Exergy analysis on a heat pump working between a heat sink and a heat source of finite heat capacity rate

Miquel PITARCH[(a)], Estefanía HERVAS-BLASCO[(a)], Emilio NAVARRO-PERIS[(a)*] , José M. CORBERÁN[(a)]

[(a)] Institut Universitari d'Investigació d'Enginyeria Energètica, Universitat Politècnica de València, Camí de Vera s/n, València, 46022, Spain

Tel: +34 963879123

enava@ter.upv.es

Abstract

The optimum performance of a pure subcritical refrigeration cycle depends significantly on the temperature lift of the heat source and sink. Therefore, the maximization of the system efficiency has to be linked to them. This paper shows an exergy analysis of each heat pump component (condenser, evaporator, expansion valve and compressor) considering that the heat source and sink are not at constant temperature. The performed study shows the components with more possibilities for improvement. Based on this analysis, the optimization of cycle parameters like subcooling and superheat as a function of the external conditions have been done. In addition, this work has demonstrated that the components having a higher influence in the system irreversibility's depends significantly on the temperature lift of the secondary fluids. Finally, the obtained results show potentials improvements of the efficiency up to 23% if the system is able to operate in the optimal subcooling and superheat.

Keywords: exergy; heat pump; optimal working point; subcooling; superheat

NOMENCLATURE

COP: Coefficient of Performance, [-]

$\dot{E}x$: Rate of exergy [kW]

ex : Specific exergy [kJ kg^{-1}]

h: Specific enthalpy [kJ kg^{-1}]

$\dot{m}$: Mass flow rate [kg s^{-1}]

p: Pressure [bar]

$\dot{Q}$: Rate of heat transfer, [kW]

RI: Relative irreversibility [%]

s: Specific entropy [kJ kg^{-1} K^{-1}]

Sc: Subcooling, [K]

Sh: Superheat [K]

S.Fluid: Secondary Fluid

T: Temperature [ºC]

$\dot{W}$: Rate of work, power [kW]

Greek symbols

ψ: Exergy efficiency [%]

Δ: Variation

Subscripts

0: Reference environment

1,2,3,4: State points of the cycle (Figure 1b)

comp: Compressor

cond: Condenser, or gas cooler

des: Destruction

evap: evaporator

EV: Expansion Valve

h: Heating

H: Heat sink

i: Refers to either the condenser, gas cooler or the expansion valve

in: inlet

opt: Optimal

out: Outlet

ref: Refrigerant

sf: Secondary fluid

hs: Heat Source

1. INTRODUCTION

Heat pumps are widely used for cooling and heating applications. Researchers are struggling to optimize the heat pump system for each application in order to obtain the maximum possible performance. There are three main ways to improve the performance of a heat pump working at a given external conditions. The first is to design more efficient heat pump components, such as the heat transfer in the heat exchangers or the efficiency of the compressor. The second is to modify the heat pump cycle according to the external conditions, so it works more efficiently, for instance, the two-stage cycle. The third is the optimization of the working cycle conditions such as subcooling, superheat, or the rejection pressure in a transcritical cycle depending on the external conditions (Inokuty, 1928), (Cecchinato et al., 2010). No matter how much the heat pump performance is improved, there will be a limit that cannot be exceeded. The maximum possible performance depends on the external conditions, and it is often defined by the ideal refrigerant cycle of Carnot or Lorentz, depending on the application (Itard and Machielsen, 1994), (van de Bor and Infante Ferreira, 2013).

In the most common applications for air conditioning and heat pumps, the temperature lifts of the secondary fluids usually is limited to 5-10 ºC .This fact has made that when exergy analysis are performed, the assumption of 0 temperature lift in the source and sink has been commonly made. Nevertheless, the increased interest in energy efficiency has made that the potential application field of heat pumps has been extended to, for instance, sanitary hot water production, dryers or energy recovery systems. In these fields, the temperature lift could increase significantly (up to 60-80 ºC) and the assumption of considering the temperature lift in the source and sink as negligible can introduce important deviations from the real optimum performance of the system.

From the point of view of the cycle, the analysis of the temperature lift can be related to the analysis of the optimum subcooling for a given temperature lift in the heat sink and the analysis of the optimum superheat for a given temperature lift in the heat source. The influence of these two parameters in the cycle has been studied previously in the literature,(Fernando et al., 2004), (Choi and Kim, 2004), (Corberán et al., 2007) analized the optimum charge of a heat pump, which is indirectly related to the degree of subcooling. Other researchers theoretically studied the effect of subcooling in an air conditioning system for different refrigerants have reported COP improvement with a certain subcooling (Hjerkinn, 2007), (Redón et al., 2014), (Xu, 2014), (Pottker and Hrnjak, 2015); they pointed out the existence of an optimal subcooling that maximizes COP for their equipment. In (Koeln and Alleyne, 2014), the authors developed a sophisticated methodology in order to look for the optimal subcooling by “try and error”, since they did not found the relationship between the optimal subcooling and the external conditions.

In the same line are the works that study the influence of subcooling on vapor-compression system from the exergy point of view. (Yang and Yeh, 2015), (Selbaş et al., 2006) focus on the thermos-economic optimization of the heat exchanger, and the condensing temperature is kept constant as a function of subcooling. Thus, they do not simulate a real system as a function of subcooling (condensing temperature depends on subcooling (Yang and Yeh, 2015), (Pitarch et al., 2018)). Furthermore, they look at the system as a whole and not to each component individually. In (Yang and Yeh, 2015), the analysis of an optimal subcooling is done for different refrigerants including the transcitical, CO_2. They concluded that the condensing temperature is the main influencer on the optimal degree of subcooling but the study does not considers the variation of external conditions.

In general, the studies found are focused on air conditioning and heat pump applications with small temperature lift. Although they arrive to relevant conclusions fixing the optimum subcooling to 5-10 K, they lack of generality for other applications where the temperature lift could change significantly (Yataganbaba et al., 2015).

Furthermore, most of the recent studies published related to exergy analysis in vapor compression systems have been focused mainly on the performance of new refrigerants mixtures and hydrocarbons (Yataganbaba et al., 2015), (Ajuka et al., 2017), (Valencia et al., 2017) or on their integration in complex systems like solar with heat pumps, CHPs, geothermal district plants, where usually the conclusions applied to their specific application and study (Arat and Arslan, 2017), (Di Somma et al., 2017).

One of the first attempt to analyze the problem of optimal subcooling dependence with the external conditions was made by (Pitarch et al., 2018). They found that the optimal subcooling highly depends on the secondary fluid temperatures at the condenser, for instance the optimal subcooling was around 10 K when heating water from 50 ºC to 60 ºC and around 42 K when heating from 10 ºC to 60 ºC with 25% of COP improvement respect to the case without subcooling. In (Pitarch et al., 2017), they theoretically showed (for an infinite condenser area assumption) that the optimal subcooling was given for the two pinch points situation. In addition, they found that the optimal subcooling mainly depends on the temperature lift of the secondary fluid at the condenser and not on the absolute inlet and outlet temperatures. In (Hervas-Blasco et al., 2017), the authors analyzed the double pinch point situation by means of experiments and a heat pump model. They confirm that in a subcritical system, there is an optimal subcooling and it is close to the double pinch point situation. They also gave a simple way to control subcooling by setting the temperature difference between the refrigerant outlet and secondary fluid inlet at the condenser approximately in 3 K based on experimental results for a wide range of operating conditions.

Nevertheless, among all the works cited previously and found in literature dealing with the optimization of the subcooling, none of them investigates the influence of subcooling on the efficiency of each heat pump component and its dependence on the secondary temperature lift. This consideration is of major relevance in order to understand how the different components behave with subcooling and to show the direction for potential improvements. The only works developed related to that have been in the frame of transcritical cycles (not strictly subcooling). For instance, (Sarkar et al., 2004) and (Chen and Gu, 2005) made an exergy analysis of the CO_2 transcritical cycle in order to look for improvement strategies of these cycles; they found that the expansion valve holds the highest irreversibility. Afterwards, in (Yang et al., 2005), a comparison of two transcritical cycles based on an exergy analysis, one with expansion valve and other with an expander, is done.

