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Design recommendations for plate heat exchangers in heat pumps using pure and mixed refrigerants

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ABSTRACT

This paper aims at deriving design recommendations for plate heat exchanger (PHE) evaporators and condensers in heat pump (HP) systems using different refrigerants. A coupled HP - PHE simulation framework was used to carry out cycle and component design for a case study, with different combinations of PHE geometrical parameters, and propane, butane and propylene/butane (0.5/0.5) as refrigerants. An operating model subsequently evaluated the heat pump thermodynamic and economic performance with the different designed PHEs: the trade-off between heat transfer area and refrigerant pressure drops was therefore taken into account for both components. Only evaporator pressure drops affected the Coefficient of Performance (COP), and different refrigerants presented dissimilar slopes of degradation. Condenser design did not influence the COP, and the economic optimum corresponded to the lowest size. The evaporator design at minimum cost was instead found as a trade-off between heat transfer area and pressure drops for all the working fluids. Despite the different rates of COP drop, the three refrigerants resulted in the same optimal pressure drop at the evaporator, equal to around 5 kPa.

Keywords: Pressure drops, Zeotropic mixtures, Evaporators, Condensers, Economic analysis

1. INTRODUCTION

Optimal design of evaporators and condensers in heat pumps plays a key-role for overall system thermodynamic and economic optimization. Plate heat exchangers (PHEs) offer a compact design that is feasible for both pure fluids and zeotropic mixtures. Contrary to the most common configurations of micro-channel heat exchangers (HEXs), PHE allows counter-current flow arrangement, which is essential for temperature glide matching between mixed refrigerants and heat source/sink. Nonetheless, HEX design often relies on criteria based on heuristics, such as maximum allowable inlet velocities or maximum pressure drop, possibly leading to suboptimal solutions, depending on the working fluid (Mancini et al., 2018). Previous studies (Muralikrishna and Shenoy, 2000; Wang and Sundén, 2003) attempted at establishing trade-off criteria between pressure drops and investment in heat transfer area, yet mostly focusing on single-phase application or without considering the impact on the thermodynamic cycle performance. On the other hand, HEX design optimization (with or without system integration) was addressed by several works in literature, also focusing on PHEs (Walraven et al., 2014; Xu et al., 2015). However, a simultaneous component/system optimization may entail a high computational cost, especially during the preliminary system design phase and working fluid selection. For example, screening of multiple combinations of pure fluids at different compositions to evaluate the performance of mixed refrigerants (Zühlsdorf et al., 2019), may require too many evaluations of system performance.

In this context, the present work aims at: (i) comparing the relative impact of condenser and evaporator design on the heat pump thermodynamic and economic performance, by looking at both area investment and pressure drops. (ii) Estimating the differences between three working fluids (pure and mixed refrigerants), for a case study. (iii) Discussing similarities and differences considering fluids thermo-physical properties. (iv) Attempting at the definition of general design recommendations for PHE evaporator and condenser in HP systems to be used during the preliminary system design phase.

2. METHODS

This section describes the chosen case study, the overall procedure and the numerical modelling.

2.1. Case Study

A case study previously published in (Zühlsdorf et al., 2019) was chosen for the analysis. A heat pump (HP) was integrated in a data centre facility for waste heat recovery purposes, with the aim of supplying heat to the district heating (DH) network. Propane, butane and a mixture of propylene/butane at (0.5/0.5) mass composition were found among the best performing working fluids. The heat pump was sized by fixing the heat transfer rate at the evaporator, i.e. 500 kW to use as waste heat. Moreover, the heat source was considered to be water entering the evaporator at 50 °C and leaving at 25 °C, while the heat sink was considered to be water entering the condenser at 50 °C to be heated up to 75 °C, i.e. the temperature level required by the DH network.

