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HFO1234ze(E) Boiling Inside a Brazed Plate Heat Exchanger

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ABSTRACT

This paper investigates the effects of heat flux, saturation temperature, and outlet conditions on HFO1234ze(E) boiling inside a Brazed Plate Heat Exchanger (BPHE). The heat transfer coefficients show great sensitivity to heat flux and outlet conditions and weak sensitivity to saturation temperature (pressure). The frictional pressure drop shows a linear dependence on the refrigerant kinetic energy per unit volume. The two-phase flow boiling heat transfer coefficients were compared with a new model for refrigerant boiling inside BPHE (Longo et al., 2015): the mean absolute percentage deviation between calculated and experimental data is 7.2%. The present data points were compared with those of HFC134a and HFO1234yf previously measured inside the same BPHE under the same operating conditions: HFO1234ze(E) exhibits heat transfer coefficients very similar to HFC134a and HFO1234yf and frictional pressure drops slightly higher than HFC134a and HFO1234yf.

1. INTRODUCTION

HFC134a has been probably the most important refrigerant of the two past decades as it dominated the applications in domestic refrigeration, mobile air conditioning and large chillers and it took part as component in several refrigerant mixtures such as HFC404A, and HFC407C. Unfortunately HFC134a exhibits a relatively large, 1300, Global Warming Potential (GWP), and it will be subjected to a gradual reduction in the use up to a complete phase out in the near future according to the different national and international regulations. For example the most recent release of the EU F-gas regulation (Regulation (EU) No 517/2014) established the complete phase out of HFC134a in domestic refrigeration and mobile air-conditioning systems since January 1st, 2015 and in centralised refrigeration systems since January 1st, 2022. Therefore it is essential to identify low GWP replacements for HFC134a.

The HydroFluoroOlefin (HFO) refrigerants HFO1234yf and HFO1234ze(E) seem to be the most promising candidates as they exhibit very low GWP values (1 or less) together with pressure and volumetric properties closely near to those of HFC134a. The unique drawback of HFO refrigerants seems to be their mild flammability (Class A2L of ANSI / ASHRAE Standard 34, 2013). The Brazed Plate Heat Exchangers (BPHEs), which involve a reduction of the refrigerant charge of one order of magnitude as compared to the traditional tubular heat exchangers, are particularly interesting for limiting the risk of flammable or mildly flammable refrigerants such as HFOs (Palm, 2007). In fact the first attempt to reduce the risk of flammable refrigerants is to decrease the refrigerant charge. The authors of the present paper had already tested HFO1234yf both in condensation and vaporisation inside a BPHE and compared its performance to those of HFC134a (Longo, 2012; Longo and Zilio, 2013). They had also carried out experimental tests on HFO1234ze(E) condensation inside a BPHE (Longo et al., 2014). This paper presents the heat transfer coefficients and pressure drops measured during HFO1234ze(E) vaporisation inside a BPHE: the effects of heat flux, saturation temperature, and outlet conditions are investigated.
2. EXPERIMENTAL SET-UP AND PROCEDURES

The experimental facility, shown in Figure 1, consists of three different circuits: one for the refrigerant and two for the secondary fluids (water and an aqueous ethylene glycol solution). The evaporator tested is a BPHE consisting of 10 plates, 72 mm in width and 278 mm in length, which present a macro-scale herringbone corrugation with an inclination angle of 65° and a corrugation amplitude of 2 mm. Figure 2 and Table 1 give the main geometrical characteristics of the BPHE tested, whereas Table 2 outlines the main features of the different measuring devices in the experimental rig. A detailed description of the experimental rig, the measurement devices and the operating procedures is reported by Longo and Gasparella (2007). The experimental results are reported in terms of boiling heat transfer coefficients and frictional pressure drop.

![Figure 2: Schematic view of the plate](image)

![Table 1: Geometrical characteristics of the BPHE](table)

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid flow plate length L(mm)</td>
<td>278.0</td>
</tr>
<tr>
<td>Plate width W(mm)</td>
<td>72.0</td>
</tr>
<tr>
<td>Area of the plate A(m²)</td>
<td>0.02</td>
</tr>
<tr>
<td>Enlargement factor Φ</td>
<td>1.24</td>
</tr>
<tr>
<td>Corrugation type</td>
<td>Herringbone</td>
</tr>
<tr>
<td>Angle of the corrugation β(°)</td>
<td>65</td>
</tr>
<tr>
<td>Corrugation amplitude b(mm)</td>
<td>2.0</td>
</tr>
<tr>
<td>Corrugation pitch p(mm)</td>
<td>8.0</td>
</tr>
<tr>
<td>Number of plates</td>
<td>10</td>
</tr>
<tr>
<td>Number of effective plates N</td>
<td>8</td>
</tr>
<tr>
<td>Channels on refrigerant side</td>
<td>4</td>
</tr>
<tr>
<td>Channels on water side</td>
<td>5</td>
</tr>
</tbody>
</table>
### Table 2: Specification of the different measuring devices

