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A novel methodology to assist manufacturers in the heat exchanger sizing for variable-speed heat pumps based on part load conditions and economic assessment

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ABSTRACT

Matts Bäckstöm assessed the selection criteria of heat exchangers for single-speed heat pumps in 1940, and his methodology and conclusions have been kept practically unchanged for almost a century. However, heat pump systems have evolved, especially with the introduction of variable-speed heat pumps. These new systems introduced a new degree of freedom to the heat pump design – the compressor speed – which is not considered by traditional design criteria.

This study proposes novel design criteria to optimize the heat exchanger size when the heat pump unit can work under part-load conditions. The proposed method models how heat exchangers and variable-speed compressors perform under different loads and sizes. It considers part-load requirements based on climatic data (which are typically available in the standards) and considers economic factors such as initial investment and operation costs (which are dependent on the components's size). Given the mentioned information, a multi-variable optimization algorithm is implemented to find the optimum heat exchanger size that minimizes the total cost. To validate the proposed approach's effectiveness, it was exemplified by determining the optimum size of a coil and a plate heat exchanger for a domestic variable-speed heat pump. The results demonstrate that using traditional criteria could lead to over-dimensioning heat exchangers in variable-speed heat pumps which could incur in an increase of costs of 5% for the studied case.

Introduction

Heat pumps (HP) have become an increasingly popular choice in the heating and air conditioning sector due to their efficiency, reliability and affordability. That is why the European Union has recognized them as a key technology to reduce CO_2 emissions and achieve the objectives proposed in the European Green Deal to become climate-neutral by 2050 [1].

Heat Pumps, especially those with an external unit, work under varying climatic conditions, which strongly affects their performance. Consequently, an annual performance evaluation is needed to fairly compare HP among them and to their counterparts (gas and electric boilers). In Europe, the standards [2,3] establish a letter code (from A+++ to G) for each device to rank the different solutions based on its yearly efficiency, which is calculated according to three different weather profiles: Cold, Intermediate and Warm, based on Helsinki (Finland), Strasbourg (France) and Athens (Greece) reference temperatures respectively. Fig. 1 shows the required heating hours for the three different climate profiles and which profile applies to each region.

The standard also provides information about the relation between the required heating capacity at each external temperature and the rated conditions of the heat pump. Eq. (1) represents this relation consisting of a straight line starting from the rated heating capacity at the nominal temperature and reaching null heating requirements at external temperatures of 16 °C.

$$\dot{Q}_{req} = \frac{t_{ext} - 16}{t_{nom} - 16}\dot{Q}_{nom} \tag{1}$$

As shown in Fig. 1, thermal loads significantly fluctuate throughout the year, and thus, capacity control in HP is necessary. Historically, ON/OFF cycling was the preferred control strategy; however, since the introduction of variable-speed compressors (VSC), the modulation of the compressor speed to modify the HP capacity has become more attractive

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Nomenclature

Abbreviations

COP	Coefficient Of Performance
CV	Coefficient of Variance of the RMSE ($\frac{RMSE}{\overline{X}}$)
HP	Heat Pump
HX	Heat Exchanger
RMSE	Root Mean Square Error
VSC	Variable Speed Compressor
VSHP	Variable Speed Heat Pump

Symbols

A	heat exchange area [m^2]
a	annuity factor [year-1]
C	cost [SEK]
$c_{\#}$	model coefficient for VSC
$\dot{E}$	Heat Pump Power Consumption [kW]
f_c	compressor speed [Hz]
I	Investment [SEK]
$k_{\#}$	model coefficients for HX
$\dot{Q}$	heating capacity [kW]
t	temperature [°C]
T	temperature [K]
ΔT_{evap}	$\lvert T_{ext} - T_{evap} \rvert$ [K]
ΔT_{cond}	$\lvert \mathrm{T}_{\mathrm{cond}} - T_{water_{service}} \rvert$ [K]
U	heat transfer coefficient [W/(m^2 °C)]

Greek letters

κ	marginal cost [SEK/(kW/K)]
η_{Ct}	Carnot's efficiency [-]
τ	working hours

Subindex

aux	auxiliary heating system
bin	related to each external temperature
compr	related to the compressor
cond	related to condenser
e	related to energy
evap	related to the evaporator
ext	of air at outdoor conditions
HX	heat exchanger
max	maximum
min	minimum
nom	nominal, at rated conditions
opt	optimum
req	required, obtained from the standard

as it can result in significant energy savings and better comfort [4,5].

However, with variable-speed heat pumps (VSHP), a new degree of freedom in the analysis of these devices arises – the speed – which directly affects the methodologies to model, select and design its components.

In recent years, numerous papers have been published studying VSHPs, which are collected in the review articles of Tolga [6], Lin Lee et al. [7] and Zhang et al [8]. Additionally, other studies focused on modelling VSCs proposing correlations to predict compressor mass flow and energy consumption (Shao et al. [9], Mendoza-Miranda et al. [10], Ossorio and Navarro-Peris [11]). And others focused on inverter power losses (Cuevas and Lebrun [12] and Ossorio and Navarro-Peris [11]).

Regarding sizing criteria, Bagarella et al. [13] studied the influence of cycling losses in the sizing strategy of on–off and variable-speed compressors. However, there is a lack of studies evaluating the sizing criteria of heat exchangers for variable-speed heat pumps, as this topic is only treated for fixed-speed heat pumps.

For the selection of HX in fixed-speed heat pumps, the work of Matts Bäckstöm has to be highlighted, which already in 1940 assessed the selection criteria of heat exchangers in his study "Economic Optimum Problems in Connection with Refrigeration" [14]. In his research, Bäckstöm selected the temperature difference between the HX and the source or sink as an optimization parameter and proposed a techno-economic methodology considering investment capital and operational cost. This methodology is still in use today and is reflected in "Refrigerating Engineering" by Granryd and Palm [15], in which a table with optimum temperature differences is displayed for either evaporators and condensers as a function of its technology and operation time.

More recent literature also assessed the selection of heat exchangers. Mancini et al. [16] focused on the design of plate evaporators for zeotropic mixtures. Wang and Sundén [17], Caputo et al. [18] and Unuvar and Kargici [19] studied the optimal design of plate heat exchangers for general use. Jiang et al. [20] proposed a general guideline for selecting heat pump systems based on economic analysis. Finally Dai et al. [21 22] annalized the design of CO2 heat pumps in China for space heating including in the study calculations for heating loads and calculations for HX required area.

However, the mentioned methodologies do not consider new degrees

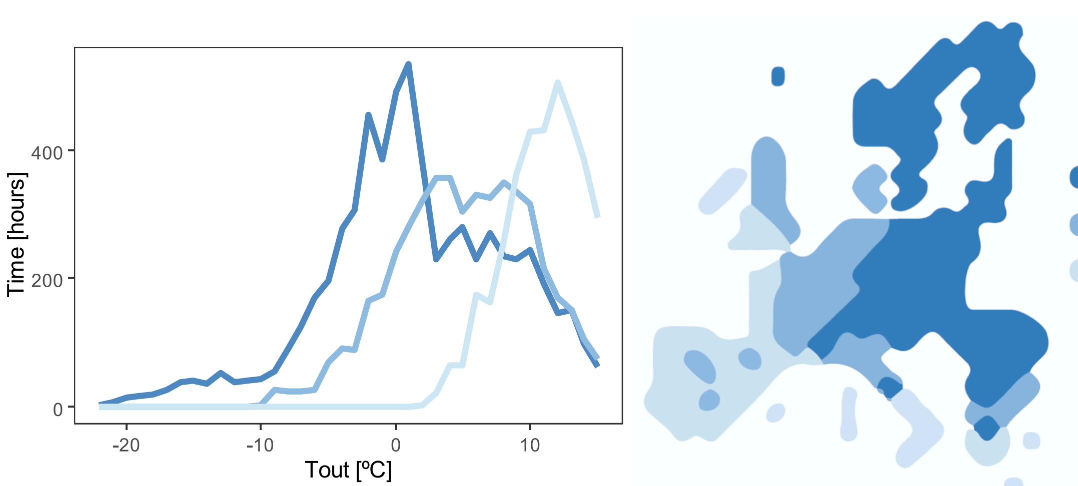

Fig. 1. Climate profiles according to the standard [3].

of freedom as compressor speed and/or design the system for a fixed and stationary condition (not including the variation of the external temperatures and loads). Moreover, it should be pointed out that, if a heat exchanger is designed for its maximum capacity, it would be overdimensioned when the VSHP works at part load (most of its lifetime).