Table 1. Heat pump design and operating parameters for the three considered refrigerants

Working fluid	$\dot{Q}_{\mathrm{cond}}$, kW	$\dot{Q}_{\mathrm{eva}}$, kW	p_{cond}, bar	p_{eva}, bar	COP, -	$\dot{V}_{\mathrm{out,eva}}$, m^3/h
Propane	638	500	28.1	8.8	4.4	370
Butane	630	500	9.5	2.2	4.5	1100
Propylene/Butane	600	500	16.4	5.6	5.6	460

2.2. Coupled heat pump-plate heat exchanger simulation framework

The simulation framework structure is shown in Figure 1. The models were built in MATLAB (Mathworks, 2017), and fluid properties were estimated by REFPROP 10 (Lemmon et al., 2018). The reader is referred to (Mancini et al., 2018) for a detailed description of the PHE design models, and to (Mancini et al., 2019) for a comprehensive description of the heat pump design and operating models, with detailed PHE characterization.

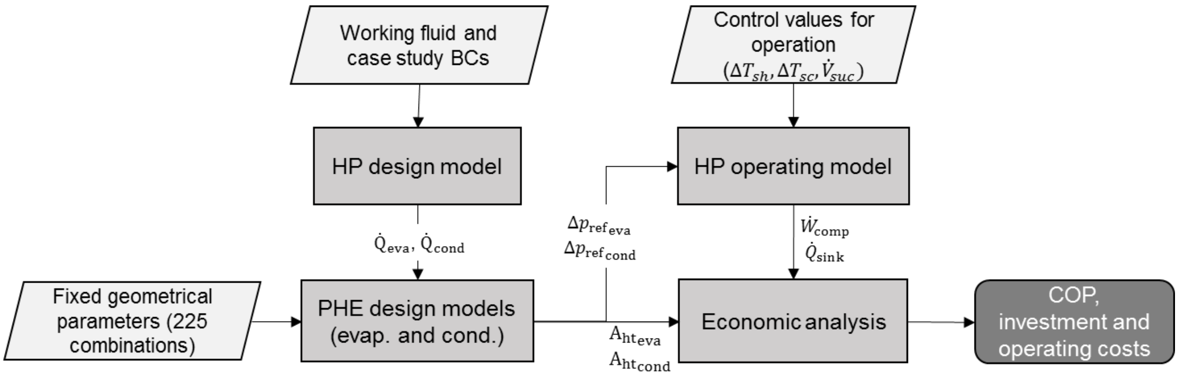

Figure 1: Workflow of the overall method

First, the working fluid and the case study boundary conditions (BCs) were defined and used in the HP design model. The PHE evaporator and condenser designs were subsequently carried out, for different combinations of geometrical parameters. The aim of the parametric analysis was to obtain PHE designs with different heat transfer areas and pressure drops. The resulting configurations were finally used in a HP operating model, which additionally received as input the control values chosen for the operation, namely a fixed superheat, subcooling and suction volume flow rate at the compressor inlet. The resulting HP operating conditions, affecting the compressor power requirement and the heat supply of the condenser, were used in an economic analysis to estimate the COP and operating costs.

2.2.1. Parametric Analysis on PHE geometry

A parametric study was conducted on the geometry of both evaporator and condenser. It was decided to fix the corrugation geometry of the PHEs and to carry out a sensitivity study on the plate size and count. The plate width W and number of channels N_{ch} were varied for both the components, while the plate length was estimated as an output of the design model. The values used in the parametric analysis are reported in Table 2.

Table 2: Plate width and number of channels values used in the parametric study

Parameter	Working fluid	Values	Unit
W_{eva}	All fluids	0.250 – 0.400 – 0.600	m
$N_{\mathrm{ch,eva}}$	Propane	50 – 100 – 150 – 200 – 250	-
	Butane	70 – 120 – 170 – 220 – 270	
	Propylene/butane	80 – 130 – 180 – 230 – 280	
W_{cond}	All fluids	0.200 – 0.400 – 0.600	m
$N_{\mathrm{ch,cond}}$	Propane	50 – 100 – 150 – 200 – 250	-
	Butane	70 – 120 – 170 – 220 – 270	
	Propylene/butane	50 – 100 – 150 – 200 – 250	

2.2.2. Evaluation of thermodynamic and economic performances

The thermodynamic and economic performance of the heat pump for different PHE geometries of both evaporator and condenser were assessed by means of performance indicators. The COP, estimated by Eq. (1), indicated the thermodynamic performance.

