\Sigma$</td>
<td>surface tension, N/m</td>
</tr>
<tr>
<td>$E$</td>
<td>void fraction, $-$</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>heat transfer coefficient, W/m$^2$K</td>
</tr>
<tr>
<td>$\alpha_v$</td>
<td>a basic single-phase heat transfer coefficient of vapor calculated using Eq. (19), W/m$^2$K</td>
</tr>
</tbody>
</table>

### Abbreviations

- MAPD: mean absolute percentage deviation
- PHE: plate heat exchanger
1. Introduction

Utilization of zeotropic mixtures as working fluids in thermal systems has been widely studied, revealing a great potential for improved temperature matching between the fluids exchanging heat and thus for improved system performance [1,2]. The benefits of using zeotropic mixtures in e.g. heating, ventilating, air conditioning and refrigerating (HVAC&R) and power generation systems have been demonstrated in the open literature. For instance, utilization of zeotropic mixtures enables a 30 % to 35 % improvement of the thermodynamic performance and thus a 8 % to 10 % cost reduction for a heat pump system [3] and a 13.6 % higher net power output for an organic Rankine cycle unit [4].

According to vapor-liquid equilibrium properties, the mixtures can be classified into two groups, i.e. azeotropic mixtures and zeotropic mixtures. Azeotropic mixtures have identical compositions of the liquid and the vapor phases and no temperature glide, behaving as a pure fluid during the two-phase heat transfer. In contrast, zeotropic mixtures have different composition in the liquid and vapor phases and exhibit a non-isothermal process during phase change, characterized by a temperature glide $T_g$ (i.e. the temperature difference between the bubble point and dew point temperatures of the bulk mixture) [5]. Consequently, the zeotropic mixture condensation differs from the pure fluid condensation due to the “mixture effects” that are summarized as follows. First, as the condensation process is non-isothermal, sensible heat transfer in both the liquid and the vapor phase is required, i.e. the condensate and the remaining vapor need to be continually cooled down to the decreasing equilibrium temperature. The sensible heat transfer rate, particularly for the vapor phase, is much lower than the phase change process of condensation. This entails an extra heat transfer resistance. Second, the less volatile component of the vapor mixture at the vapor-liquid interface condenses more, and thus the more volatile component has a local concentration that is higher than the concentration of the more volatile component of vapor bulk. Due to the concentration gradient, mass diffusion of the more volatile component occurs towards the bulk and of the less volatile component towards the interface. Compared with the pure fluid condensation (a complete phase-change and identical concentration process), the extra heat and mass transfer resistances (the mixture effects) result in a degradation of the condensation rate, i.e. the heat transfer coefficient of the mixture at a certain composition is lower than the weighted average of the heat transfer coefficients of the pure components.

As for prediction methods of the mixture condensation heat transfer, Colburn and Drew [6] proposed the basic transport equations accounting for the heat and mass transfer resistances based on application of the film condensation model and vapor/liquid equilibrium at the interface. Although the approach suggested by Colburn and Drew [6] presents a physically realistic treatment of the mixture condensation process, the calculation of mass transfer in the vapor phase makes the method
difficult to apply for the practical industrial design. A model suggested by Silver-Bell-Ghaly [7] simplifies the calculation of the mass transfer resistance, under the hypothesis that complete mixing both in the liquid and in the vapor phase is reached, and that overall equilibrium is maintained [8]. Consequently, the mass transfer resistance is replaced by an estimation of the heat transfer resistance in the vapor [8]; see a detailed description of the method suggested by Silver-Bell-Ghaly [7] in Section 2.4. Plate heat exchangers (PHEs) are commonly used as condensers in various industrial applications. However, only very few research works presented the condensation characteristics of zeotropic mixtures in PHEs and made an effort to develop/identify prediction methods for the heat transfer and pressure drop. Thonon and Bontemps [9] investigated condensation of butane/propane mixtures of two mass fractions of 49%/51% and 28%/72% experimentally at four different condensation pressures and a range of liquid-only Reynolds numbers. Accordingly, the mixtures have a temperature glide of 8.0 °C to 12.8 °C within the working conditions used in the experiments. Mancin et al. [10] presented an experimental study of a partial condensation (outlet vapor quality $x_{out} = 0.15$ to 0.58) of a pre-defined zeotropic mixture, R407C (R32/R125/R134a: 0.23/0.25/0.52) under four different mass fluxes and a fixed condensation pressure of 1.61 MPa, where R407C has a temperature glide of about 5 °C. Compared with the well-studied thermal-hydraulic performance of zeotropic mixture condensation in conventional tubes, the state-of-the-art of mixture condensation in PHEs can be summarized as follows:

- The current experimental investigations only cover a limited range of working conditions, especially for the temperature glide of mixtures which is a crucially influential parameter for two-phase heat transfer of zeotropic mixtures [1,11].
- No comparison of the results between zeotropic mixtures and their pure components was conducted; thus an analysis of heat transfer degradation caused by mixture effects for the zeotropic mixture condensation in PHEs is lacking.
- No study presents pressure drop characteristics of zeotropic mixture condensation in PHEs.