To assess these issues, in this study, a new methodology for sizing heat exchangers for a VSHP is proposed. It combines an improved techno-economical analysis including the speed modulation and incorporates climatic weather profiles available in the standards to account for the variation of external conditions. The methodology is meant to be simple and comprehensible so it can be easily implemented in the first steps of heat pump design process. To exemplify the proposed method, it has been applied to the sizing of the heat exchangers of an air-to-water HP for domestic space heating and its results have been compared with the ones obtained by Bäckstöm classical criteria. Furthermore, for modelling the HP performance, simplified correlations for predicting temperature differences between the HX and the secondary fluids have been proposed for either the coil and the plate HX. Regarding the compressor modelling, a simple correlation is proposed, which takes into account its working limits.

Methodology

Bäckstöm

As mentioned in the introduction, in 1940 Matts Bäckström published a study which settled the bases of HX selection. The chosen selection variable was the optimum temperature difference between the sink or the source and the evaporating or condensing temperature, respectively. This optimum temperature difference is a parameter that directly depends on the heat exchanger's size and global transfer coefficient.

To choose the optimum temperature difference in the HXs, Bäckström minimized the cost function displayed in Eq. (2).

$$C = a\,I + \kappa_e \tau \dot{E} + constants \tag{2}$$

In which the first term represents the yearly capital costs (I representing the total capital investment and a the annuity factor) and the second representing the yearly operational cost (κ_e representing the energy cost, τ the working hours and $\dot{E}$ the power consumption of the unit). To define the variables as a function of temperature differences in the HXs, the investment was expressed with the marginal investment cost κ_{HX}, representing how capital costs increase with the HX size (Eq. (3)).

$$I = \kappa_{HX}\, UA = \kappa_{HX} \frac{\dot{Q}_{HX}}{\Delta T_{HX}} \tag{3}$$

And for energy consumption, the expression in Eq. (4) was used, assuming a constant Carnot efficiency.

$$\dot{E} = \frac{\dot{Q}_{cond}}{COP} = \frac{\dot{Q}_{cond}}{\eta_{Ct}\frac{T_{cond}}{T_{cond}-T_{evap}}} = \dot{Q}_{cond}\frac{T_{cond}-T_{evap}}{\eta_{Ct} T_{cond}} \tag{4}$$

Eq. (5) can be obtained substituting both Eq. (3) and Eq. (4) in Eq. (2). And finally, Eq. (5) can be solved for an optimum temperature difference (ΔT) in either the condenser and in the evaporator.

$$\begin{aligned} C &= a\left[\frac{\kappa_{evap}\dot{Q}_{evap}}{\Delta T_{evap}} + \frac{\kappa_{cond}\dot{Q}_{cond}}{\Delta T_{cond}}\right] \\ &\quad + \kappa_{el}\tau \dot{Q}_{cond}\frac{(t_{sink}+\Delta T_{cond}) - (t_{source}-\Delta T_{evap})}{\eta_{Ct} T_{cond}} \end{aligned} \tag{5}$$

Eq. (5) depends on the following:

- Economically linked variables which are the result of an economic assessment as: a (the annuity factor), κ_{evap}, and κ_{cond} (the marginal cost of the HX per "size") and κ_e (the cost of the energy needed to operate the HP).
- Charachteristic parameters of the HP as $\dot{Q}_{evap}$, $\dot{Q}_{cond}$ and η_{Ct} (which is considered by Bäckström constant in the range of interest).
- External working parameters as t_{sink}, t_{source} (working temperatures) and τ (total operation hours), which depend on the application and on the working condition.
- HX temperature differences ΔT_{cond} and ΔT_{evap},which mainly depend on the size and typology of the heat exchangers. They are the optimization variables that are chosen to minimize the total cost (C).

Once this function is built, Bäckström performs the partial derivatives of the cost function with ΔT_{cond} and ΔT_{evap} respectively and equals the new expressions to zero to look for minimum cost. This methodology results in two formulas that can be used to calculate the optimum ΔT directly and in a semi-independent manner (it is not entirely independent as for the optimization of one HX information of the optimum ΔT of the other HX is needed, which is approximated or guessed).

The main limitation of this methodology is that the optimization is made for a single condition and, as it is known, air–water HP working conditions significantly vary over time, especially in the external air unit as external temperature changes. One simplistic approach could be to design the HP at the rated or at the most repeated condition, which has been a common methodology.

However, a better solution would be to combine Bäckström methodology with the information provided in the HP rating standard. The standard offers information regarding working hours and required heating capacity at each external condition Eq. (1). For example, the conditions in Fig. 2 would apply for a HP working in an intermediate climate with a rated heating capacity of 6.5 kW at −10 °C.

Consequently, the required heating capacity and expected heating hours in a year can be obtained for a given HP and for each external temperature. And with the given information, the cost equation can be rewritten as shown in Eq. (6):

$$\begin{aligned} C &= a\left[\frac{\kappa_{evap}\dot{Q}_{evap,nom}}{\Delta T_{evap,nom}} + \frac{\kappa_{cond}\dot{Q}_{cond,nom}}{\Delta T_{cond,nom}}\right] \\ &\quad + \sum_{bin}\kappa_e \tau_{bin}\dot{Q}_{cond,bin}\frac{(t_{sink,bin}+\Delta T_{cond,bin}) - (t_{source,bin}-\Delta T_{evap,bin})}{\eta_{Ct}(t_{sink,bin}+\Delta T_{cond,bin}+273.15)} \end{aligned} \tag{6}$$

In this way, the operational costs are divided into bins (one for each external temperature), and for each bin, the standard gives information of τ, $\dot{Q}_{cond}$ and t_{source}. Regarding t_{sink}, the standard defines four different conditions (35, 45, 55 and 65 °C) depending on the application. The only still missing parameter is $\dot{Q}_{evap,nom}$ that can be approximated with Eq. (7):

$$\dot{Q}_{evap,nom} = \dot{Q}_{cond,nom} - \dot{W}_{compr} = \dot{Q}_{cond,nom}\left(1 - \frac{1}{\eta_{Ct} COP_{Ct}}\right) \tag{7}$$

Consequently, if the economic parameters are also known, the only non-defined information is how ΔT evolve in the HX for different external conditions (ΔT_{bins}). Nevertheless, a heat exchanger model can be used to predict ΔT_{bins} based on a ΔT_{nom} and on the working condition of each bin. With that, the only non-defined parameters would be the optimum temperature differences in both HXs ($\Delta T_{evap,nom}$, $\Delta T_{cond,nom}$), which are the optimization variables of the proposed methodology.

