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Published in:
Energy

Link to article, DOI:
[10.1016/j.energy.2018.05.026](https://doi.org/10.1016/j.energy.2018.05.026)

Publication date:
2018

Document Version
Peer reviewed version

[Link back to DTU Orbit](#)

Citation (APA):
Mancini, R., Zühlsdorf, B., Jensen, J. K., Markussen, W. B., & Elmegaard, B. (2018). Deriving guidelines for the design of plate evaporators in heat pumps using zeotropic mixtures. *Energy*, 156, 492-508.
<https://doi.org/10.1016/j.energy.2018.05.026>

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Accepted Manuscript

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PII: S0360-5442(18)30846-6

DOI: [10.1016/j.energy.2018.05.026](https://doi.org/10.1016/j.energy.2018.05.026)

Reference: EGY 12859

To appear in: *Energy*

Received Date: 1 December 2017

Revised Date: 25 April 2018

Accepted Date: 3 May 2018

Please cite this article as: Mancini R, Zühlsdorf B, Jensen JonaskJæ, Markussen WB, Elmegaard B, Deriving guidelines for the design of plate evaporators in heat pumps using zeotropic mixtures, *Energy* (2018), doi: 10.1016/j.energy.2018.05.026.

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Deriving guidelines for the design of plate evaporators in heat pumps using zeotropic mixtures

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Abstract

This paper presents a derivation of design guidelines for plate heat exchangers used for evaporation of zeotropic mixtures in heat pumps. A mapping of combined heat exchanger and cycle calculations for different combinations of geometrical parameters and working fluids allowed estimating the trade-off between heat transfer area and pressure drops on the thermodynamic and economic performance indicators of the cycle. Compressor running costs constituted the largest cost share, and increased due to a steep decrease of the heat pump coefficient of performance at high refrigerant pressure drops. It was found that the pressure drop limit leading to infeasible designs was dependent on the working fluid, thereby making it impossible to define a guideline based on maximum allowable pressure drops. It was found that economically feasible designs could be obtained by correlating the vapour Reynolds number and the Bond number at the evaporator inlet as $Re_V^{-0.42} Bo^{0.26} \approx 0.040$. The use of the proposed guideline was illustrated for the mixture Propane/Iso-Pentane (0.5/0.5), leading to evaporator designs with net present values deviating maximum -4.4% from the best value found in the mapping. The presented methodology can be applied in different scenarios to develop similar guidelines, thereby decreasing the cost of combined cycle and component optimizations.

Keywords: plate heat exchanger, zeotropic mixture, economic analysis, design guideline, dimensionless numbers, pressure drops, heat transfer area

1. Introduction

Zeotropic mixtures are blends of two or more components, with different mass fractions of the liquid and vapour phases at thermodynamic phase equilibrium. Therefore, the temperature at bubble and dew points differ at any saturation pressure and the mixture undergoes a temperature glide during phase change. The use of zeotropic mixtures as working fluids for thermodynamic cycles offers a possibility of optimizing the cycle efficiency by reducing the thermodynamic irreversibility in the heat exchangers (HEXs). Due to non-isothermal evaporation and condensation, the exergy destruction in the HEXs can be reduced by matching the working fluid temperature glide with the heat source and heat sink temperature profiles.

Zühlsdorf et al. [1–3] demonstrated the advantage of using zeotropic mixtures in heat pumps for different applications. A good glide match between the evaporating fluid and the heat source resulted in a beneficial influence on the cycle thermodynamic performance and better improvements were obtained for larger heat source temperature glides [2]. The improvement of using mixtures in a booster heat pump for a district heating system was estimated equal to up to 30 % compared to pure working fluids. A larger overall improvement up to 40 % was achieved for a reduced degree of required superheat imposed in the case of mixtures [3].

One drawback of using zeotropic mixtures is the degradation of the heat transfer coefficient compared to pure fluids, which was observed during both evaporation and condensation in different experimental campaigns, as reported in [4, 5]. In the case of evaporation, several reasons contribute to the heat transfer degradation: (i) an earlier suppression of the nucleate boiling contribution due to an additional mass diffusion resistance created by the more readily

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evaporation of the more volatile component [6, 7]; (ii) large variation of the refrigerant physical properties during evaporation, due to variable compositions of liquid and vapour phases, which, according to Jung et al. [8, 9], accounts for the 80% of the total heat transfer degradation; (iii) worse transport properties of mixtures compared to pure fluids [5]. A number of studies quantified the heat transfer degradation differently: Ross et al. [6] observed a reduction of up to 50 % compared to pure fluids, Jung et al. [8, 9] reported varying reduction rates between 19 % and 36 % depending on the mixture composition, while Torikoshi and Ebisu [10] calculated a degradation of 20 % and 30 % compared to the heat transfer coefficient estimated by the ideal mixing rule. It is therefore of paramount importance to optimize the design of the heat transfer equipment when zeotropic mixtures are employed, in order to avoid investing in higher heat transfer areas for the heat exchangers.

Plate Heat Exchangers (PHEs) offer a modular and flexible solution for such applications, since it is possible to achieve high heat transfer coefficients within a compact design due to the flow turbulence generated by the characteristic plate corrugation patterns. PHEs are comprised of thin parallel plates stacked together in order to form channels for fluid flow, which can also be arranged in a counter-current manner for achieving a temperature glide match between the mixture and the secondary fluid. Gasketed-type PHEs consist of plates sealed by gaskets and held together by a frame. For higher operating temperature and pressure, the plates can be sealed together by brazing. At the current state-of-the-art, the operating conditions of gasketed-type PHEs are limited to 20.4 bar and 150 °C, whilst brazed heat exchangers can be operated up to 40 bar [11, 12], thereby offering a reasonable range of operation at typical heat pump working conditions.

When designing heat exchangers for a given application, different criteria can be adopted to select the geometrical configuration. The pressure drop of one or both fluids can be limited to a maximum allowable value [11, 12], and the heat transfer area can be minimized for a full utilization of the available pressure loss, as applied in [13]. For single phase HEXs, such pressure drop limitations could also be translated into maximum gas and liquid phase velocities at the inlet, and typical design values can be found in literature for a number of heat exchanger configurations [12]. These values are often based on heuristics from manufacturers and the extension to other types of applications (e.g. zeotropic mixtures and/or phase change) is not trivial.

Following other design approaches, the heat exchanger can be optimized by carrying out a cost minimization problem without a maximum pressure drop limitation, and evaluating the trade-off between heat transfer area and pressure drops. Different previous studies have approached the problem by considering solely the cost related to the heat exchanger, namely the investment cost and the pumping and compression costs related to the two streams, for a general heat exchanger configuration [14, 15], for shell and tube heat exchangers [16] and for plate heat exchangers [17, 18]. However, the economic analysis lacked assessment of the impact of the heat exchanger pressure drops on the other components, as well as on the overall cycle thermodynamic performance.

In literature a number of studies can be found on simultaneous optimization of plate heat exchangers used as evaporators and/or condensers and thermodynamic cycle design, mostly focusing on low temperature applications and pure fluids. Some of the works are related to the assessment of the impact of some specific cycle parameters on the PHE design [19, 20], whilst other studies performed combined cycle-PHE optimization procedures with the aim of maximizing the cycle efficiency [21], and by including also an economic analysis [22, 23]. The pressure drops were mostly considered as pumping cost on the heat source/sink side, and none of the studies assessed the impact of the working fluid pressure drops on the outlet condition of the evaporator. Moreover, a complete and combined component-cycle optimization comes at a demanding computational cost, especially during the preliminary design phase, where many different working fluids are usually compared and ranked.