Moreover, the two above-mentioned research works both used the method suggested by Silver-Bell-Ghaly [7] to predict the heat transfer results, and the predictive performances significantly differ for the experimental results obtained in the two studies. Specifically, Thonon and Bontemps [9] found that the method suggested by Silver-Bell-Ghaly [7] provides a good prediction for the experimental data at high flow rates but overestimated the data at low flow rates, while the method completely failed to predict the test data from Mancin et al. [10]. Although the method suggested by Silver-Bell-Ghaly [7] was able to predict the partial test data from Thonon and Bontemps [9], the authors did not provide information on the calculation method of the void fraction and the slope of the condensation curve, which were used for obtaining the thermal resistance caused by the mixture effects. Therefore, it is still unclear how to apply practically the method for zeotropic mixture condensation in PHEs.
from the research works presented by Thonon and Bontemps [9]. No prediction method for the test
data at low flow rates was further developed in the paper. On the other hand, Mancin et al. [10]
derived a new correction factor that is a function of the difference between the saturation temperature
$T_{sat}$ and the wall temperature $T_{wall}$ by a regression of experimental data, acting on the pure fluid model
for predicting the mixture heat transfer coefficient $\alpha_{mix}$. It was given as

$$\alpha_{mix} = \alpha_{pure}[1.074(T_{sat} - T_{wall})^{-0.386}].$$

Furthermore, the prediction methods suggested in the two studies [9,10] have not been evaluated by
data that are independent from the other studies due to a lack of research work in the field. In
conclusion, the thermal-hydraulic characteristics of zeotropic mixture condensation in PHEs needs to
be investigated further, and prediction models for heat transfer coefficients and frictional pressure
drop need to be identified and developed.

This paper presents an experimental analysis of flow condensation heat transfer and pressure
drop characteristics of zeotropic mixtures in a plate heat exchanger. The objectives are to obtain
prediction methods for the heat transfer coefficient and frictional pressure drop and to quantify the
degradation of the heat transfer coefficient by using mixtures. Accordingly, a series of experiments
was carried out using binary mixtures of two pure hydrofluorocarbon fluids, R134a and R245fa, at
five different mass fractions. These fluids are widely used in industry. In addition, there is a relatively
high difference in atmospheric boiling point between the two fluids (i.e. -26.3 °C for R134a and 15.3
°C for R245fa), enabling a high temperature glide of the R134a/R245fa zeotropic mixture. The paper
provides the following novel contributions to state-of-the-art:

- An experimental campaign with wider ranges of working conditions than those of previous works
  (see the details in Section 2.1).
- A comprehensive analysis of the heat transfer degradation mechanisms of zeotropic mixtures.
- An investigation of the pressure drop characteristics of zeotropic mixture condensation.
- A new heat transfer model based on the method suggested by Silver-Bell-Ghaly [7].

The paper proceeds with a description of the method in Section 2, including the experimental
campaign, calculation of mixture composition, data analysis and prediction methods of the heat
transfer coefficient and frictional pressure drop. Section 3 presents the experimental results of the
thermal-hydraulic performance, prediction result and thermal resistance. Sections 4 and 5 discuss the
important findings and report the conclusions of the study, respectively.

2. Methods

2.1 Experimental campaign

In the past we carried out a series of condensation experiments using pure fluids [12,13]. The
same test facilities and a small-scale PHE (employed as the test section condenser) were used in the
present study. Figure 1 shows a piping and instrumentation diagram, a photo and the details of the
Figure 1. The details of the test facility and the PHE: (a) a piping and instrumentation diagram, (b) a photo of the test rig, and (c) an illustration of a PHE chevron-corrugated plate and the values of the geometrical parameters of the plate.

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Measured values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length L (mm)</td>
<td>317</td>
</tr>
<tr>
<td>Width W (mm)</td>
<td>76</td>
</tr>
<tr>
<td>Port-to-port length Lp (mm)</td>
<td>278</td>
</tr>
<tr>
<td>Port-to-port width Wp (mm)</td>
<td>40</td>
</tr>
<tr>
<td>Diameter of inlet/outlet port Dp (mm)</td>
<td>36</td>
</tr>
<tr>
<td>Chevron angle β (°)</td>
<td>65</td>
</tr>
<tr>
<td>Corrugation pitch λ (mm)</td>
<td>7</td>
</tr>
<tr>
<td>Amplitude of corrugation b (mm)</td>
<td>1</td>
</tr>
</tbody>
</table>
plate geometry. As shown in Figure 1(a), the test rig 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 fluid, respectively. Thermocouples are installed at the inlet and outlet of the PHEs to measure the temperature of the working fluid. Moreover, a differential pressure transducer is used to measure the pressure difference of the working fluid between the inlet and outlet of the condenser. The condenser has 16 plates in total, 8 cooling water passes and 7 working fluid passes. The hydraulic diameter of a cross-corrugated channel between the two plates is 3.4 mm. Further details of the test facility and the PHE can be found in Refs. [13,14]. In order to implement a reasonable comparison between the previous results of pure fluids and the results of mixtures obtained in this study, each data point was measured following the same process as that of the pure fluid, i.e. an almost complete condensation process. Specifically, the working fluid mixtures at the inlet and outlet of the condenser were controlled. At the inlet it was slightly superheated vapor with a superheated degree $\Delta T_{\text{sup}}$ up to 5 K, while at the outlet it was close to saturated liquid with a vapor quality $x_{\text{out}}$ below 0.05 for avoiding a single-phase region of subcooled liquid in the condenser. We tested the R134a/R245fa mixtures with five different quantitative compositions (see the detailed values in Section 3.1) with the temperature glides varying from 3.0 °C to 13.9 °C, which is beneficial for studying the effect of temperature glide on zeotropic mixture condensation. All five mixtures were tested with the bubble point temperatures $T_b$ of 30 °C, 40 °C and 50 °C that are prevailing working conditions in the condensers of HVAC&R and power generation plants [5,15,16]. The corresponding condensation pressures $P_{\text{con}}$ are 3.0 bar to 11.9 bar. The mass fluxes $G_{\text{wf}}$ used in the experimental test are 21.0 kg/m²s to 79.6 kg/m²s.