<table>
<thead>
<tr>
<th>Device</th>
<th>Type</th>
<th>Uncertainty (k= 2)</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermometer</td>
<td>T-type thermocouple</td>
<td>0.1 K</td>
<td>-20 / 80°C</td>
</tr>
<tr>
<td>Differential thermometer</td>
<td>T-type thermopile</td>
<td>0.05 K</td>
<td>-20 / 80°C</td>
</tr>
<tr>
<td>Abs. pressure transducer</td>
<td>Strain-gage</td>
<td>0.075% f.s.</td>
<td>0 / 1.0 MPa</td>
</tr>
<tr>
<td>Diff. pressure transducer</td>
<td>Strain-gage</td>
<td>0.075% f.s.</td>
<td>0 / 0.3 MPa</td>
</tr>
<tr>
<td>Refrigerant flow meter</td>
<td>Coriolis effect</td>
<td>0.1% measured value</td>
<td>0 / 300 kg/h</td>
</tr>
<tr>
<td>Water flow meter</td>
<td>Magnetic</td>
<td>0.15% f.s.</td>
<td>100 / 1200 l/h</td>
</tr>
</tbody>
</table>

### 3. DATA REDUCTION

The boiling heat transfer coefficient \( h_r \) is computed from the overall heat transfer coefficient \( U \) by determining the water side heat transfer coefficient \( h_w \).

\[
h_r = \left( \frac{1}{U} - s / \lambda_p - 1 / h_w \right)^{-1}
\]  

(1)

The overall heat transfer coefficient in the evaporator \( U \) is equal to the ratio between the heat flow rate \( Q \), the nominal heat transfer area \( S \) and the logarithmic mean temperature difference \( \Delta T_{ln} \).

\[
U = \frac{Q}{S \Delta T_{ln}}
\]  

(2)

The heat flow rate is derived from a thermal balance on the waterside of the evaporator:

\[
Q = m_w c_{pw} |\Delta T_{w}|
\]  

(3)

Where \( m_w \) is the water mass flow rate, \( c_{pw} \) the water specific heat capacity and |\( \Delta T_{w} | \) the absolute value of the temperature variation on the water side of the evaporator. The nominal heat transfer area of the evaporator

\[
S = N A
\]  

(4)

is equal to the nominal projected area \( A = L \times W \) of the single plate multiplied by the number \( N \) of the effective elements in heat transfer, as suggested by Shah and Focke (1988). When the evaporator works only in two-phase heat transfer the logarithmic mean temperature difference is equal to:

\[
\Delta T_{ln} = \left( \frac{1}{(T_{w} - T_{sat})} \right) / \ln \left( \frac{T_{w} - T_{sat}}{T_{sat} - T_{sat}} \right)
\]  

(5)

Where \( T_{sat} \) is the average saturation temperature of the refrigerant derived from the average pressure measured on refrigerant side and \( T_{wi} \) and \( T_{wo} \) the water temperatures at the inlet and the outlet of the evaporator. When the evaporator works both in vaporisation and super-heating, Dutto et al. (1991) and Fernando et al. (2004) suggested the following expression for the logarithmic mean temperature difference:

\[
\Delta T_{ln} = \frac{Q}{(Q_{boil} / \Delta T_{ln,boil}) + (Q_{sup} / \Delta T_{ln,super})}
\]  

(6)

where

\[
Q_{boil} = m_w c_{pw} (T_{w} - T_{sat})
\]  

(7)

\[
Q_{sup} = m_w c_{pw} (T_{w} - T_{sat})
\]  

(8)

are the heat flow rate exchanged in the boiling and super-heating zones respectively, \( \Delta T_{ln,boil} = (T_{w} - T_{sat}) / \ln \left( \frac{T_{w} - T_{sat}}{T_{sat} - T_{sat}} \right) \) \( \Delta T_{ln,super} = (T_{w} - T_{sat}) / \ln \left( \frac{T_{w} - T_{sat}}{T_{sat} - T_{sat}} \right) \) (9) are the logarithmic mean temperature difference in the boiling (eq. 9) and super-heating (eq. 10) zones respectively, whereas \( T_{w} \) is the water temperature between the super-heating and the boiling zone and \( T_{sat} \) is the refrigerant temperature at the outlet of the evaporator. The water temperature between the super-heating and the boiling zone is calculated from:

\[
T_{w} = T_{w} - m_w c_{pw} (T_{w} - T_{sat}) / (m_w c_{pw})
\]  