$$\mathrm{COP} = \frac{\dot{Q}_{\mathrm{sink}}}{\dot{W}_{\mathrm{comp}}} \qquad \text{Eq.(1)}$$

The specific cost of heat, calculated by means of Eq. (2) and expressed in €/MWh, was instead chosen to quantify the economic performance (Zühlsdorf et al., 2019). In Eq. (2), CF represents the fuel cost to run the compressor, TCI the total capital investment and CRF the capital recovery factor, estimated as function of an effective interest rate. OH represents the HP yearly operating hours.

$$\mathrm{c_h} = \frac{\mathrm{CF_{el}} + \mathrm{TCI} \cdot \mathrm{CRF}}{\dot{Q}_{\mathrm{sink}} \cdot \mathrm{OH}} \qquad \text{Eq.(2)}$$

The use of the specific cost of heat allows estimating the price at which the heat must be sold to the DH network in order to compensate for the investment and running cost of the HP. In this specific case study, the heat source was assumed as free waste heat from the data centre facility.

3. RESULTS

Section 3.1 deals with the impact of PHE designs on the heat pump COP, while Section 3.2 introduces the results of the economic analysis. Section 3.3 presents optimal designs obtained for the different refrigerants. Design recommendations are finally summarized in Section 3.4.

3.1. Impact of Evaporator and Condenser Design on Thermodynamic Performance

Figure 2 shows how the refrigerant pressure drops of the evaporator affected the heat pump COP. Each point corresponds to one combination of the design variables reported in Table 2, e.g. different evaporator and condenser geometries. The different marker styles and colours represent the working fluid. The COP value was reported by scaling it with respect to the design value. The thermodynamic performance degradation due to pressure drops for the different PHE evaporator designs was fitted linearly for each working fluid, and Figure 2 also reports the value of the slopes. For example, choosing a design with 30 kPa pressure drops entails around 4 % COP drop for butane, 3 % for propylene/butane, while it stays lower than 2 % for propane.

It must be noted that only few design points are visible in the figure compared to the 225 different combinations evaluated in the sensitivity study. This is due to the condenser design, which does not affect the COP, hence the points with the same evaporator geometry and different condenser designs overlap. Figure 3 (a) further illustrates this result, reporting the COP for the three working fluids vs. the total refrigerant pressure drops at the condenser. There is no dependency between the COP and the condenser design, and the colour scale – indicating different normalized evaporator pressure drops – shows that maximum COP was obtained for minimum evaporator pressure drops in all the cases. Note that Figure 3 (a) shows that the relative ranking of the working fluids in terms of COP was slightly affected by the PHE designs. Propylene/butane remained the refrigerant with the highest COP regardless of the evaporator and condenser configuration, while butane and propane were found to overlap depending on the evaporator design (pressure drops).

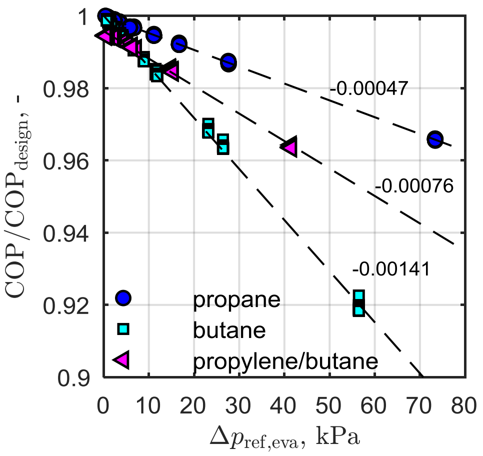

Figure 2: Normalized COP as a function of total refrigerant pressure drop in the evaporator

Table 3: Bubble and dew temperatures and thermo-physical properties at design pressure

Working fluid	p_{eva}, bar	ρ_{V}, kg/m^3	dT_{bubble}/dp K/kPa
Propane	8.8	19.1	0.044
Butane	2.2	5.7	0.140
Propylene/Butane	5.6	13.6	0.054