The study presented in this paper addresses the following aspects: (i) It presents a methodology for deriving design guidelines for plate heat exchangers integrated in a thermodynamic cycle, namely a heat pump. (ii) The methodology is based on assessing the impact of both plate heat exchanger size and pressure drops on the thermodynamic and economic performance of the heat pump; the pressure losses are not only included as pumping cost of the heat source side, but also imply a modification of the thermodynamic state points of the cycle at the evaporator outlet, which accordingly affects the heat pump design, investment and operating cost. (iii) It utilizes the aforementioned methodology to derive design guidelines for PHE evaporator design in heat pumps using zeotropic mixtures as working fluids. The obtained results are intended for employment in practical engineering during the process of component selection for similar applications, hence avoiding the cost of combined cycle and component analysis.

The methodology is based on complementing a vapour compression heat pump sizing model together with a detailed numerical model of the evaporator, accounting for the variation of the heat transfer coefficient and fluid prop-

erties during the evaporation process and estimating the impact of heat transfer area and pressure drops on the cycle thermodynamic and economic performance indicators. The methodology was applied to the case of evaporator design for a heat pump, and eight different working fluids were selected based on a previous study [1], which demonstrated the thermodynamic and economic feasibility of using zeotropic mixtures in heat pumps for waste heat recovery in a spray drying facility.

2. Methods

The methodology adopted in the present study is based on a parametric analysis on the main design parameters of a plate heat exchanger to assess the impact of the different design configurations on the thermodynamic and economic performance indicators of a thermodynamic cycle, namely a heat pump. Fig.1 shows the schematic of the work flow of the methodology. Two different models were built and integrated in the Matlab environment [24], i.e. a cycle simulation model for a heat pump, explained in details in Section 2.3, and a detailed PHE model, presented in Section 2.4. After the working fluid selection process, explained in Section 2.2, the preliminary sizing of the heat pump was done and the design parameters were calculated, i.e. desired heat exchanger capacity, mass flow rates, pressures and temperatures. The values were subsequently sent to the plate heat exchanger model, which additionally received as inputs the geometrical parameters from which the required heat transfer area and resulting pressure losses were estimated. The outputs were returned to the heat pump model, where the sizing of the cycle was re-evaluated. In this second iteration, the sizing process took into account the resulting heat exchanger size and pressure drops for the economic calculation, as it is briefly described in Section 2.6. The process was repeated for all the combinations of geometrical parameters chosen for the parametric analysis, which is introduced in Section 2.1. Moreover, the same process was repeated for all the eight working fluids considered in the case study, by considering the same combinations of PHE design parameters and calculating the Coefficient of Performance (COP) and Net Present Value (NPV). As shown in Fig.1, all the data points were collected and used as basis for deriving a general design guideline, valid for all the working fluids and the boundary conditions of the present case study. The aim was to correlate the point with optimal economic performances to the PHE design parameters. In order to generalize the results, non-dimensional parameters were employed as explained in Section 2.8.

Figure 1: Work flow of the overall methodology

2.1. Parametric analysis and PHE geometry

The parametric analysis was carried out by varying the main design variables of a PHE. Fig.2 shows the main geometrical parameters, namely plate size, number and corrugation geometry. The plate size is given by the width W and the length L ; L_{HT} defines the effective length for heat transfer and the total heat transfer area depends on the number of channels N_{ch} employed. The corrugation characteristics are determined by the corrugation pitch Λ , the corrugation height b and the chevron angle β . The corrugation thickness t is a trade-off between mechanical resistance to stresses and conductive thermal resistance. The corrugation parameters determine the hydraulic diameter of the channels, thereby defining the flow conditions of the working fluid. The hydraulic diameter was estimated by using Eq.(1) [11], where Φ is the enlargement factor.

$$D_h = \frac{2b}{\Phi} \quad (1)$$

The enlargement factor represents the ratio between the actual heat transfer area and the projected area of the plate (without corrugation), and it is expressed by Eq.(2) [11] as function of corrugation height and pitch.

$$\Phi = \frac{1}{6} \left(1 + \sqrt{1 + \left(\frac{\pi b}{\Lambda} \right)^2} + \sqrt{1 + \frac{1}{2} \left(\frac{\pi b}{\Lambda} \right)^2} \right) \quad (2)$$

The port diameter D_p determines the inlet/outlet velocities of the refrigerant and the heat source, mainly affecting the port pressure losses; the PHE can be manufactured with different values of the diameter for the two working fluids,

depending on the desired velocity and the phase.

Figure 2: Schematic view of a chevron type PHE [25]

In the parametric study, the design variables were varied among the values reported in Table 1, investigating all the possible combinations between them, for a total of 1440 different PHE configurations. The thickness was fixed to a value commonly found in literature [11]. The port diameter was considered as fixed depending on the magnitude of the plate width. The plate length was calculated as output of the plate heat exchanger design model in order to match the evaporator capacity of the case study for all the combinations of PHE design variables. The plates were considered to be manufactured in stainless steel, with thermal conductivity equal to 16.2 W/(m K). The free flow area and the heat transfer area were calculated as function of the design parameters, as reported in Eq.(3) and (4) [11], respectively.

$$A_0 = bN_{ch}W \quad (3)$$

$$A_{ht} = 2(W \cdot L_{HT} + b \cdot L_{HT}) \cdot N_{ch} \Phi \quad (4)$$

The effective length L_{ht} was employed for heat transfer calculations, while the port-to-port length was used in order to calculate the frictional pressure losses. The relation between port-to-port length and effective length is given by Eq.(5) [11].

$$L_p = L_{HT} + \frac{D_{p-in}}{2} + \frac{D_{p-out}}{2} \quad (5)$$

In order to avoid unrealistic results and to minimize maldistribution effects along the plate width, solutions with length to width ratio lower than 2 were considered infeasible and excluded from further analysis.

Table 1: Geometrical parameters of the PHE varied in the parametric study

Parameter	Value	Unit
W	0.15, 0.25, 0.35, 0.45, 0.55	m
N_{ch}	25, 50, 75, 100, 150, 200	-
b	2, 4, 6, 8	mm
Λ	2, 4, 6, 8	mm
β	30, 45, 60	°
t	0.5	mm
D_p	0.03, 0.06, 0.1	mm

2.2. Case study and working fluid selection

The framework of the analysis was given by a case study [1] assessing the integration of high temperature heat pumps in a spray drying facility. Waste heat was recovered by integrating a heat pump, with the aim of pre-heating air up to 120 °C. Different zeotropic mixtures were compared in terms of COP and NPV for a single-stage configuration

of a vapor compression heat pump, in which the working fluid was varied based on binary mixtures formed by combinations of a number of natural refrigerants. The refrigerant screening included hydrocarbons, Dimethyl Ether (DME), Diethyl ether (DEE) and carbon dioxide CO₂, chosen for their low Global Warming Potential (GWP) and Ozone Depletion Potential (ODP), and being miscible between each other for wide ranges of temperature and pressure without leading to any chemical reactions [1, 2]. Table 2 summarizes the best performing mixtures, with considered mass composition and all the evaporation pressures were found to be well below the maximum operating pressure for PHEs. The table also shows the preliminary COP, calculated without accounting for pressure losses of the evaporator.

Table 2: Summary of the considered working fluids

Working fluid	p_{ev} , bar	COP, -
Propane/Iso-Pentane (0.5/0.5)	4.9	3.08
Propane/n-Pentane (0.8/0.2)	8.4	3.04
Propane/n-Pentane (0.4/0.6)	3.0	3.02
Butane/Hexane (0.9/0.1)	2.5	3.07
DME/n-Pentane (0.4/0.6)	2.6	3.26
DME/n-Pentane (0.7/0.3)	5.0	3.24
DME/Iso-Pentane (0.5/0.5)	4.0	3.15
Propylene/Iso-Pentane (0.4/0.6)	3.9	3.14

2.3. Heat pump cycle model

The thermodynamic cycle was modelled in steady state and it consists of a single-stage configuration of a vapour compression heat pump. Fig.3 shows the sketch of the unit, with the different components integrated in the cycle, namely compressor, condenser, throttling valve and evaporator.