Evaporator model

The objective of this section is to provide a correlation that predicts the temperature difference between the refrigerant and the external environment (ΔT) in a coil HX as a function of:

- **heating capacity**: which can be easily retrieved from the standard for each external condition

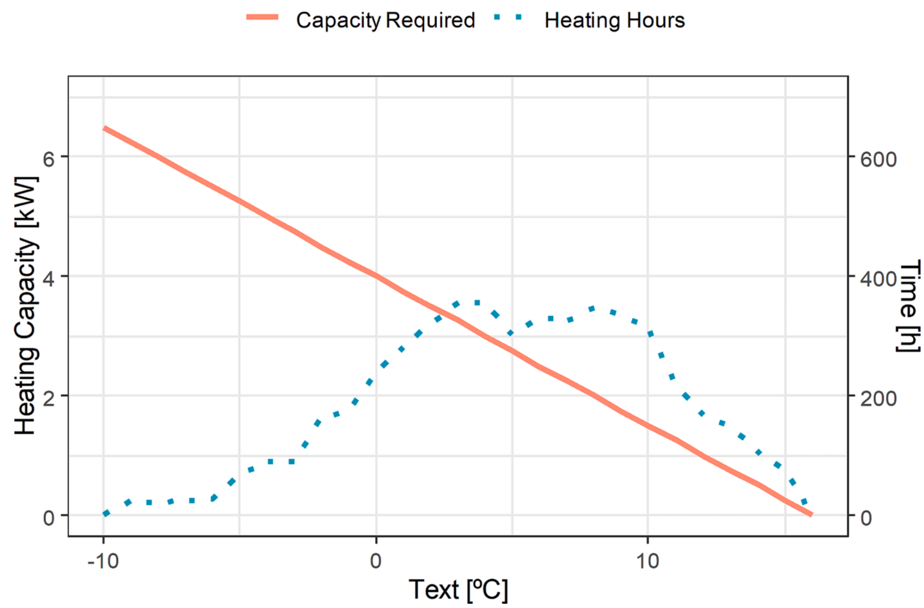

Fig. 2. Operating hours and heat capacity required as a function of external temperature for a HP with a rated capacity of 6.5 kW working in an intermediate climate.

- **Sizing parameter**: for example, coil width, a design parameter that increases total exchange area without significantly affecting the heat exchange mechanism and flow patterns.

IMST-ART, a detailed heat pump model was used to study the effect of these variables in ΔT. IMST-ART [23] is an advanced HP simulation software that combines accurate heat and mass exchange correlations to model HP systems using as input only geometric and easy-to-obtain data from catallogues. This tool allows a detailed description of HXs (row number, tubes per row, tube spacing, number of parallel circuits and their distribution, fin pitch, material properties…). IMST-ART implements heat transfer coefficient and pressure drop correlations discretized for virtual discrete volumes in the heat exchanger and solves a complete simulation of the hole system in less than 20 s. Additionally, it has been proven to be robust and has been validated against extensive experimental data at steadty-state conditions [24–27] with prediction errors between 5 % and 10 % of Maximum Relative Error. It should be remarked that for this study the frost formation won't be considered to keep it simpler and more readable.

Fig. 3 displays the IMST-ART results obtained for ΔT at different

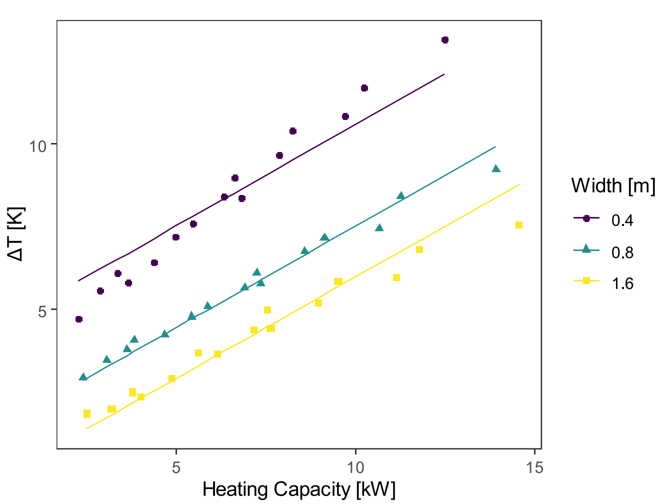

Fig. 3. Dependence of ΔT in the evaporator as a function of heating capacity and coil width and prediction of the used model.

heating capacities and different coil sizes. The modified sizing parameter was the coil width, and the different heating capacities were introduced by varying in the detailed model the external temperatures and compressor speeds. Regarding the other HX parameters defined in the simulations, the studied evaporator had 3 rows, 8 parallel circuits, 32 tubes per row, a longitudinal tube spacing of 19 mm, a transversal tube spacing of 25 mm and a fin pitch of 2 mm. The air velocity was considered to be 1.5 m/s and the air flow is variable with the coil width; as the coil gets wider, more fans will be installed.

The results show a linear correlation between Heating Capacity and ΔT in the evaporator. Regarding the effect of the coil size, as width diminishes, the total exchange area decreases and ΔT increases accordingly, just as expected. What is remarkable is that the dependence between heat capacity and ΔT (slope) does not seem to be significantly affected by the width change. With those results, the following simplified evaporator model is proposed:

$$\Delta T_{evap} = \left(k_{00} + \frac{k_{01}}{Width}\right) + k_1 \dot{Q}_{cond} \tag{8}$$

The term between brackets is a constant that only depends on the size and consequently, for a given HX, it disappears if the expression is given in differences with respect to a nominal condition ($\Delta T_{e,nom}$ and $\dot{Q}_{cond,nom}$). The result of expressing the model with respect to nominal contions is shown in Eq. (9).

$$\Delta T_{evap,bin} = \Delta T_{evap,nom} + k_1 \left(\dot{Q}_{cond,bin} - \dot{Q}_{cond,nom}\right) \tag{9}$$

Eq. (9) only uses one coefficient and can be substituted in the general cost expression [Eq. (6)] so all the terms inside the summation are referenced to the optimization parameter that is $\Delta T_{evap,nom}$. Additionally, once the optimum ΔT_{nom} has been obtained, Eq. (10) can be used to predict an optimum coil width.

$$Width_{opt} = \frac{k_{01}}{\Delta T_{evap,nom_{opt}} - k_1 \dot{Q}_{cond,nom} - k_{00}} \tag{10}$$

Condenser model

With the condenser, a similar methodology was used to obtain a simplified correlation to estimate ΔT as a function of heating capacity and a sizing parameter.

First, the detailed heat pump model IMST-ART was used to study the dependence of operating variables on ΔT. IMST-ART allows the simulation of different lengths, widths, number of plates and can also define the plate's pitch, thickness and used material. Additionally, a parametric study of the inlet water temperature and its mass flow can also be carried out. In case IMST-ART is not available, other alternatives as SSP calculator from SWEP [28] could be used to obtain the fitting coefficients.

Fig. 4 shows the obtained ΔT simulated with IMST-ART in the condenser at different heating capacities, number of plates and water temperature steps. The modelled condenser had a length and width between ports of 0.478 m and 0.073 m respectively and the plates had a pitch of 2.35 mm and thickness of 0.4 mm. Regarding the waterside, the outlet water temperature was 35 °C and the water mass flow was the one required to reach the desired temperature step in the waterside (ΔT_{water}).

The results also show a linear dependence of ΔT_{cond} with heating capacity. Modifying the number of plates mainly affects the slope of the dependence with heating capacity, not the intercept (contrary to the coil heat exchanger). In fact, it can be checked that a common intercept exists, which has been proven to depend mainly on the water temperature difference. With the information provided by the simulations, the simplified model in Eq. (11) was proposed:

$$\Delta T_{cond} = k_0 + \left(k_{10} + \frac{k_{11}}{Plates}\right)\dot{Q}_{cond}$$

$$\text{with } k_0 = \begin{cases} \Delta T_{water} \\ k_{00} + k_{01}\Delta T_{water} \end{cases} \tag{11}$$

The definition of the intercept k_0 has two different approaches:

- The simplified one considers it to be equal to ΔT_{water}, it is the simpler one and fits well the general trend but incurs more significant errors at high and low ΔT_{water}. Its fitting results are displayed in Fig. 4 with dotted lines.
- On the other side, if more precision is required, k_0 can be defined as a linear function of ΔT_{water} giving excellent results. Its performance can be checked out in Fig. 4 with solid lines.