Regarding to superheat, it is usually considered undesirable from performance point of view, but a small amount of superheat is desirable for the compressor reliability (Jolly et al., 2000). Up to the knowledge of the authors, there is no literature available about the influence of superheat on system performance as a function of evaporator secondary fluid temperature lifts as it has been done in recent years for subcooling in the condenser. Usually, a small secondary fluid temperature lift at the evaporator is preferred. This might be the reason of the lack on superheat studies. However, there are applications where the required temperature lift at the evaporator is high. For instance, in heat recovery applications (Hervas-Blasco et al., 2017), it could be more profitable to have a high temperature lift at the heat source in order to extract as much energy as possible.

In this work, a novel exergy analysis for vapor-compression systems as a function of the temperature lift of the sink and the source is done. In order to perform this analysis, the water-to-water heat pump presented in (Pitarch et al., 2018) has been selected as reference. A model of this heat pump has been built and validated against a set of experimental test of the heat pump. The model has shown an error lower than 3% in the prediction of the relevant variables. Later on, the model has been used to analyze the system in a wide range of external conditions. From this analysis, the relative influence of each component to the losses of the system has been defined. The subcooling and superheat depending on the temperature lift of the source and the sink in order to operate at optimum COP conditions has been obtained.

This study will allow improving the performance of refrigeration systems controlling properly the subcooling and superheat degree of the system. It is important to remark that the proper control of these variables can be performed without major changes on the vapor compression system.

2. METHODOLOGY

2.1 Theoretical basis

The exergy at a given point is a measure of the maximum potential work that can be done respect to a reference state. This type of analysis is a way to evaluate the good use of the available energy. In this sense, it is possible to understand the direction for potential improvements (Yumrutaş et al., 2002). In real working cycles, several sources of irreversibility (exergy destruction) prevent achieving the maximum performance, that is, the ideal performance.

The main sources of irreversibility on a vapor compression system are:

- Compressor: The compression does not take place at constant entropy.
- Condenser: The temperature profile of refrigerant and secondary fluid does not match perfectly (finite temperature differences).
- Expansion Valve: The pressure drop is not isentropic, but isenthalpic.
- Evaporator: The temperature profile of refrigerant and secondary fluid does not match perfectly (finite temperature differences).
- For all components and in the pipe exist the irreversibility of the pressure drop due to friction.

For the exergy analysis in this study, the following assumptions have been taken into account:

- The pressure drop in the refrigerant and secondary fluid side is small enough to be negligible.
- Both, the throttling valve and the expansion valve are taken as a unique isenthalpic expansion from point 3 to 5 (Figure 1).
- The expansion and compression are considered adiabatic.
- The irreversibility at the liquid receiver are neglected.
- The kinetic, potential and chemical irreversibility are neglected.

This study evaluates the exergy destruction (irreversibility) and their corresponding exergy efficiencies component by component (Dinçer and Rosen, 2012).

Eq. 1 and eq. 2 represent respectively, the total and the specific exergy of a fluid:

$$\dot{E}x = \dot{m} \cdot ex \tag{1}$$

$$ex = (h - h_0) - T_0 \cdot (s - s_0) \tag{2}$$

Where h is the specific enthalpy, s is the specific entropy, and T_0, h_0 and s_0 are the temperature, the enthalpy and the entropy at the dead state. T_0 has been chosen as 0 ºC as most of this work is focus on applications with liquid water but other reference can be selected, and the reference pressure for the refrigerant is taken as the saturation pressure at T_0 (Dinçer and Rosen, 2012).

The exergy destruction for the compressor accounting with the electrical and mechanical losses can be calculated as follows:

$$\dot{E}x_{des,comp} = \dot{W}_{comp} + \dot{m}_{ref} \cdot (ex_1 - ex_2) = \dot{W}_{comp} - \dot{m}_{ref} \cdot [(h_2 - h_1) - T_0 \cdot (s_2 - s_1)] \tag{3}$$

Where $\dot{W}_{comp}$ is the compressor consumption of the heat pump and its value could be estimated from representative catalogue data, m_{ref} the refrigerant mass flow and h_i and s_i the respective relative enthalpies and entropies of the refrigerant at the inlet and outlet of the compressor according to Figure 1.

The exergy destruction of the condenser and evaporator is obtained from eq. 4 and eq. 5. It has been calculated accounting with the electrical and mechanical losses, but neglecting the heat interaction with the ambient (adiabatic):

$$\dot{E}x_{des,cond} = \dot{m}_{ref} \cdot [(h_2 - h_3) - T_0 \cdot (s_2 - s_3)] - \dot{Q}_h \cdot \frac{\overline{T}_H - T_0}{\overline{T}_H} \tag{4}$$

$$\dot{E}x_{des,evap} = \dot{Q}_c \cdot \frac{\overline{T}_C - T_0}{\overline{T}_C} - \dot{m}_{ref} \cdot [(h_1 - h_5) - T_0 \cdot (s_1 - s_5)] \tag{5}$$

Where $\dot{Q}_h$ and $\dot{Q}_C$ are the heating and cooling capacity of the heat pump. The temperatures $\overline{T}_H$ and $\overline{T}_C$ are the entropy-averaged temperatures (average temperature in the temperature-entropy diagram) of the secondary fluid at the condenser and evaporator respectively. For constant heat capacity rate and constant pressure processes, the entropy-averaged temperature can also be written as in Eq. 6 and 7 (Hasan et al., 2002).

$$\overline{T}_H = \frac{h_{Sfcondout} - h_{Sfcondin}}{s_{Sfcondout} - s_{Sfcondin}} = \frac{T_{Sfcondout} - T_{Sfcondin}}{\ln(T_{Sfcondout} / T_{Sfcondin})} \tag{6}$$

$$\overline{T}_C = \frac{h_{Hsevapin} - h_{Hsevapot}}{s_{Hsevapin} - s_{Hsevapout}} = \frac{T_{Hsevapin} - T_{Hsevapout}}{\ln(T_{Hsevapin} / T_{Hsevapout})} \tag{7}$$

The exergy destruction at the expansion valve is:

$$\dot{E}x_{des,EV} = \dot{m}_{ref} \cdot T_0 \cdot (s_5 - s_3) \quad (8)$$

Based on the previous expressions, the exergy efficiency for each component is defined according to the Eq. 9-12.

$$\psi_{comp} = \frac{\dot{m}_{ref} \cdot [(h_2 - h_1) - T_0 \cdot (s_2 - s_1)]}{\dot{W}_{comp}} \quad (9)$$

$$\psi_{cond} = \frac{\dot{Q}_h \cdot \frac{T_H - T_0}{T_H}}{\dot{m}_{ref} \cdot [(h_2 - h_3) - T_0 \cdot (s_2 - s_3)]} \quad (10)$$

$$\psi_{evap} = \frac{\dot{m}_{ref} \cdot [(h_1 - h_5) - T_0 \cdot (s_1 - s_5)]}{\dot{Q}_c \cdot \frac{T_C - T_0}{T_C}} \quad (11)$$

$$\psi_{EV} = \frac{ex_5}{ex_3} = \frac{(h_5 - h_0) - T_0 \cdot (s_5 - s_0)}{(h_3 - h_0) - T_0 \cdot (s_3 - s_0)} \quad (12)$$

The total exergy efficiency of the heat pump defined in eq. 13.

$$\psi_{HP} = \frac{\dot{W}_{comp} - \dot{E}x_{des,tot}}{\dot{W}_{comp}} \quad (13)$$

Where $\dot{E}x_{des,tot}$ is the total exergy destruction or irreversibility, i.e. the sum of exergy destruction of the heat pump components.