Figure 3: Schematic of the heat pump model

Table 3 shows the design parameters of the cycle. The heat source side was completely defined by the boundary conditions, while the outlet temperature of the condenser was set as a free variable. The evaporation and condensation pressures of the working fluid were defined by the minimum required pinch point temperature difference between the working fluid and the fluid at the secondary side. The amount of subcooling was defined by the pinch point temperature difference and the sink inlet temperature in order to obtain the maximum efficiency. A minimum superheating of 5 K was included in the evaporator, in order to ensure a dry compression for all the fluids. The compressor was modelled by assuming a constant isentropic efficiency, while the motor efficiency accounted for the power generation losses

[26]. The thermodynamic performance was evaluated by estimating the COP, defined in Eq.(6), as the ratio between the thermal energy provided to the heat sink and the compressor power.

$$\text{COP} = \frac{\dot{Q}_{\text{sink}}}{\dot{W}_{\text{comp}}} \quad (6)$$

The working fluid properties were calculated by Refprop [27], using recommended standard equation of states and mixing parameters, while heat sink and source properties were computed using CoolProp [28].

Table 3: Boundary conditions for the heat pump

Parameter	Value	Unit
Heat source		
Medium	Water	
T_{in}	65	°C
T_{out}	40	°C
$\dot{m}$	14.8	kg/s
$\dot{Q}$	1544	W
Heat sink		
Medium	Water	
T_{in}	75	°C
$\dot{m}$	10.6	kg/s
Compressor		
η_{is}	0.8	-
η_{motor}	0.95	-
ΔT_{sh}	5	K
Heat exchangers		
ΔT_{pinch}	10	°C

2.4. PHE design model

The PHE design model was based on a one-dimensional discretization of the heat exchanger along the flow direction, at constant enthalpy steps, hence with constant heat flow rate for each control volume (CV). The heat exchanger solver was based on a successive substitution approach, with heat transfer area and pressure drops set as unknown. A total number of $n = 50$ control volumes was chosen, as trade-off between accuracy and computational cost of the design model. The internal solver iterated on the length of each CV, as well as on the pressure drops of both refrigerant and heat source, with a tolerance set on the relative residuals equal to 10^{-2} .

The PHE model was solved by imposing steady-state mass, momentum and energy conservation equations, which were solved for each CV and both fluids. The logarithmic mean temperature difference method was applied locally, for the computation of the UA value in each CV. Counter-current flow of the refrigerant and heat source was imposed, longitudinal conduction through the walls and heat loss to the external environment were neglected. Thermodynamic state variables and fluid properties were computed for each CV based on a first order linear interpolation of the values at the nodes. The local heat transfer coefficients and pressure drops were estimated using experimental correlations. The choice of appropriate prediction methods for the considered case study is discussed in Section 2.5.

2.5. Choice of prediction methods

The local heat transfer coefficient and frictional pressure drops were computed for both fluids using experimental correlations, thereby conveying focus on the choice of suitable prediction methods for the working fluids and the boundary conditions of the case study. Literature presents several correlations developed for the estimation of the heat transfer coefficient for refrigerant flow boiling in PHEs. There is however a lack of a suitable prediction method for the evaporation heat transfer coefficient in PHEs for zeotropic mixtures of natural refrigerants. Amalfi et al. [29] developed a flow boiling correlation based on an extensive database collected from several studies in literature, claiming to have a better agreement than other existing correlations. The only zeotropic mixture included in the database was the near-azeotropic mixture R410a, which presents a small temperature glide during evaporation. Such

correlation is therefore not directly applicable to the estimation of the heat transfer coefficient of mixtures. Moreover, it does not take the mixture degradation of heat transfer into account.

Mixture degradation of heat transfer was estimated by a number of experimental studies focusing on in-tube flow boiling of different zeotropic mixtures, including ammonia-water. An extensive literature review can be found in [30]. The developed correlations were however derived for tubular geometry, with a different flow mechanism compared to PHEs. In order for a correlation to be fully applicable, the effect of the geometry must be included as well. It was therefore decided to apply a theoretical method, which was first developed for mixture condensation by Silver [31] and Bell-Ghaly [32]. Sardesai et al. [33] extended the theoretical derivation to convective boiling heat transfer, deriving the formulation expressed by Eq.(7) [33].

$$h_{TP} = \frac{1 + h_{NB-mix}/h_C}{1/h_C + \bar{z}/h_V} \quad (7)$$

Here, h_{NB-mix} is the nucleate boiling contribution of the mixture, h_C is the convective two-phase contribution and h_V is the single-phase vapour heat transfer coefficient estimated for the vapour flowing alone in the channel. $\bar{z}$ is a correction term taking into account the ratio of sensible over latent heat transfer, and it is evaluated by Eq.(8) [32].

$$\bar{z} = x \cdot \frac{dT}{dh} \cdot c_{p,V} \quad (8)$$

In the estimation of h_{NB-mix} , the mixture effects on degradation of nucleate boiling and loss of effective wall superheat is accounted for by applying a suitable correction factor. Thome and Shakir [34] approach was used for this purpose, expressing the relation between ideal and mixture heat transfer coefficient as [34] :

$$\frac{h_{NB-mix}}{h_{NB-id}} = \left\{ 1 + \frac{h_{NB-id}}{\dot{Q}''} (T_{dew} - T_{bubble}) \left[1 - \exp \left(\frac{\dot{Q}''}{\rho_L h_{lat} \beta_L} \right) \right] \right\}^{-1} \quad (9)$$

By using the theoretical method proposed by Sardesai [33], it was possible to choose prediction methods specifically developed for chevron-type PHEs for the estimation of the different terms h_{NB-id} , h_C and h_V . The method by Amalfi et al.[29] was used to estimate the contribution of convective boiling, while Cooper [35] was employed for the nucleate boiling term. Martin [36] correlation was employed for the single-phase heat transfer coefficient of the vapour flowing alone. The same correlation was used for the refrigerant heat transfer coefficient in the superheated region, as well as for the water single-phase heat transfer coefficient along the whole heat exchanger.

Pressure drops were computed by considering all the contributions to the steady state momentum equation, i.e. friction, gravity(static), acceleration and inlet/outlet ports terms:

$$\Delta p_{tot} = \Delta p_{fr} + \Delta p_{gr} + \Delta p_{acc} + \Delta p_{ports} \quad (10)$$

The pressure drop over the total length of the HEX was calculated by considering each control volume separately. In the two-phase region, the acceleration and static contributions were calculated by the homogeneous model, thereby using Eq.(11) and Eq.(12) for each CV, respectively [37].

$$\Delta p_{acc} = G^2 \left[\left(\frac{x}{\rho_V} + \frac{1-x}{\rho_L} \right)_{OUT} - \left(\frac{x}{\rho_V} + \frac{1-x}{\rho_L} \right)_{IN} \right] \quad (11)$$

$$\Delta p_{gr} = \rho_m g \Delta L \quad (12)$$

The two-phase mean density was determined by:

$$\rho_m = \left[\frac{x}{\rho_V} + \frac{1-x}{\rho_L} \right]^{-1} \quad (13)$$

Eq.(13) was also used for the computation of the frictional pressure drop in the two-phase region. The frictional contribution was calculated by means of the two-phase Fanning friction factor, as expressed by Eq.(14) [37].

$$\Delta p_{fr} = 2 f_{TP} \frac{\Delta L G^2}{D_h \rho_m} \quad (14)$$

The friction factor was computed by using the correlation by Amalfi et al. [29]. In the single-phase region, the friction factor correlation by Martin [36] was used for both the refrigerant superheating and the water flow. Last, the port pressure drops were computed using the Shah and Focke [38] correlation, as expressed by Eq.(15), where G_p

indicates the mass flux at the ports, depending on the port diameter.

$$\Delta p_{\text{ports}} = 0.75 \left[\left(\frac{G_p^2}{2\rho} \right)_{\text{IN}} + \left(\frac{G_p^2}{2\rho} \right)_{\text{OUT}} \right] \quad (15)$$

For the refrigerant flow, port pressure losses were considered only at the outlet control volume, since the inlet loss was included in the refrigerant expansion at the throttling valve. Both inlet and outlet pressure drops due to ports were evaluated for the heat source, thereby contributing to the pumping power.