The stronger dependence of the COP to the evaporator pressure drops was related to a number of reasons. First, a higher value of pressure drops entailed a decrease in the heat output of the condenser. This was due to the impact of evaporator operation on the operating parameters of the cycle, i.e. refrigerant mass flow rate and pressures. The mass flow rate was in fact diminished at fixed superheat and subcooling, and the HP operated at a lower pressure ratio, i.e. a lower compressor power was required. However, the decrease in total heat flow rate of the condenser affected the COP negatively to a larger extent, as shown in Figure 3 (b): the condenser total heat flow rate – normalized by the design value – is plotted as a function of the evaporator pressure drops, with different colours representing different magnitudes of normalized condenser pressure drops. Similarly to the COP degradation reported in Figure 2, butane reports the steepest degradation, followed by propylene/butane and propane. The slopes of degradation were found to be similar to the COP drop plotted in Figure 2. Condenser pressure drops, represented by the different colour scale, were found not to affect the total heat flow rate at the condenser, and different condenser designs overlap in Figure 3 (b) similarly to Figure 2.

The higher drop of heat pump COP for butane compared to the other two fluids is due to another main reason: evaporator pressure drops also imply a saturation temperature drop, which increases the exergy destruction due to finite temperature differences in the evaporator component, thus contributing to the COP degradation. Depending on some relevant fluid thermo-physical properties, refrigerants can be more or less subject to this effect.

(Brignoli et al., 2017) identified the ones characterizing refrigerants sensitivity to pressure drops, namely vapour density and saturation temperature drop due to pressure drop. These are reported in Table 3 for the three working fluids at design conditions. Lower vapour density implies higher pressure drops for the same mass flux and PHE geometry. Therefore, a design that is feasible for propane and propylene/butane, possibly enhancing the heat transfer coefficient, might entail too large pressure drops for butane, with vapour density much lower than the other two refrigerants. This was one of the reasons why the bounds for plate width and number of channels variation in the sensitivity study (see Table 2) were chosen differently for the working fluids, i.e. to ensure feasible PHE designs with similar values of pressure drops.

The refrigerant vapour density is thus a property that must be carefully taken into account when choosing the working fluid and subsequently selecting the HEX design. Higher saturation temperature drop due to pressure drop, estimated by the Clapeyron relation (Moran, 2017), entails a larger drop of the saturation temperature due to pressure drop, thereby increasing the exergy losses due to the temperature difference between the refrigerant and the heat source side in the evaporator. Table 3 shows how the ranking between the working fluids in terms of vapour density and dT_{bubble}/dp respects the slopes of the COP degradation reported in Figure 2. Therefore, despite both propane and propylene/butane reported similar slopes for the degradation of condenser heat flow rate, the mixture was found to be more sensitive to saturation temperature drop due to pressure drop, resulting in a steeper COP drop.