Regarding ΔT_{water} and the service temperature, the standard defines them as a function of:

- Application: the reference service temperature is 35, 45, 55 or 65 °C for low, intermediate, medium and high-temperature applications, respectively.
- Water pump: if it can vary its speed, then $\Delta T_{water} = 5K$ is assumed for all conditions. Else, the water flow is established to provide $\Delta T_{water} = 5K$ for the standard rating condition given in EN 14511–2 and then that water flow is kept constant for the rest of the conditions.

Given the model in Eq. (11), its slope (k_1) can be calculated with Eq. (12) given a nominal condition (ΔT_{nom}, $\dot{Q}_{cond,nom}$).

$$k_1 = \left(k_{10} + \frac{k_{11}}{Plates}\right) = \frac{\Delta T_{cond,nom} - k_0}{\dot{Q}_{cond,nom}} \tag{12}$$

And if Eq. (11) and Eq. (12) are combined and rearranged Eq. (13) is obtained:

$$\Delta T_{cond,bin} = \Delta T_{cond,nom}\frac{\dot{Q}_{cond,bin}}{\dot{Q}_{cond,nom}} + k_0\left(1 - \frac{\dot{Q}_{cond,bin}}{\dot{Q}_{cond,nom}}\right) \tag{13}$$

With this equation, each ΔT_{bin} is calculated based on ΔT_{nom}. Consequently, this equation can be substituted into the modified Bäckstöm expression in Eq. (6) so the only unknown temperature difference on the condenser side is $\Delta T_{cond,nom}$.

Once the optimum $\Delta T_{cond,nom}$ has been obtained, Eq. (12) can be used to calculate the optimum number of plates in the condenser (taking into account that it should be a pair number). It should be noted that if the simplified model is considered ($k_0 = \Delta T_{water}$), Eq. (13) won't contain any fitting coefficient and thus, no condenser model would be needed to calculate the optimum $\Delta T_{cond,nom}$.

Compressor model

The compressor is the device in charge of modulating the heating capacity to adapt it to the load. The modulation is carried out by varying the compressor speed in a range that can differ from model to model but typically ranges from less than 30 Hz to values slightly higher than 120 Hz. A typical modulating scheme is displayed in Fig. 5.

Fig. 5 shows three different operating zones. In the middle part, the VSHP manages to modulate its capacity to exactly fit the thermal load.

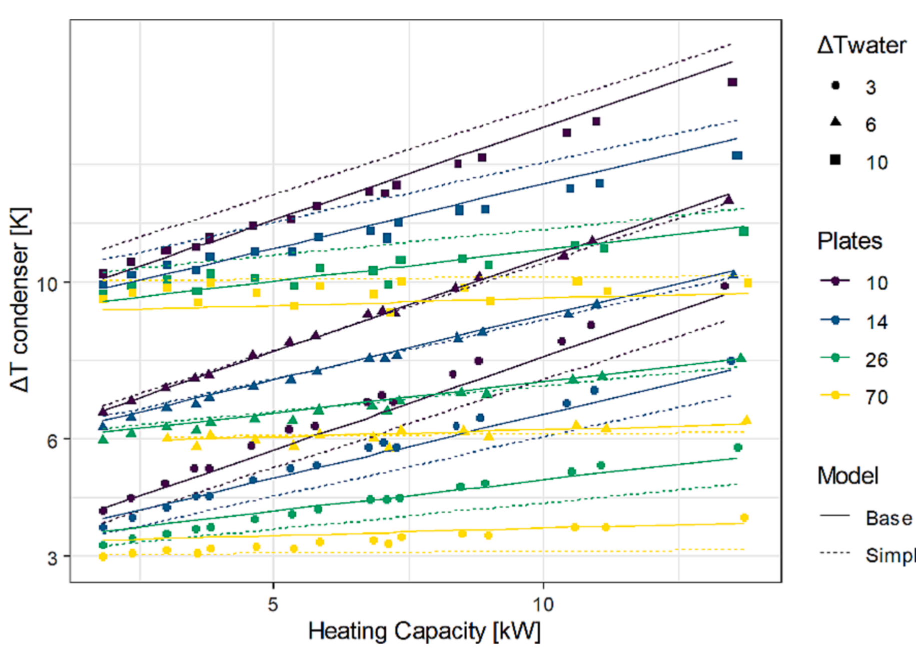

Fig. 4. Dependence of ΔT as a function of heating capacity and number of plates and prediction of the used models.

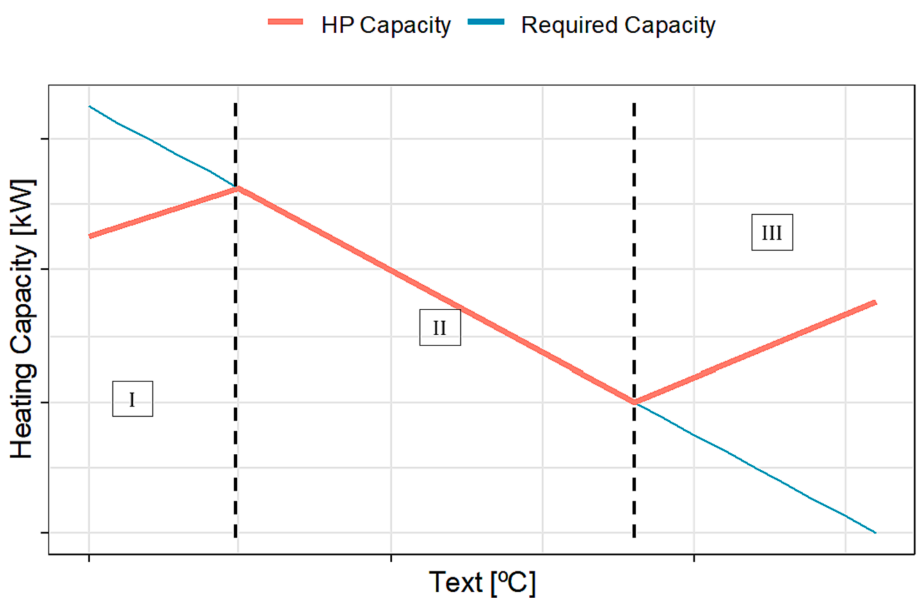

Fig. 5. Heat pump operational zones.

At low external temperatures, the compressor speed reaches its maximum, the heat pump is unable to provide enough heating capacity and a backup heating system is required (typically a boiler or an electric heater). On the other hand, when the outdoor temperature is warmer, the compressor reaches its lowest speed, starts providing more heat than required and starts to cycle between ON/OFF states.

To correctly model the three differentiated areas, a compressor model is required to correlate the compressor speed with the provided heating capacity. In order to provide a simplified expression, the heating capacity was modelled only as a function of evaporating temperature and speed, as shown in Eq. (14).

$$\dot{Q}_{cond} = c_0 + c_1 f_c + c_2 t_{evap} f_c + c_3 t_{evap}^2 f_c \tag{14}$$

It can be justified to model the capacity only with evaporating temperature as it determines the refrigerant mass flow which directly affects the provided capacity. Condensing temperature also affects heating capacity, but the effect is weaker and in the described application the condensing temperature remains relatively constant as the service water temperature does not vary significantly over the year.

