ZEOTROPIC MIXTURES STUDY IN PLATE HEAT EXCHANGERS AND ORC SYSTEMS

Quentin Blondel[1], Nicolas Tauveron[1]*, Guillaume Lhermet[1] and Nadia Caney[1]

[1]Univ. Grenoble Alpes, F-38000 Grenoble, France, CEA, LITEN, LCST, F-38054, Grenoble, France

*Corresponding Author: nicolas.tauveron@cea.fr

ABSTRACT

To our best knowledge, there are no correlations in the scientific literature to predict the two-phase heat transfer coefficients of Hydrofluoroethers pure fluids and zeotropic mixtures used in plate heat exchangers. This work has allowed the development of evaporation and condensation correlations for such fluids. Correlations based on dimensionless numbers and working fluids critical properties gave promising results, with a prediction of the two-phase heat transfer coefficient around 15% for heat source and 50% for cooling source which lead respectively to a prediction of 4% and 6% for the global thermal power exchanged. The coupled operation between the heat exchangers and the organic Rankine cycle was also studied. For this purpose, the influence of the heat source variation on the performances of the hot exchanger and the overall cycle was analyzed. Results obtained showed a positive improvement simultaneously of the hot exchanger performances and the ORC performances. Regarding the use of zeotropic mixtures as working fluids in ORC system, it is often proposed in the literature as being able to significantly increase the performance of this thermodynamic cycle due to better heat transfer at the hot source. Nevertheless, the great majority of these studies are numerical and do not take into account the interactions and system effects existing within the thermodynamic cycle when using a mixture instead of a pure fluid. This work has allowed to highlight this complexity and to provide a better understanding of the use of zeotropic mixtures.

1 INTRODUCTION

The energy sector is facing major challenges in the upcoming century as energy demand is rising and its major impact on the global warming issue needs to be addressed. Among the solutions to overcome these challenges, renewable energies and process energy efficiency could be partially fulfilled by the use of the Organic Rankine Cycle (ORC) technology (Quoilin *et al.*, 2013). Replacement working fluids, such as Hydrofluoroethers (HFE), Hydrofluoroolefins (HFO) or Hydroclorofluoroolefins (HCFO), are one of the main research topics in ORC field due to current environmental concerns. Linked to these issues, zeotropic mixtures of replacement fluids are expected to be an interesting solution to improve ORC performances and reduce working fluids noxiousness. However, heat transfer correlations of these fluids are scarce; thereby, proper predictive models for heat exchanger and global ORC system are not always accurate. Moreover, the number of experimental works available in the literature that would allow to correct and improve these correlations is very limited (Bamorovat Abadi and Kim, 2017).

This study attempts to provide a better understanding of these mechanisms. For this purpose, semi-empirical models of the hot and cold heat exchangers of an ORC were developed and validated with experimental data. The choice of HFE fluids type (pure fluids and mixtures) as working fluids has been first made due to the possibility to have mixture data in the available thermodynamic software. Indeed, it was very important to have reliable thermodynamic properties for the mixture to study properly their influence on components and global ORC system. There are various publications in the scientific literature where general coefficients are used to determine mixture properties; it is especially true for "new fluids". However, as we wanted to use not debatable data, the choice of mixture was very limited at the time of the beginning of this work. Secondly, this choice was linked to the environmental concern (GWP, ODP, degradation at comparable levels as HFO or HFCO fluids). The present fluids

used are Novec649 and HFE7000 as well as zeotropic mixtures composed of these two pure fluids. In order to improve the precision of the models, a third pure fluid, HFE7100, was also studied. At the same time, data from experimental tests conducted in pure fluids and zeotropic mixtures of HFE fluid type are studied, this in order to understand the influence of zeotropic mixtures on the behavior of the thermodynamic cycle. The analysis of the results obtained intends to improve comprehension of the influence of this type of working fluid on the global cycle performances but also on the heat exchanger, an inner component of the ORC. Indeed, HFE zeotropic mixtures were never studied numerically or experimentally to our best knowledge (Landelle *et al.*, 2017).

2   MATERIAL AND METHODS

The Process Flow Diagram (PFD) and a picture of the installation are shown in Figure 1. Within it, the working fluid, in its liquid state, flows to the high pressure level thanks to the volumetric pump. The fluid is then heated, evaporated and superheated in contact with the heat source through the evaporator. It is expanded in the turbine, producing mechanical work and then converted into electricity. At low pressure, the fluid is cooled, condensed and sub-cooled by the cooling source within the condenser and then pumped again to close the cycle. A small centrifugal pump is placed downstream the volumetric pump to slightly increase its inlet pressure and avoid any risk of cavitation.
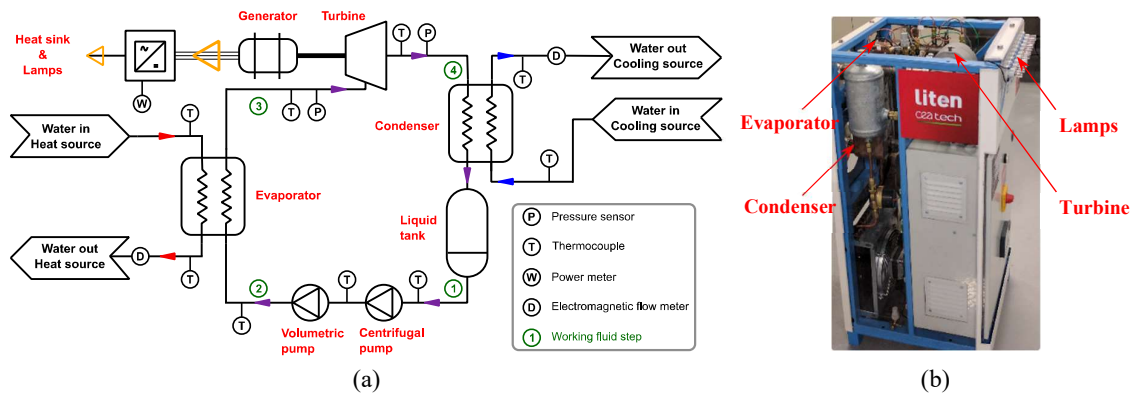