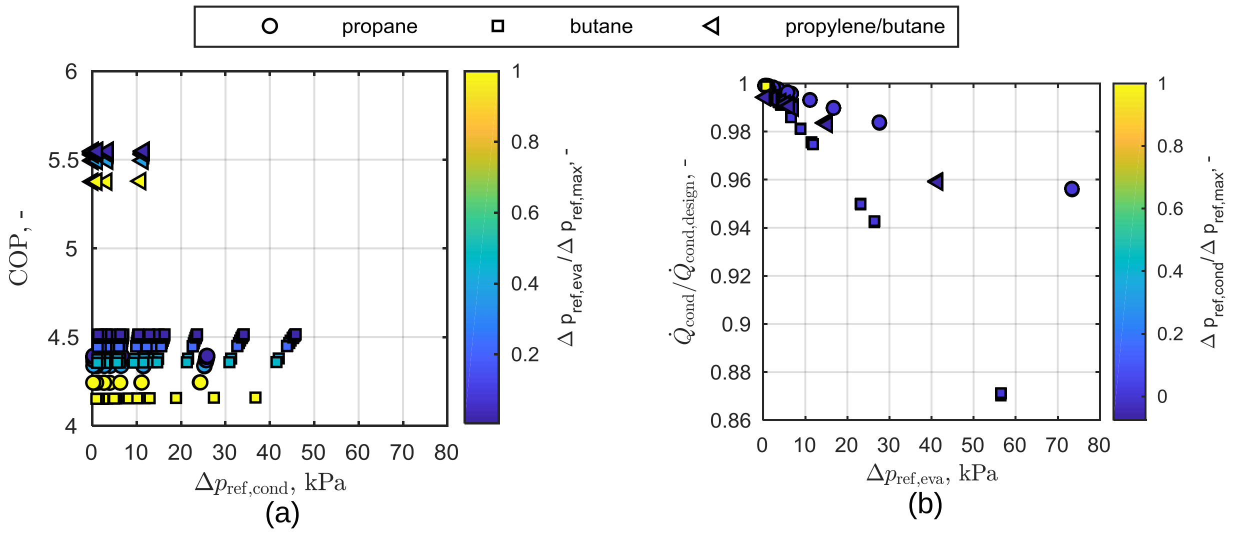

Figure 3: (a) COP as a function of total refrigerant pressure drop in the condenser; (b) Normalized condenser heat flow rate as function of total refrigerant pressure drop in the evaporator

3.2. Impact of Evaporator and Condenser Design on Economic Performance

The economic performance was evaluated by looking at the specific cost of heat, which is influenced by both condenser and evaporator design. The TCI is dependent on both HEX sizes (related to the heat transfer area), while the operating costs are strongly related to the heat pump COP. The economic performance indicator is reported in Figure 4 (a) for the three refrigerants as function of the pressure drop at the evaporator. The three working fluids present similar trends, with a trade-off and a minimum for a certain value of refrigerant pressure drops. Figure 4 (a) reports a close-up for the case of propane, where the colours represent different values of condenser heat transfer area. The results show that evaporator and condenser have different types of impact on the specific cost of heat. For a fixed condenser design – thus a fixed value of heat transfer area shown by the colours of Figure 4 (b) – the economic performance was a trade-off between heat transfer area and pressure drops of the evaporator. The minimum cost was determined by a certain value of evaporator pressure drops, which was similar for all the different condenser designs.

On the other hand, the condenser design influences the economic performance in a different manner: it was always desirable to employ condenser designs with minimum heat transfer area (shown by the darker blue colour in Figure 4 (b)). In fact, since the COP was not found to be affected by condenser pressure drops, the operating costs were not influenced by its design. The TCI was instead related to the heat transfer area, thereby suggesting a minimization of the condenser size. A relevant finding shown by Figure 4 is thus given by the fact that the evaporator and condenser geometry optimization, despite aimed at minimizing the cost of the overall system, could be considered as decoupled problems. The designer can choose to optimize the evaporator geometry and to find the design leading to the minimum cost, regardless of the condenser design (and the opposite holds). One additional relevant aspect highlighted by Figure 4 (a) is that all the refrigerants presented a minimum of specific cost of heat for similar values of refrigerant pressure drops. These values lie in the interval 5 – 10 kPa, which is relatively low, compared to usual manufacturer guidelines setting the limitation to around 50 kPa (SWEP International AB, 2015).

The results might therefore suggest that in this particular case study – regardless of the refrigerant chosen – an optimal value of pressure drops could be defined preliminarily to the refrigerant selection and HEX sizing. In this way, it is possible to define a guideline – based on optimal pressure drops – to design the PHE evaporators and condensers prior to the working fluid screening. This would enable the screening to avoid suboptimal solutions or the computational cost of optimizing the HEX design for all the working fluids, some of which the screening will in any case subsequently disregard. This result might seem contradictory if one thinks about the different slopes of COP degradation for the different fluids, previously shown in Figure 2. However, Figure 4 (a) reports an effective steeper increase of the specific cost of heat for butane rather than propane and propylene/butane. This increase started however after reaching the minimum specific cost of heat (after around 30 kPa), thus the different slopes of the working fluids for the COP degradation were not relevant to define the optimum. Moreover, since the optimal values of refrigerant pressure drops was found to be as low as 5 kPa to 10 kPa, the COP degradation (shown in Figure 2) admissible for this case was found to be lower than 2 %.