2.6. Economic model

After the sizing of the PHE, the heat pump model was re-evaluated by taking the updated temperature and pressure at the evaporator outlet into account, as shown in Fig.1. Therefore, the COP was slightly affected by the change in compressor power resulting from the mixture pressure drop. Different aspects of the HEX design influenced the value of the NPV of the heat pump. The NPV was calculated by considering different cost and revenue streams, by using Eq.(16) [1]. The considered costs were the Total Capital Investment (TCI), accounting for the Capital Investment (CI) of each system component, the Operation and Maintenance Cost (OMC), the electricity cost (FC_{hp}) due to the compressor running and the required water pumping cost (FC_w). The revenue stream was considered as the natural gas saving (FC_{ng}), which otherwise would be necessary to produce the thermal energy output of the heat pump.

$$\text{NPV} = -\text{TCI} - \text{OMC} - \frac{FC_{\text{hp}}}{\text{CRF}} - \frac{FC_w}{\text{CRF}} + \frac{FC_{\text{ng}}}{\text{CRF}} \quad (16)$$

The interest and inflation rates were assumed equal to 7 % and 2 %, respectively, and then used for the estimation of the Capital Recovery Factor (CRF), with a plant economic lifetime of 20 years [26] [39]. Operation and Maintenance Costs were assumed to be the 20 % of the investment cost as one time cost at the time of the investment [39]. The saving of natural gas was calculated by estimating the useful thermal energy produced by the heat pump, equal to the heat sink capacity and by considering a boiler efficiency η_{boiler} equal to 0.9 [26]. The price of natural gas was considered as 0.0303 €/kWh [40].

The TCI accounted for the investment of condenser, compressor and evaporator. The sizing of the condenser was carried out by considering constant heat transfer coefficients and by using the logarithmic mean temperature difference methods for the three different sections of desuperheating, condensation and subcooling. The TCI of each individual component was calculated from the Purchased Equipment Cost (PEC), scaled according to the component size with scaling factors and reference values of PECs reported by Ommen et al. [26]. The TCI was increased by a factor 4.16 compared to the PEC, to account for the investment of the expansion of an existing facility [39].

The running cost of the compressor, given by FC_{hp} was adapted to the updated compressor power by considering an estimation of 7400 hr/yr [1] as annual operating time τ_h of the heat pump and an electricity cost c_{el} of 0.0783 €/kWh [40]. The additional fuel cost term due to the water pressure drops, was calculated by using Eq.(17) as a function of the water mass flow rate, water density, total water pressure drops, and a pump efficiency η_{pump} equal to 0.95.

$$FC_w = \frac{\dot{m}_s}{\rho_s} \frac{\Delta p_{s-\text{tot}}}{\eta_{\text{pump}}} c_{\text{el}} \tau_h \quad (17)$$

Fig.4 shows the log(p)-h diagram of the heat pump with and without pressure drops in the evaporator. It can be noticed how the outlet of the evaporator is affected by the pressure losses, thus changing the compressor power required as well as the investment cost, which is based on the suction line. The pressure drop also influences the evaporator inlet location in the diagram, but to a lower extent compared to the outlet. In order to compare the results for different working fluids, performing with different COP and maximum NPV, it was decided to normalize each NPV for the maximum value calculated for the specific fluid. The value of NPV of a certain design point of the HEX i for a given working fluid wf was therefore normalized as $\text{NPV}_{i-\text{wf}}^*$, defined by Eq.(18).

$$\text{NPV}_{i-\text{wf}}^* = \frac{\text{NPV}_{i-\text{wf}}}{\max(\text{NPV}_{\text{wf}})} \quad (18)$$

Figure 4: Example of $\log(p)$ - h diagram without pressure drops (blue) and with 50 kPa pressure losses in the evaporator (sky-blue)

2.7. Data analysis

The different HEX configurations and working fluids were compared based on dimensionless numbers, in order to derive a guideline describing the design points corresponding to maximum NPV. Non-dimensional parameters were chosen in order to describe the impact of the fluid properties, of the boundary conditions imposed by the cycle (refrigerant evaporation pressure, mass flow and inlet quality) and of the PHE geometry. The main dimensionless numbers governing the heat transfer and pressure drop correlations for the two-phase flow of the mixture were used (see Section 2.4), and they are reported in Table 4.

Table 4: Non-dimensional parameters used for the analysis of the results

Symbol	Name	Formula
θ^*	Dimensionless inclination angle	$\frac{90 - \beta}{70}$
Bd	Bond number	$\frac{gD_h^2(\rho_L - \rho_V)}{\sigma}$
We _m	Weber number	$\frac{G^2 D_h}{\sigma \rho_m}$
ρ^*	Liquid-to-vapour density ratio	$\frac{\rho_L}{\rho_V}$
Re _{LO}	Liquid only Reynolds number	$\frac{GD_h}{\mu_L}$
Re _V	Vapour alone Reynolds number	$\frac{Gx D_h}{\mu_V}$

The numbers were used to correlate the normalized net present value to the different heat exchanger designs. The parameters were evaluated at the refrigerant conditions at the evaporator inlet, since the scope was to derive guidelines applicable before the actual component design. The inlet conditions are fixed by the boundary conditions

of the heat pump, thereby being known for all the fluids a priori. It was therefore decided to exclude the Boiling number from the data analysis, since it requires information on the heat flux distribution along the heat exchanger and a preliminary estimation of the heat transfer area is needed. The Weber number We_m is defined by evaluating the equivalent two-phase density, expressed by Eq. 13.

The correlation between the dimensionless parameters and the normalized NPV was obtained by adopting a power law of the form expressed by Eq.(19).

$$NPV^* = a \cdot \theta^{*b} \cdot Bd^c \cdot We_m^d \cdot Re_V^e \cdot Re_{LO}^f \cdot \rho^{*g} \quad (19)$$

The normalized NPV was first correlated as function of all the dimensionless parameters considered in Table 4. All the design points with positive NPV were considered in the fitting, and the results for all the working fluids were considered simultaneously in one fitting. After deriving the coefficient for Eq. (19), the most relevant parameters were identified. This was done by evaluating the obtained exponent and by comparing different combinations of non-dimensional numbers. The correlation that best fitted the NPV trend for all the fluids, as well as leading to a similar optimal range for the different mixtures, was chosen and presented as design guideline.

2.8. Uncertainty analysis

For each correlation chosen for the PHE design model, for both heat transfer and pressure drop, a certain accuracy was reported by the authors. This measure represents an estimation of how well the prediction method fits the experimental database from which the correlation was derived. Therefore the accuracy does not necessarily indicate the uncertainty of the prediction methods when they are applied to different working fluids, geometry and operating conditions. It is however relevant to evaluate the change of the obtained results when a certain deviation is applied to the calculation of heat transfer coefficient and pressure drops. For this purpose the accuracy bounds of each correlations were considered as a source of uncertainty in an uncertainty analysis, with the aim of evaluating to which extent the results would change for similar variations in the inputs. The bounds are reported in Table 5, with the exception of h_V . Since Martin [36] expresses h_V as a function of f_V , the uncertainty of h_V was directly included by considering the accuracy bound for the vapour friction factor f_V .