(11)
where \( m_r \) is the refrigerant mass flow rate and \( c_{pV} \) is the specific heat capacity of the refrigerant vapour. This approach computes the overall heat transfer coefficient of the evaporator \( U \) as the average value between the overall heat transfer coefficient of the boiling zone \( U_{boil} \) and that of the super-heating zone \( U_{sup} \) weighted on the base of the respective heat transfer area. In this way it is possible to directly compare the heat transfer performance of an evaporator working only in two-phase heat transfer with that of an evaporator working also in vapour super-heating. The water side heat transfer coefficient \( h_w \) is computed by the following non-dimensional equation:

\[
h_w = \frac{0.277}{\left( \frac{\mu_w}{d_h} \right) Re_w^{0.766} Pr_w^{0.333}}
\]

\( 5 < Pr_w < 10 \quad 200 < Re_w < 1200 \)

implemented by means of a modified Wilson plot technique as suggested by Muley and Manglik (1999). The detailed description of this procedure is reported by Longo and Gasparella (2007). The refrigerant vapour quality at the evaporator inlet and outlet \( X_{in} \) and \( X_{out} \) are computed starting from the refrigerant temperature \( T_{pb.in} \) and pressure \( p_{pb.in} \) at the inlet of the pre-evaporator (sub-cooled liquid condition) considering the heat flow rate exchanged in the pre-evaporator and in the evaporator \( Q_{pb} \) and \( Q \) and the pressure at the inlet and outlet \( p_{in} \) and \( p_{out} \) of the evaporator as follows:

\[
X_{in} = f(J_{in}, p_{in})
\]

\[
X_{out} = f(J_{out}, p_{out})
\]

\[
J_{in} = J_{pb.in}(T_{pb.in}, p_{pb.in}) + \frac{Q_{pb}}{m_r}
\]

\[
J_{out} = J_{in} + \frac{Q}{m_r}
\]

\[
Q_{pb} = m_{pb.w} c_{pw} |D T_{pb.w}|
\]

where \( J \) is the specific enthalpy of the refrigerant, \( m_r \) the refrigerant mass flow rate, \( m_{pb.w} \) the water mass flow rate and \( |D T_{pb.w}| \) the absolute value of the temperature variation on the water side of the pre-evaporator.

The frictional pressure drop on the refrigerant side \( \Delta p_f \) is computed by subtracting the manifolds and ports pressure drops \( \Delta p_c \), the momentum pressure drops \( \Delta p_a \) and the gravity pressure drops \( \Delta p_g \) from the total pressure drop measured \( \Delta p_t \):

\[
\Delta p_t = \Delta p_f - \Delta p_c - \Delta p_a - \Delta p_g
\]

The momentum and gravity pressure drops are estimated by the homogeneous model for two-phase flow as follows:

\[
\Delta p_a = G^2 (v_V - v_L) |\Delta \chi|
\]

\[
\Delta p_g = g \rho_n L
\]

where \( G \) is the refrigerant mass flux, \( v_L \) and \( v_V \) are the specific volume of liquid and vapour phase, \( |\Delta \chi| \) is the absolute value of the vapour quality change between inlet and outlet and

\[
\rho_n = \frac{X_m / \rho_v + (1 - X_m) / \rho_l}{1 - X_m + X_m}
\]

is the average two-phase density between inlet and outlet calculated by the homogeneous model at the average vapour quality \( X_m \) between inlet and outlet. The manifold and port pressure drops are empirically estimated, in accordance with Shah and Focke (1988), as follows

\[
\Delta p_c = 1.5 \frac{G^2}{(2 \rho_n)}
\]

The refrigerant properties are evaluated by the NIST Standard Reference Database REFPROP 9.1 (Lemmon et al., 2013).
Table 3: Operating conditions during experimental tests