Figure 1: Experimental ORC (a) Process Flow Diagram (PFD), (b) Picture

Six different organic fluids have been studied as ORC working fluids. There are three pure fluids with the Novec649, HFE7000 and HFE7100, and three zeotropic mixtures wich are blends of the Novec649 and HFE7000. These fluids are used as working fluids because of their relatively environmental friendly properties and theirs good performances as working fluids regarding the available literature. The last aspect of choice of HFE fluid type (or closed to it with the Novec649) is the possibility to create zeotropic mixtures when these pure fluids are mixed together, mixtures which are one of the points of interest of this study. Table 1 below summarizes the main properties of the working fluids.

Table 1: Main properties of the six working fluids studied

Properties		Novec649	HFE7000	HFE7100	80%$_{Novec649}$ - 20%$_{HFE7000}$	50%$_{Novec649}$ - 50%$_{HFE7000}$	25%$_{Novec649}$ - 75%$_{HFE7000}$
Fluid type		Dry	Dry	Dry	Dry	Dry	Dry
Fluid class		Fluoroketone	Hydrofluoroether	Hydrofluoroether	-	-	-
Critical temperature	(℃)	168,7	164,6	195,3	165,5	162	161,9
Critical pressure	(bar)	18,7	24,8	22,3	20,3	22	23,3
Normal glide	(K)	0	0	0	2,9	2	0,3
ODP		0	0	0	0	0	0
GWP		1	530	320	107	266	398
Inflammability		No	No	No	No	No	No

An important test campaign (229 tests) was conducted and allowed to compare correctly the performances of the system with the different working fluids, and to study the influence of zeotropic

mixtures on this thermodynamic cycle. The experimental data obtained allowed the development of heat transfer correlations for HFE fluids in pure fluids and zeotropic mixtures. The development of these correlations is detailed in section 3. The analysis of the system performances following the use of zeotropic mixtures as ORC working fluids is presented in section 4. All of these results and analyses are based on experimental data and in order to provide the most accurate measurements possible, many of the sensors have been calibrated. Typical uncertainties for the main sensors are 0.3% for electrical power output measurements, 0.3% for flow measurements, 0.2°C for temperature sensors and 0.5% for pressure sensors. Measurement uncertainties influence on the experimental data and correlations developed in this work have been taken into account and calculated according to the theory of uncertainties propagation using the root-sum-square method (Taylor and Kuyatt, 1994). The uncertainty U_Y of the variable Y was calculated as a function of the uncertainties U_{Xi} for each measured variable Xi, as defined in equation (1).

$$U_Y = \sqrt{\sum_i \left(\frac{\partial Y}{\partial X_I}\right)^2 \cdot U_{X_i}^2} \tag{1}$$

The heat transfer correlations developed in this study have been defined for brazed plate heat exchangers whose characteristics are described in Table 2.

Table 2: ORC heat exchangers characteristics

Exchanger	Characteristics	
Evaporator (two exchangers in series)	Type	: Brazed plate heat exchanger
	Manufacturer	: SWEP
	Reference	: B10TH + B16H
	Number of plates	: 20 + 30
	Number of passes	: 1
	Heat transfer area	: 0.516 m^2 + 1.048 m^2
Condenser	Type	: Brazed plate heat exchanger
	Manufacturer	: SWEP
	Reference	: B12H
	Number of plates	: 40
	Number of passes	: 1
	Heat transfer area	: 1.112 m^2

3   NEW ZEOTROPIC MIXTURES CORRELATIONS FOR HEAT EXCHANGERS

The study of heat exchangers with HFE pure fluids and mixtures, deals here with the analysis of the phase change steps, in condensation and evaporation, and to the development of two-phase correlations for these fluids within plate heat exchangers. This choice is linked to the fact that the energy necessary for the transition of the working fluid from one single-phase state to another represents an important part (in evaporation), or even the main part (in condensation), of the heat exchanges carried out within the exchangers of the ORC installation studied. Moreover, the work of Watson (1943) has highlighted the very strong link between the two-phase characteristics of the working fluids and their critical properties. Finally, the interest of the two-phase heat transfer is also based on the specificity of the zeotropic mixtures, because of the temperature glide. The existing number of correlations for two-phase heat transfer is substantial (Eldeeb *et al.*, 2016), both for evaporation (Ayub *et al.*, 2019), or condensation (Tao and Infante Ferreira, 2019) of working fluids in a plate exchanger. However, HFE organic fluids, or fluids closed to it, have been scarcely studied within the literature. In fact, to our best knowledge, there is no appropriate correlation for this type of fluid in the case of plate heat exchangers, despite the large number of existing correlations. So, different correlations initially developed for phase change in tubes, in evaporation and condensation, are studied in order to define the most relevant ones to use in comparison with the experimental results. Indeed, tubes heat transfer correlations are considered relevant for plate heat exchangers due to the fluids flow and the heat transfer phenomena similarity in tubes and plate heat exchangers channels (Huang, 2010; Tao et Infante Ferreira, 2019). Thus, four correlations have been studied in evaporation and six correlations in condensation. These different correlations are detailed in the Table 3 below, a specific correlation for zeotropic mixtures is

also considered in this work. This correlation, associated with a two-phase correlation of evaporation or condensation (when it is possible), allows taking into account more precisely the differences of volatilities specific to the constituent species of a zeotropic mixture.

