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Published in:

Proceedings of the 13th IIR-Gustav Lorentzen Conference on Natural Refrigerants

Link to article, DOI:

[10.18462/iir.gl.2018.1261](https://doi.org/10.18462/iir.gl.2018.1261)

Publication date:

2018

Document Version

Peer reviewed version

[Link back to DTU Orbit](#)

Citation (APA):

Mancini, R., Jensen, J. K., Markussen, W. B., & Elmegaard, B. (2018). Comparison of Heat Transfer and Pressure Drop Correlations for Evaporation of Zeotropic Mixtures in Plate Heat Exchangers. In *Proceedings of the 13th IIR-Gustav Lorentzen Conference on Natural Refrigerants* (Vol. 1, pp. 659-668). International Institute of Refrigeration. <https://doi.org/10.18462/iir.gl.2018.1261>

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COMPARISON OF HEAT TRANSFER AND PRESSURE DROP CORRELATIONS FOR EVAPORATION OF ZEOTROPIC MIXTURES IN PLATE HEAT EXCHANGERS

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ABSTRACT

Zeotropic mixtures offer a possibility of optimizing heat pump cycle design by matching the working fluid temperature glide to the heat source and sink temperature profiles. Suitable heat transfer and pressure drop prediction methods are of paramount importance to evaluate the performance of plate heat exchangers (PHE) using such fluids. It is therefore relevant to evaluate the uncertainty in PHE performance estimation when zeotropic mixtures are used as working fluids. In this work, different correlations for the (Silver, 1947) and (Bell and Ghaly, 1973) method were compared to evaluate the evaporation heat transfer coefficient of mixtures of CO₂ and hydrocarbons at different mass compositions and subject to heat source temperature glides of 10 K, 15 K and 20 K. Moreover, the impact of using different pressure drop models and correlations was included in the sensitivity study on the evaporator performance. Maximum deviations of 6 % and 10 % were obtained on the total heat transfer rate, depending on the prediction method chosen for the heat transfer coefficient and pressure drop, respectively. Larger discrepancies were found in the estimation of the mean UA value and total frictional pressure drops. Working fluids subject to the largest glide of 20 K resulted to be more sensitive to the choice of the prediction method. No recommended set of correlation was found, but similarities and differences between the different mixtures were identified and discussed.

Keywords: Plate heat exchangers, Zeotropic mixtures, Evaporators, Prediction methods

1. INTRODUCTION

In the past two decades, increasing attention was posed on the utilization of natural refrigerants like hydrocarbons (HCs), proposed for their zero Ozone Depletion Potential and negligible Global Warming Potential. Hydrocarbons can be directly employed as pure fluids, but another possibility is to combine them in order to create refrigerant mixtures. For instance, the benefit of using mixtures of carbon dioxide (CO₂) and HCs leads to the advantage of reducing the typically high operating pressure of CO₂ systems and decreasing the flammability entailed by hydrocarbon utilization (Kim et al., 2008). Moreover, the so-called zeotropic mixtures offer the additional advantage of decreasing the irreversibility caused by temperature differences in the evaporator and the condenser by matching the heat source and sink temperature glides with the working fluid, due to its varying temperature during phase change. (Zühlsdorf et al., 2017a, 2017b) showed the possibility of increasing the heat pump coefficient of performance (COP) up to 27 % in two different case studies where different combinations of HCs and CO₂ were considered at different mass compositions.

The benefit obtained by using mixtures on the thermodynamic performance of the cycle entails however an increase in the necessary heat exchangers heat transfer area, due to mixture degradation of the heat transfer coefficient compared to pure fluids, observed in a number of experimental campaigns (Radermacher and Hwang, 2005). A number of reasons contribute to heat transfer degradation: (i) an earlier suppression of the nucleate boiling contribution. (ii) Formation of a mass diffusion resistance due to the more readily evaporation of the more volatile component. (iii) Reduction of effective super-heating and of the effective heat flow rate available for evaporation. (iv) Worse mixture transport properties (Collier and Thome, 1994).

This poses a major attention to the design and performance of heat transfer equipment when zeotropic mixtures are employed. Plate heat exchangers (PHEs) offer a suitable solution for the high heat transfer coefficient

achieved within their compact design and the possibility to arrange the flow in a counter-current mode, which is essential to achieve glide matching. Scientific literature collects different studies focused on experimental campaigns for deriving heat transfer coefficient and pressure drop correlations for two-phase flow in PHEs. (Vakili-Farahani et al., 2015) and (Eldeeb et al., 2015) carried out extensive literature reviews on the published experimental campaigns and prediction methods for evaporation and condensation in PHEs. In particular, (Vakili-Farahani et al., 2015) underlined the discrepancy of the different data and the difficulties of creating prediction models that are generally valid for PHEs, similarly to the ones existing for in-tube flow boiling. (Amalfi et al., 2016a) developed a correlation for heat transfer coefficient and friction factor during flow boiling in PHEs, based on the many experimental data sets available in literature. The developed prediction model showed better agreement with the experimental data than existing correlations. The databank was however limited to pure working fluids, with the exception of the near-azeotropic mixture R410a, the azeotropic mixture R507a and ammonia/water. The prediction method developed by (Amalfi et al., 2016a) is therefore not directly applicable to mixtures of natural refrigerants.