<table>
<thead>
<tr>
<th>Runs</th>
<th>$T_{\text{sat}}$(°C)</th>
<th>$p_{\text{sat}}$(MPa)</th>
<th>$X_{\text{in}}$</th>
<th>$X_{\text{out}}$</th>
<th>$\Delta T_{\text{sup}}$(°C)</th>
<th>$G_r$(kg m$^{-2}$s$^{-1}$)</th>
<th>$G_w$(kg m$^{-2}$s$^{-1}$)</th>
<th>$q$(kWm$^{-2}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>138</td>
<td>9.9–20.2</td>
<td>0.30–0.43</td>
<td>0.19–0.30</td>
<td>0.79–1.00</td>
<td>4.6–10.3</td>
<td>11.1–31.4</td>
<td>49.0–141.9</td>
<td>3.7–16.7</td>
</tr>
</tbody>
</table>

4. ANALYSIS OF THE RESULTS

The experimental data consists of 138 vapourisation runs carried out at three different vaporisation temperatures (10, 15, and 20 °C) and four different evaporator outlet conditions (saturated mixtures with vapour quality of 0.80 and 1.00, super-heated vapour with super-heating of 5 and 10 °C), Table 3 shows the main operating conditions during the experimental tests: refrigerant saturation temperature $T_{\text{sat}}$ and pressure $p_{\text{sat}}$, inlet and outlet refrigerant vapour quality $X_{\text{in}}$ and $X_{\text{out}}$, outlet refrigerant super-heating $\Delta T_{\text{sup}}$, refrigerant mass flux $G_r$, water mass flux $G_w$, and heat flux $q$. The operating conditions investigated are typical for evaporators of vapour compression chillers and heat pumps in air conditioning application (Palm and Claesson, 2006). A detailed error analysis performed in accordance with Kline and McClintock (1953) indicates an overall uncertainty within ±12.0% for the refrigerant heat transfer coefficient measurement and within ±6.6% for the total pressure drop measurement.

Figures 3, 4, and 5 show the boiling heat transfer coefficients against heat flux at three different evaporation temperatures (10, 15 and 20 °C) and four different evaporator outlet conditions (vapour quality around 0.80 and 1.00, vapour super-heating around 5 and 10 °C). The heat transfer coefficients show great sensitivity to heat flux and outlet conditions and weak sensitivity to saturation temperature (pressure). The boiling heat transfer coefficients with 0.80 outlet vapour quality are 6-11% higher than those with 1.00 outlet vapour quality, 13-16% higher than those with 5 °C of outlet vapour super-heating, and 39-46% higher than those with 10 °C of outlet vapour super-heating. The inception of the dry-out might justify the slight decrease of the boiling heat transfer coefficient when outlet vapour quality increases from 0.80 to 1.00, whereas the increase of the outlet vapour super-heating involves a considerable degradation of the boiling heat transfer coefficients.

The experimental two-phase flow boiling heat transfer coefficients were compared against traditional equations for nucleate boiling, such as Cooper (1984) and Gorenflo (1993), and also against a recent model specifically developed for refrigerant vaporisation inside BPHEs by Longo et al. (2015).

Figure 3: Boiling heat transfer coefficient on refrigerant side vs. heat flux at 10°C
The absolute mean percentage deviation between calculated and experimental data is 13.4%, 13.1%, and 7.1% for Cooper (1984) equation, Gorenflo (1993) equation, and Longo et al. (2015) model, respectively. Figure 6 shows the comparison between the experimental two-phase flow boiling heat transfer coefficients and the calculated values by Longo et al. (2015).

**Figure 4**: Boiling heat transfer coefficient on refrigerant side vs. heat flux at 15°C

**Figure 5**: Boiling heat transfer coefficient on refrigerant side vs. heat flux at 20°C
Figure 6: Comparison between experimental and calculated heat transfer coefficients by Longo et al. (2015) model

Figure 7 shows the two-phase flow boiling frictional pressure drop against the kinetic energy per unit volume of the refrigerant flow computed by the homogeneous model:

\[ KE/V = G^2 / (2 \rho_m) \]  \hspace{1cm} (23)

It is possible to observe a fairly linear dependence of the frictional pressure drop on the kinetic energy per unit volume of the refrigerant flow as already found by Jassim et al. (2006) in adiabatic two-phase flow of HFC134a through a BPHE with herringbone and bumpy corrugation. The following best fitting linear correlation was derived from present experimental data:

\[ \Delta p_f [kPa] = 1.667 KE/V [J m^{-3}] \]  \hspace{1cm} (24)

This linear correlation reproduces present set of experimental data points with a mean absolute percentage deviation around 7.2%.