And the total energy efficiency of the system is based on the heating COP defined in Eq. 14.

$$COP_h = \frac{\dot{Q}_h}{\dot{W}_{comp}} \quad (14)$$

Another important parameter to analyze the system is the relative irreversibility of each component. That is, the irreversibility contribution of each component over the system or total. The relative irreversibility for the component *i* is defined as:

$$RI_i = \frac{\dot{E}x_{des,i}}{\dot{E}x_{des,tot}} \tag{15}$$

This ratio is since a component could have the highest exergy efficiency and still could account with the major part of the system irreversibility (highest potential of improvement regardless the exergy efficiency).

2.2 Case study

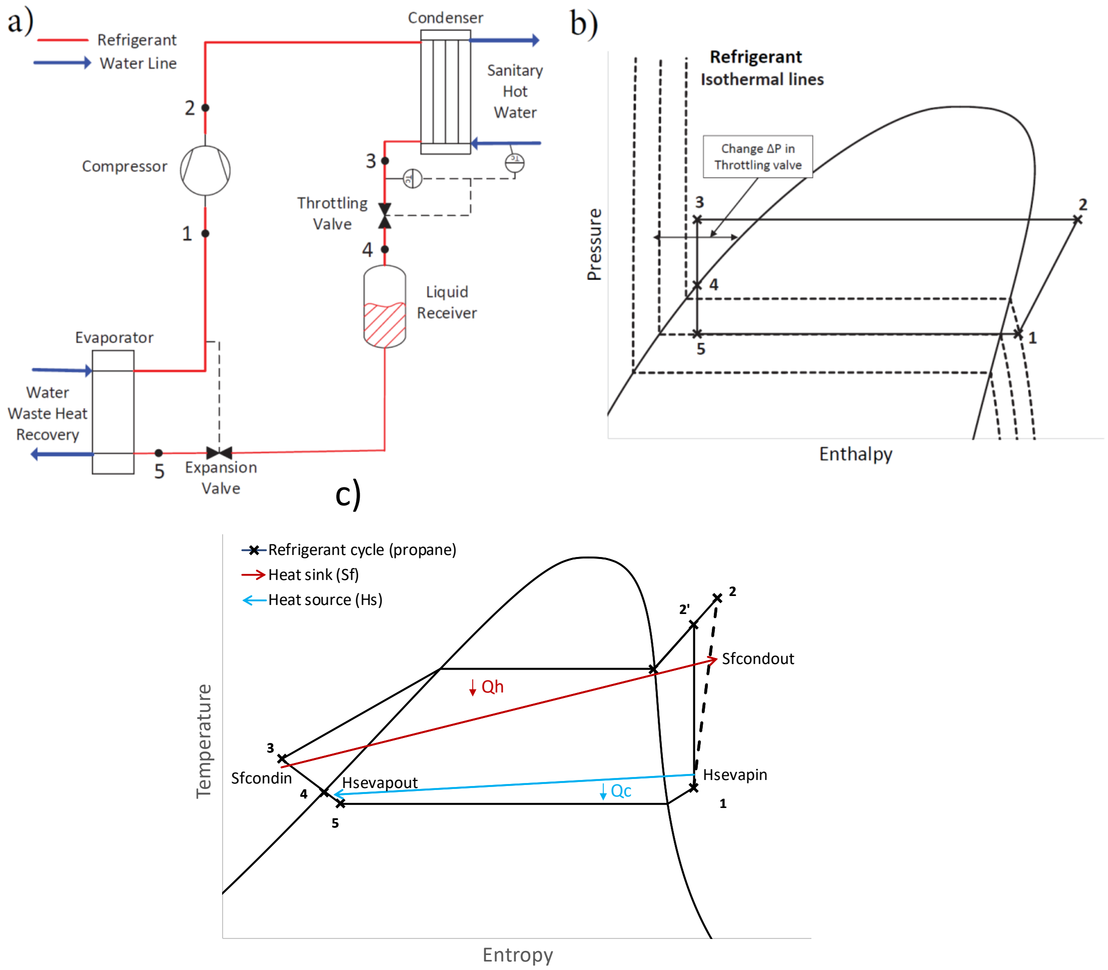

Figure 1: Heat Pump with subcooling at the condenser controlled by a throttling valve a) Scheme of the real system, b) P-h diagram and c) T-s diagram of the heat sink, heat source and refrigerant.

2.2.1. Description of the analyzed system

This work is done based on the water-to-water heat pump able to control the subcooling in the condenser and the superheat at the evaporator (see Figure 1). The heat pump is provided with two expansion devices: a throttling valve, located between the condenser and the liquid receiver, and an expansion valve, placed between the liquid receiver and the evaporator. The liquid receiver is used to accommodate the changes in the active charge in the system derived from the variations in the degree of subcooling at the condenser. The throttling valve is the active control component that allows setting the subcooling at the condenser independently from the external conditions. This is made as a consequence of the fact that the liquid receiver at outlet of the throttling valve forces the refrigerant to be in saturated liquid conditions. Therefore, a change in the pressure drop in the valve will produce a change in the subcooling at the outlet of the condenser."

The expansion valve is the active control component that allows setting the superheat at the evaporator outlet. Therefore, in this system, it is possible to control subcooling and superheat independently at any external conditions.

For more information about the system, please refer to (Pitarch et al., 2018).

Table 1 shows the characteristics of each component

Table 1: Components of the HP system

Component	Type	Size
Compressor	Scroll (2900 rpm)	29.6 m^3h^{-1}
Condenser	BPHE Counterflow	3.5 m^2
Evaporator	BPHE Counterflow	6 m^2
Liquid Receiver	-	7 l
Expansion Valve	Electronic EX-6	93 kW
Throttling Valve	Electronic EX-5	39 kW

2.2.2. Model Validation

A model of the heat pump has been developed using the commercial software IMST-ART (IMST-ART v3.80) in order to study the influence of subcooling and superheat at different external conditions. IMST-ART (Corberan et al., 2002) is a dedicated software for modeling heat pump systems as a whole, according to the state-of-the-art. It incorporates a number of sub-models representing the different parts of the heat pump: compressor, condenser, evaporator and expansion valve.

Compressor model

The compressor model used in IMST-ART is based on the manufacturer AHRI curves for the used compressor working with the used refrigerant.

Expansion valve

The expansion valve includes both, the throttling and the expansion valve according to Figure 1a and it is modeled as an isenthalpic pressure drop in order to fulfill the desired superheat.

Condenser and evaporator

The selected condenser has a horizontal port distance and vertical port distance of 60 mm and 470 mm respectively and 62 plates.

The selected evaporator has 120 plates and a horizontal port distance and vertical port distance of 50 mm and 466 mm respectively.

IMST-ART is able to consider most of the geometrical and operation parameters of current evaporators and condensers. The global solution method employed is called SEWTLE (Semi-Explicit method for Wall Temperature Linked Equations)(José M. Corberán, Pedro Fernández d, 2001).

The model of this heat pump was already validated in terms of heating COP, capacity and condensing pressure in (Pitarch et al., 2018).

The experimental results are based on the measures detailed explained (Pitarch et al., 2018). Table 2 collects the main sensors with the relative and absolute uncertainty intrinsic to the sensor. Please, for more information regarding to uncertainties and error studies, refer to (Pitarch et al., 2018).