2.2 Estimation of the mixture composition

The use of zeotropic mixtures in thermodynamic systems leads to a composition shift, i.e. a difference between the actual circulating composition in a steady-state operating system and the originally charged composition [17]. The composition shift is mainly caused by liquid holdup in the two-phase flow regions of the heat exchangers [17–19]. Specifically, in the two-phase heat transfer of the evaporator or condenser, the vapor has a higher concentration of the more volatile component, while the liquid has a lower concentration of the more volatile component. Moreover, the vapor has a much faster motion compared to the liquid phase due to the difference in their densities, and such velocity slip generates a liquid holdup in heat exchangers, resulting in a higher concentration of more of the volatile component in the circulating composition than that in the originally charged concentration.

The traditional measurement method of circulating composition is gas chromatography. Due to its disadvantages during the sampling process, including i) the change of mixture composition, ii) long sampling time and thus the difficulty of real-time measurement for the composition, and iii)
mandatory vapor phase for the sample, Bao and Zhao [20] suggested to calculate the real-time composition based on the measurement of the density of the circulating fluid. The latter approach was adopted in this paper. Accordingly, the density of liquid-phase mixture was measured at the outlet of the working fluid pump using a Coriolis mass flowmeter. Moreover, a calculated density of liquid-phase mixture can be obtained using Refprop 10.0 [21] with measured temperature and pressure and an assumed composition. Finally, based on an iterative procedure, a composition enabling a minimum deviation between the measured density and calculated density is defined as the actual circulating composition in the system.

2.3 Data reduction and uncertainty analysis

The condensation heat transfer coefficient of zeotropic mixtures $\alpha_{\text{mix}}$ was calculated as

$$\frac{1}{\alpha_{\text{mix}}} = \frac{1}{U} - \frac{1}{\alpha_w} \frac{\delta_{\text{wall}}}{k_{\text{wall}}},$$ (2)

where the $\delta_{\text{wall}}$ and $k_{\text{wall}}$ are the thickness and the thermal conductivity of the plate, respectively. The heat transfer coefficient of single-phase cooling water $\alpha_w$ was calculated by

$$\alpha_w = 0.4225Re_w^{0.733}Pr_w^{1/3}(\mu_w/\mu_{\text{wall}})^{0.14}(k/D_h),$$ (3)

where $Re_w$ and $Pr_w$ are the Reynolds number and the Prandtl number of water, respectively, and $\mu_w$ and $\mu_{\text{wall}}$ are the dynamic viscosity of water based on the mean water and wall temperatures, respectively. Equation (3) was developed using the modified Briggs and Young Wilson plot method in the previous work [13]. The detailed working conditions of the water side, including the temperatures at the inlet and outlet of the condenser, $T_{w,in}$ and $T_{w,out}$ and the volume flow rate of water $\dot{V}_m$ as well as the $Re_w$, $Pr_w$ and $\alpha_w$ are summarized in Table 1.

<table>
<thead>
<tr>
<th>$T_{w,in}$ (°C)</th>
<th>$T_{w,out}$ (°C)</th>
<th>$\dot{V}_m$ (L/min)</th>
<th>$Re_w$</th>
<th>$Pr_w$</th>
<th>$\alpha_w$ (kW/m²K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>16.8 to 41.1</td>
<td>21.3 to 47.4</td>
<td>16.2 to 16.8</td>
<td>859 to 1444</td>
<td>4.0 to 7.0</td>
<td>30.9 to 40.9</td>
</tr>
</tbody>
</table>

The overall heat transfer coefficient $U$ was calculated based on the log mean temperature difference (LMTD) method [22], which is defined as

$$U = \frac{c_{p,w}\dot{m}_w(T_{w,out}-T_{w,in})}{A \cdot \text{LMTD}},$$ (4)

where $T_{w,in}$ and $T_{w,out}$ are the temperatures of the cooling water at inlet and outlet of the condenser, respectively, $\dot{m}_w$ is the mass flow rate of the cooling water, $c_{p,w}$ is the average specific heat of the water in the condenser, and $A$ is the heat transfer area of the condenser. The LMTD was calculated as

$$\text{LMTD} = \frac{(T_{wf,out}-T_{w,in})-(T_d-T_{w,out})}{\ln[(T_{wf,out}-T_{w,in})/(T_d-T_{w,out})]},$$ (5)

where the $T_{wf,out}$ is the temperature of the mixture at the outlet of condenser and $T_d$ is the dew point.
temperature of the mixture. The temperature glide of the zeotropic mixture in the condensation process is contradictory to the hypothesis of the LMTD method, i.e. the temperature change of the fluid is solely caused by the heat exchange [23]. Qiao [23] therefore mathematically derived a correction factor $\beta/\alpha$ for modifying the LMTD, which can be applied to this study and shown as

$$LMTD = \frac{(T_{w_{out}}-T_{w_{in}} + \beta/\alpha)-T_{a_{in}}+T_{a_{out}}+\beta/\alpha)}{\ln[(T_{w_{out}}-T_{w_{in}} + \beta/\alpha)/(T_{a_{in}}+T_{a_{out}}+\beta/\alpha)]} - \frac{\beta}{\alpha}. \quad (6)$$