Evaporation and condensation literature correlations are first analyzed with the original fluid, the Novec649, by their two-phase heat transfer coefficients $\alpha_{tp,correl}$ which are compared to the heat transfer coefficients $\alpha_{tp,expe}$ obtained experimentally. The calculation model was developed for each of the literature correlations, under EES software (F-Chart Software, 2018) and coupled with the REFPROP thermodynamic fluid database (NIST, 2013). Input data such as thermofluidic conditions (pressures, flow rates, temperatures), geometry of heat exchanger (number of plates, plate area, number of channels) were included in the model. When comparing the two-phase heat transfer coefficients from literature correlations with experimental coefficients, the « $Gullapalli_{evap}$ » correlation best predicted the two-phase evaporation coefficients, while in the case of the two-phase condensation heat transfer coefficients the « $Shon$ » correlation was found to be the most appropriate, as presented with Figures 2 and 3.
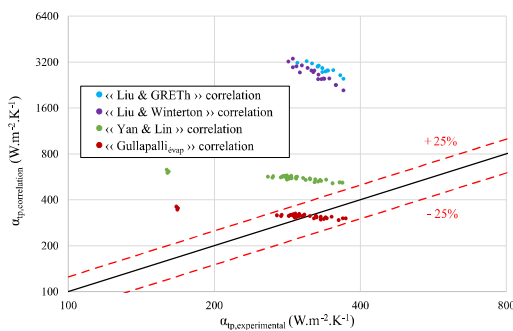

Figure 2: Experimental and literature correlated two-phase heat transfer coefficients comparison (evaporation of Novec649)
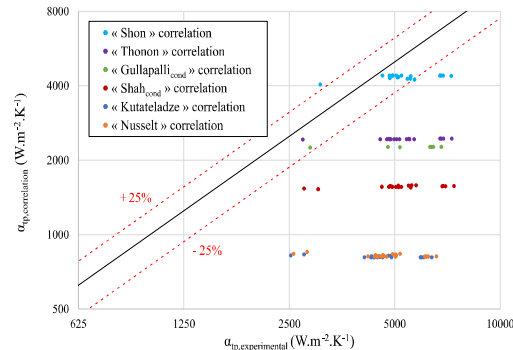
Figure 3: Experimental and literature correlated two-phase heat transfer coefficients comparison (condensation of Novec649)

Thus, these two correlations are the ones considered the most relevant to predict the two-phase heat transfer coefficients of the present HFE working fluids: the sizing fluid Novec649, the HFE7000 and the zeotropic mixtures of these two pure fluids and also the HFE7100. Indeed, one of the goals of this work is to determine global and unified two-phase correlations for all six working fluids studied. The « $Silver, Bell \& Ghaly$ » correlation is also used with the « $Shon$ » correlation in the case of condensation. In the case of an evaporation correlation, the pairing with the « $Silver, Bell \& Ghaly$ » correlation is only possible if the evaporation correlation makes the distinction between the parts of nucleate boiling and convective boiling. However, the « $Gullapalli_{evap}$ » correlation does not make this distinction, therefore the « $Silver, Bell \& Ghaly$ » correlation cannot be associated with it. Although the « $Gullapalli_{evap}$ » and « $Shon, Silver, Bell \& Ghaly$ » correlations are the optimal correlations among those in the literature, a significant dispersion of the predicted two-phase coefficients is still found to be present. Because of this, the coefficients from the literature are supplemented with multiplying factors Ω_{correl} as presented with equations (2) and (3).

$$\alpha_{e,correl} = \alpha_{tp,Gullapalli_{evap}} \cdot \Omega_{e,correl} \quad (2) \qquad \alpha_{c,correl} = \alpha_{tp,Shon,Silver,Bell,Ghaly} \cdot \Omega_{c,correl} \quad (3)$$

This semi-empirical corrective function principle, applied to a two-phase heat transfer coefficient correlation, is notably used among heat exchanger manufacturers to account for the effects of exchanger geometry, fluid type, regime and channel flow distribution (Gullapalli, 2013). In this way, it is possible to adapt a specific pure fluid correlation to mixtures with comparable characteristics by properly adjusting the multiplying function. It is therefore necessary to determine on which parameters this corrective factor depends in order to correct the literature correlation and to predict the experimental two-phase heat transfer coefficient for all the working fluids. It could be observed that the corrective factor was strongly correlated to the boiling number Bo, as presented in the case of evaporation with Figure 4.