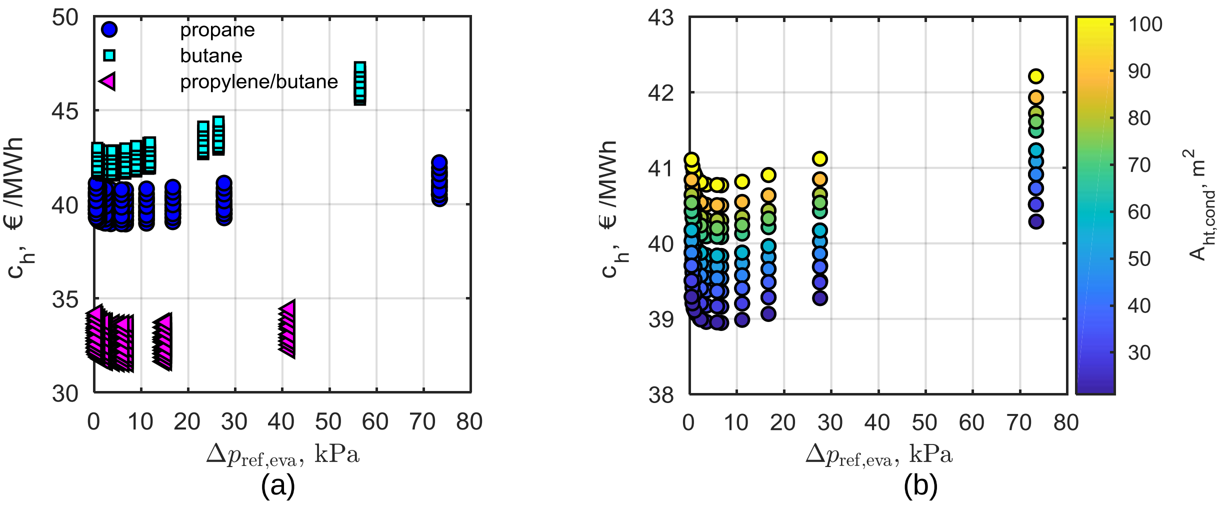

Figure 4: Specific cost of heat as function as a function of total refrigerant pressure drops in the evaporator for: (a) all fluids, (b) propane (close-up)

3.3. Economic Optimum: PHE Designs

Table 4 shows the geometrical specification of the PHE evaporator and condenser designs corresponding to the economic optimum shown in Figure 4. Moreover, heat transfer area, total refrigerant pressure drops and heat pump COP and specific cost of heat are reported. The designs minimizing the specific cost of heat are in agreements with the results presented above. The PHE evaporator design was limited to pressure drops from 4.2 kPa for butane and 7.2 kPa for propylene/butane. The optimal values are thus close for all the working fluids. Similar plate size was found for propane (P) and propylene/butane (P/B), while a plate with a lower aspect ratio (length-to-width ratio) was obtained for butane (B). Due to the lower vapour density, butane is in fact subject to higher pressure drops for the same velocity and channel geometry. The condenser design was instead found to be similar for propane and butane, e.g. high pressure drops over 100 kPa were allowed. With similar channel geometry, the condenser design for propylene/butane led instead to low pressure drops, e.g. 11.5 kPa. This was likely due to the lower dependency of the total heat flow rate at the condenser on the pressure drops, as previously explained in Section 3.1. Note that the condenser design corresponding to the economic optimum is an extreme design case of very high aspect ratio, due to the large bounds for variation of width and channels imposed in the sensitivity study. Such designs might be unrealistic compared to commercial plate size. The economic optimum would in any case correspond to the same evaporator design and more realistic plate size for the condenser could be easily obtained by choosing solutions with higher condenser heat transfer area (e.g. going up in the parallel curves reported in Figure 4).

Table 4: PHE designs corresponding to the economic optimum

	Evaporator design					Condenser design						
	W m	L m	N_{ch} -	A_{ht} m^2	Δp_{ref} kPa	W m	L m	N_{ch} -	A_{ht} m^2	Δp_{ref} kPa	COP -	c_h €/MWh
P	0.250	0.460	150	20.5	6.7	0.200	1.768	50	21.1	113.3	4.4	38.9
B	0.400	0.447	170	36.0	4.2	0.200	1.472	80	28.1	140.2	4.5	41.5
P/B	0.250	0.655	180	35.1	7.1	0.200	1.774	100	42.3	11.5	5.5	31.6