The $\beta/\alpha$ is a function of the difference of the saturation temperature $\Delta T_{sat}$ caused by the pressure drop across the heat exchanger, defined by [23]

$$\frac{\beta}{\alpha} = -\frac{\Delta T_{sat}}{U(UA \left[ \frac{1}{C_w} + \frac{1}{C_{wf}} \right])}, \quad (7)$$

where the $C_w$ and $C_{wf}$ are the heat capacity rates of water and working fluid, respectively. In the present study, the $\beta/\alpha$ is small varying from -0.19 to -0.01 due to the low pressure drop across the condenser, resulting in a negligible change in LMTD compared with original values calculated by Eq. (5). Therefore, we used the original definition (Eq. (5)) to calculate the LMTD in the paper.

The outlet vapor quality $x_{out}$ is defined as

$$x_{out} = 1 - \frac{m_{c_p,w}(T_{w_{out}}-T_{w_{in}})-m_{c_{w}}(h_{v,sup}-h_{d})}{m_{c_{w}}h_{lg}}, \quad (8)$$

where $h_{v,sup}$ is the enthalpy of superheated vapor at the inlet of the condenser, $h_{d}$ is the enthalpy at dew point, and $h_{lg}$ is the latent heat of condensation. The mean vapor quality of the inlet and outlet of the condenser $x_m$ was defined as

$$x_m = \left(1 + x_{out}\right)/2. \quad (9)$$

where the inlet vapor quality was defined to be 1. As described in the paper, we keep a slight superheating of up to 5 K at the inlet of the condenser. Consequently, the heat transfer rate of the vapor desuperheating accounts for only a small fraction, 1.2 % to 3.6 %, of the heat transfer rate of the whole condensation process, implying that not treating the vapor desuperheating region separately is expected to have a minor impact on the results.

The frictional pressure drop $\Delta P_{fri}$ of the working fluid mixtures was determined by

$$\Delta P_{fri} = \Delta P_{tot} - \Delta P_{p} + \Delta P_{ele} + \Delta P_{dec}, \quad (10)$$

where the total pressure drop $\Delta P_{tot}$ is measured by the differential pressure transducer. The deceleration and elevation pressure drops $\Delta P_{dec}$ and $\Delta P_{ele}$ were estimated from the homogenous model of the two-phase flow [24]:

$$\Delta P_{dec} = G_{wf}^2 (1 - x_{out})(\frac{1}{\rho_v} - \frac{1}{\rho_l}) \quad (11)$$

and

$$\Delta P_{ele} = g \rho_m L_p \quad (12)$$

where $\rho_l$ and $\rho_v$ are the liquid-phase and vapor-phase densities calculated using the mean vapor quality,
1. $G_{wf}$ is the mass flux of working fluids calculated based on the cross-sectional area of the PHE channel, and $L_p$ is the port-to-port length of the plate. The mean two-phase density in the condenser $\rho_m$ was calculated at the mean vapor quality $x_m$, obtained by

$$\frac{1}{\rho_m} = \frac{x_m}{\rho_v} + \frac{1-x_m}{\rho_l}. \quad (13)$$

4. The pressure drop across the inlet and outlet ports of the condenser $\Delta P_p$ is defined as

$$\Delta P_p = 0.75G_p^2 \left( \frac{1}{2\rho_{in}} + \frac{1}{2\rho_{out}} \right), \quad (14)$$

where the mass flux at port $G_p$ is calculated based on the cross-sectional area of the plate port, $\rho_{in}$ is the density of superheated vapor at the inlet and $\rho_{out}$ is the mean two-phase density at the outlet calculated based on the outlet vapor quality of the mixture. The friction factor $f$ of the working fluid was determined by

$$f = \frac{\Delta P_{fri} \rho_p}{2L_p G_{wf}^2}. \quad (15)$$

Table 2 summarizes the uncertainties of direct measurement, including the fluid density, temperature, flow rate, pressure and pressure differences, as well as the percent uncertainties of the three main parameters, circulating composition, heat transfer coefficient and frictional pressure drop that are calculated based on the directly measured values in accordance with the method suggested by Kline and McClintock [25]. It is worth noting that on the basis of the calculation method of circulating composition presented in Section 2.2, the uncertainty of the circulating composition is associated with the uncertainty of measured and calculated densities. Furthermore, the uncertainty of the calculated density is co-determined by the uncertainty resulting from the prediction models used in Refprop [21] and the measurement uncertainty of pressure and temperature that are used as the parameters for the density calculation.