The difference between heat transfer in pure and multi-component working fluids was investigated in literature. (Silver, 1947) and (Bell and Ghaly, 1973) developed a method to account for the heat transfer degradation entailed by the mixture. The method uses weighted contributions of nucleate boiling, convective boiling and vapor phase heat transfer coefficients. In order to apply such method, different prediction methods can be chosen for the two-phase convective and nucleate boiling coefficients, as well as for the single-phase heat transfer coefficient. The objective of this paper was to estimate the impact of using different combinations of the heat transfer correlations developed for PHEs in the Silver and Bell-Ghaly method.

Moreover, the impact of employing different methods and correlations to calculate the frictional contribution to pressure drop was evaluated. The pressure drop literature review was based on the studies collected by (Amalfi et al., 2016b), and applied to the considered working fluids. Both sensitivity studies were carried out for a case study in which the heat exchanger (HEX) performance was estimated and the geometry of the PHE was fixed, thereby assessing the impact on the total heat flow rate at the evaporator.

2. METHODS

2.1 Case study and cycle boundary conditions

The comparison was based on a case study of a single-stage vapor compression heat pump, presented in (Zühlsdorf et al., 2017b), designed for delivering 2200 kW at the condenser, by heating up water from 40 °C to 80 °C. The inlet temperature of the heat source (water) was fixed to 40 °C, and different heat source temperature glides were assessed, namely 10 K, 15 K and 20 K, corresponding to Case I, II and III, respectively. Optimal zeotropic working fluid mixtures were found for the different temperature glides. These are reported in Table 1 with related mass composition, evaporation pressure, heat pump Coefficient of Performance (COP) and heat load at the evaporator.

Table 1: Evaporator boundary conditions for the different working fluids (Zühlsdorf et al., 2017b)

Case I, $\Delta T_{gl} = 10$ K				Case II, $\Delta T_{gl} = 15$ K				Case III, $\Delta T_{gl} = 20$ K			
Mixture	p_{ev} bar	COP -	\dot{Q}_{ev} kW	Mixture	p_{ev} bar	COP -	\dot{Q}_{ev} kW	Mixture	p_{ev} bar	COP -	\dot{Q}_{ev} kW
CO ₂ / DME (0.1/0.9)	7.5	5.62	1809	Propylene/ Butane (0.3/0.7)	4.0	5.40	1792	DME/ Pentane (0.3/0.7)	1.2	5.24	1780
DME/ Butane (0.5/0.5)	4.8	5.61	1807	Propane/ Butane (0.3/0.7)	3.9	5.38	1791	DME/Iso- Pentane (0.3/0.7)	1.7	5.19	1776
Ethane/ Propane (0.1/0.9)	12.3	5.60	1807	CO ₂ /DME (0.1/0.9)	7.0	5.32	1787	Propylene / Butane (0.5/0.5)	5.0	5.15	1773
Ethane/ Propylene (0.1/0.9)	14.5	5.57	1805	DME/Iso- Pentane (0.8/0.2)	4.5	5.31	1786				

The impact of the chosen prediction methods was assessed in a relevant practical situation, in which the geometry of the PHEs was entirely fixed and the suction volume of the refrigerant flow at the compressor inlet (corresponding to the evaporator outlet) and the super-heating were fixed by the control. The required number of channels was estimated for each working fluid with a preliminary evaporator sizing based on constant heat transfer coefficients in the different regions of evaporation and super-heating. The other HEX parameters were defined according to an existing evaporator model (V65) manufactured by SWEP (SWEP International AB, 2015). Its design parameters are reported in Table 2. The plate size was given by plate width W and port-to-port length L_p , while the corrugation characteristics were the chevron angle β , the corrugation height b and pitch Λ . The plate thickness t gives the trade-off between conductive resistance and mechanical strength. The PHE was assumed to be constructed in stainless steel, with a thermal conductivity equal to 16.2 W(mK)^{-1} .

Table 2: Geometry of the PHE – SWEP Evaporator type V65 (SWEP International AB, 2015)

W , m	L_p , m	β , °	b , mm	Λ , mm	D_p , m	t , mm
0.363	0.731	35	1.9	7	0.1	0.5

It was assumed to employ a thermostatic expansion valve, thus calculating the refrigerant super-heating in relation to inlet pressure. The refrigerant mass flow and saturation pressure were calculated in order to match the control values, and a numerical model was employed to estimate the PHE performance.

2.2 Plate heat exchanger numerical model

The numerical model of the PHE was based on an internal solver for the solution of heat transfer and fluid flow in the PHE, coupled with a Newton-Raphson solver iterating on the refrigerant operating conditions. Fluid thermo-physical properties were calculated using Refprop (Lemmon et al., 2013) and Coolprop (Bell et al., 2014), for mixtures and water, respectively. The tolerance of the Newton-Raphson solver was set to 10^{-3} on the relative residuals of the refrigerant suction volume and super-heating.