Table 3: Two-phases correlations (a) evaporation, (b) condensation, (c) mixture

Authors	Evaporation correlations	Naming in the present work
Liu et Winterton (1991)	$\alpha_{tp} = [(S_{Liu} \cdot \alpha_{nb})^2 + (F_{Liu} \cdot \alpha_{cb})^2]^{1/2} \qquad \alpha_{cb} = 0{,}023 \cdot Re_{lo}^{0,8} \cdot Pr_l^{0,4} \cdot \frac{\lambda_l}{D_h}$ $\alpha_{nb} = 55 \cdot \left(\frac{P}{P_{crit}}\right)^{0,12} \cdot \dot{q}''^{\,0,67} \cdot \left[-log_{10}\left(\frac{P}{P_{crit}}\right)\right]^{-0,55} \cdot M^{-0,5}$ $F_{Liu} = \left[1 + x \cdot Pr_l \cdot \left(\frac{\rho_l}{\rho_v} - 1\right)\right]^{0,35} \quad S_{Liu} = \left(1 + 0{,}055 \cdot F_{Liu}^{\,0,1} \cdot Re_{lo}^{0,16}\right)^{-1}$	*« Liu & Winterton »*
Liu et Winterton (1991) & GRETh (1999)	$\alpha_{tp} = [(S_{Liu} \cdot \alpha_{nb})^2 + (F_{Liu} \cdot \alpha_{cb})^2]^{1/2} \quad \alpha_{cb} = 0{,}347 \cdot Re_l^{0,653} \cdot Pr_l^{1/3} \cdot \frac{\lambda_l}{D_h}$ $\alpha_{nb} = 55 \cdot \left(\frac{P}{P_{crit}}\right)^{0,12} \cdot \dot{q}''^{\,0,67} \cdot \left[-log_{10}\left(\frac{P}{P_{crit}}\right)\right]^{-0,55} \cdot M^{-0,5}$ $F_{Liu} = \left[1 + x \cdot Pr_l \cdot \left(\frac{\rho_l}{\rho_v} - 1\right)\right]^{0,35} \quad S_{Liu} = \left(1 + 0{,}055 \cdot F_{Liu}^{\,0,1} \cdot Re_{lo}^{0,16}\right)^{-1}$	*« Liu & GRETh »*
Yan et Lin (1999)	$\alpha_{tp} = 1{,}926 \cdot Bo_{eq}^{\,0,3} \cdot \left[(1-x) + x \cdot \left(\frac{\rho_l}{\rho_v}\right)^{0,5}\right] \cdot Re_{lo}^{0,5} \cdot Pr_l^{1/3} \cdot \frac{\lambda_l}{D_h}$	*« Yan & Lin »*
Gullapalli (2013)	$\alpha_{tp} = 1{,}926 \cdot Bo_{eq}^{\,0,3} \cdot \left[(1-x) + x \cdot \left(\frac{\rho_l}{\rho_v}\right)^{0,5}\right] \cdot Re_l^{0,5} \cdot Pr_l^{1/3} \cdot \frac{\lambda_l}{D_h}$	*« Gullapalli_{evap} »*

(a)

Authors	Condensation correlations	Naming in the present work
Nusselt (1916)	$\alpha_{tp} = 1{,}1 \cdot \lambda_l \cdot \left[\frac{\mu_l^2}{\rho_l \cdot (\rho_l - \rho_g) \cdot g}\right]^{-1/3} \cdot Re_l^{-1/3}$	*« Nusselt »*
Kutateladze (1963)	$\alpha_{tp} = 0{,}756 \cdot \lambda_l \cdot \left[\frac{\mu_l^2}{\rho_l \cdot (\rho_l - \rho_g) \cdot g}\right]^{-1/3} \cdot Re_l^{-0,22}$	*« Kutateladze »*
Shah (1979)	$\alpha_{tp} = \psi_{Shah} \cdot \alpha_{lo} \qquad \alpha_{lo} = 0{,}347 \cdot Re_{lo}^{0,653} \cdot Pr_l^{1/3} \cdot \frac{\lambda_l}{D_h}$ $\psi_{Shah} = \frac{1}{(x_2 - x_1)} \cdot \left[-\frac{(1-x)^{1,8}}{1{,}8} + \frac{3{,}8}{\left(\frac{P}{P_{crit}}\right)^{0,38}} \cdot \left(\frac{x^{1,76}}{1{,}76} - \frac{0{,}04 \cdot x^{2,76}}{2{,}76}\right)\right]_{x_1}^{x_2}$	*« Shah_{cond} »*
Thonon (2002)	$\alpha_{tp} = 1564 \cdot Re_{eq}^{-0,76} \cdot \alpha_{lo} \qquad \alpha_{lo} = 0{,}347 \cdot Re_{lo}^{0,653} \cdot Pr_l^{1/3} \cdot \frac{\lambda_l}{D_h}$	*« Thonon »*
Gullapalli (2013)	$\alpha_{tp} = \left(\alpha_{gr}^2 + \alpha_{sh}^2\right)^{1/2} \qquad \alpha_{gr} = 1{,}1 \cdot \lambda_l \cdot \left[\frac{\mu_l^2}{\rho_l \cdot (\rho_l - \rho_g) \cdot g}\right]^{-1/3} \cdot Re_l^{-1/3}$ $\alpha_{sh} = \left[1 + x \cdot \left(\frac{\rho_l}{\rho_g} - 1\right)\right]^{0,5} \cdot \alpha_{lo}$	*« Gullapalli_{cond} »*
Shon (2018)	$\alpha_{tp} = 2{,}337 \cdot Re_{eq}^{1,024} \cdot Re_{lo}^{-0,294} \cdot Bo_{eq}^{\,0,361} \cdot Pr_l^{1/3}$	*« Shon »*

(b)

Authors	Mixture correlation	Naming in the present work
Silver (1947) & Bell et Ghaly (1973)	$\frac{1}{\alpha_{tp,m}} = \frac{1}{\alpha_{tp}} + \frac{Y}{\alpha_v} \qquad Y = x \cdot cp_v \cdot \frac{\Delta T_g}{\Delta h_{lv}} \qquad \alpha_v = 0{,}347 \cdot Re_v^{0,653} \cdot Pr_v^{1/3} \cdot \frac{\lambda_v}{D_h}$	*« Silver, Bell & Ghaly »*