2.4 Prediction methods

The model suggested by Silver-Bell-Ghaly [7] was adopted in the study to predict the condensation heat transfer coefficient of zeotropic mixtures $\alpha_{mix}$, defined by
\[
\frac{1}{\alpha_{\text{mix}}} = \frac{1}{\alpha_{\text{pure}}} + \frac{Z}{\alpha_v}, \tag{16}
\]

where \(\alpha_{\text{pure}}\) is calculated based on the pure fluid model using mixture properties, \(Z\) is the ratio of the sensible heat transfer coefficient of the pure fluid model using mixture properties, \(\alpha_v\) is the heat transfer coefficient of the vapor phase and the total heat transfer coefficient during the condensation, and \(\alpha_v\) is the heat transfer coefficient of the vapor phase. The ratio \(Z/\alpha_v\) represents the heat transfer resistance caused by the mixture effects.

Del Col et al. [8] simplified \(Z\) in Eq. (16), i.e. the ratio of the sensible heat transfer of the vapor phase to the total heat transfer during the condensation, calculated by

\[
Z = x c_{p,v} T_g / \dot{h}_{lg}, \tag{17}
\]

where \(c_{p,v}\) is the specific heat of the vapor phase. In this work, the local vapor quality \(x\) is replaced by the mean vapor quality \(x_m\). The parameter \(\alpha_v\) in Eq. (16), the heat transfer coefficient of the vapor phase, refers to the vapor-liquid interface and can be calculated based on the method suggested by Thome et al. [26]:

\[
\alpha_v = f_i \alpha_v^0, \tag{18}
\]

where \(\alpha_v^0\) represents a basic single-phase heat transfer coefficient of vapor and \(f_i\) is an interfacial roughness correction factor, accounting for heat transfer enhancement due to the interfacial shear between the condensate and the vapor. Thome et al. [26] demonstrated that \(f_i\) is an important factor influencing the convective condensation and argued that the shear of the high-speed vapor is transmitted to the liquid film across the interface. Hence, it increases the magnitude and number of waves generated at the interface, which in turn increases the available surface area for condensation, tending to increase the heat transfer. The basic vapor single-phase heat transfer coefficient is calculated using the well-known Dittus and Boelter [27] correlation:

\[
\alpha_v^0 = 0.023 Re_v^{0.8} Pr_v^{1/3} (k_v/D_h), \tag{19}
\]

where \(k_v\) is the thermal conductivity of the vapor phase and \(Re_v\) and \(Pr_v\) are the Reynolds number and the Prandtl number of the vapor phase, respectively. The Reynolds number of the vapor phase \(Re_v\) is calculated based on the mean velocity of the vapor phase \(u_v\) [8]:

\[
Re_v = \rho_v u_v D_h / \mu_v, \tag{20}
\]

where \(\mu_v\) and \(\rho_v\) are the dynamic viscosity and density of the vapor phase, and the mean velocity of the vapor phase \(u_v\) is calculated by

\[
u_v = \frac{x \rho_\text{swf}}{\rho_v \varepsilon}, \tag{21}
\]

where \(\varepsilon\) is the vapor void fraction. Zhu and Haglind [28] recently conducted a computational fluid dynamics analysis to study the two-phase flow in a PHE cross-corrugated channel. Based on the simulation data, the authors modified the void fraction model suggested by Zuber and Findlay [29], to derive a new model for the cross-corrugated channel PHE, defined as
\[ \varepsilon = \frac{u_{v,s}}{c_0 u_{m,s} + c_1 u_{dr}}, \]  
\[ (22) \]

with

\[ c_0 = 1.03 \left( \frac{1-x}{x} \right)^{0.02} \quad \text{and} \quad c_1 = \begin{cases} 1 & \text{for } \varepsilon \geq 0.25 \\ \left( \frac{1-x}{x} \right)^{0.125} & \text{for } \varepsilon < 0.25 \end{cases} \]  
\[ (23) \]

The vapor superficial velocity \( u_{v,s} \) and mean superficial velocity \( u_{m,s} \) are calculated by

\[ u_{v,s} = x G_{wf}/\rho_v \]  
\[ (24) \]

and

\[ u_{m,s} = G_{wf}/\rho_m. \]  
\[ (25) \]

The parameter \( u_{dr} \) denotes the drift velocity between two phases, given in the form

\[ u_{dr} = 1.53 \left[ \frac{g \sigma (\rho_l - \rho_v)}{\rho_l^2} \right]^{0.25}, \]  
\[ (26) \]

where \( \sigma \) is the surface tension of the liquid phase. The roughness factor \( f_i \) is defined as [26]

\[ f_i = 1 + \left( \frac{u_v}{u_l} \right)^{0.5} \left[ \frac{(\rho_l - \rho_v) g \delta_{film}}{\sigma} \right]^{0.25}, \]  
\[ (27) \]

where the \( \delta_{film} \) is the liquid film thickness, estimated using [30]

\[ \delta_{film} = D_h(1 - \varepsilon)/4, \]  
\[ (28) \]

and the mean velocity of the vapor phase \( u_l \) is calculated by

\[ u_l = \frac{(1-x) G_{wf}}{\rho_v (1-\varepsilon)}. \]  
\[ (29) \]

In our previous work [12], a heat transfer correlation for pure working fluids was derived. It was used in this study to calculate the value of \( \alpha_{pure} \) in Eq. (16) using the properties of mixtures

\[ \alpha_{pure} = 0.4703 R_{eq}^{0.5221} P_{wl}^{1/3} B_{d}^{0.1674} \rho^{*0.2126} (k_i/D_h). \]  
\[ (30) \]