The internal solver was based on a successive substitution approach iterating on wall temperature and pressure drops. A one-dimensional (1D) discretization of the HEX was employed along the flow direction with equidistant heat transfer area steps. The solution strategy was built based on a combination of the SEWTLE solver presented in (Corberán et al., 2000) and an improved solution strategy based on alternate refrigerant and heat source iterations as suggested in (Eldeeb et al., 2016). A total number of 100 control volumes (CVs) was employed as a trade-off between accuracy and computational cost, with solver tolerance set to 10^{-5} for the norm of relative residuals of wall temperature and refrigerant and source pressure drops.

2.3 Prediction methods – heat transfer coefficient

The heat transfer coefficient in the two-phase region was estimated by using the (Silver, 1947) and (Bell and Ghaly, 1973) method, with the formulation expressed by Eq. (1).

$$h_{TP} = \frac{1 + \frac{h_{NB_{mix}}}{h_C}}{\frac{1}{h_C} + \frac{\bar{z}}{h_V}} \quad (1)$$

h_{TP} indicates the two-phase heat transfer coefficient and it is a function of the nucleate boiling (NB) contribution $h_{NB_{mix}}$, the convective boiling (CB) heat transfer coefficient h_C , and the vapor phase heat transfer coefficient h_V . The nucleate boiling contribution $h_{NB_{mix}}$ was calculated based on a dedicated experimental correlation, and corrected by a factor taking the degradation entailed by the mixtures into account. For this purpose the (Thome and Shakir, 1987) correlation was employed, as reported in Eq.(2). The mixture degradation is as a function of the difference between dew and bubble point temperatures ($T_{dew} - T_{bubble}$), the specific heat flux per unit area \dot{Q}'' , latent heat of evaporation h_{lat} and liquid density ρ_L . β_L represents the liquid mass transfer coefficient, equal to 0.0002 m/s.

$$\frac{h_{NB_{mix}}}{h_{NB_{id}}} = \left(1 + \frac{h_{NB_{id}}}{\dot{Q}''} (T_{dew} - T_{bubble}) \left(1 - e^{-\frac{\dot{Q}''}{\rho_L h_{lat} \beta_L}} \right) \right)^{-1} \quad (2)$$

Last, the correction factor \bar{z} of Eq.(1) was introduced by (Silver, 1947) and (Bell and Ghaly, 1973), as function of vapor quality x , vapor specific heat capacity c_{pV} and the slope of evaporation curve dT/dh , indicating the rate of sensible heat transfer over the total heat load.

$$\bar{z} = x \frac{dT}{dh} c_{pV} \quad (3)$$

The sensitivity analysis was based on the correlations reported in Table 3. (Hsieh and Lin, 2003) was developed as a superposition model, with distinct contributions for nucleate and convective boiling. (Longo et al., 2015) was developed by considering a transition point between nucleate and convective dominated regions, based on the Boiling number. Therefore, each term of both correlations was considered in the appropriate category, as reported in Table 3. The remaining two-phase correlations were categorized as convective or nucleate boiling methods, depending on the related authors observations on the heat transfer, which could be dominated by heat flux (nucleate boiling) or by vapor quality and mass flow (convective boiling). All possible combinations of the three terms were analyzed, leading to 75 different possibilities for each fluid.

Table 3: Summary of the considered experimental correlations for the different terms of Eq. (1)

	h_c	$h_{NB_{id}}$	h_v
	Two-phase convective	Two-phase nucleate boiling	Single-phase
a	(Danilova et al., 1981)	A (Cooper, 1984)	(Chisholm and Wanniarachchi, 1992)
b	(Hsieh and Lin, 2003)	B (Hsieh and Lin, 2003)	(Wanniarachchi et al., 1995)
c	(Han et al., 2003)	C (Palm and Claesson, 2006)	(Martin, 1996)
d	(Longo et al., 2015)	D (Gorenflo and Kenning, 2010)	
e	(Amalfi et al., 2016a)	E (Longo et al., 2015)	

The heat transfer coefficient of the heat source (water) was estimated using the single-phase correlations reported in the last column of Table 3. The single-phase heat transfer coefficient of the refrigerant in the superheated region was estimated by the same correlation. A weighted interpolation was used in the transition between evaporation and super-heating from vapor quality of 0.9.

2.4 Prediction methods – frictional pressure drops

The pressure drops of the heat source and of the refrigerant were estimated as the sum of the contributions of acceleration, friction, gravity and manifolds. The acceleration and gravity contributions were estimated by assuming homogeneous flow (Vakili-Farahani et al., 2015), while manifold pressure drops were calculated using the correlation by (Shah and Focke, 1988). An upward refrigerant flow and downward water flow were considered, as typical in the considered PHE configuration (SWEP International AB, 2015).