(c)

It is however noticed that a dispersion related to the fluid type employed still exists; this notably with a distribution related to the critical temperatures of the fluids. Thus, by integrating the critical properties of the six organic fluids studied, it was possible to define the corrective factor according to a single function, as described in the case of evaporation with Figure 5.
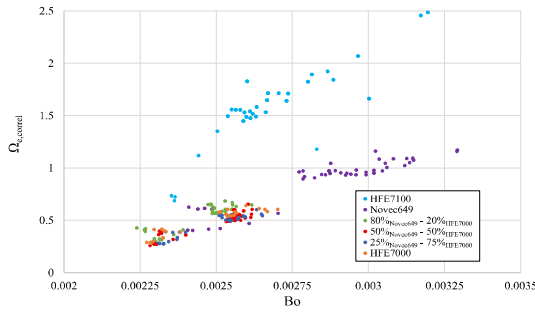

Figure 4: Variation of the corrective factor $\Omega_{e,correl}$ correlated to the Boiling number **Bo**
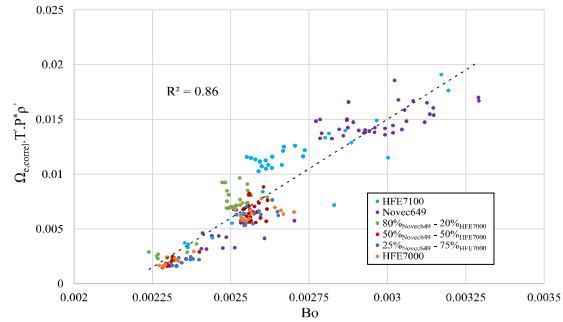
Figure 5: Variation of the corrective factor $\Omega_{e,correl}$ correlated to the dimensionless numbers related to the type of fluid and heat transfer

The correlated corrective factors are presented with equations (4) and (5), respectively for evaporation and condensation. Equations (6) to (9) detail the dimensionless numbers on which these corrective factors depend.

$$\Omega_{e,correl} = \frac{18{,}026 \cdot Bo - 0{,}0391}{T' \cdot P^* \cdot \rho'} \quad (4) \qquad \Omega_{c,correl} = \frac{36{,}842 \cdot Bo + 0{,}1088}{T'} \quad (5)$$

$$Bo = \frac{\dot{q}''}{\dot{G} \cdot \Delta h_{lv}} \quad (6) \qquad T' = \frac{T_{sat,mean}}{T_{crit}} \quad (7) \qquad P^* = \frac{P}{P_{crit}} \quad (8) \qquad \rho' = \frac{\rho_v}{\rho_{crit}} \quad (9)$$

Once these corrective factors are determined, the correlated two-phase heat transfer coefficients (equations (2) and (3)) are calculated and can be compared to the experimental two-phase coefficients, as presented with Figures 6 and 7 for the results of zeotropic mixtures. Table 4 summarizes the different confidence intervals of heat transfer coefficients related to experimental uncertainties for all working fluids in evaporation and condensation.
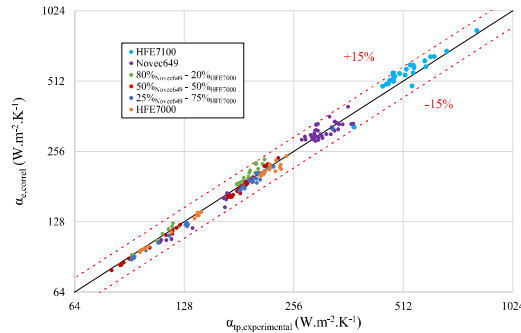

Figure 6: Experimental and correlated two-phase heat transfer coefficients comparison (evaporation)
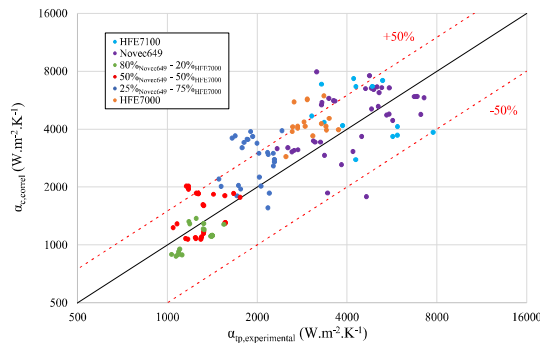

Figure 7: Experimental and correlated two-phase heat transfer coefficients comparison (condensation)

Table 4: Typical uncertainties of heat transfer coefficients in evaporation and condensation

Uncertainties		Novec649	HFE7000	HFE7100	80%$_{\text{Novec649}}$ - 20%$_{\text{HFE7000}}$	50%$_{\text{Novec649}}$ - 50%$_{\text{HFE7000}}$	25%$_{\text{Novec649}}$ - 75%$_{\text{HFE7000}}$
Evaporation	$U_{\alpha_{tp,experimental}}$	7.1%	6.5%	7.9%	6.3%	6.5%	6.5%
Condensation	$U_{\alpha_{tp,experimental}}$	20.8%	26.6%	19.6%	11.5%	16.7%	18.3%