\( P_{wi} \) is the liquid Prandtl number and \( R_{eq} \) is the equivalent Reynolds number, defined as

\[ R_{eq} = G_{eq} D_h/\mu_l, \]  
\[ (31) \]

where \( \mu_l \) is the liquid viscosity, and the equivalent mass flux \( G_{eq} \) is defined as

\[ G_{eq} = G_{wf} [1 - x_m + x_m (\rho_l/\rho_v)^{0.5}]. \]  
\[ (32) \]

The parameter \( B_d \) is the Bond number, and \( \rho^* \) is the density ratio of the liquid phase to vapor phase defined as

\[ B_d = g (\rho_l - \rho_v) D_h^2 / \sigma \]  
\[ (33) \]

and

\[ \rho^* = \rho_l/\rho_v. \]  
\[ (34) \]

Generally, the Silver-Bell-Ghaly method [7] applied in this study includes the following modifications: i) the modification of Thome and co-workers [8,26] for mixture heat transfer resistance, ii) a pure fluid correlation (Eq. (30)) for PHE condensation heat transfer [12], and iii) the void fraction model developed by Zhu and Haglind for a PHE cross-corrugated channel [28].
In our previous work [12] also a pressure drop correlation for pure working fluids was developed, defined as

\[ f = 11557.62Re_{eq}^{-1.0041}Bd^{0.3002}\rho^{0.4268}. \]  

We adopted this correlation to calculate the predicted values of pressure drop using the properties of the mixtures.

3. Results

3.1 Actual circulating compositions of mixtures

Figure 2 compares the circulating composition and charged composition of the five zeotropic mixtures. The results indicate a 1.4 % to 2.9 % higher mass fraction of R134a in the circulating composition. The implications on the estimation of the heat transfer coefficient are quantified in Section 4. In addition to the measured density, the enthalpy change of the mixture between the subcooled liquid at the inlet of the preheater and the superheated vapor at the outlet of evaporator was used to calculate the circulating composition, further validating the feasibility of the method. The results indicate a good agreement of the two circulating compositions, calculated using the density and enthalpy change of the working fluid mixtures (the deviation between the two compositions ranges from 0.5 % to 2 %).

![Figure 2. Comparison of originally charged composition and actual circulating composition.](image)

3.2 Heat transfer

Figure 3 depicts the heat transfer coefficient variation of the zeotropic mixtures as a function of the mass fluxes with different bubble point temperatures. The heat transfer coefficients range from 1538 W/m² K to 3474 W/m² K. As shown in the figure, the heat transfer coefficient increases with increasing mass flux, which is a typical characteristic for shear-controlled condensation. This is consistent with the results of pure fluids R134a and R245fa presented in Ref. [12]. However, the effect of temperature on the mixture heat transfer differs significantly from that of the pure fluid. Specifically, the heat transfer coefficients of pure fluids increased more dramatically with decreasing condensation temperature. The experimental results [12] show that the heat transfer coefficient of
R245fa and R134a increases by 42 % and 16 %, respectively, when varying the condensation temperature from 50 °C to 30 °C. Conversely, the condensation temperature shows less influence on the heat transfer coefficient of zeotropic mixtures, and the heat transfer coefficient of the mixture of R134a/R245fa with 0.431/0.569 in mass fraction is even almost independent of the temperature.

Figure 3. Heat transfer coefficients of the zeotropic mixtures as a function of mass flux with different bubble point temperatures.

In order to further study the mixture effect, Figure 4 presents the heat transfer coefficient of
zeotropic mixtures as a function of R134a mass fraction at different mass fluxes and bubble point temperatures, accompanied with the values of the pure fluids R134a and R245fa (R134a mass fraction equal to 1 and 0, respectively). The heat transfer degradation caused by the mixture effects was quantified as the decrease of mixture heat transfer coefficient compared with an ideal value (dotted line) calculated by a linear mass fraction weighting of pure component values (ideal mixing rule). Moreover, a solid line in the figure represents a polynomial fit of the seven data points, indicating a variation tendency of the mixture heat transfer coefficient with the mass fraction of R134a. The results indicate that the heat transfer degradation of the mixture increases with decreasing bubble point temperature and mass flux and shows a non-monotonic increase with the mass fraction of R134a. Among the five mass fractions studied, the R134a mass fraction of 0.431 and 0.923 resulted in the highest and the lowest heat transfer degradation, respectively. Notably, the decrease in heat transfer coefficient can go up to 48% for the mixture of R134a/R245fa with 0.431/0.569 in mass fraction at $T_b = 30 \, ^\circ C$ and $G_{wf} = 31 \, kg/m^2s$.

Figure 4. Heat transfer coefficient of zeotropic mixtures as a function of mass fraction of R134a at different mass fluxes and bubble point temperatures.