3.4. Design Recommendations

The results presented so far led to the following findings, summarized as design recommendations for evaporator and condenser design in heat pump systems:

- Design criteria such as maximum allowable pressure drops and fixed velocities possibly lead to suboptimal solutions depending on the case study and the working fluid.
- Evaporator and condenser designs can be carried out as decoupled geometry optimization problems, e.g. the design of each component does not affect the other.
- Evaporator design influences the heat pump COP to a large extent, while the condenser design is not found to be relevant for the thermodynamic performance.
- Both components influence the economic performance: the minimum cost is found by defining optimal trade-offs between heat transfer area and pressure drop in the evaporator, while it is always desirable to minimize condenser heat transfer area.
- The economic optimum could be defined by a maximum allowable COP degradation, which theoretically corresponds to different allowable pressure drops for each refrigerant.
- The same optimal evaporator pressure drop was found for different refrigerants for this case study. This is not necessarily a general conclusion, given the very low allowable COP degradation and the dependence of the results from the economic boundary conditions.

4. DISCUSSION

The results presented in this paper were based on numerical modelling of PHE and HP, in both design and performance mode. The PHE models were based on a 1D discretization along the fluid-flow direction, thus neglecting the occurrence of channel-to-channel maldistribution effects. (Mancini et al., 2019) showed that refrigerant pressure drops have a major impact on the heat transfer degradation due to maldistribution effects, causing an additional COP degradation. This implies that the optimal value of refrigerant pressure drops found in this study could be even lower for those refrigerants (like butane) that are particularly sensitive to maldistribution effects. The results of economic analysis are strongly dependent on the boundary conditions assumed for the case study. The heat from the source was considered to come at no cost, and the COP degradation solely implied an increase of the compressor running cost. A COP decrease could also imply a reduction of the heat source utilization, which might partially counteract the cost increase in those cases where the heat source is not free (e.g. district heating applications). Last, the importance of tailoring design criteria depending on the working fluid and application is stressed: designers must be aware that limiting inlet velocities or imposing allowable pressure drops can lead to suboptimal solutions, since the slope of COP degradation has a strong dependence on refrigerant properties. For example, for the economic analysis of propane shown in Figure 4(b), choosing an evaporator design with pressure drops around 50 kPa would entail an increase of almost 1 €/MWh in the specific cost of heat.

5. CONCLUSIONS

This paper presented an approach to derive recommendations for PHE design in HP using pure and mixed refrigerants. A sensitivity study was carried out by simultaneously varying plate size and count for both evaporator and condenser, and PHE geometrical configurations with different trade-offs between heat transfer area and pressure drops were obtained. This study constitutes an attempt to derive design guidelines during a preliminary cycle design phase, where a full component optimization is computationally expensive, and adopting heuristics criteria might lead to suboptimal solutions. The results showed that only evaporator pressure drops affected the COP, and the different refrigerants were subject to different degradation slopes. Butane, with lowest vapour density and higher saturation temperature drop due to pressure drop, resulted in the steepest drop, with 30 kPa generating a 4 % COP drop. Condenser design solely influenced the economic performance, thus a minimization of the size was always preferable. The trade-off between evaporator size and pressure drops was found to be independent from the condenser design, and the analysed case study resulted in optimal COP degradation of maximum 2 % for all the working fluids, thereby defining optimal pressure drop values in a narrow range, around 5 kPa.

ACKNOWLEDGEMENTS

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NOMENCLATURE

A_{ht}	heat transfer area (m^2)	Abbreviations	
c_h	specific cost of heat (€×MWh^{-1})	BC	boundary conditions
$\mathrm{CF_{el}}$	annual cost of electricity (€)	HEX	heat exchanger
COP	coefficient of performance (-)	HP	heat pump
CRF	capital recovery factor (-)	PHE	plate heat exchanger
L	plate length (m)	Greek symbols	
N_{ch}	number of channels (-)	ρ	density (kg×m^{-3})
OH	operating hours (h)	Δ	Difference (-)
p	pressure (bar)	Subscripts	
$\dot{Q}$	heat flow rate (W)	comp	compressor
T	temperature (K)	cond	condenser
TCI	total capital investment (€)	eva	evaporator
W	plate width (m)	V	vapour

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