Illustrated in Figure 6, the developed evaporation correlation allows to predict correctly the two-phase heat transfer coefficient, with a relative dispersion lower than 15% for all the results. In the case of condensation, the dispersion of results is greater (Figure 7). This phenomenon can be explained by higher value of the heat transfer coefficients for the working fluid in condensation than in evaporation. Thus, when the flow rate of the cold source decreases, it tends to be the limiting parameter of the heat transfer. This behavior is different from that observed in evaporation, because of the very low two-phase transfer coefficients of the working fluid compared to those of water. However, when comparing the experimental thermal power and the correlated thermal power exchanged with all fluids within the condenser, represented in Figure 8, small differences between the values obtained is found. The calculated thermal power accuracy is in the range of 6% at most. For its part, the evaporation correlation leads to a thermal power deviation of 4% at most (Figure 9).
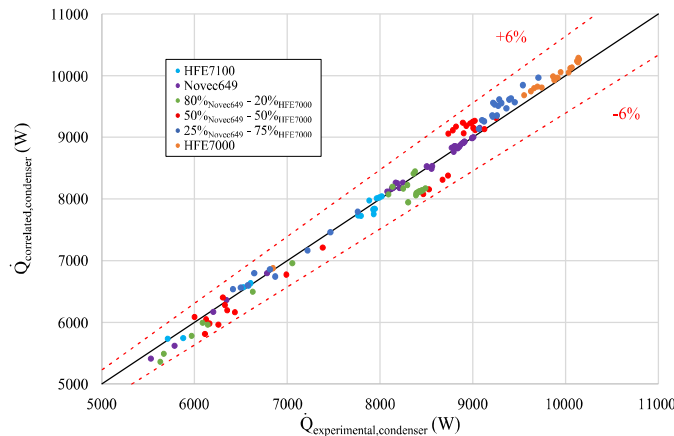

Figure 8: Comparison of the experimental and correlated global power exchanged in the condenser
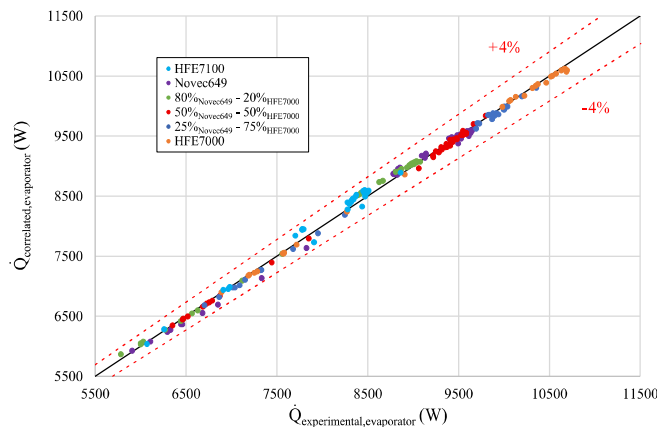

Figure 9: Comparison of the experimental and correlated global power exchanged in the evaporator

4   HEAT SOURCE AND ZEOTROPIC MIXTURE INFLUENCE ON ORC SYSTEM

The two-phase semi-empirical correlations of HFE type pure fluids and zeotropic mixtures developed in the previous section 3 are based on literature correlations adapted and validated for the

working fluids considered thanks to experimental data from the ORC test bench. Beyond these correlations, the work carried out on this experimental installation intends to better understanding the behavior and the performances of an ORC when using zeotropic mixtures as working fluid.

Thus, this section 4 focuses more specifically on the influence of the heat source on the ORC cycle and the possible links and optimizations between the overall ORC performance and the performance of the exchanger located at the heat source. The specific interest about the hot exchanger of the system is natural when studying a heat to power system such as an organic Rankine cycle. Moreover, this interest is increased due to the potential improvement of heat transfer performances brought by the temperature glide of the zeotropic mixtures. For this purpose, the performances of this component and of the global system are studied according to their exergy efficiencies, as defined by equations (10) and (11), and their values are compared in Figure 10.

$$\eta_{II,closed} = \frac{Usefull\ product}{Available\ resource} = \frac{\dot{W}_{el,tur}}{\dot{m}_h \cdot \left[h_{h,in,evap} - h_{h,out,evap} - T_{ref} \cdot \left(s_{h,in,evap} - s_{h,out,evap}\right)\right]} \tag{10}$$

$$\eta_{II,evap} = \frac{\dot{m}_f \cdot \left[h_{f,out,evap} - h_{f,in,evap} - T_{ref} \cdot \left(s_{f,out,evap} - s_{f,in,evap}\right)\right]}{\dot{m}_h \cdot \left[h_{h,in,evap} - h_{h,out,evap} - T_{ref} \cdot \left(s_{h,in,evap} - s_{h,out,evap}\right)\right]} \tag{11}$$