Table 2: Sensors and uncertainties of the experimental results

Magnitude	Model	Relative uncertainty	Absolute uncertainty	Units

Pressure	*Differential 1151 Smart Rosemount*	*0.1256 % of Span*	*4.684E-04*	*bar*
	Differential P Siemens Sitrans P	*0.1417 % of Span*	*3.542E-04*	*bar*
	Differential P Setra	*0.25 % of Span*	*1.723E-03*	*bar*
	P 1151 Smart GP7 Rosemount	*0.1239 % of Span*	*2.602E-02*	*bar*
	P 1151 Smart GP8 Rosemount	*0.1547 % of Span*	*7.889E-02*	*bar*
	P 3051 TG3 Rosemount	*0.1351 % of Span*	*3.782E-02*	*bar*
Temperature	*Thermocouple T Type*		*1*	*K*
	RTD		*0.06*	*K*
Flow	*Coriolis SITRANS F C MASS 2100*	*0.29 % of Reading*		
	Magnetic SITRANS FM MAG5100 W	*0.36 % of Reading*		
Power	*DME 442*	*0.3 % of Reading*		

Figure 2 compares the model results with the experiments, for the relative irreversibility of each component and their respective exergy efficiency as well as the efficiency of the heat pump system. The results are presented as a function of subcooling for the external conditions at the condenser (Tw,c) from 10 ºC to 60 ºC, and at the evaporator an inlet water temperature (Tw,ei) of 20 ºC with a mass flow rate of 7000 kg/h. Table 3 collects the experimental test point used in the validation.

As it can be seen in Figure 2, the highest discrepancies are found for the relative irreversibility at the condenser and evaporator. Nevertheless, all the deviations are below 5%. The optimal subcooling (maximum exergy efficiency of the heat pump) takes place around 42 K for both, the experimental, and the model heat pump.

One should notice that, for this particular case (validation), the secondary fluid outlet temperature at the evaporator changes as a function of subcooling. The heat pump capacity is affected by the subcooling and the secondary fluid mass flow rate through the evaporator is fixed to 7000 kg/h. For the rest of the points showed in the results, the secondary fluid outlet temperature at the evaporator is fixed to a given value and the secondary fluid mass flow rate through the evaporator will depend on the heat pump capacity. In this way, it is easier to extend the results to other heat pump sizes and situations.

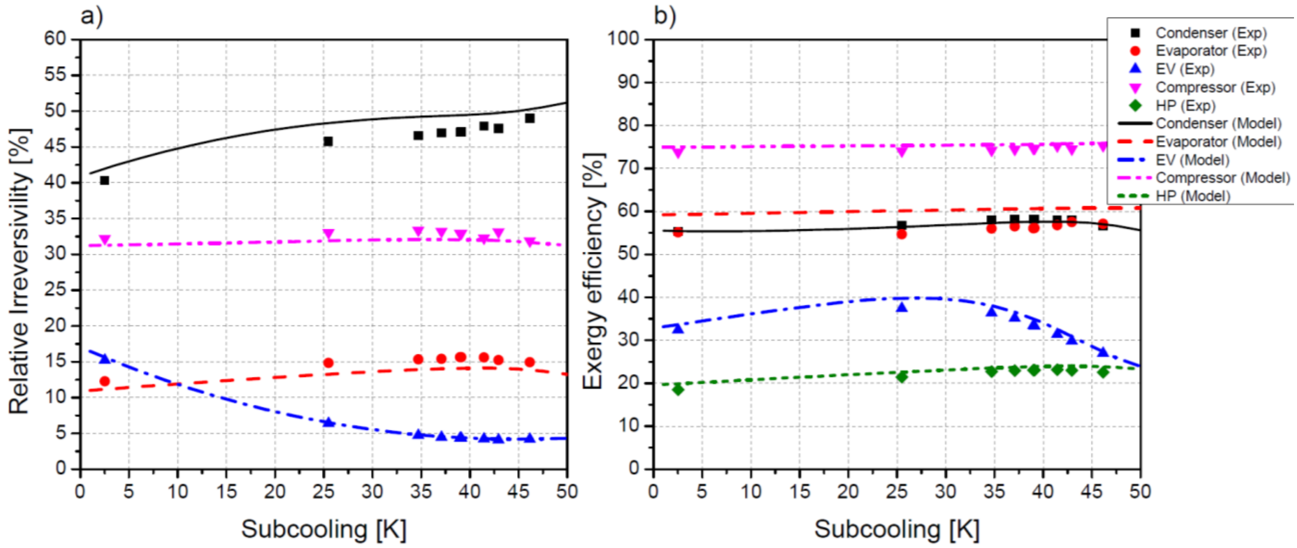

Figure 2. a) Relative irreversibility of each heat pump component. b) Exergy efficiency for each heat pump component and the heat pump system. Water temperatures at the inlet/outlet of the condenser 10 ºC/60 ºC. The inlet water temperature at the evaporator inlet 20 ºC, with a mass flow rate of 7000 kg/h

3. RESULTS AND DISCUSSION

This section is divided in four parts. The first one analyses the exergy destruction of each component and their respective exergy efficiency as a function of the temperature lift in the condenser for different subcoolings. The secondary fluid temperatures at the evaporator are fixed to 20 ºC and 15 ºC for the inlet and outlet, respectively. The refrigerant superheat at the evaporator outlet is fixed to 10 K. The second part focuses on the exergy destruction of each component and their respective exergy efficiency as a function of the temperature lift in the evaporator for different superheats. Based on the previous sections, the third part analyses the influence of both secondary fluids combined with subcooling and superheat. Finally, the last part supplies some examples of different applications where this study can be used in order to improve the efficiency of the system. Table 3 summarizes all the conditions used in the performed study.

Table 3: Temperatures of the secondary fluids at the evaporator and condenser superheat and subcooling used

	Evaporator		DT evap [K]	Sh [K]	Condenser		DT cond [K]
	Tin [ºC]	Tout [ºC]			Tin [ºC]	Tout[ºC]	
1st PART	20	15	5	10	10	60	50
					30		30
					50		10
					30	80	50
					50		30
					70		10
2nc PART	20	1	19	0-20	10	60	50
		5	15				
		10	10				
		15	5				
		5	15		50		10
		10	10				
		15	5				
3rdPART	20	5	15	0-20	10	60	50
		15	5		50		10

The outlet water temperature in the condenser has been selected in order to cover some applications were the use of heat pumps is not currently widespread. For instance, in domestic hot water production, or in the production of water up to 80 ºC that could be interesting for the substitution of boilers in systems using high temperature radiators as terminal units

3.1 Exergy analysis as a function of temperature lift in the condenser

In previous works it has been demonstrated that for a given temperature lift in the condenser secondary fluid and for a heat exchanger of infinite area there is an optimum subcooling which is found where the condenser have two pinch points where the temperature difference between both fluids is 0K. For heat exchangers of finite area this relation is approximately maintained but the temperature difference is higher than 0K. Nevertheless, the influence of the temperature lift in the condenser secondary fluid in the whole system is not obvious.

Figure 3 shows the relative irreversibility of each component and their exergy efficiency as a function of subcooling for different water inlet temperatures at the condenser. The water outlet temperature is fixed to 60 ºC and the evaporator water inlet and outlet temperatures are 20 ºC and 15 ºC respectively. Looking at the total exergy efficiency of the heat pump, it can be seen that the optimal subcooling when heating water from 10 ºC to 60 ºC (50 K water temperature lift) is around 42 K. The efficiency of the compressor and the evaporator remains almost constant with subcooling, the efficiency of the condenser slightly increases up to a maximum around 42 K. The maximum efficiency of the expansion valve is around 27 K of subcooling, having a lower efficiency at the optimum point of the heat pump. As it can be seen in the relative irreversibilities, the condenser is the component with the highest irreversibilities followed by the compressor, which account with around the 32% from the total irreversibility. This is in agreement with the fact that the curve of heat pump exergy efficiency follows the shape of the curve of the condenser efficiency.

When heating water from 30 ºC to 60 ºC, the optimal subcooling is around 26 K. The total exergy efficiency is higher than in the case 10 ºC to 60 ºC, this means that the heat pump is working closer to the maximum performance for this condition. The condenser efficiency has increased significantly (up to 80%) and it is practically constant until 26 K of subcooling. The efficiency of the evaporator and compressor have similar values to the observed previously and the expansion valve have a maximum efficiency approximately 26 K of subcooling. Now the compressor has the highest irreversivilities of the system, followed by the condenser.