The frictional pressure drop was estimated by using three different models found in literature. First, correlations calculating the two-phase Fanning friction factor were considered as indicated in the first column of Table 4. The second adopted model was the (Lockhart and Martinelli, 1949) method, consisting in the estimation of two-phase multipliers correlating the single-phase liquid and vapor pressure drops to the two-phase pressure drop. The (Palm and Claesson, 2006) coefficients were used, thereby assuming a Chisholm parameter of 4.67, found using experimental data for PHEs. Last, the kinetic energy model was used, assuming proportionality between the kinetic energy per unit volume and the frictional pressure losses, as proposed by (Longo and Gasparella, 2007). All the models were employed for the estimation of the frictional pressure drop

locally, thereby summing all the contributions of the CVs to obtain the total refrigerant and heat source frictional pressure drop in the PHE. In the super-heating region, the single-phase Fanning friction factor proposed by the single-phase correlations reported in the last columns of Table 3 was used.

Table 4: Prediction methods for the two-phase frictional refrigerant pressure drops

Two-phase Fanning friction factor		Lockhart-Martinelli multipliers	
F 1	(Yan and Lin, 1999)	LM 1	(Palm and Claesson, 2006)
F 2	(Hsieh et al., 2002)		
F 3	(Hsieh and Lin, 2003)	Kinetic energy proportionality	
F 4	(Han et al., 2003)	KE 1	(Longo and Gasparella, 2007)
F 5	(Amalfi et al., 2016a)		

3. RESULTS

Subsection 3.1 presents the results of the sensitivity analysis on the heat transfer coefficient, thereby illustrating the impact of the different correlations combinations on the mean UA value and on the UA development in the PHE. Subsection 3.2 presents the results of the sensitivity analysis of the different calculation methods for frictional pressure drops. Subsection 3.3 illustrates how the UA value and the frictional pressure drops impact the output of the performance model, i.e. the heat transfer rate at the evaporator.

3.1 Impact on the UA values

Fig. 1, Fig. 2 and Fig. 3 show the heat transfer correlation sensitivity analysis results for Case I, II and III, respectively. Fig. 1 (a), Fig. 2(a) and Fig. 3 (a) report the mean UA value for all the combinations: the abscissa indicates different combination of nucleate and convective boiling correlations, while the three different marker styles represent the three single-phase correlations. Different colors represent the different working fluids of the three cases. Fig. 1 (b), Fig. 2 (b) and Fig. 3 (b) represent the development of the UA value along the evaporator, as function of the cumulated heat flow rate. The results are shown for only two correlation combinations for each fluid, represented by a continuous and a dotted line. The former characterizes the combination estimating the highest mean heat transfer coefficient for the specific fluid, while the latter represents the correlations giving the lowest mean heat transfer coefficient.

Better heat transfer performance was obtained for smaller temperature glides, with mean UA values up to 400 kW/K for Case I against values not higher than 300 W/K for Case III. By looking at the UA distribution in the evaporator in Fig. 1 (b), Fig. 2 (b) and Fig. 3 (b), Case I yields a steeper heat transfer growth for increasing vapor quality compared to Case II and III for the best performing correlation combinations. This is due to the higher degradation of heat transfer for increasing temperature glide for both the case of nucleate and convective boiling, as it is expressed by Eq. (2) and (3), due to the larger sensible heat flow rate needed for increasing the mixture temperature during phase change. The fluid ranking in terms of heat transfer performance remains the same in all cases, regardless of the correlation applied. By looking at Case I in Fig. 1 (b), the UA value of DME/Butane (50/50) outperforms the other fluids for both the best and worst correlation combinations.

Similarities were found for the single-phase and nucleate boiling correlations. The single-phase correlation by (Chisholm and Wanniarachchi, 1992) led to higher heat transfer compared to (Martin, 1996) and (Wanniarachchi et al., 1995), which yielded the lowest UA. This holds for all the mixtures, as shown by the different markers in Fig. 1 (a), Fig. 2 (a) and Fig. 3 (a). (Hsieh and Lin, 2003) (B) underestimated the NB contribution compared to the other correlations, while (Palm and Claesson, 2006) (C) overestimated the nucleate boiling. The results, shown in Fig. 1 (b), Fig. 2 (b) and Fig. 3 (b), are the same for all the fluids.

By looking at convective boiling contributions, (Hsieh and Lin, 2003) (b) correlation led to the best heat transfer coefficient for all the working fluids of Case I, while (Longo et al., 2015) (d) yielded the best

performances for Case II and III. The lowest CB contribution to heat transfer was estimated by (Amalfi et al., 2016a) (e) and (Danilova et al., 1981) (a), depending on the working fluid.