3.3 Pressure drop
Figure 5 shows the frictional pressure drop of zeotropic mixtures versus mass flux at different bubble point temperatures. The values range from 27.2 kPa/m to 76.9 kPa/m. The results suggest that the frictional pressure drop increases when the mass flux increases and when the bubble temperature decreases. Moreover, unlike for the heat transfer results, the frictional pressure drop of zeotropic mixtures increases monotonously with the decreasing mass fraction of R134a, and all the mixture values are between the values of the two pure fluids. This phenomenon is exemplified in Figure 6, presenting the average pressure drop calculated based on the same range of mass flux ($G_{wf} \approx 32$ kg/m²s to 68 kg/m²s) for a bubble point temperature of 50 °C.
Figure 5. Frictional pressure drop of the zeotropic mixtures as a function of mass flux with different bubble point temperatures.
Figure 6. A comparison of average frictional pressure drop at the bubble point temperature of 50 °C.

3.4 Prediction results of heat transfer and pressure drop

In order to quantify the deviations between the experimental and the predicted values, a parameter, the mean absolute percentage deviation (MAPD), was introduced. It was defined as

\[
\text{MAPD} = \frac{1}{n} \sum_{i=1}^{n} \left| \frac{\text{data}_{i,\text{pred}} - \text{data}_{i,\text{exp}}}{\text{data}_{i,\text{exp}}} \right| \times 100\%.
\] (36)

Figure 7 shows a comparison between the measured and predicted values for the heat transfer coefficient. As shown in the figure, the Silver-Bell-Ghaly method [7] adopted in this study (see details in Section 2.4) provides a good prediction for the test results, covering 97 % of the experimental data within a ±30 % bandwidth. Consequently, a 12.2 % MAPD was found for the prediction of heat transfer data.

Figure 7. Predicted heat transfer coefficients of zeotropic mixtures using the Silver-Bell-Ghaly method [7] versus the experimental results.

Figure 8 shows a comparison of experimental results with the predicted values calculated using Eq. (35). The results suggest that Eq. (35) is able to predict about 98 % of the experimental data within a ±30 % bandwidth, with a MAPD of 8.6 %, indicating that the correlation is applicable for mixtures.
Figure 8. Predicted frictional pressure drop of zeotropic mixtures using the Eq. (35) versus the experimental results.

3.5 Thermal resistance

The approach based on a modification of the Silver-Bell-Ghaly method [7], as presented in Section 2.4, showed a good prediction for the heat transfer data. Thus, the parameter ratio $Z/\alpha_v$ in the approach was used to estimate the thermal resistance caused by the mixture effect $R_{mix}$, defined as

$$R_{mix} = \frac{Z}{\alpha_v} \cdot \frac{1}{A} = \frac{x_{e_v} e_T \varphi}{h L} \cdot \frac{1}{\frac{1}{f_1 a_v^0} \cdot \frac{1}{A}}. \quad (37)$$

Figure 9 exemplifies the variation of heat transfer resistance of mixture effects versus the bubble point temperatures for the zeotropic mixture of R134a/R245fa at 0.640/0.360 mass fraction. The results in Figure 9 indicate that the thermal resistance increases with decreasing bubble point temperature (condensation temperature). This explains the phenomenon that the heat transfer coefficient of mixtures relies less on the condensation temperature than pure fluids do (see details in Section 3.2). On the one hand, the decrease of the condensation temperature results in a lower vapor density and thus a higher vapor velocity, promoting more shear stress at the interface and enhancing the heat transfer. On the other hand, the decrease of the condensation temperature entails a larger thermal resistance caused by mixture effects, degrading the heat transfer. The positive and negative effects on heat transfer triggered by the variation of condensation temperature may offset each other, resulting in the condensation temperature appearing to show less impact on the heat transfer.

Figure 9, furthermore, reveals that the thermal resistance caused by mixture effects decreases with increasing mass flux, which has already been demonstrated by the results of heat transfer degradation shown in Figure 4. This is because the higher vapor velocity at larger mass flux intensifies the heat transfer of the vapor phase, resulting in a larger value of $f_1 a_v^0$. This result is consistent with those found in horizontal smooth tubes by Shao and Granryd [31] and Smit et al. [32], where the authors found that the heat transfer degradation is larger at the lower mass fluxes dominated by a
Figure 9. Thermal resistance of R134a/R245fa at 0.640/0.360 mass fraction caused by mixture effects at different bubble point temperatures and mass fluxes.

Figure 10. Thermal resistance of zeotropic mixtures caused by mixture effects as a function of mass fluxes at the bubble point temperature of 50 °C.

Figure 10 presents the effects of mixture composition on thermal resistance $R_{\text{mix}}$ calculated using Eq. (37) under the condition of $T_b = 50$ °C. As shown in the figure, the comparison of results of the thermal resistance of different mixture compositions is in line with the results shown in Figure 4, e.g. the mixtures with R134a mass fractions of 0.431 and 0.923 have the highest and the lowest thermal resistance $R_{\text{mix}}$, respectively, resulting in the highest and the lowest heat transfer degradation. Furthermore, it is worth noting that the temperature glide determined by the mixture composition (see Table 3) is correlated with the thermal resistance (see Figure 10). In the research works presented by Shao and Granryd [26] and Smit et al. [27], the highest heat transfer degradation was also found for the mixtures with the largest temperature glides, i.e. a mass fraction of 0.265/0.735 for an R32/R134a mixture with a temperature glide of 5.8 K and a mass fraction of 0.5/0.5 for an R22/R142b mixture with a temperature glide of 6.54 K, respectively. This observation suggests that the temperature glide of a zeotropic mixture is a crucially influential parameter for the heat transfer resistance caused by...
Table 3. Temperature glide of zeotropic mixtures in the present experimental test with \( T_b \) = 50 °C.