Table 5: Input uncertainties for the uncertainty analysis

Parameter	Correlation	Bounds	Reference
h_C	Amalfi	$\pm 22.1 \%$	Mean absolute error [29]
h_{NB-id}	Cooper	$\pm 59.0 \%$	Mean absolute error, reported in [37]
$\frac{h_{NB-mix}}{h_{NB-id}}$	Thome and Shakir	$\pm 11.1 \%$	Mean absolute error, reported in [41]
f_{TP}	Amalfi	$\pm 21.5 \%$	Mean absolute error [29]
f_V	Martin	- 50.0 % + 100 %	Accuracy bounds, reported in [11]

The Monte Carlo (MC) method [42] was applied in order to carry out the uncertainty analysis, with the aim of estimating the probability density of the model outputs, namely A_{ht} , $\Delta p_{r,tot}$, $\Delta p_{s,tot}$ and NPV. The inputs were assumed to be uniformly distributed between the uncertainty bounds reported in Table 5. A latin hypercube sampling (LHS) technique was adopted to create 500 different samples in the input space, proved to be more reliable compared to random sampling [43]. The MC simulations were performed following the approach by Sin and Gernaey [44], and the results were obtained as mean values, standard deviations and 95 % coverage intervals. The analysis was applied to Propane/Iso-Pentane (0.5/0.5), by fixing the PHE geometry to one of the optimal designs found by applying the derived design guideline.

3. Results

In this section, the main results are presented. Sections 3.1 and 3.2 present the trade-off between heat transfer area and pressure drops and the impact on the different revenue and cost streams. Sections 3.3 and 3.4 report the fitting with the non-dimensional parameters and the derivation of the final design guideline.

3.1. Heat transfer area and pressure drops

Fig.5 shows the NPV as function of the heat transfer area (a) and total refrigerant pressure drops (b) for all the fluids considered in the analysis. It is shown that both heat transfer area and pressure drops have an impact on the economic performance of the heat pump. However, the NPV decreases to a lower extent for higher values of A_{ht} compared to an increase of the refrigerant pressure drops. Fig 5(b) shows that for all the eight cases there is a trade-off which coincides with minimizing the refrigerant pressure drops to a very low value. Moreover, Fig 5(b) shows that the pressure drops impact is different depending on the fluid: the mixtures Butane/Hexane (0.9/0.1) and DME/n-Pentane (0.4/0.6) are the most sensitive to pressure drops, performing with negative NPVs at 60 kPa and 120 kPa, respectively. On the contrary, Propane/n-Pentane (0.8/0.2) shows a weaker dependence, with NPV values always positive in the considered pressure drop range.

Figure 5: Effect of heat transfer area of the evaporator (a) and refrigerant total pressure losses (b) on the non-dimensional NPV for all the working fluids

3.2. Cost breakdown

Fig.6 shows the breakdown of the NPV as calculated by Eq.16. The abscissa reports the refrigerant total pressure drop, with trends similar for all the working fluids. The blue line indicates the revenue stream, which contributed positively to the NPV and was determined by the natural gas saving. The black line represents the sum of all the expenses, which impacted negatively the NPV. The breakdown of the different terms of the total cost is indicated by the filling colors of the cost line.

The dominant contribution was given by the compressor running cost (grey area), which was negatively affected by the refrigerant pressure drop, causing an increase of the required compressor work. This is explained by Fig.7, where the COP of the heat pump is plotted against the total refrigerant pressure drop. For a fixed design thermal load

of the heat pump, a lower COP entails a higher required compressor work, as shown by Eq.(6). A sharp decrease is observed for all the fluids, with different slopes, similarly to the trends found in Fig.5 (b). The different trends for the different fluids found in the cost breakdown of Fig.6 can therefore be explained by a different sensitivity of the COP to the pressure drop. The mixture Propane/n-Pentane (0.8/0.2) was found to be least affected by pressure drop, while Butane/Hexane (0.9/0.1) presented the steepest slope.

In the cost breakdown of Fig.6, the TCI is the second highest contribution, indicated by the light-blue area. The TCI undergoes a slight increase for low values of refrigerant pressure drop, and then increases more sharply for higher pressure drop. Since operation and maintenance was considered as a fixed share of the capital investment, OMC undergoes the same trend. An explanation for the TCI behaviour can be found in Fig.8(a) and (b), showing the capital investment of the evaporator, condenser and compressor as functions of the total refrigerant pressure drop. All the working fluids are reported in the same plot with different marker colors. The evaporator and condenser investment costs, directly related to the heat transfer area, are one order of magnitude lower than the compressor investment if the pressure drops are not minimized. When the evaporator investment increases at lower pressure drops (entailing a higher investment due to the larger heat transfer area), the CI of the HEX reaches the same order of magnitude of the lower investment entailed by the compressor. The change in HEX CI due to varying heat transfer area is similar for all the fluids, with an overlap of the different curves.

On the other hand, the compressor CI in Fig.8 (b) undergoes a steep increase for higher values of pressure drop and presents different trends depending on the mixture. The sharp increase is due to the change in the compressor suction line, which is affected by the evaporator outlet. In fact, for higher refrigerant pressure drops, a lower refrigerant outlet density is obtained, entailing a higher volume flow rate, which is directly proportional to the compressor size and cost. The different slopes for the different fluids are related to a different sensitivity of the refrigerant properties to the change in evaporator outlet pressure. The trends of the fluids follow the results found for COP in Fig.7, with Propane/n-Pentane (0.8/0.2) showing the weakest dependence on the pressure drops and opposite trends for mixtures Propane/n-Pentane (0.4/0.6) and Butane/Hexane (0.9/0.1), where the pressure drops have a major influence on both the compressor running cost and TCI.

One very relevant aspect to highlight is the intersection point between revenue and costs streams, in Fig.6, representing the pressure drop limit above which the design of the PHE leads to infeasible solutions, i.e. with negative NPV. This limit value is not the same for all the working fluids, supporting the thesis that it is not necessarily optimal to design HEXs by imposing a maximum allowable pressure drop regardless of the working fluid or boundary conditions.

The water pumping cost was found to be negligible compared to the all the other contributions. No clear relation is therefore expected between heat source pressure drops and NPV.

Last, it must be noted that such analysis highlights considerations, which might be useful when choosing an optimal working fluid for a specific case study. In fact, by looking at Fig.5 (a) and Fig.7, the mixture DME/-Pentane (0.7/0.3) outperforms the other fluids both in terms of maximum NPV and COP found in the mapping, thereby suggesting the choice of such fluid as preferred option. However, the mixture Propane/n-Pentane (0.8/0.2) showed the weakest dependence on refrigerant pressure drops, thus offering an additional flexibility during the evaporator design process. It would therefore be up to the designer to evaluate and prioritize the different aspects.

Figure 6: Breakdown of the contributions to the NPV, divided in cost and revenue. The colours indicate the different shares of the terms contributing to the cost stream

- Pro-iPe(0.5/0.5) • Pro-nPe(0.4/0.6) • DME-nPe(0.4/0.6) • DME-iPe(0.5/0.5)
- Pro-nPe(0.8/0.2) • But-Hex(0.9/0.1) • DME-nPe(0.7/0.3) • Propy-iPe(0.4/0.6)

Figure 7: COP as function of the total refrigerant pressure drops for all the working fluids

- Pro-iPe(0.5/0.5) • Pro-nPe(0.4/0.6) • DME-nPe(0.4/0.6) • DME-iPe(0.5/0.5)
- Pro-nPe(0.8/0.2) • But-Hex(0.9/0.1) • DME-nPe(0.7/0.3) • Propy-iPe(0.4/0.6)

Figure 8: Capital investment of the heat exchangers (a) and compressor (b), as function of the total refrigerant pressure drops for all the working fluids

3.3. Correlation between NPV and non-dimensional parameters

By exponentially correlating the design points by means of Eq.(19) to the chosen non-dimensional parameters, the NPV* is found to be proportional to the combinations of dimensionless numbers reported by Eq.(20), indicated by K .

$$K = \theta^{*0.045} \text{Bd}^{0.051} \text{We}_m^{-0.011} \rho^*{}^{-0.066} \text{Re}_{LO}^{0.009} \text{Re}_V^{-0.12} \quad (20)$$

Fig.9 reports the normalized net present value as function of K , for Propane/Iso-Pentane (0.5/0.5). The different colors represent the chevron angle of the PHE, namely 30° , 45° and 60° corresponding to a value of non-dimensional inclination angle of 0.86, 0.64 and 0.43, respectively. The trends for the other working fluids are not shown in the paper, but they are similar. The obtained design points are well correlated by K , and it is recommended to employ HEXs with $K = [0.4 - 0.6]$ at the evaporator inlet, in order to obtain solutions with NPV in the best 20 %. By increasing the value of K , the economic performance is negatively affected by the increase in heat transfer area, while for $K < 0.4$ an increase of refrigerant pressure losses entails a steep decrease of NPV.