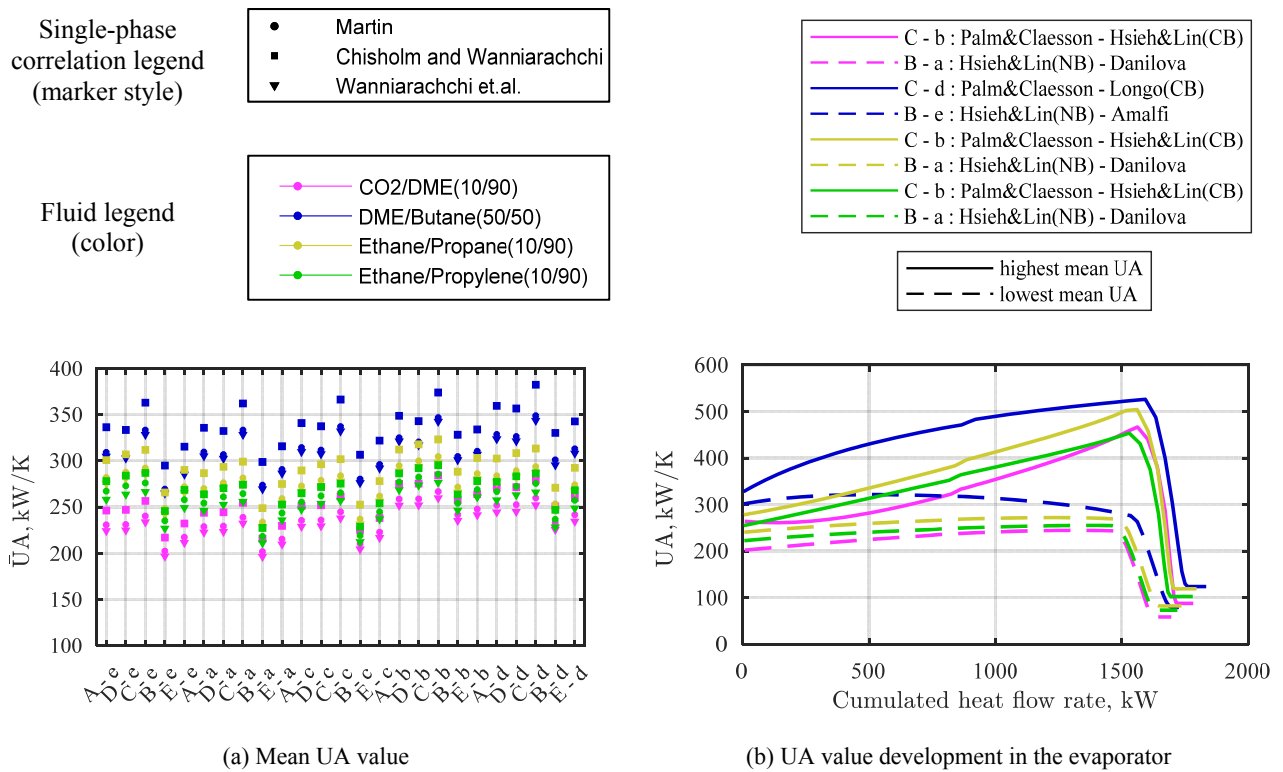


Fig. 1: Case I results (a) Impact of the different heat transfer correlations on the mean UA value; (b) UA as function of the cumulated heat flow rate in the evaporator.