The model performance can be visualized in Fig. 6, in which the dots represent the results obtained from IMST-ART and the lines represent the model predictions. The maximum relative error is close to 5 % and the RMSE is lower than 100 W.

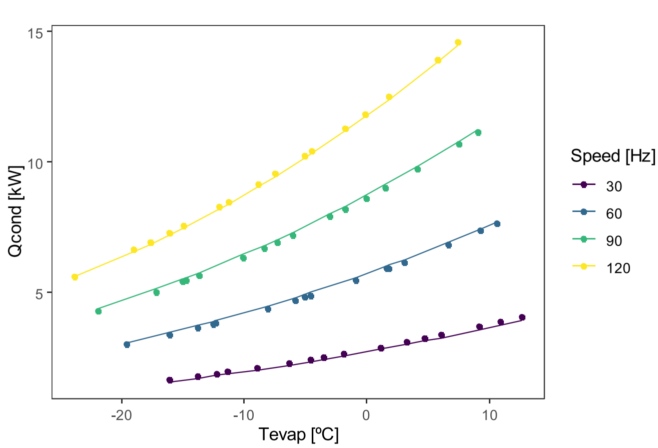

Fig. 6. Compressor model performance.

Consequently, knowing the heating requirements and the evaporating temperature ($T_{evap} = T_{ext} - \Delta T_{evap}$), it is possible to calculate the compressor working speed with Eq. (14). However, ΔT_{evap} is not known a priori as it is a result of applying the evaporator model, which in turn needs to know the operation zone of the HP. Therefore, an iterative calculation is needed, which will be detailed in the following section.

Additionally, when using Eq. (14), the required working speeds can result in values higher than the compressor's maximum speed. In that case, the compressor model uses as input $f_{c,max}$ to calculate the maximum heating capacity that the compressor is able to produce at that condition. This procedure will also be used when the required speed is less than $f_{c,min}$ to calculate the actual capacity produced.

For zone I, the backup system will start working to reach the desired capacity and thus, another power-consuming element will appear in the system which, in the case of an electrical heater, can be modelled by adding the term appearing in Eq. (15) to the general cost equation in Eq. (6).

$$\sum_{bin} \kappa_e \tau_{bin} \dot{Q}_{aux,bin} \tag{15}$$

Being $\dot{Q}_{aux,bin}$ the extra capacity needed to reach the required one ($\dot{Q}_{req,bin}$). It can be calculated with Eq. (16).

$$\dot{Q}_{aux,bin} \begin{cases} \dot{Q}_{req,bin} - \dot{Q}_{cond,bin} & if\ \dot{Q}_{req,bin} > \dot{Q}_{cond,bin} \\ 0 & if\ \dot{Q}_{req,bin} < \dot{Q}_{cond,bin} \end{cases} \tag{16}$$

For zone III, the compressor will oscillate between on and off states, so the total operating time in the third zone should be reduced using the following formula Eq. (17).

$$\tau_{bin} \dot{Q}_{cond,bin} = \tau_{req} \dot{Q}_{req,bin} \tag{17}$$

Optimum temperature difference

Once a simplified model is proposed for each key element of the system, all the information can be put together to find the optimum economic HX size.

As mentioned in the previous section, an iterative calculation is needed to determine the different compressor zones. A diagram showing the general calculation flow is displayed in Fig. 7 for better understanding.

First a HX size is assumed for the evaporator and condenser which

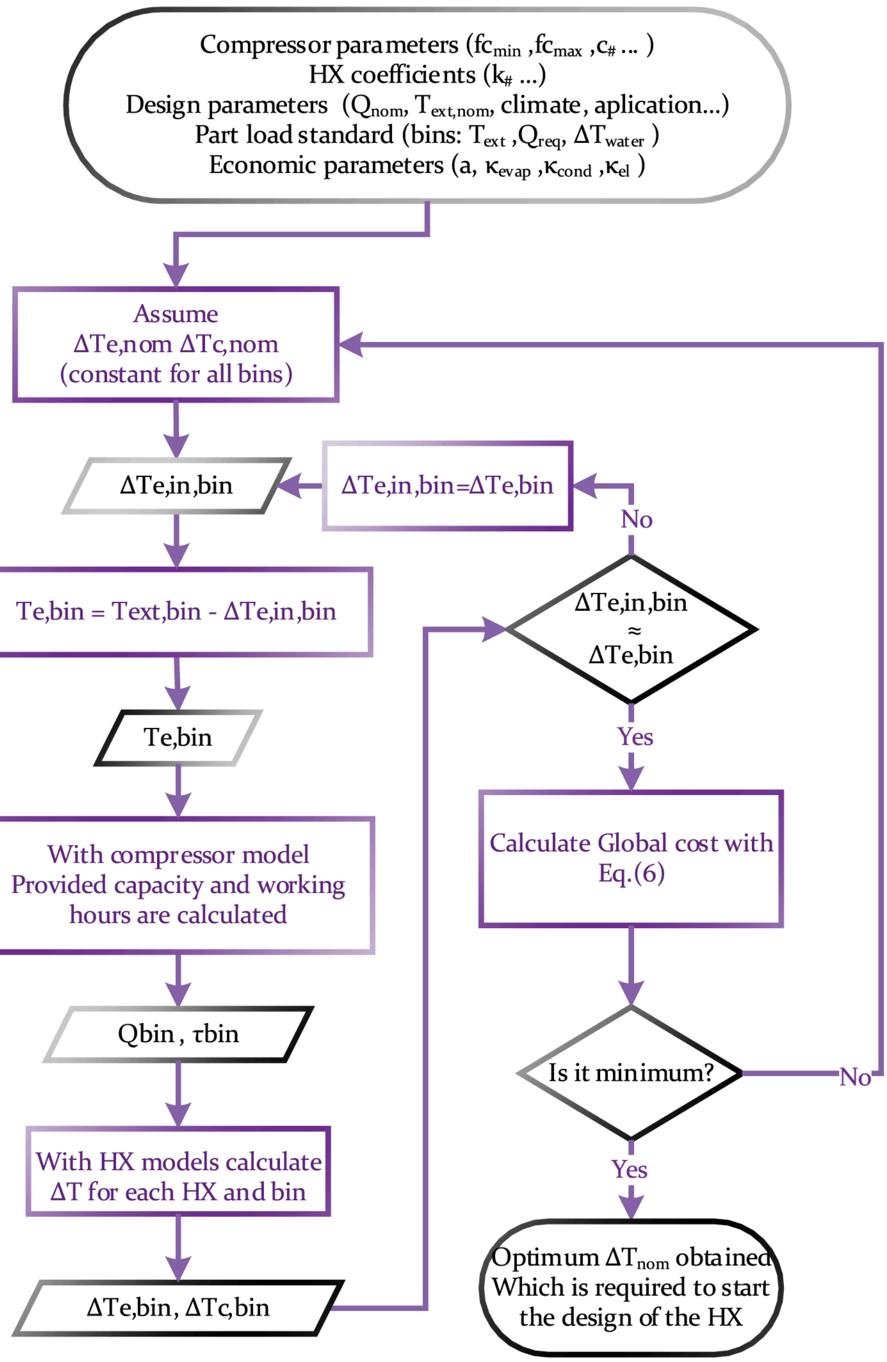

Fig. 7. Calculation flow for sizing HXs.

translates in assuming a ΔT_{nom}, which is the temperature difference between the refrigerant and the secondary fluid at nominal conditions ($T_{ext,nom}$, $\dot{Q}_{nom}$). Initially, ΔT_{nom} is considered to be constant under all external conditions. Then, the evaporating temperature is calculated and the compressor zones are established. The new Q_{bins} are calculated for zone I and III with $f_{c,max}$ *and* $f_{c,min}$ respectively, and total operating time is corrected with Eq. (17) in zone III. Once Q_{bins} are calculated, both HX models are used to calculate their respective ΔT which won't be constant for all conditions anymore. With this new ΔT_{evap} the compressor model and HX correlations are used again in an iterative way until the solution converges (the convergence is often achieved in a couple of iterations). Once the performance of the HP is modelled, the cost formula in Eq. (6) can be used to estimate the total cost. The described procedure is repeated changing ΔT_{nom} in evaporator and condenser until the minimum cost is reached. Once $\Delta T_{nom,opt}$ has been found for both HX, Eq. (10) and Eq. (12) can then be used to obtain the optimum width of the coil and the optimum number of plates in the condenser.