The results presented in Figure 10 refer to the tests carried out with the two pure fluids Novec649, HFE7000 and with the three zeotropic mixtures composed of these two fluids under similar operating conditions. During these tests, the cooling source was fixed at a flow rate of 900 l.h^{-1} while the heat source flow rate varied between 1500 l.h^{-1} and 3500 l.h^{-1}. This flow rate variation was performed in order to look at the heat source influence on the system. It can be observed that this influence is similar for all the working fluids. The optimization of the global ORC performances is directly linked to the optimization of the heat source and the performances of the corresponding exchanger. Indeed, the available resource is identical for the heat exchanger and the thermodynamic cycle. Therefore, when the heat source flow rate varies, the efficiencies of this exchanger and the ORC vary accordingly (equations (10) and (11)). On the other hand, each of the useful products remains the same. In the case of the heat exchanger, this follows from the conservation of the heat exchange balance, and in the overall case of the ORC, from the fact that the variation of the heat source does not reduce the thermodynamic cycle pressure ratio and thus the production of electrical energy. However, it can also be noted that the zeotropic mixtures do not allow to reach better performances for the ORC system and for the heat exchanger. This behavior could be explained by the characteristics related to the coupled operation of the ORC cycle and the working fluids, as shown in Figure 11, and by the properties of the zeotropic mixtures, as shown in Figure 12.
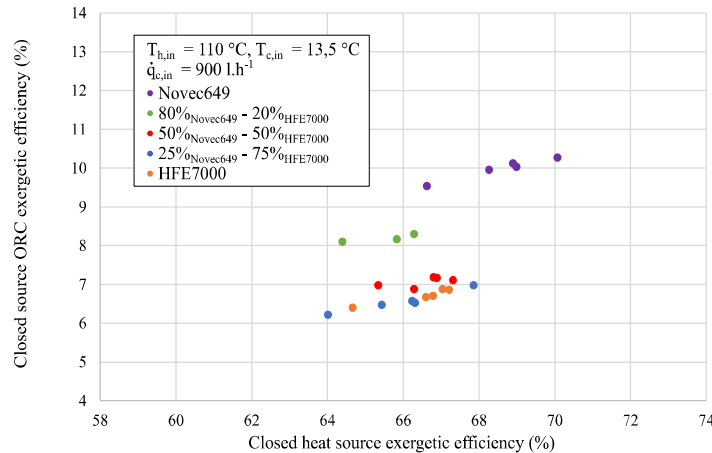

Figure 10: ORC and heat source performances comparison

For the same test carried out with a hot flow rate of 1500 l.h^{-1} and a cold flow rate of 900 l.h^{-1}, the values of the evaporation logarithmic mean temperature differences *ΔT$_{ln}$* for the different working fluids are presented in Figure 11.
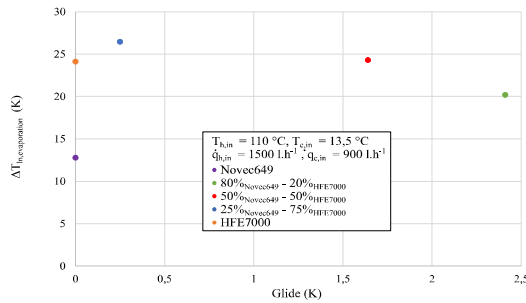
Figure 11: Evaporation logarithmic mean temperature differences comparison
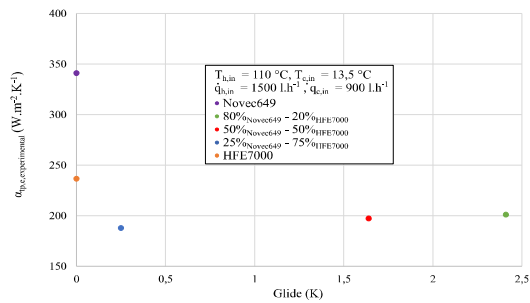
Figure 12: Evaporation two-phase heat transfer coefficient comparison

This characteristic data of the two-phase heat transfer is a relevant indicator to estimate the possible optimization brought by the use of zeotropic mixtures. Indeed, the zeotropic temperature glide is expected to lead to lower values of evaporation ΔT_{ln} in comparison with those of pure fluids, because of the possibility offered to the two fluids within the exchanger to better match their temperature variation. However, the present results show that the Novec649 is the working fluid with the lowest value of ΔT_{ln}. This behavior comes from the fact that for the tests performed with zeotropic mixtures composed of Novec649 and HFE7000, the thermodynamic cycle although modified due to the zeotropic glide does not decrease the ΔT_{ln} and therefore does not lead to optimize the two-phase heat transfer. Beyond the potential improvement of the organic Rankine cycle performances offered by the use of zeotropic mixtures due to their temperature glide, these heat transfer performances depend directly on the intrinsic properties of the working fluids used and more particularly on the zeotropic mixtures heat transfer coefficients studied. Figure 12 compares the values of the different working fluids two-phase heat transfer coefficients $\alpha_{tp,e}$ for the same tests defined in the previous Figure 11. Thus, it can be observed that the pure fluids Novec649 and HFE7000 have the highest evaporation two-phase coefficients in comparison with the zeotropic mixtures. This behavior could be explained by the difference in volatility of the zeotropic mixtures constituent species considered, which leads to a degradation of the two-phase heat transfer coefficient of this mixture (Silver, 1947; Bell and Ghaly, 1973), as presented in Figure 6 of the previous section 3.

5   CONCLUSIONS

The influence of pure fluids and zeotropic mixtures of HFE fluid type used as working fluids in an experimental ORC plant using brazed plate heat exchangers has been studied. Two heat transfer correlations have been adapted and lead to a correct prediction of HFE zeotropic mixture heat transfer performances. Indeed, although the evaporation heat transfer correlation is more accurate than the condensation one, both lead to a good accuracy of the calculated thermal powers, in the range of 6% in the condensation case whereas this accuracy is in the range of 4% in the evaporation case. It could be shown that the optimization of the ORC's heat source directly implies to an optimization of the global system performances and this whatever the working fluid, with pure fluids or in zeotropic mixtures. However, the tests carried out and the present results put forward that the zeotropic mixtures of HFE type do not make it possible to reach better performances in comparison with the pure fluid Novec649. At the same time, it should be noted that zeotropic mixtures studied here present low glide values (lower than 3 K). HFE zeotropic mixtures investigations with higher glide values are necessary to expand this study. Other tests are planned to confirm (or not) this analysis and to extend it to different levels of heat source temperature.