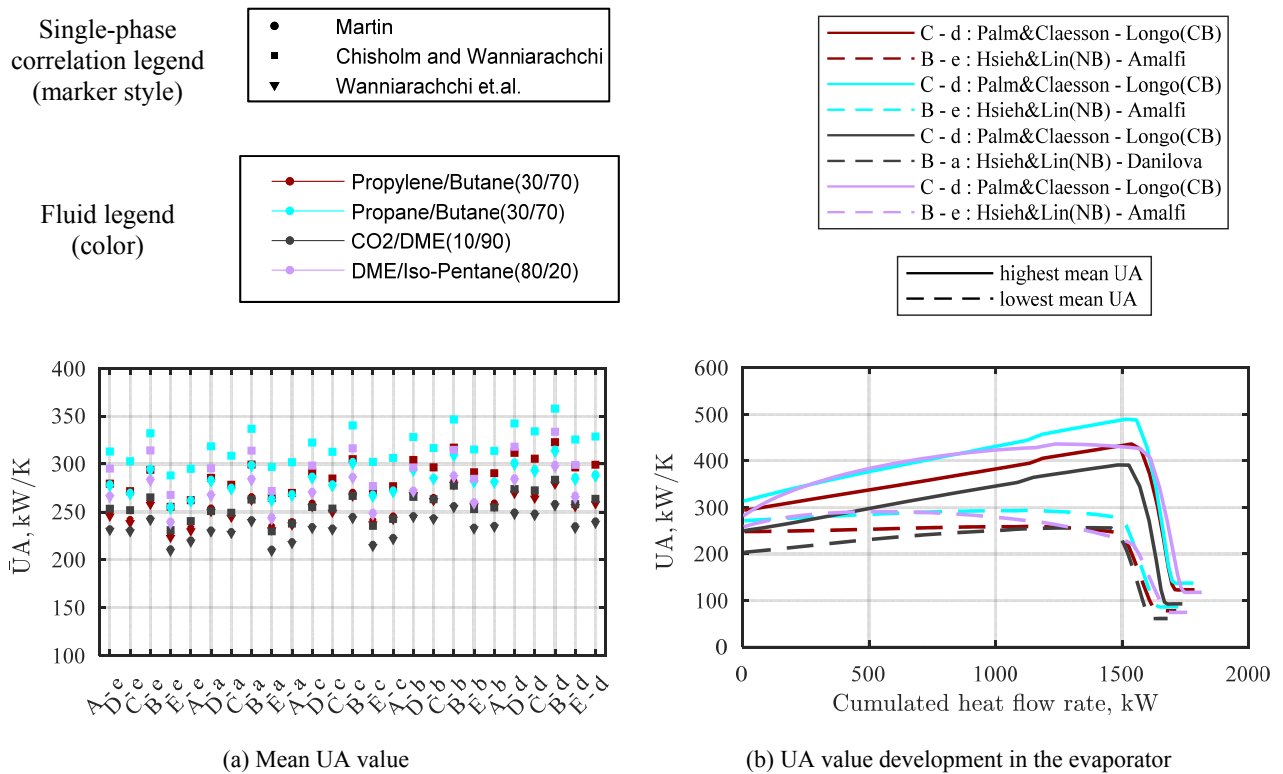


Fig. 2 Case II results (a) Impact of the different heat transfer correlations on the mean UA value; (b) UA as function of the cumulated heat flow rate in the evaporator.

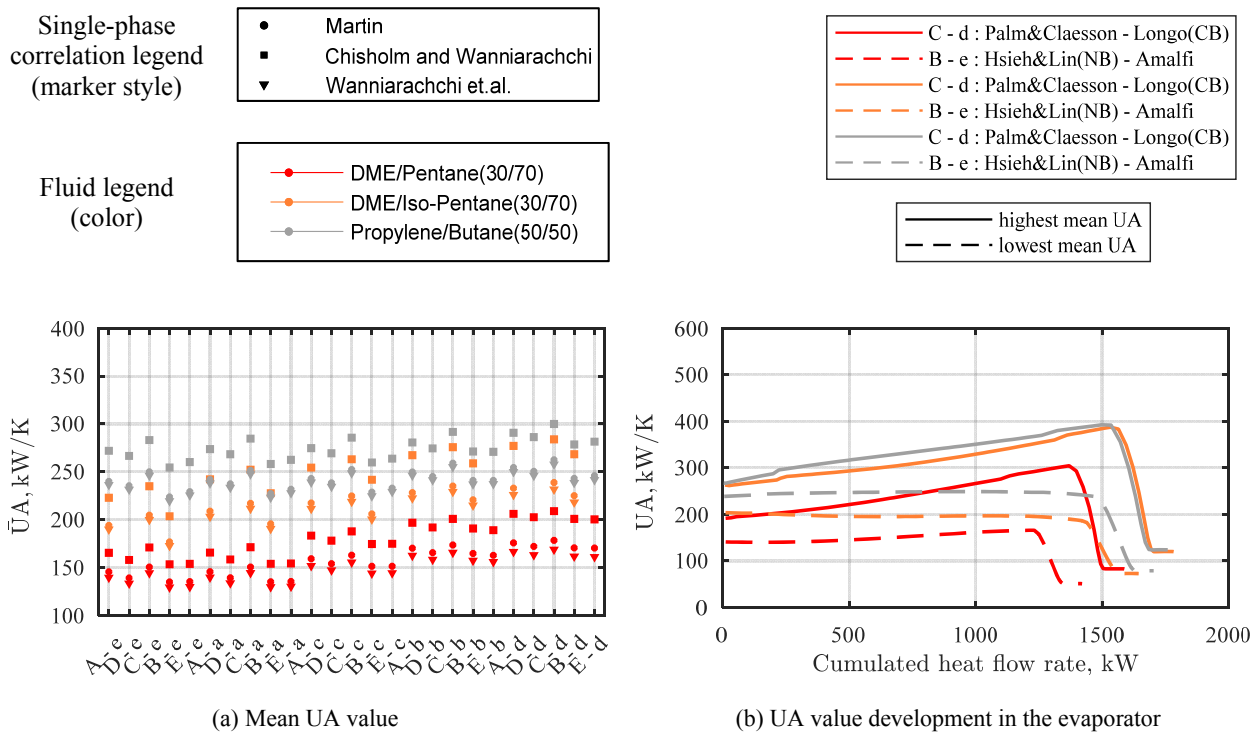


Fig. 3 Case III results (a) Impact of the different heat transfer correlations on the mean UA value; (b) UA as function of the cumulated heat flow rate in the evaporator.

3.2 Impact on the frictional pressure drops

Fig. 4 (a), (b) and (c) report the results of the pressure drop sensitivity analysis for Case I, II and III, respectively. The mixture temperature glide does not affect the results, opposite to the heat transfer coefficient behavior. The model by (Lockhart and Martinelli, 1949) with (Palm and Claesson, 2006) coefficients, the model by (Longo and Gasparella, 2007) and the friction factor correlations F 4 (Han et al., 2003) and F 5 (Amalfi et al., 2016a) led to similar pressure drop values. On the other hand, the Fanning friction factor correlations F 1, F 2 and F 3, namely (Yan and Lin, 1999), (Hsieh et al., 2002) and (Hsieh and Lin, 2003) over-estimated the pressure drop compared to the other correlations. DME/Iso-pentane(30/70) was the working fluid most sensitive to the choice of the correlation, with a large difference between (Lockhart and Martinelli, 1949) model and (Yan and Lin, 1999) correlation results. It must be noted that some results are not presented for DME/Pentane (30/70) in Case III, due to model convergence issues for very high pressure drops. It can be therefore be concluded that this fluid was very sensitive as well to the choice of the prediction method.