This procedure can be performed independently for the evaporator and condenser, assuming approximate data for the other heat exchanger. However, if higher accuracy is needed, multivariable optimization algorithms can be used so both optimizations are carried out simultaneously.

Results

Optimization of HX size

The described methodology has been exemplified with the HX optimization of an air-to-water HP for a low-temperature application with a required nominal capacity of 6.5 kW at external conditions of −10 °C and working in an intermediate climate profile. A preliminary analysis has decided to use the typology of HXs described in the methodology section, whose model parameters are displayed in Table 1:

The supplied hot water is provided at 35 °C and is circulated with a variable speed pump that ensures a water temperature difference of 5 K. The compressor starts cycling at speeds lower than 30 Hz and it is able to provide the desired rating capacity at its maximum speed of 120 Hz, so there is no need for auxiliary systems. The different working zones of the compressor in the exemplified application are displayed in Fig. 8. The cycling starts at external temperatures close to 5 °C, and from there, the operating hours are reduced by applying Eq. (17).

The rest of the input parameters were considered to be the same used in "Refrigeration Engineering" [15] when exemplifying Bäckstöm methodology so a direct comparison could be performed. Regarding economic parameters: $a = 0.25$, $k_e = 0.5\frac{SEK}{KWh}$, $k_{evap} = 2000\frac{SEK}{KW\cdot K}$ and $k_{cond} = 1430\frac{SEK}{KW\cdot K}$. And last but not least, Carnot efficiency was considered constant and equal to 0.45 (as also stated in [15]).

Given the input information, the iterative procedure described in Fig. 7 was coded in R-language [29], and a genetic algorithm [30] was implemented for the multivariable optimization so both temperature differences in the evaporator and condenser are optimized simultaneously for more accurate results.

In Fig. 9, the evolution of the costs with the ΔT in the HX is displayed for the optimum solution. The total cost is the result of adding the capital and operational costs and it is represented with a solid line.

An increase in ΔT implies using a smaller HX, which reduces the capital costs. On the contrary, a high ΔT increases the difference between evaporating and condensing conditions reducing COP and increasing the operational cost. If both effects are added the total costs are obtained, whose minimum is found with the optimization algorithm and is marked with a dot in Fig. 9. For the studied case, the optimum temperature difference in the coil evaporator is 7.78 K and 10.1 K in the plates condenser resulting in a total cost of 2803 SEK.

With these results, Eq. (10) and Eq. (12) can be used to select the HX dimensions, which results in a coil width close to 0.5 m (0.448 m) for the evaporator and a condenser with six plates. The resulting number of plates is relatively small so other plate HX with a lower length between ports could be selected.

Note that the difference in the slope of the operational cost is steeper in the case of the evaporator, even if the energy cost is the same for both cases. That results in a flatter total cost for high ΔT in the condenser, which in turn makes the optimum cost less sensitive to variations of the selected ΔT in the optimum range. The explanation for this different behaviour is that, as ΔT_{evap} changes, the compressor working zones change accordingly, varying the working hours and affecting the operational costs. On the other hand, the operational costs are more stable with the variation of ΔT_{cond} as it does not affect the compressor working zones.

A rigorous direct comparison with the pure method presented by Bäckstöm can't be made as the variation of the external conditions can't be considered. However, if Eq. (5) is applied and it is assumed that the HP is operated consistently at the nominal conditions ($t_{source} = -10^{\circ}C$ and $t_{sink} = 35^{\circ}C$) the solution in Fig. 10 is obtained.

The optimum ΔT in the evaporator and condenser would be 4.46 °C and 4.98 °C respectively and the annual minimum cost ascends to 7130 SEK, double as much as the part-load solution. This is unrealistic due to the fact that the HP is always working at its maximum consumption zone, which also affects the sensibility of the solution, which is steeper for both cases compared to the part-load solution.

To conclude this section, it will be assessed the potential cost savings associated with the proposed methodology when compared to conventional approaches. In order to do that, the optimal solution obtained with Bäkstrom criteria has been integrated to the cost function of the proposed new method (which is capable of accurately calculate the total cost considering all part load available information).

As a result, if the new part-load cost equation is evaluated with a ΔT close to 5 K in either the condenser and evaporator (representing the Bäkstrom-designed configuration), the obtained total expense incurred amounts to 2943 SEK. This represents a 5 % cost increase compared with the optimized solution. It is essential to highlight that this incremental cost is specific to the particular application under analysis and is expected to vary across different applications, potentially reaching substantially higher values.

Influence of the economic parameters on the optimization

The optimization solution depends on the chosen economic parameters that must be obtained from an economic analysis. In the following, the influence of these parameters on the optimal solution will be studied.

The described optimization methods were applied varying each economic parameter ceteris paribus and the results are displayed in a grid of plots in Fig. 11. In each row a different cost equation was used: in the first row, Bäckstöm original formula was used without considering part-load [Eq. (5)], while in the second, part-load was considered using Eq. (6). Regarding the columns, each column represents the parametric study of a different economic variable: k_e, k_{evap} and k_{cond} respectively. In each plot, the different cost curves obtained in the parametric study are represented with solid lines and the optimums are marked with a dot.

The most evident result is that the optimum ΔT displaces to the right (to smaller HXs) as the cost of the HXs (k_{evap} and k_{cond}) increases and the contrary occurs with the cost of energy; as the cost of energy grows, bigger HXs are preferred as they are more energy efficient.

For the same parameter values, optimum ΔT are always smaller when part-load is not considered and the gap is enhanced as HXs cost increase or when energy costs decrease. The highest difference in ΔT_{opt} is found when k_{cond} is modified, the explanation of this comes to the fact that the cost curves of the condenser are flatter. On the other hand, the variable that has a higher impact on the total cost is the cost of energy.

It should be pointed out that the energy cost also affects the HX cost. However, in the parametric study this was not analyzed and k_e was modified maintaining constant the HX cost.

Discussion and limitations of results

The HX design methodology proposed in the study followed a detailed and rigorous modelling of the system and it was exemplified with a particular case scenario. To further enhance the precision when applying the methodology the following modifications could be applied:

Table 1
Model coefficients of the HXs and compressor.