The use of a design guideline based on K , which depends on six different parameters, might however lead to have redundant information on the boundary conditions and the fluids in the parameters. The problem was therefore simplified by relating the NPV* to a selected number of non-dimensional parameters only. In order to do so, the results were assessed based on the coefficients and on additional considerations.

First, the dependence of the NPV on the liquid only Reynolds number Re_{LO} is weak, since its exponent in Eq.(20) is two orders of magnitude lower than the vapour alone Reynolds number and one order of magnitude lower compared to the other parameters, thereby suggesting that a good fitting could be obtained by neglecting the influence of this non-dimensional number. Furthermore, it was decided to exclude the liquid to vapour density ratio ρ^* , since it contains information solely depending on the inlet densities and other dimensionless number, as the Reynolds numbers, the Weber number and the Bond number already contain the density characteristics of both vapour and liquid phases.

Lastly, by looking at the impact of the dimensionless inclination angle θ^* in Fig.9, the plot suggests that it is always optimal to minimize the chevron angle (hence enhancing the degree of turbulence of the fluid flow in the channels) if the PHE design is carried out in the optimal region or else for high value of K , i.e. designs with higher heat transfer area. On the contrary, with refrigerant dominant pressure losses ($K < 0.4$), it is slightly better to employ higher chevron angles for reaching less turbulent flow, thereby decreasing the pressure losses. It is therefore unnecessary to include the chevron angle into the design guideline, since it is possible to tune the other PHE design parameters in order to be in the optimal design region, where it is recommended to employ low values of β .

Figure 9: Normalized NPV as function of all the dimensionless parameters correlated exponentially for Propane/Iso-Pentane (0.5/0.5). The different colors represent the different chevron angles adopted

3.4. Correlation between NPV and selected dimensionless numbers

The results are presented as function of the three remaining dimensionless numbers, namely the Bond number Bd , the Weber number We_m and the Reynolds number of the vapour flowing alone Re_V . Table 6 shows the coefficients obtained by fitting the normalized net present values with different combinations of the three dimensionless parameters.

Four different cases were assessed by combining the three parameters all at once and by considering combinations of only two of them. The aim was to find which correlation attained the best representation of NPV*, thereby being appropriate for deriving the design guideline. The values reported in the first row of the table (Case I), obtained by considering all the three non-dimensional numbers, suggest that there is not a predominant contribution of one of them, since the coefficients lie in a similar range.

Table 6: Coefficients obtained by fitting the NPV* to different combinations of dimensionless parameters

Case	Parameters combination	a	b	c
Case I	$Re_V^a \cdot Bd^b \cdot We_m^c$	1.21	-0.19	-0.87
Case II	$Bd^b \cdot We_m^c$	0.14	-0.25	
Case III	$Re_V^a \cdot We_m^c$	0.74	-	-0.62
Case IV	$Re_V^a \cdot Bd^b$	-0.42	0.26	-

Fig. 10 shows the NPV* as a function of the different exponential combinations of dimensionless numbers, with the ordinate axis reporting the design points of all the working fluids. It can be observed that the normalized net present value presents a clear trend for all of the combinations. This is possibly related to the redundancy of information on geometry, fluid properties and boundary conditions contained in the three parameters.

By looking at the formulas reported in Table 4, We_m and Re_V are dependent on the hydraulic diameter D_h and on the free flow area A_0 , while the Bond number Bd solely depends on D_h . Any combinations of the parameters therefore contain all the information regarding the PHE geometry, i.e. the corrugation geometry, plate width and number of channel, with the exception of plate length (estimated by the PHE sizing for matching the thermal load and thus not needed to be included in the guideline) and chevron angle, excluded from the guideline.

Moreover, by looking at the fluid properties of the three non-dimensional numbers, it can be seen that surface tension σ and the liquid and vapour phase densities ρ_L and ρ_V are contained in any of considered combinations, together with the inlet vapour quality x and mass flow $\dot{m}_r$.

In order to choose which combination of parameters to consider for the final design guideline, the position of the optimal values of the different coefficients was compared for the different working fluids. The aim was to understand which design guideline resulted in a recommendation which was narrower to describe the optimal NPV* for all the working fluids.

Table 7 reports the minimum, maximum and mean values of the different cases for the best solutions, which were selected as NPV* deviating at most 5 % from each best solution. The results are reported for all the working fluids, as well as the overall maximum, minimum and mean value. The deviation (last row) was estimated by considering how far the minimum and maximum value were from the overall mean.

The results suggest that the best agreement between the different mixtures is obtained by the combination of Re_V and Bd , whose deviations are -34 % and 48 % for the minimum and maximum value, respectively. The interval covered by Butane/Hexane(0.9/0.1) is shifted towards the left compared to the other working fluids.

Figure 10: Normalized NPV as function of different combinations of non-dimensional numbers for all the working fluids reported in Table 6; (a)Case I; (b)Case II; (c)Case III; (d)Case IV

Table 7: Minimum, maximum and average values of different combinations of dimensionless numbers for the best results (5 % NPV*), for each working fluid and overall

Working fluid	Case I $Re_V^{1.21} \cdot Bd^{-0.19} \cdot We_m^{-0.87}$			Case II $Bd^{0.14} \cdot We_m^{-0.25}$			Case III $Re_V^{0.74} \cdot We_m^{-0.62}$			Case IV $Re_V^{-0.42} \cdot Bd^{0.26}$		
	min	max	mean	min	max	mean	min	max	mean	min	max	mean
Propane/Iso-Pentane (0.5/0.5)	2110	3310	2620	0.48	0.75	0.59	91.4	152	118	0.030	0.047	0.038
Propane/n-Pentane (0.8/0.2)	2240	3800	2900	0.48	0.77	0.60	104	166	128	0.031	0.046	0.037
Propane/n-Pentane (0.4/0.6)	1420	2780	2130	0.37	0.70	0.54	68	129	99	0.026	0.045	0.036
Butane/Hexane (0.9/0.1)	1560	1570	1560	0.47	0.47	0.47	78.9	80.9	79.8	0.034	0.034	0.034
DME/n-Pentane (0.4/0.6)	2150	2800	2380	0.65	0.86	0.72	103	144	118	0.044	0.058	0.049
DME/n-Pentane (0.7/0.3)	1640	2430	1930	0.46	0.67	0.53	81.3	121	96	0.032	0.045	0.037
DME/Iso-Pentane (0.5/0.5)	1810	2920	2190	0.55	0.88	0.66	88	151	111	0.038	0.059	0.046
Propylene/Iso-Pentane (0.4/0.6)	2100	3330	2670	0.52	0.84	0.67	93	160	124	0.034	0.054	0.043
Overall	1420	3800	2300	0.37	0.88	0.60	68	166	109	0.026	0.059	0.040
Deviation in %	-38%	65%		-38%	46%		-38%	-52%		-34%	-48%	

Fig.11 shows the positive solutions of NPV for all the working fluids as function of the obtained parameter $Re_V^{-0.42} Bd^{0.26}$. The red line represents the mean value, equal to 0.040, while the dotted black lines represent the minimum (0.026) and maximum value (0.059). The plots show that all the fluids, except Butane/Hexane (0.91/0.1), present a good agreement with the obtained guideline: the optimal NPV points are all included in the interval [0.040-0.059], with the peak lying around 0.040 for almost all the fluids. DME/n-Pentane (0.4/0.6) and DME/Iso-Pentane (0.5/0.5) have the maximum slightly shifted towards 0.050. They perform however with NPV* very close to the optimum for 0.040 (equal or above 80 % of the optimal value). Propane/n-Pentane(0.4/0.6) shows solutions with NPV* going down to -30 % for the optimal value for 0.040. The worst performing points are probably related to higher values of chevron angle (see Fig.9), thus optimal designs can be achieved as well with the proposed guideline. In agreement with the numbers reported in Table 7, Butane/Hexane (0.91/0.1) is slightly shifted towards the left compared to the other fluids, i.e. with optimal points in the interval [0.026-0.040]. Nonetheless it has been decided not to shift the design interval towards values optimal for this fluid, since it was shown that this mixture performed with generally lower NPV in the mapping compared to all the other fluids (see Fig.6).