3.3 Impact on the performance model output

Fig. 5 reports the percentage deviation from the mean of all correlation combinations of the most deviating correlation combinations for heat transfer, which were identified in Fig. 1 (b), Fig. 2 (b) and Fig. 3 (b). The deviations are reported for the mean UA value, indicating the impact of the chosen correlations on the heat transfer coefficient calculation, and the consequent effect on the total heat flow rate at the evaporator, output of the performance model. Case I and Case II presented maximum differences in mean UA values around $\pm 30\%$, while Case III reported slightly higher deviations, with a maximum around $\pm 35\%$. A smaller effect was however found on the total heat flow rate. Despite larger discrepancies were obtained on the mean UA value, maximum deviations around $\pm 2\%$ for Case I and II and $\pm 6\%$ for Case III, can be observed in Fig. 5.

Fig. 6 reports the results of the frictional pressure drops. The maximum deviations were reported from the mean value obtained for each working fluids. As expected from the frictional pressure drop values presented in Fig. 4, a deviation of up 130% was obtained for some of the fluids if correlations F 1, F2 and F 3 were employed. A much lower impact was however found on the total heat flow rate at the evaporator, similarly to the heat transfer correlation case. Case III was the most sensitive to the choice of the correlations, reporting maximum deviation on the total heat transfer rate around $\pm 10\%$ for DME/Iso-Pentane(30/70).

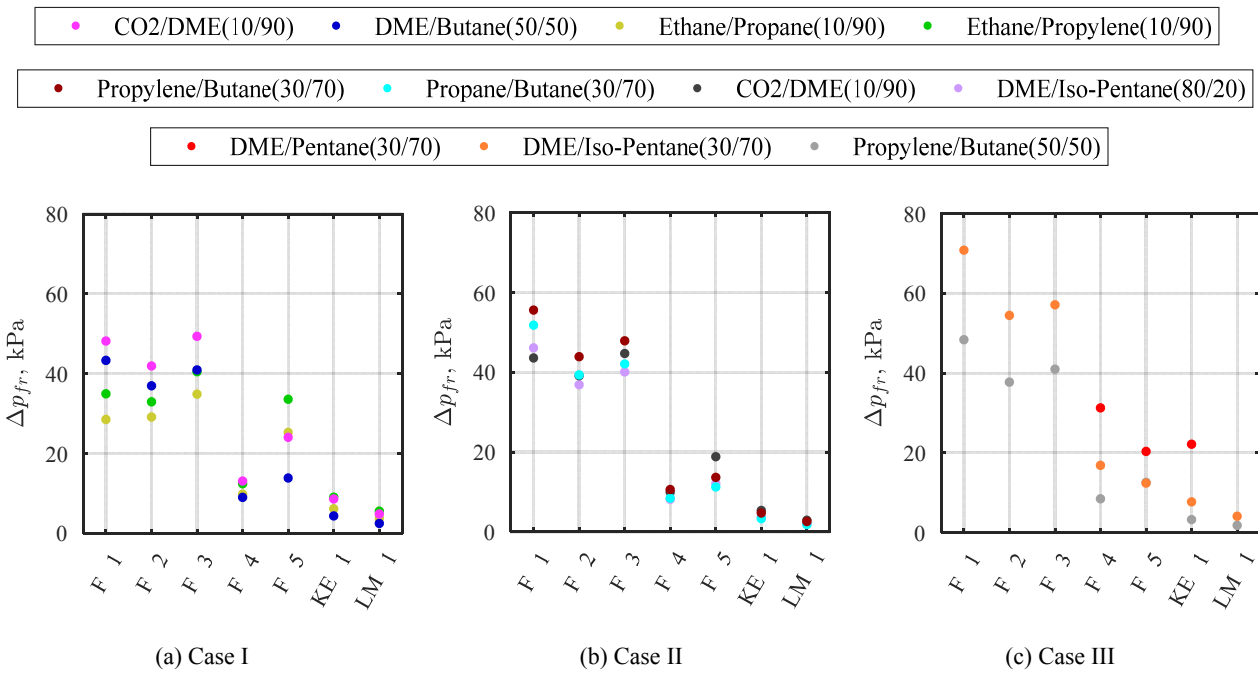


Fig. 4 Frictional pressure drop estimated with the different correlations, for Case I (a), II (b) and III(c)