	Model coefficients					
	k0		k1			
	$k00$	$k01$	$k10$	$k11$	RMSE	CV
Coil Evaporator	0.7814	0.77332	0.73083		0.328	4.61
Plate Condenser	0.8963	0.81633	−0.03324	5.3225	0.258	3.852
	c0	**c1**	**c2**	**c3**	**RMSE**	**CV**
Compressor	0.1795	3.12E-03	3.21E-05	−0.3137	0.129	1.904

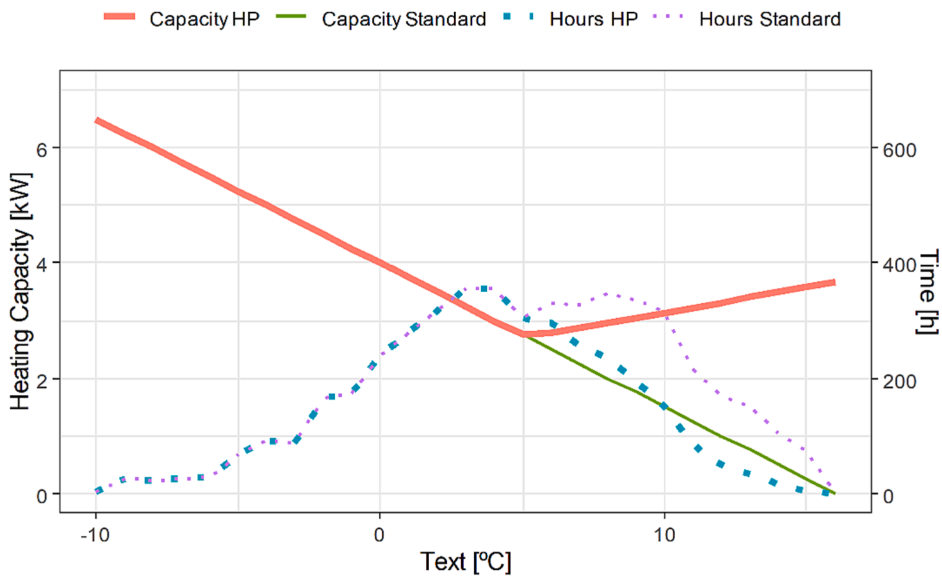

Fig. 8. Compressor zones in the optimum condition and working hours for each bin.

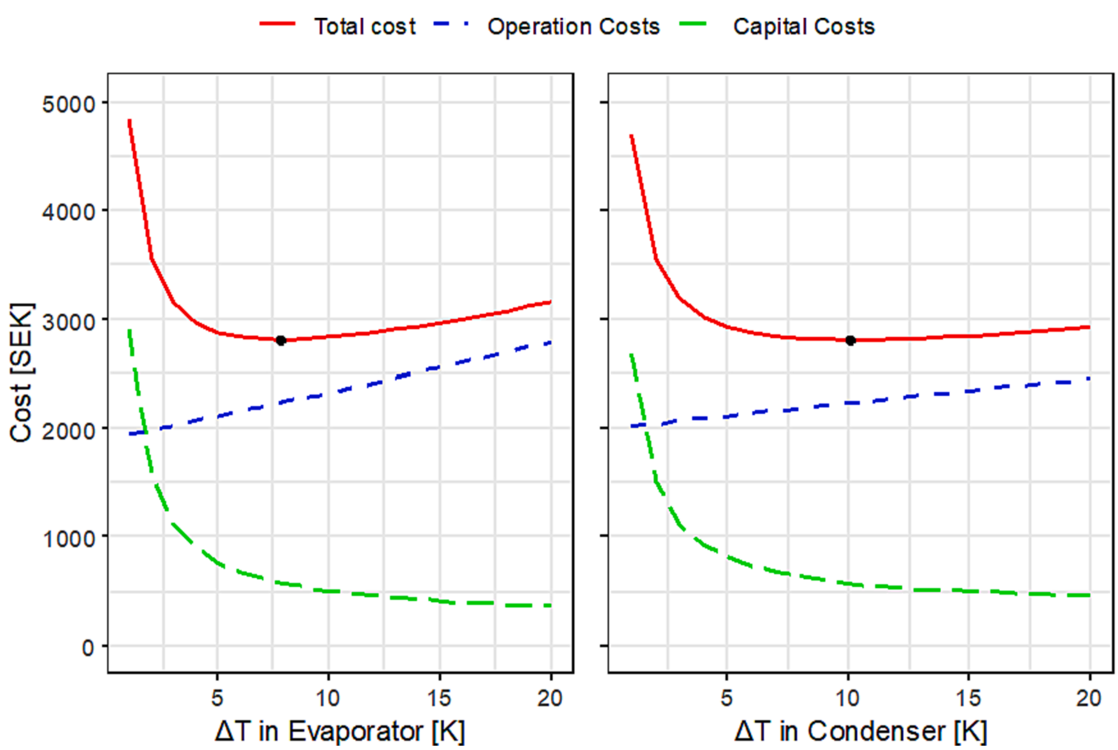

Fig. 9. . Optimization of temperature differences in heat exchangers using Eq. (6).

utilize local standards and climatic registries to calculate the operating hours at each external temperature according to the final location (e.g. ANSI/ASHRAE 116–2010 for EEUU), incorporate updated economic data for components using the particular manufacturer information and, last but not least, consider the dependence of Carnot's Efficiency with compressor speed rather than treating it as a constant [9].

Additionally, compressor cycling (Fig. 5: Zone III) introduces a COP reduction which has not been taken into account in the present study for better readability. To include it, the standard [3] proposes the correction factor described in Eq. (18), which can be multiplied to each bin of Eq. (6) (being $Cd = 0.25$ the default value for air-to-water HPs for water heating).

$$\frac{COP_{bin,corrected}}{COP_{bin}} = \left(1 - Cd\left(1 - \frac{Q_{req,bin}}{Q_{cond,bin}}\right)\right) \quad (18)$$

It is also important to highlight that simplified correlations were proposed for coil and plate heat exchangers Eq. (8). These models could be applied to other typologies or capacity ranges, but prior validation is required.

Finally, the method was presented for heating heat pumps. However, the methodology is meant to be general and applicable to most heat pumps applying little modifications. For cooling heat pumps the same methodology could be followed redefining the operational costs term Eq. (4): the desired heating capacity is substituted with cooling capacity

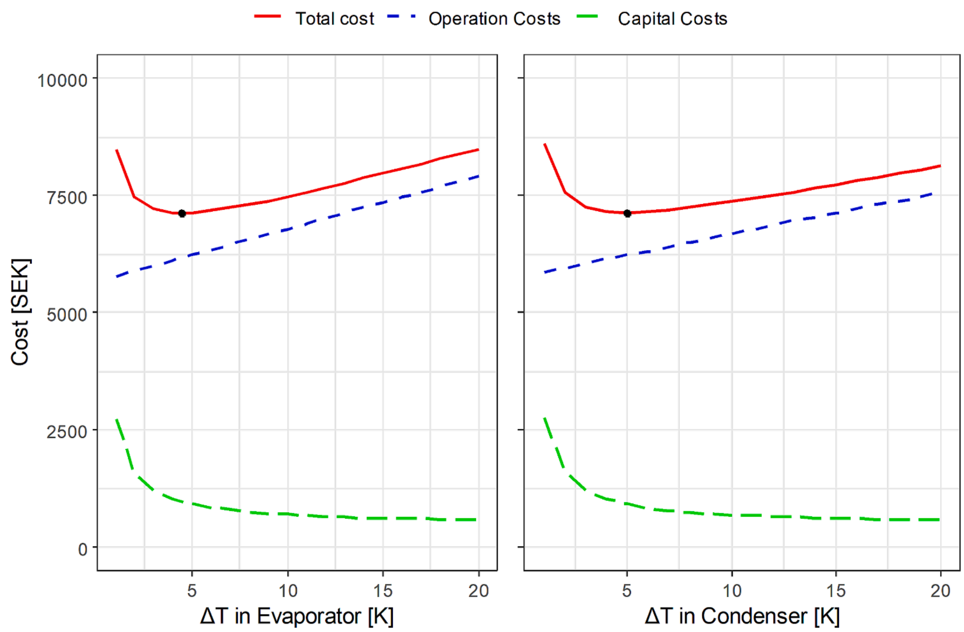

Fig. 10. Optimization of temperature differences in heat exchangers with Bäckstöm method.