NOMENCLATURE

Symbols				*Subscripts*	
α	Heat transfer coefficient	$(W.m^{-2}.K^{-1})$		c	Condensation
Bo	Boiling number	$(-)$		closed	Closed source
Δh_{lv}	Latent heat	$(J.kg^{-1})$		correl	Correlation
ΔT_{ln}	Logarithmic mean temperature difference	(K)		crit	Critical
η	Efficiency	$(\%)$		e, evap	Evaporation
$\dot{G}$	Mass flux	$(kg.s^{-1}.m^{-2})$		el	Electrical

h	Enthalpy	$(J.kg^{-1})$	f	Fluid	
$\dot{m}$	Mass flow rate	$(kg.s^{-1})$	h	Hot	
Ω	Corrective factor	(–)	II	Second law - Exergy	
P	Pressure	(bar)	in	Input	
P^*	Saturation to critical pressures ratio	(–)	mean	Mean	
$\dot{Q}$	Thermal power	(W)	out	Output	
$\dot{q}"$	Heat flux	$(W.m^{-2})$	ref	Reference	
ρ'	Dew point to critical densities ratio	(–)	sat	Saturation	
ρ_v	Dew point density	$(kg.m^{-3})$	tp	Two-phase	
S	Entropy	$(J.kg^{-1}.K^{-1})$	tur	Turbine	
T	Temperature	(°C), (K)			
T'	Saturation to critical temperatures ratio	(–)			
$\dot{W}$	Turbine electrical power	(W)			

REFERENCES

Ayub ZH, Khan TS, Salam S, Nawaz K, Ayub AH, Khan MS. Literature survey and a universal evaporation correlation for plate type heat exchangers. International Journal of Refrigeration 2019;99:408–18.

Bamorovat Abadi G, Kim KC. Investigation of organic Rankine cycles with zeotropic mixtures as a working fluid: Advantages and issues. Renewable and Sustainable Energy Reviews 2017;73:1000–13.

Bell KJ, Ghaly MA. An approximate generalized design method for multicomponent/partial condenser. AIChE Symp Ser 1973;69:72–9.

Eldeeb R, Aute V, Radermacher R. A survey of correlations for heat transfer and pressure drop for evaporation and condensation in plate heat exchangers. International Journal of Refrigeration 2016;65:12–26.

F-Chart Software. EES: Engineering Equation Solver, F-Chart Software: Engineering Software. Professional V10.417-3D, 2018.

GRETh, « TM11 : Pertes de pression et transfert de chaleur dans les échangeurs à plaques en simple phase », Manuel Technique du GRETh, 1999.

Gullapalli VS. Estimation of Thermal and Hydraulic Characteristics of Compact Brazed Plate Heat Exchangers. Phd Thesis. Lund University, 2013.

Huang J. Performance analysis of plate heat exchangers used as refrigerant evaporators. Thesis. 2010.

Kutateladze SS. Fundamentals of heat transfer. New York: Academic Press; 1963.

Landelle A., Tauveron N., Haberschill P., Revellin R., Colasson S., Performance Evaluation and Comparison of Experimental Organic Rankine Cycle Prototypes from Published Data, Energy Procedia, 2017:105:1706-1711.

Liu Z, Winterton RHS. A general correlation for saturated and subcooled flow boiling in tubes and annuli, based on a nucleate pool boiling equation. International Journal of Heat and Mass Transfer 1991;34:2759–66.

NIST. Reference Fluid Thermodynamic and Transport Properties – REFPROP. Version 9.1, 2013.

Nusselt W. The condensation of steam on cooled surfaces. Z Ver Dtsch Ing 1916;60:541–6.

Quoilin S, Broek MVD, Declaye S, Dewallef P, Lemort V. Techno-economic survey of Organic Rankine Cycle (ORC) systems. Renewable and Sustainable Energy Reviews 2013;22:168–86.

Shah MM. A general correlation for heat transfer during film condensation inside pipes. Int. J. Heat Mass Transf. 1979;22:547–56.

Shon BH, Jung CW, Kwon OJ, Choi CK, Tae Kang Y. Characteristics on condensation heat transfer and pressure drop for a low GWP refrigerant in brazed plate heat exchanger. International Journal of Heat and Mass Transfer 2018;122:1272–82.

Silver L. Gas cooling with aqueous condensation. Trans Inst Chem Eng 1947;25:30–42.

Tao X, Infante Ferreira CA. Heat transfer and frictional pressure drop during condensation in plate heat exchangers: Assessment of correlations and a new method. International Journal of Heat and Mass Transfer 2019;135:996–1012.

Taylor BN, Kuyatt CE. NIST Guidelines for Evaluating and Expressing the Uncertainty of NIST Measurement Results Cover. NIST 2015. 1994 https://www.nist.gov/pml/nist-technical-note-1297/nist-guidelines-evaluating-and-expressing-uncertainty-nist-measurement (accessed June 24, 2021).

Thonon B, Bontemps A. Condensation of Pure and Mixture of Hydrocarbons in a Compact Heat Exchanger: Experiments and Modelling. Heat Transfer Engineering 2002;23:3–17.

Watson KM. Thermodynamics of the Liquid State. Ind Eng Chem 1943;35:398–406.

Yan Y-Y, Lin T-F. Evaporation Heat Transfer and Pressure Drop of Refrigerant R-134a in a Plate Heat Exchanger. J Heat Transfer 1999;121:118–27.

ACKNOWLEDGEMENT