<table>
<thead>
<tr>
<th>R134/R25fa mixture</th>
<th>( T_g ) (K)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.225/0.775</td>
<td>10.3</td>
</tr>
<tr>
<td>0.431/0.569</td>
<td>12.7</td>
</tr>
<tr>
<td>0.640/0.360</td>
<td>10.9</td>
</tr>
<tr>
<td>0.795/0.205</td>
<td>7.3</td>
</tr>
<tr>
<td>0.923/0.077</td>
<td>3.2</td>
</tr>
</tbody>
</table>

4. Discussion

As stated in Section 3.1, there was a difference of (0.014 to 0.029) between the R134a mass fraction of the originally charged composition and the actual circulating composition. In Table 4, the deviation between the heat transfer coefficients when using the charged composition and the circulating composition for the calculation is summarized. As shown, a maximum deviation ranging from 4.5 % to 8.2 % is identified for the five zeotropic mixtures with different compositions. This indicates a non-negligible effect on heat transfer caused by the difference between the circulating composition and the charged composition. Hence, it is essential to use the circulating composition for experimental analysis of mixtures.

Table 4. Deviation between the heat transfer coefficients calculated using charged composition and circulating composition.

<table>
<thead>
<tr>
<th>R134/R25fa mixture</th>
<th>0.225/0.775</th>
<th>0.431/0.569</th>
<th>0.640/0.360</th>
<th>0.795/0.205</th>
<th>0.923/0.077</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.6 % to 6.0 %</td>
<td>2.2 % to 4.5 %</td>
<td>3.6 % to 7.0 %</td>
<td>4.7 % to 8.0 %</td>
<td>2.9 % to 8.2 %</td>
<td></td>
</tr>
</tbody>
</table>

The findings for the pressure drop characteristic of zeotropic mixtures shown in Sections 3.3 and 3.4 (i.e. the variation of pressure drop with the mixture composition and the prediction method) indicate that the mixture effects have less impact on the pressure drop than on the heat transfer. The pressure drop of a zeotropic mixture can be approximately estimated by linear interpolation of the pressure drop of the two pure components composing the mixture.

A previous study evaluating the predictive performance of existing heat transfer correlations for pure fluids [33] indicates that the existing correlations generally fail to predict the results that are outside of the original working conditions used for the development of them. Therefore, when using the Silver-Bell-Ghaly method [7] (Eq. (16)) to predict the condensation heat transfer coefficient of zeotropic mixtures in PHEs, an appropriate pure fluid correlation for calculating the \( \alpha_{pure} \) needs to be selected based on the application. The correlation (Eq. (30)) used to calculate the \( \alpha_{pure} \) in this study was developed in our previous work [12] based on a comprehensive database including the experimental results of seven pure working fluids obtained with condensation temperatures of (30 to 90) °C. The correlation is aimed to be applicable for the PHE condenser in organic Rankine cycle and heat pump systems. Moreover, the good prediction for heat transfer data indicates that Eq. (37) is appropriate for estimating the condensation heat transfer resistance caused by the mixture effects in
5. Conclusions

An experimental analysis was conducted studying the condensation heat transfer and pressure drop performances of R134a/R245fa zeotropic mixtures with five different compositions in a plate heat exchanger, covering bubble point temperatures of 30 °C to 50 °C and various mass fluxes. A modified Silver-Bell-Ghaly method [7] was used to predict the mixture condensation heat transfer. The results indicate the necessity to use the actual circulating composition for the experimental analysis of zeotropic mixtures.

The experimental results suggest that the heat transfer coefficient and frictional pressure drop of mixtures both increase with decreasing mass flux, indicating a shear-controlled condensation process. Moreover, the frictional pressure drop of mixtures increases with decreasing condensation temperature, while the heat transfer coefficient shows less dependence on the condensation temperature due to a combined effect of the mixture property (decreased vapor density with the temperature) and the heat transfer resistance caused by mixture effects on the heat transfer.

The results indicate that a heat transfer degradation of the mixtures exists compared with an ideal mixing value calculated by a linear mass fraction weighting of the pure component values. The heat transfer degradation increases with decreasing condensation temperature and mass flux, and the mixtures with 0.431 and 0.923 mass fractions of R134a result in the highest and the lowest heat transfer degradation, respectively. The variation of the mixture heat transfer degradation with the other variables (temperature, mass flux and R134a mass fraction) is in line with the results of thermal resistance caused by the mixture effects. The maximum decrease in the heat transfer coefficient is 48 % for the mixture of R134a/R245fa with 0.431/0.569 in mass fraction at \( T_b = 30 \) °C and \( G_{nf} = 31 \) kg/m²s. Moreover, the temperature glide was identified as an influential parameter for the condensation heat transfer of zeotropic mixtures, i.e. a higher temperature glide leads to a larger heat transfer degradation.

The results suggest that the modified Silver-Bell-Ghaly method [7] provides a good prediction for the experimental data with a mean absolute percentage deviation of 12.2 %. Moreover, a pure fluid pressure drop correlation developed in our previous study [12] is able to predict the experimental results with a mean absolute percentage deviation of 8.6 %.

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