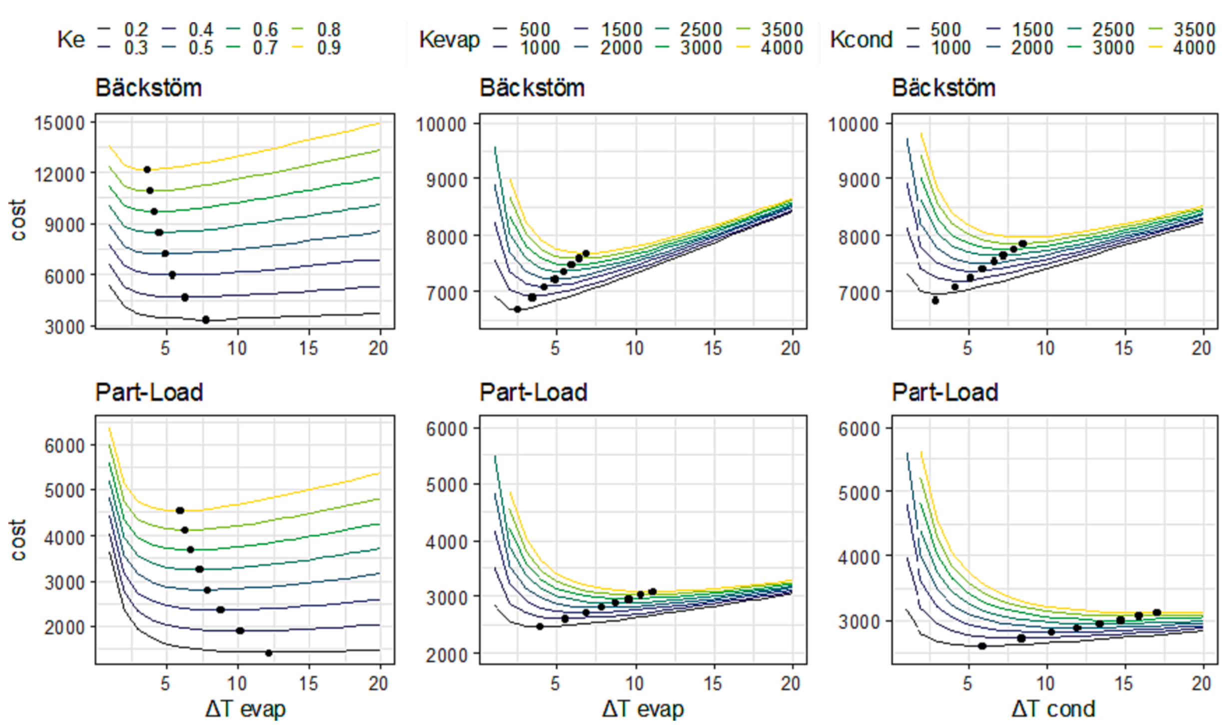

Fig. 11. Study of the influence of the different economic parameters in the optimization of HXs.

and the heating COP is substituted by refrigerating COP which results in Eq. (19).

$$\dot{E} = \frac{\dot{Q}_{evap}}{COP_{refr}} = \frac{\dot{Q}_{evap}}{\eta_{Ct\_refr}\dfrac{T_{evap}}{T_{cond}-T_{evap}}} = \dot{Q}_{evap}\frac{T_{cond}-T_{evap}}{\eta_{Ct\_refr}\,T_{evap}} \tag{19}$$

With the mentioned changes the general cost equation would result in Eq. (20).

$$C = a\left[\frac{\kappa_{evap}\,\dot{Q}_{evap,nom}}{\Delta T_{evap,nom}} + \frac{\kappa_{cond}\,\dot{Q}_{cond,nom}}{\Delta T_{cond,nom}}\right] + \sum_{bin}\kappa_e\tau_{bin}\,\dot{\boldsymbol{Q}}_{\boldsymbol{evap,bin}}\frac{(t_{sink,bin}+\Delta T_{cond,bin})-(t_{source,bin}-\Delta T_{evap,bin})}{\boldsymbol{\eta_{Ct\_refr}}(\boldsymbol{t_{source,bin}}+\Delta\boldsymbol{T_{evap,bin}}+273.15)} \tag{20}$$

Moreover, the Standard EN 14825 [3] also includes a climate profile for space cooling with the reference amount of hours expected at each external temperature and part load capacity at each condition [Fig. 12] which is necessary to define the different bins in Eq. (20).

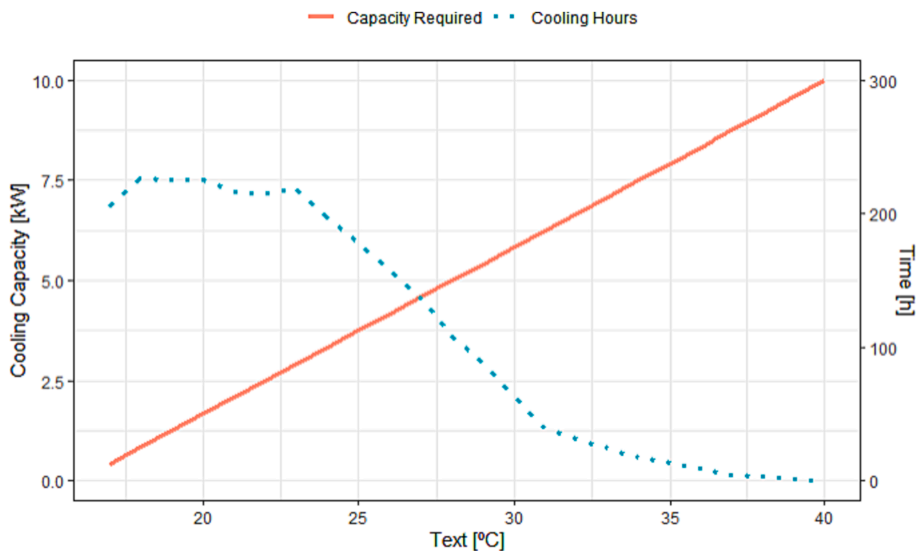

Fig. 12. Operating hours and cooling capacity required as a function of external temperature for a HP with a rated capacity of 10 kW.

Conclusion

A new systematic methodology for selecting heat exchangers in variable-speed heat pumps working at part load has been proposed. Some of the more relevant aspects of the procedure are:

- The evolution of the external working temperatures over time is considered as it affects the working time, the required capacity and the heat pump's performance.
- The chosen heat exchangers' design parameters were the coil's width and the number of plates. Simple heat exchanger models are proposed that predict the temperature difference as a function of the operating conditions and the design parameters.
- The methodology takes into account the compressor working range and establishes three differentiated working zones. A simplified compressor model is presented for the calculation of the provided capacity at the different zones.
- The obtained optimal solutions suggest using smaller heat exchangers compared to when classical rules are followed. Additionally, the cost function seems flatter, so a slight deviation from the optimum will have a smaller effect on total cost.
- For the studied case, savings close to 5 % are obtained when using the proposed methodology compared with the classical one. However, the savings could vary for other applications reaching higher values.

Overall, this study provides a general approach to assist system designers in heat exchanger selection based on a cost-based analysis, which could lead to a better design of future heat pumps, increasing total efficiency and making these systems more attractive compared with classical boilers or electrical heaters.

CRediT authorship contribution statement

Rubén Ossorio: Methodology, Software, Validation, Writing – original draft, Writing – review & editing, Visualization. **Emilio Navarro-Peris:** Conceptualization, Supervision, Project administration. **Javier Marchante-Avellaneda:** Methodology, Software, Writing – review & editing.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article.

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