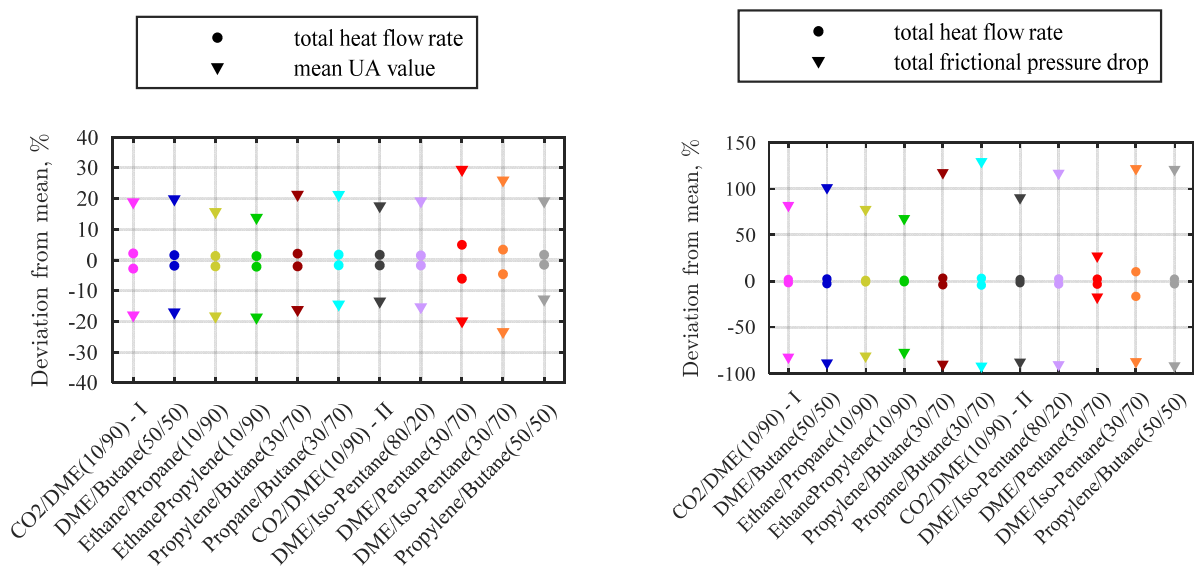


Fig. 5 Deviation from the mean of the UA value and of the total evaporator heat flow rate

Fig. 6 Deviation from the mean of frictional pressure drops and of the total evaporator heat flow rate

4. DISCUSSION

The presented results suggest that the choice of the prediction method had a significant impact on the estimation of both the heat transfer coefficients and pressure drops. It is however not possible to conclude with a clear recommendation on a set of correlations. The working fluids presented similarities in terms of correlations overestimating or underestimating both UA and frictional pressure drop compared to the mean results and this must be taken into account when choosing a certain prediction method for other mixtures of natural refrigerants. It would however be necessary to compare the results with experimental data specifically obtained for zeotropic mixtures evaporation in PHEs, in order to identify a specific recommended set of correlations for both heat transfer and pressure drops.

Moreover, the results suggested that despite the large discrepancies on the absolute values of UA and pressure drops, a much smaller deviation was obtained on the output of the performance model, i.e. the heat transfer rate at the evaporator. It must be therefore taken into account that different degrees of uncertainties are related with the correlations choice depending on the scope of the analysis. For a user interested in analyzing the PHE performance impact on a thermodynamic cycle, the choice of the prediction method would not lead to large differences (max 6 % and 10% were found for heat transfer and pressure drop correlations, respectively). On the other hand, it would be very relevant to consider the much larger differences between correlations for a user focusing on detailed heat transfer and pressure drop mechanisms in the component.

A typical practical situation was assessed in the present study, i.e. evaluating the performance of the evaporator (with fixed geometry) as the operating conditions were adjusted according to control. The design problem represents an additional situation in which experimental correlations are used. In this case, the operating conditions are fixed and the PHE heat transfer area must be found to meet the design thermal load. A sensitivity analysis to assess the prediction methods impact on the design problem will be therefore part of future work. Last, it must be noted that the sensitivity analysis was limited to assess the impact of different correlations for the (Silver, 1947) and (Bell and Ghaly, 1973), and no correlation which was specifically derived for zeotropic mixtures was included in the analysis. This highlights one more time the importance of focusing further work on deriving experimental correlations for evaporation of different zeotropic mixtures in PHEs.

5. CONCLUSION

The impact of heat transfer and pressure drop correlations was estimated for evaporation of zeotropic mixtures in PHEs. The analysis was based on a performance problem, fixing the suction volume at the evaporator outlet and the mixture super-heating for a given case study. Different mixtures of HCs and CO₂ subject to three different temperature glides were evaluated. The main findings can be summarized as follow:

- Differences of up 30 % were obtained for the mean UA value for Case I and II, while Case III resulted in slightly larger deviation of up 35 %.
- Working fluids experiencing larger temperature glides presented a higher degradation of heat transfer, as well as a stronger dependence on the chosen prediction method.
- Differences of up 130 % were obtained for the frictional pressure drop estimation, with DME/Iso-pentane(30/70) being most sensitive to the correlation applied
- Smaller deviations were obtained on the performance model output, namely 6 % and 10 % for the heat transfer and pressure drop correlation choice, respectively.
- It was not trivial to identify a recommended combination of correlations, despite the similarities between the working fluids. Experiment validation is needed to clarify the results.

ACKNOWLEDGEMENTS

The research work was conducted within the frame of the THERMCYC project (Advanced thermodynamic cycles utilizing low temperature heat sources) funded by Innovation Fund Denmark: The Danish Council for Strategic Research in Sustainable Energy and Environment. The support is gratefully acknowledged.

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