HFO1234ze(E) and HFO1234yf are probably the most promising replacements for HFC134a, therefore it is interesting to compare their thermal and hydraulic performance to those of HFC134a. Present HFO1234ze(E) data points were compared with those of HFO1234yf and HFC134a previously measured by the present authors (Longo, 2012; Longo and Gasparella, 2007) inside the same BPHE under the same operating conditions. HFO1234ze(E) exhibits heat transfer coefficients very similar to HFC134a and HFO1234yf and frictional pressure drops slightly higher than HFC134a and HFO1234yf. This can be attributed mainly to the lower absolute pressure and the higher vapour specific volume of HFO1234ze(E) with respect to both HFC134a and HFO1234yf.
6. CONCLUSIONS

This paper investigates the effects of heat flux, saturation temperature, and outlet conditions on HFO1234ze(E) boiling inside a BPHE. The heat transfer coefficients show great sensitivity to heat flux and outlet conditions and weak sensitivity to saturation temperature (pressure). The two-phase flow boiling heat transfer coefficients are 39-46% higher than those with 10 °C of outlet super-heating. The frictional pressure drop shows a linear dependence on the kinetic energy per unit volume of the refrigerant flow and therefore a quadratic dependence on refrigerant mass flux. The two-phase flow boiling heat transfer coefficients are in fair agreement with a recent model for refrigerant vaporisation inside BPHEs (Longo et al., 2015): the absolute mean percentage deviation between calculated and experimental data is 7.1%.

HFO1234ze(E) exhibits boiling heat transfer coefficients very similar to HFC134a and HFO1234yf and frictional pressure drop slightly higher than HFC134a and HFO1234yf. Based on the discussed results, it can be concluded that HFO1234ze(E) has potential to be a substitute of HFC134a, and a suitable alternative to HFO1234yf also in applications adopting BPHE as evaporator.

NOMENCLATURE

\[ A \] \quad \text{nominal area of a plate} \quad (m^2)

\[ b \] \quad \text{height of the corrugation} \quad (m)

\[ c_p \] \quad \text{specific heat capacity} \quad (J \text{ kg}^{-1} \text{K}^{-1})

\[ d_h \] \quad \text{hydraulic diameter, } d_h = 2b \quad (m)

\[ \text{f.s.} \] \quad \text{full scale}

\[ g \] \quad \text{gravity acceleration} \quad (m \text{s}^{-2})

\[ G \] \quad \text{mass flux, } G = m / (n_s W b) \quad (\text{kg m}^{-2} \text{s}^{-1})

\[ h \] \quad \text{heat transfer coefficient} \quad (W \text{ m}^{-2} \text{K}^{-1})

\[ J \] \quad \text{specific enthalpy} \quad (J \text{ kg}^{-1})
$k$ coverage factor

$KE/V$ kinetic energy per unit volume (J m$^{-3}$)

$L$ flow length of the plate (m)

$m$ mass flow rate (kg s$^{-1}$)

$n_{ch}$ number of channels

$N$ **number of effective plates**

$p$ pressure (Pa)

$P$ **corrugation pitch** (m)

$Pr$ Prandtl number, $Pr = \mu c_p / \lambda$

$q$ heat flux, $q = Q / S$ (W m$^{-2}$)

$Q$ heat flow rate (W)

$R_a$ arithmetic mean roughness (ISO 4271/1) (μm)

$Re$ Reynolds number, $Re = G d_h / \mu$

$R_p$ roughness (DIN 4762/1) (μm)

$S$ nominal heat transfer area, m$^2$

$s$ plate wall thickness (m)

$T$ temperature, K

$U$ overall heat transfer coefficient (W m$^{-2}$K$^{-1}$)

$v$ specific volume (m$^3$kg$^{-1}$)

$W$ width of the plate (m)

$X$ vapour quality, $X = (J - J_L) / \Delta J_{LV}$

**Greek symbols**

$\beta$ inclination angle of the corrugation (°)

$\lambda$ thermal conductivity (W m$^{-1}$K$^{-1}$)

$\mu$ viscosity (kg m$^{-1}$s$^{-1}$)

$\rho$ density (kg m$^{-3}$)

$\Delta$ difference

$\Delta J_{LV}$ latent heat of vaporisation (J kg$^{-1}$)

$\Phi$ enlargement factor

**Subscript**

$a$ momentum

$c$ manifolds and ports

$f$ frictional

$g$ gravity

$in$ inlet

$L$ liquid phase

$LV$ liquid vapour phase change

$ln$ logarithmic

$out$ outlet

$p$ plate

$pb$ pre-evaporator

$r$ refrigerant

$sat$ saturation

$sup$ super-heating

$t$ total

$V$ vapour phase

$w$ water

$wi$ water inlet

$wm$ water between the super-heating and the boiling zone

$wo$ water outlet
REFERENCES


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