After the assessment of the results from Table 7 and Fig.11, it was decided to propose the guideline presented by Eq.21 as design recommendation.

$$Re_V^{-0.42} Bd^{0.26} \approx 0.040 \quad (21)$$

If a higher value is obtained, the PHE design is expected to lead to higher equipment investment (related to the heat transfer area), while a lower value will lead to high refrigerant pressure drop, resulting in a steep decrease of both COP and NPV.

Figure 11: Normalized NPV as function of $Re_V^{-0.42}Bd^{0.26}$ for all the working fluids. The red line represents the mean value $Re_V^{-0.42}Bd^{0.26} = 0.04$, while the dotted lines represent max and min, 0.059 and 0.026 respectively.

4. Discussion

Section 4.1 presents the discussion of an application of the derived guideline, Section 4.2 reports the results of the uncertainty analysis while Section 4.3 briefly comments on the deviation of the obtained guideline if an alternative prediction method is used for the refrigerant pressure drop estimation, which resulted to have the largest impact on the heat pump economic performance. Section 4.4 highlights the limitation of the study and potential future work.

4.1. Application of the derived design recommendation

The case of the first working fluid mixture was considered, namely Propane/Iso-Pentane (0.5/0.5). The design, based on the derived guideline consists of the following steps:

1. Fix the boundary conditions of the thermodynamic cycle and estimate the inlet condition at the evaporator (quality, mass flow, evaporation pressure).
2. Calculate the following fluid properties at the inlet condition: ρ_L , ρ_V , σ , μ_V .
3. Choose a value for the chevron angle β . It is recommended to employ low values (e.g. 30 – 35°), if the design parameter is not constrained.
4. Decide which design parameters are fixed by external boundaries (fixed plate size and/or numbers and/or corrugation geometry).
5. Tune the remaining free design parameters in order to obtain $Re_V^{-0.42}Bd^{0.26} \approx 0.040$. Calculate the required plate length by using the evaporator heat flow rate, fixed by the cycle.

Table 8 shows different combinations of geometrical parameters, each line corresponding to a different PHE design. Some geometrical parameters were fixed a priori, while other were calculated by means of the design guideline using the methodology presented above. The goal of fixing and releasing different geometrical parameters was to show the different possible scenarios of a designer, which might be constrained in some of the design variables. The plate length was estimated by the sizing model, matching the case study thermal load.

In the first three rows of the table, the corrugation geometry, namely pitch Λ and height b , were fixed together with the corrugation angle. The number of channels N_{ch} and plate width W were found in order to obtain values of $Re_V^{-0.42}Bd^{0.26}$ equal to 0.040. It can be noticed that by respecting the proposed design guideline, the NPV deviates maximum -4.4 % from the best value of the parametric analysis. The COP in the third design point is slightly lower due to the higher pressure drops of the refrigerant.

The fourth and fifth rows of Table 8 report two PHE design which were obtained by fixing the plate width and the number of plates, as well as the chevron angle. The corrugation geometry was found in order to match the design guideline and it was found that also in this case the two solutions deviate of only -1.9 % and -1.5 % from the best NPV, thereby lying in an optimal region for the PHE design. It can be noticed that all the five solutions proposed have the same magnitude of refrigerant pressure drops, ranging from 30 kPa to 35 kPa. This is the same range shown in Fig.6(a) for the same working fluid.

Table 8: Some examples of PHE design points for the mixture Propane/Iso-Pentane (0.5/0.5)

Design	W [m]	N_{ch} [–]	β [°]	b [m]	Λ [m]	L_{HT} [m]	A_{ht} [m ²]	Δp_{r-tot} [kPa]	NPV [€]	ΔNPV [%]	COP [–]	$Re_V^{-0.42}Bd^{0.26}$ [–]
	Free variables		Fixed geometry									
I	0.20	188	45	0.005	0.007	0.66	92.0	31.0	696,960	-3.6	3.02	0.040
II	0.19	196	35	0.005	0.007	0.57	80.1	31.3	711,620	-1.5	3.02	0.040
III	0.44	59	35	0.007	0.007	0.65	80.7	36.0	690,990	-4.4	3.01	0.040
	Fixed geometry		Free variables									
IV	0.20	100	35	0.009	0.008	0.77	82.1	32.6	708,580	-1.9	3.02	0.040
V	0.18	150	35	0.007	0.006	0.54	79.8	31.0	712,180	-1.5	3.02	0.040

4.2. Uncertainty analysis

The uncertainty analysis of the values of heat transfer coefficients and pressure drops was carried out for the design I reported in Table 8 for Propane/Iso-Pentane (0.5/0.5), performing with a NPV -3.6 % lower than the best solution found in the mapping. The mean values, standard deviation and 95 % coverage interval are reported in Table 9. The input uncertainties assigned to the heat transfer and pressure drop correlations resulted in a standard deviation of 10.6 % and 5.0 % for heat transfer area and refrigerant pressure drops, respectively. However, the overall impact on the heat pump NPV resulted to be only equal to a 1.8 % standard deviation. This was also translated to 95 % of all the solutions deviating maximum 3.5 % from the mean. Moreover, it can be observed that the PHE design I obtained in Table 8 deviated only 0.3 % compared to the mean value estimated performing 500 simulations from the uncertainty input space. The heat source pressure drop is not reported, since the results showed a negligible impact on the NPV.

Table 9: MC uncertainty analysis results for PHE design I for Propane/Iso-Pentane (0.5/0.5) performed with a sample population of 500 elements

	A_{ht}	$\Delta p_{r,tot}$	NPV
Mean value	93.2 m ²	33.2 kPa	694,780 €
Absolute standard deviation	9.9 m ²	1.7 kPa	12,370 €
Percentage standard deviation	10.6 %	5.0 %	1.8 %
95 % coverage interval	20.7 %	9.8 %	3.5%
Design I	92.0 m ²	31.0 kPa	696,960 €
Deviation from the mean	-1.3 %	-6.8 %	+0.3%

It must be stressed again that the input uncertainties were taken from the reference papers for the experimental correlations, and they were estimated by the authors as accuracy referred to experimental database not containing the present working fluids and operating conditions. This analysis ensures however that, despite large deviations assigned to the heat transfer and pressure drop calculations, such as the wide interval of [-50%, +100%] given as input for the estimation of f_V in Table 5, a much smaller impact was obtained on the NPV estimation, equal to only 1.8 % standard deviation.

It is nevertheless relevant to focus future work on validating the use of the chosen prediction methods for similar case studies by conducting experiments for flow boiling of zeotropic mixtures in PHEs. Moreover, despite the small uncertainties obtained on the NPV for PHE design I, the input uncertainties might lead to exclude some of the optimal solutions due to possible underestimations of the NPV.