When heating water from 50 ºC to 60 ºC, the optimal subcooling is found around 10 K. The condenser efficiency is higher than in the previous case, but decreases slightly with subcooling. The expansion valve has its maximum efficiency at 11 K of subcooling and it is the element accounting for higher relative irreversivilities after the compressor. This result is the opposite the result found for heating water from 10 ºC to 60 ºC.

Figure 4 shows the relative irreversibility of each component and their exergy efficiency as a function of subcooling for different water inlet temperatures at the condenser. The water outlet temperature is fixed to 80 ºC and the evaporator water inlet and outlet temperatures are 20 ºC and 15 ºC respectively. Therefore, it has the same conditions at the evaporator and the same water temperature lift at the condenser than in figure 3, but with a higher outlet temperature. Comparing both figures, it can be stablished that optimal subcooling depends mainly on the secondary fluid temperature lift in agreement with the experimental results presented in (Pitarch et al., 2018). From these figures,

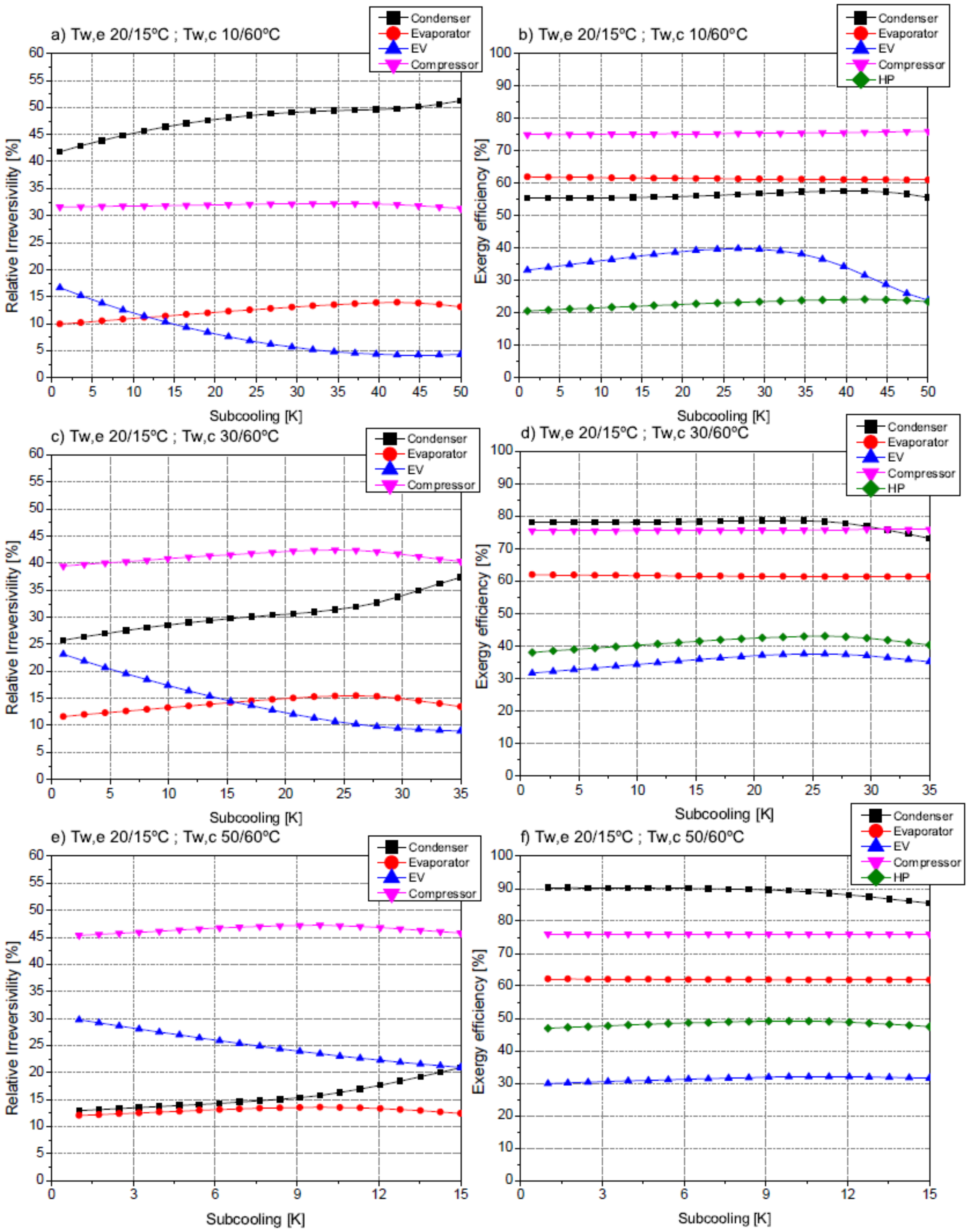

Figure 3. a), c) and e) Relative irreversibility of each heat pump component. b), d) and f) Exergy efficiency for each heat pump component and the heat pump system. Water temperatures at the inlet/outlet of evaporator 20 ºC/15 ºC, and water at the condenser outlet 60 ºC

it can be concluded that the most influent variable on the expansion valve irreversibility is the refrigerant outlet temperature at the condenser, which is directly related to the subcooling and it is limited by the inlet temperature of the secondary fluid at the condenser. The condenser exergy efficiency depends mainly on the temperature difference between the refrigerant and the secondary fluid at the condenser, hence it is affected mainly by the temperature profile of the secondary fluid (temperature lift) and refrigerant (subcooling). The evaporator is not affected significantly by neither the temperature lift nor the absolute value of the secondary fluid temperature in the condenser. Finally, the compressor efficiency depends on the condensing pressure, which is related to the absolute temperatures of the secondary fluid at the condenser. The compressor efficiency remains almost constant below the optimal subcooling, but it slightly decreases from this point due to an increase of the condensing pressure.

Table 4 contains the absolute irreversibility of each component for the studied external conditions working at the optimal subcooling. It also shows the exergy efficiency and heating COP improvement respect to the case without subcooling. One should notice the difference between the definitions of COP and exergy efficiency. The exergy efficiency gives information about the external conditions, since it measures how well the heat pump is working in comparison to the reference reversible cycle. Finally, it can be commented that for low and medium condenser temperature lift, the main component that contributes to the heat pump improvement with subcooling is the expansion valve, which efficiency always increases with subcooling up to a maximum, followed by the condenser, which efficiency only increases at a certain external conditions. For high condenser temperature lift, the condenser can have a higher influence than the expansion valve.

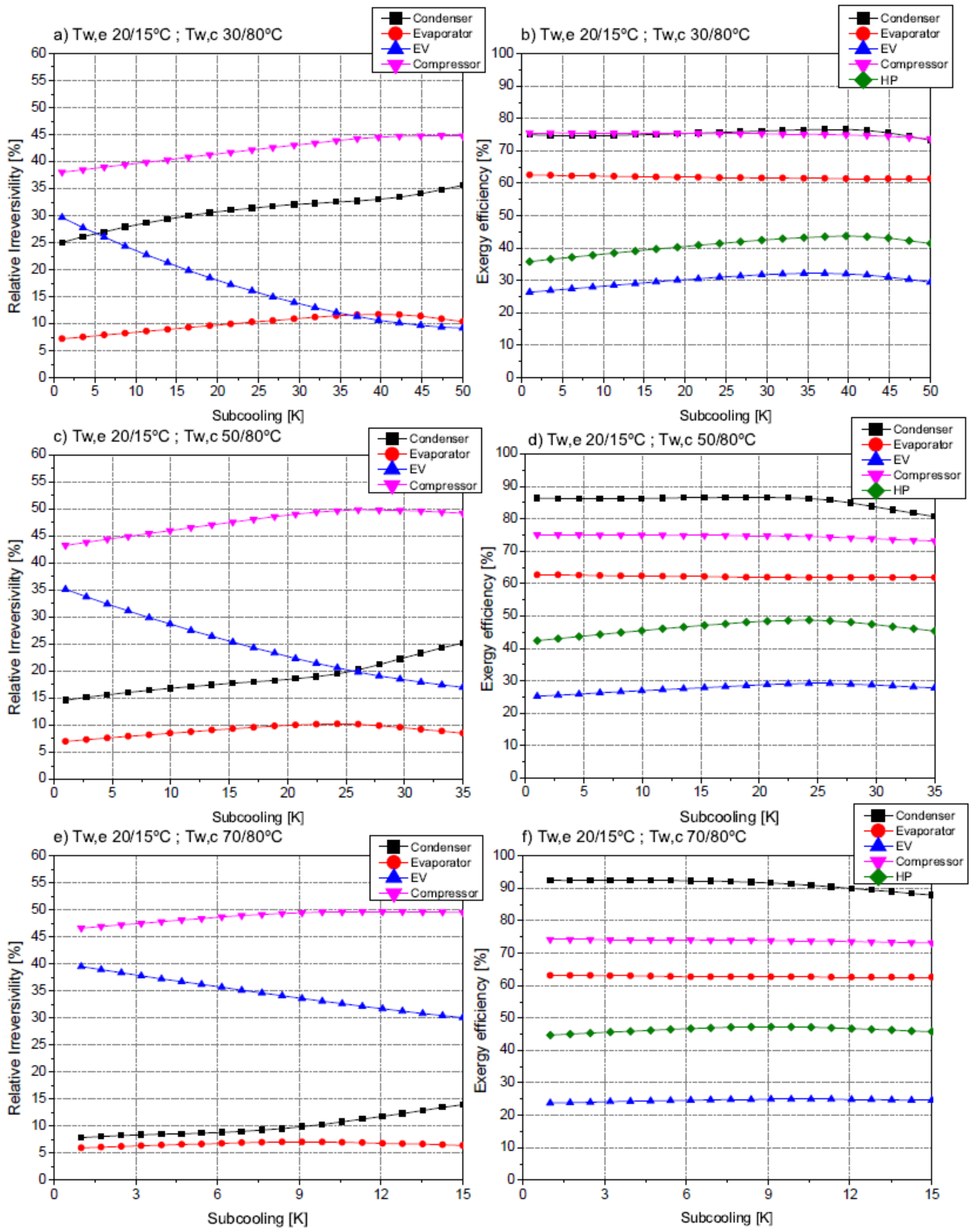

Figure 4. a), c) and e) Relative irreversibility of each heat pump component. b), d) and f) Exergy efficiency for each heat pump component and the heat pump system. Water temperatures at the inlet/outlet of evaporator 20 ºC/15 ºC, and water at the condenser outlet 80 ºC

Table 4: Exergy destruction of each component COP and heat pump efficiency improvement for different external conditions working at the optimal subcooling. Tw,e from 20 ºC to 15 ºC.

External condition	Sb_{opt} [K]	W_{comp} [kW]	$\dot{E}x_{des,cond}$ [kW]	$\dot{E}x_{des,evap}$ [kW]	$\dot{E}x_{des,EV}$ [kW]	$\dot{E}x_{des,comp}$ [kW]	$\dot{E}x_{des,tot}$ [kW]	ψ_{HP} [%] (0K Subcool-Opt. Subcool)	COP_h [-] (0K Subcool-Opt. Subcool)
10 ºC to 60 ºC	42.26	8.56	3.23	0.90	0.28	2.08	6.49	(20.7-24.2)	(4.46-5.54)
30 ºC to 60 ºC	26.05	8.80	1.60	0.78	0.51	2.12	5.01	(38.1-43.1)	(4.17-4.82)
50 ºC to 60 ºC	9.84	9.17	0.73	0.63	1.09	2.20	4.66	(47.0-49.2)	(3.82-4.02)
30 ºC to 80 ºC	39.68	11.25	2.09	0.74	0.68	2.82	6.33	(35.8-43.7)	(3.06- 3.87)
50 ºC to 80 ºC	24.26	11.68	1.17	0.61	1.23	2.97	5.98	(42.4-48.8)	(2.81- 3.30)
70 ºC to 80 ºC	9.11	12.32	0.64	0.46	2.18	3.21	6.50	(44.7-47.3)	(2.49-2.66)

Although it is necessary to study the particularity of each case, from the previous analysis the following rules of thumb can be applied:

1- For low inlet temperatures and high temperature lifts of the secondary fluid at the condenser, the components that account with higher irreversibility are the condenser and the compressor. Hence, these are a good place to look for improvement. For instance, it could be discussed if a refrigerant mixture with a high temperature glide and working with subcooling could improve performance in these cases.

2- For high inlet temperatures and low temperature lifts of the secondary fluid at the condenser, the components that account with higher irreversibility are the expansion valve and the compressor. In this case, it could be discussed the use of a liquid-suction heat exchanger or doing mechanical subcooling to improve the efficiency in the expansion valve. In (Yang et al., 2005) it is proposed to use an expander in order to recover part of the energy lost in the expansion valve in a transcritical system.

3.2 Exergy analysis as a function of temperature lift in the evaporator

In a similar way to the condenser, considering an evaporator of infinite area, the performance of the evaporator for a heat source of finite capacity and a temperature lift of ΔT is found for an evaporating temperature equal to Tw,e,in.+ΔT and a superheat equal to ΔT. When the evaporator has a finite area, the results are not so direct. This section will show a study in order to give some insight in that respect.

Figure 5a) shows the exergy efficiency of the heat pump as a function of superheat for four different temperature lifts of the evaporator secondary fluid. Tw,ei is fixed to 20 ºC and the water is heated from 10 ºC to 60 ºC in the condenser. For all conditions, the exergy efficiency slightly increases with superheat up to a maximum. From this point, the exergy efficiency decreases at a higher rate. For the condition Tw,e from 20 ºC to 15 ºC, the optimal superheat is 6 K, while the optimal superheat if the conditions at the evaporator are Tw,e from 20 ºC to 1 ºC is 16 K (with more than 3% of performance improvement). The degree of improvement with superheat is not so important as the one observed with subcooling. The dependency of the optimal superheat with the secondary fluid temperature lift can be seen better in figure 5b), where it has been included the results for a different conditions on the condenser (Tw,c from 50 ºC to 60 ºC).

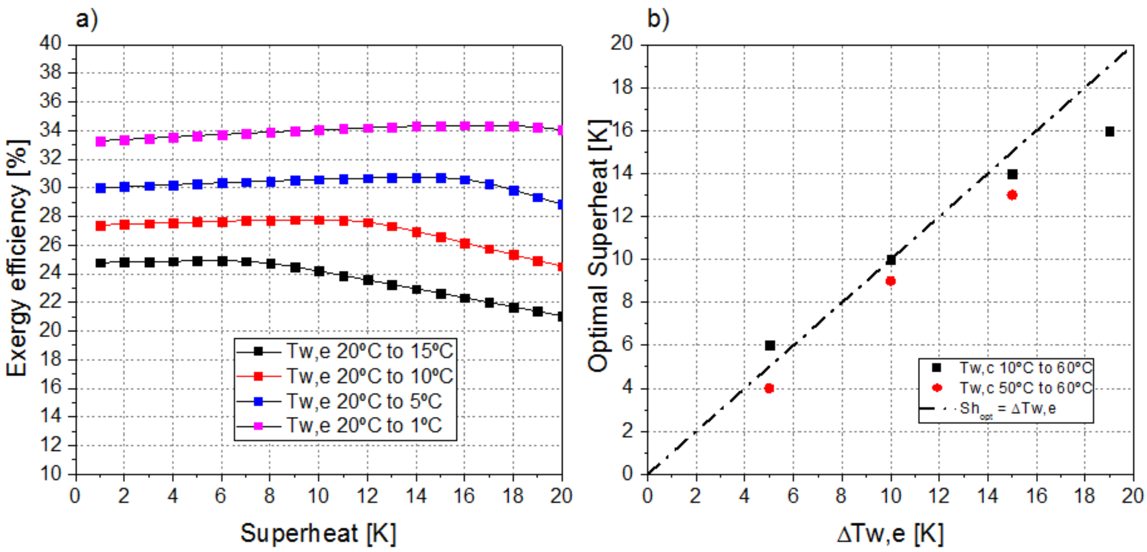

Figure 5. a) Exergy efficiency as a function of superheat, Tw,c from 10 ºC to 60 ºC b) Optimal superheat as a function of the secondary fluid temperature lift at the evaporator, Tw,ei=20 ºC