4.3. Sensitivity to pressure drop correlation

From the analysis of the results, the refrigerant pressure drop was found to have the largest impact on the NPV. Therefore, it was decided to carry out a sensitivity analysis on the prediction method chosen to evaluate the two-phase frictional contribution, constituting the major contribution to the refrigerant total pressure loss. The same parametric study was carried out, i.e. considering the same design points, yet by employing an alternative prediction method.

The Lockhart-Martinelli [45] method was applied, for which the Martinelli parameter was estimated from the ratio of the single-phase vapour and liquid pressure drops. The Martin correlation [36] was used in order to estimate the single-phase contributions, and the two-phase multiplier was calculated by using the fitting to the Martinelli's parameter developed by Chisholm [46] with a multiplication coefficient $C = 4.67$, as proposed by Palm and Claesson [47].

By using the same fitting coefficients obtained using the previous data set, i.e. -0.42 and 0.26 for Re_V and Bd respectively, the mean value was calculated for the best results, namely PHE designs based on the best 5 % NPV. A good agreement was obtained between the two different data sets, since the mean value of the guideline $Re_V^{-0.42}Bd^{0.26}$ was found equal to 0.044, thereby being +9.5 % higher than the base case.

4.4. Limitations and future work

The methodology presented in this work is based on a number of boundary conditions and assumptions. First and foremost, the methodology is based on numerical calculations and has not been documented experimentally. The study has only been applied to a limited number of cases. The work should be seen as a suggestion and exemplification of a method, rather than a complete guideline. The methodology was indeed derived for eight different working fluids integrated in the same heat pump, with fixed heat source temperature glide of 25 °C. The influence of having different glides along the evaporator might be investigated as part of future work.

The economic analysis showed that the refrigerant total pressure drops have a major impact on the NPV of the heat pump. In order to estimate the NPV, a number of economic boundaries were assumed, and the price of electricity and natural gas were based on the data available for the Danish energy sector. Furthermore, the case study was based on a waste heat recovery application, where the heat source comes at zero cost. The COP decrease, entailing a higher compressor running cost, could be partially balanced by a reduced utilization of the heat source. The economic boundaries are therefore dependent on the specific case study, and the application of the presented methodology should be adapted to the specific boundary conditions.

Last, it is worth mentioning two limitations imposed by the PHE sizing model. Flow maldistributions are not taken into consideration, neither along the plate width nor among the different HEX channels. This aspect might lead to different results concerning some design variables, e.g. plate length-to-width ratio, since it does not take into account that the flow might not be perfectly counter-current if low values of such ratio are employed. In order to avoid unrealistic results, a minimum ratio was assumed in the present analysis.

5. Conclusions

This paper presented a methodology to derive design guidelines for plate heat exchangers. The methodology was demonstrated by applying it to a case study of plate heat exchanger evaporators in heat pumps using zeotropic mixtures. The basis for the derivation was a numerical model used for simultaneous sizing of the thermodynamic cycle and of the heat exchanger, with heat transfer area and pressure drops being considered for both the calculation of the cycle COP and NPV.

The analysis showed that there is an economic trade-off between heat transfer area and refrigerant pressure drop, with the latter having a larger impact on the results. Higher refrigerant pressure drops resulted in a COP degradation, subsequent increase of the compressor running costs, as well as higher investment required for the compressor. Different working fluids presented different sensitivity to pressure drops, thus the trade-off could not be uniquely defined in terms of a maximum allowable pressure drop.

The results were therefore assessed with the aim of deriving a general design guideline, applicable for all the fluids. Different non-dimensional parameters were correlated to the normalized net present value, and it was shown that Re_V , Bd and We_m are the parameters mostly influencing the results. Moreover, an optimal value was identified for the dimensionless factor $Re_V^{-0.42}Bd^{0.26}$ for the best solutions in terms of NPV, deviating no more than 5 % from the highest NPV. It was shown that for most of the mixtures values above 80 % of the best NPV were found for $Re_V^{-0.42}Bd^{0.26} \approx 0.040$.

Butane/Hexane(0.9/0.1) presented non-favourable design points, with an optimal region shifted towards the left of the obtained guideline. Moreover, Propane/n-Pentane at (0.4/0.6) reported values of NPV above 70% of the best solution for the suggested guideline, performing slightly worse than the other working fluids for some of the designs. This was related to the effect of the chevron angle, and it was therefore recommended to employ low values of β in the optimal region.

The design steps, based on applying the aforementioned guideline, were summarized and applied to the case of for the Propane/Iso-Pentane (0.5/0.5). It was illustrated that different configurations respecting $Re_V^{-0.42}Bd^{0.26} \approx 0.040$ yielded close to optimal NPVs. These were obtained by tuning differently plate corrugation geometry, size and number of plates.

Moreover, an uncertainty analysis of the heat transfer coefficients and pressure drops estimated by experimental correlations was carried out for one of the optimal designs of Propane/Iso-Pentane (0.5/0.5), using the MC method on a population sample of 500 elements obtained using LHS. The NPV mean value was found to be 0.3 % higher than the design point, with a standard deviation of 1.8 %.

The presented methodology offers the possibility of deriving guidelines for different applications, where HEXs are integrated in a thermodynamic cycle and different working fluids are compared, avoiding computationally expensive combined cycle and component optimization to obtain feasible designs.

Acknowledgments

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

Nomenclature

A_0	free flow area[m ²]
D_h	hydraulic diameter [m]
D_p	port diameter [m]
G	mass flux [kg/(s m ²)]
L	plate length [m]
N_{ch}	number of channels [-]
T	temperature [° C]
U	overall heat transfer coefficient [W/(m ² K)]
W	plate width [m]
$\bar{z}$	Silver and Bell-Ghaly correction[-]
$\dot{Q}$	heat load [W]
$\dot{W}$	work [W]
$\dot{m}$	mass flow [kg/s]
b	corrugation height [m]
c_{el}	specific cost of electricity [€/kWh]
f	Fanning friction factor [-]
g	gravitational acceleration [m/s ²]
h	heat transfer coefficient [W/(m ² K)]
n	number of control volumes [-]
p	pressure [Pa]
t	plate thickness [m]
x	vapour quality [-]
Bd	Bond number [-]
COP	coefficient of performance [-]

CRF	capital recovery factor[1/a]
FC	fuel cost[€/a]
NPV	net present value [€]
OMC	operation and maintenance cost[€]
PEC	purchased equipment cost[€]
Re	Reynolds number [-]
TCI	total capital investment[€]
We	Weber number [-]

Abbreviations and acronyms

CI	capital investment
CV	control volume
HEX	heat exchanger
LHS	latin hypercube sampling
MC	Monte Carlo
PHE	plate heat exchanger

Greek letters

β	chevron angle [°]
Δ	difference [-]
η	efficiency [-]
Λ	corrugation thickness [m]
μ	viscosity [Pa · s]
Φ	enlargement factor [-]
ρ	density [kg/m ³]
σ	surface tension [N/m]
τ	time [hr]
θ	inclination angle (90 – β) [°]

Superscripts

"	per unit area	L	liquid
*	normalized	lat	latent
i	control volume i	LO	liquid only
j	solver iteration j	m	mean
Subscripts			
acc	acceleration	NB	nucleate boiling
C	convective	ng	natural gas
comp	compressor	out	outlet
ev	evaporation	p	port-to-port
fr	friction	r	refrigerant
gr	gravity	s	heat source
hp	heat pump	sh	super-heating
ht, HT	heat transfer	tot	total
id	ideal	TP	two-phase
id	mixture	V	vapour
in	inlet	w	water
is	isentropic	wf	working fluid

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Highlights

Paper ID: EGY-D-17-06250

Title: “Deriving guidelines for the design of plate evaporators in heat pumps using zeotropic mixtures”

- Maximum allowable pressure drop do not lead to optimal heat exchanger design
- Refrigerant pressure drop negatively impacts both COP and NPV of heat pumps
- Different optimal pressure drop are found depending on the working fluid
- A HEX design guideline was derived by correlating non-dimensional parameters