Figure 6 shows the relative irreversibility of each component and their exergy efficiency as a function of superheat and for different water outlet temperatures at the evaporator. Tw,ei is fixed to 20 ºC and the condenser water inlet and outlet temperature are 10 ºC and 60 ºC respectively. The maximum efficiency when cooling from 20 ºC to 5 ºC is given at 14 K of superheat (optimal superheat). At this point, the efficiency (exergy or COP) of the system has increased around 3% respect to the case without superheat (Table 4). Below the optimal superheat, the efficiencies of the condenser, the evaporator and the compressor slightly increase with superheat, being the compressor the one that experiences a bigger change. After the optimal superheat, the efficiency of condenser and compressor slightly decreases, while the efficiency of the evaporator decreases significantly. The efficiency of the expansion valve always decreases with superheat. The efficiency of the compressor and condenser has a similar trend for the other two conditions in the evaporator (20 ºC to 10 ºC and 20 ºC to 15 ºC). The compressor still has its maximum around 14 K or 15 K of superheat and the condenser maximum is now around 18 K, even though the optimal superheat of the system decreases with Tw,e (for instance, the optimal superheat for Tw,e from 20 ºC to 15 ºC is around 6 K). While the efficiency of the evaporator, contrary to the first condition, always decreases with superheat. This trend was similar to the observed for the condenser as a function of subcooling in the previous section.

Regarding to the relative irreversibility of each component, while the superheat is maintained below the optimum value, the superheat or the evaporator water temperature lift do not affect them significantly.

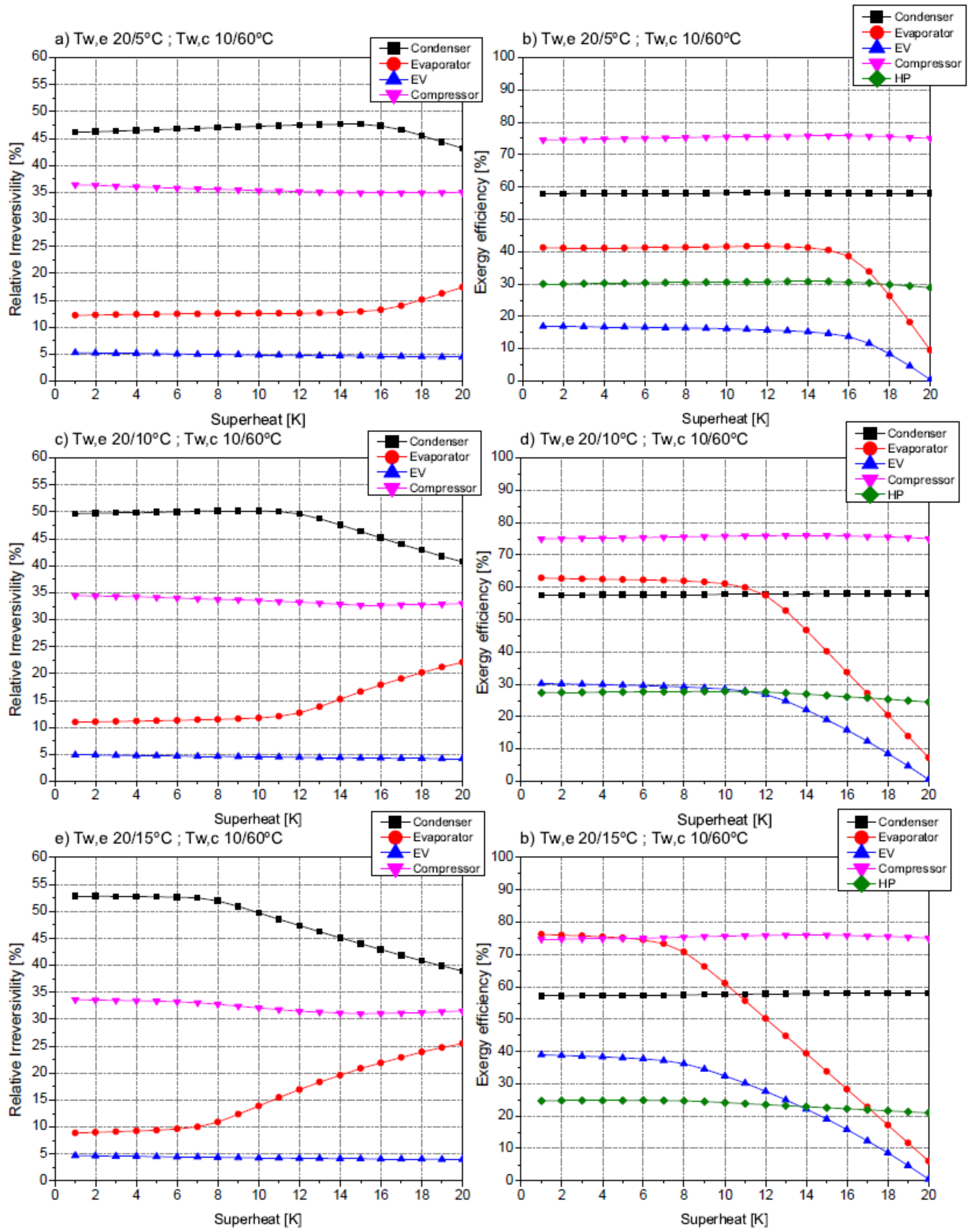

Figure 6. a), c) and e) Relative irreversibility of each heat pump component. b), d) and f) Exergy efficiency for each heat pump component and the heat pump system. Tw,ei= 20 ºC and Tw,c from 10 ºC to 60 ºC

Figure 7 is similar to Figure 6 but the condenser is heating the secondary fluid from 50 ºC to 60 ºC. The optimal superheat is slightly lower than in the previous case (10 ºC to 60 ºC). On the one hand, the evaporator, the compressor and the expansion valve efficiencies have a similar trend than in Figure 6 (case 10 ºC to 60 ºC at the condenser). On the other hand, the efficiency of the condenser always decreases with superheat for the three considered secondary fluid temperatures at the evaporator. This behaviour is different from the observed in Figure 6 and can be understood as:

The discharge temperature increases with superheat and, even though the condenser needs to use more area for the de-superheating region, the condensing pressure decreases. This is due to the internal pinch point, which is limiting the condensing pressure and it is relocated to a lower temperature when the condenser has a higher a higher area for de-superheating. The effect of superheat on the discharge temperature and the condensing pressure is the same for the cases of Figure 6 and 7. Nevertheless, as it has been seen, it affects the condenser differently: for high temperature lifts at the condenser (10 ºC to 60 ºC, Figure 6), the efficiency increases with superheat up to a maximum, while for low temperature lifts (50 ºC to 60 ºC, Figure 7), the efficiency decreases with superheat.

Regarding to the relative irreversibility of each component, below the optimal value of superheat, they are not significantly affected by superheat or the secondary fluid temperature lift at the evaporator. The major part of the irreversibility takes place in the compressor, followed by the expansion valve. Similar simulations have been carried out for Tw,ei of 10ºC and 30ºC obtaining similar conclusions to the ones presented in this article.

Therefore, it has been proved that an optimal superheat exists, and it depends mainly on the secondary fluid temperature lift at the evaporator but the temperature lift in the condenser should be taken into account too. The main component that contributes to the heat pump irreversivilities is the compressor. The condenser and evaporator efficiencies only increase at a certain external condition, while the expansion valve efficiency always decreases. It is worth it to remark that the degree of improvement is lower than the one achieved with subcooling.

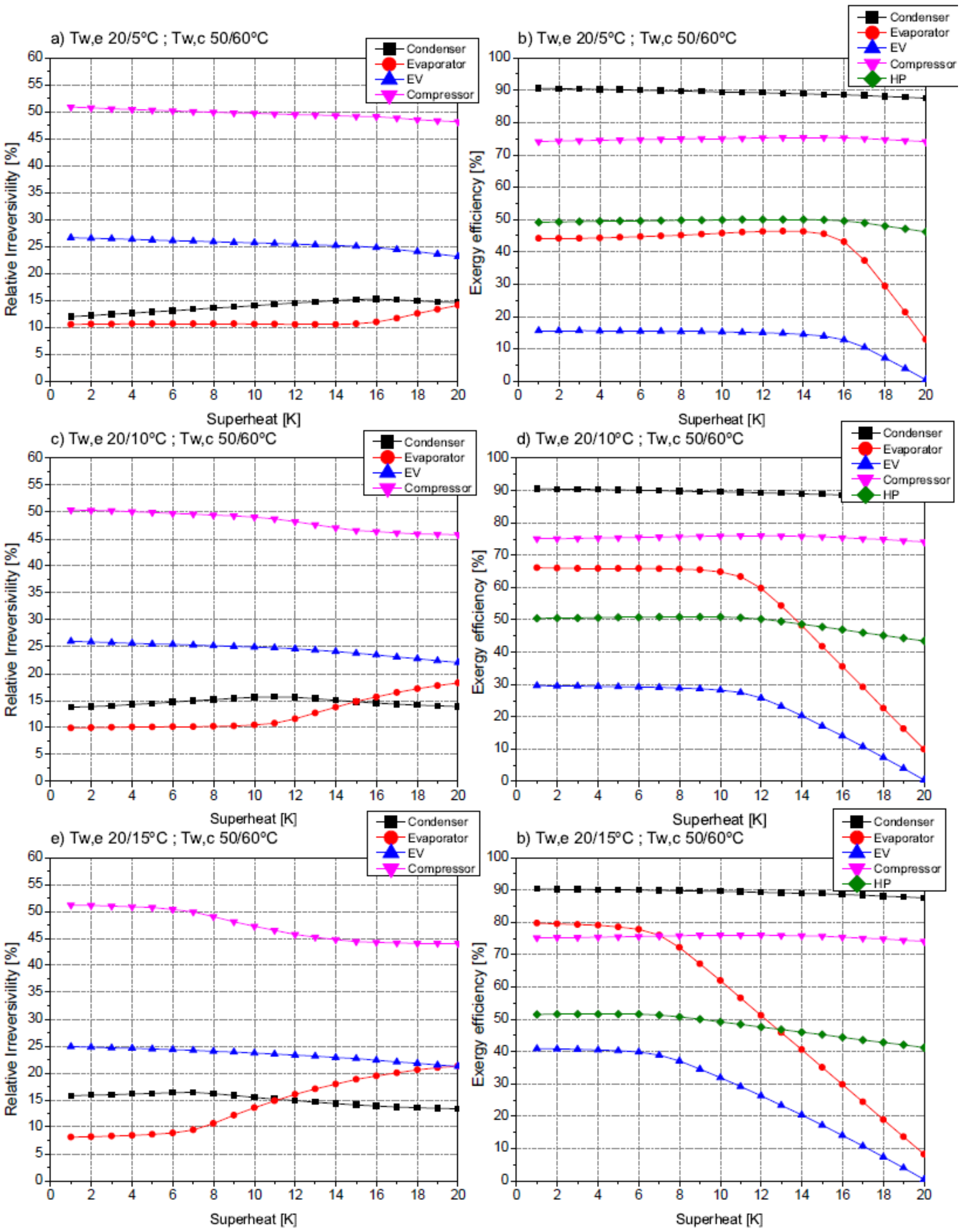

Figure 7. a), c) and e) Relative irreversibility of each heat pump component. b), d) and f) Exergy efficiency for each heat pump component and the heat pump system. Tw,ei= 20 ºC and Tw,c from 50 ºC to 60 ºC

Table 5: Exergy destruction of each component and heat pump efficiency improvement for different external conditions working at the optimal superheat and subcooling.

External condition	Sh_{opt} [K]	W_{comp} [kW]	$\dot{E}x_{des,cond}$ [kW]	$\dot{E}x_{des,evap}$ [kW]	$\dot{E}x_{des,EV}$ [kW]	$\dot{E}x_{des,comp}$ [kW]	$\dot{E}x_{des,tot}$ [kW]	ψ_{HP} [%] (0K SH-Opt. SH)	COP_h [-] (0K SH-Opt. SH)
20 ºC to 1 ºC & 10 ºC to 60 ºC	16	7.89	2.40	0.55	0.26	1.96	5.18	(33.3 -34.4)	(4.43-4.58)
20 ºC to 5 ºC & 10 ºC to 60 ºC	14	8.09	2.67	0.71	0.26	1.96	5.61	(30.0-30.7)	(4.80-4.93)
20 ºC to 10 ºC & 10 ºC to 60 ºC	10	8.42	3.05	0.72	0.28	2.04	6.08	(27.4-27.8)	(5.27-5.35)
20 ºC to 15 ºC & 10 ºC to 60 ºC	6	8.77	3.47	0.63	0.29	2.19	6.58	(24.8-24.9)	(5.72-5.75)
20 ºC to 5 ºC & 50 ºC to 60 ºC	13	8.86	0.65	0.47	1.12	2.19	4.43	(49.2-50.0)	(3.51-3.58)
20 ºC to 10 ºC & 50 ºC to 60 ºC	9	9.09	0.69	0.46	1.12	2.20	4.46	(50.4-50.9)	(3.88-3.92)
20 ºC to 15 ºC & 50 ºC to 60 ºC	4	9.33	0.73	0.38	1.11	2.30	4.52	(51.5-51.6)	(4.24-4.25)

Although it is necessary to study the particularity of each case, from the previous analysis the following rule of thumb can be applied:

- The optimum superheat in the evaporator increases as increases the temperature lift in the evaporator. This optimum superheat constitutes a value that the system never has to overcome as from this value there is an important reduction of system efficiency.

3.3 Exergy analysis of the whole system as a function of subcooling and superheat

In the previous two sections, it has been shown that there is an optimum subcooling and superheat in vapor compression cycles working between a heat sink and a heat source of finite capacity. It has been shown also that the optimum subcooling depends mainly on the temperature lift of the heat sink and the optimum superheat depends mainly on the temperature lift of heat source. However, it has been also seen that the heat sink has an influence in the optimum superheat and the heat source has an influence in the optimum subcooling. In order to understand properly the combined effect of both variables from the point of view of the system figure 8 shows the counterplot of the heat pump exergy efficiency as a function of superheat and subcooling for different conditions in the evaporator and the condenser. The efficiency degradation above the optimal subcooling is slightly higher than below this point. The efficiency degradation above the optimal superheat is considerably higher than below this point. The efficiency improvement obtained when superheat increases from 0 up to the optimum is not very significantly (this behavior is significantly different for subcooling). This might be the reason why most of the works and technical papers recommendations is to keep low superheat.

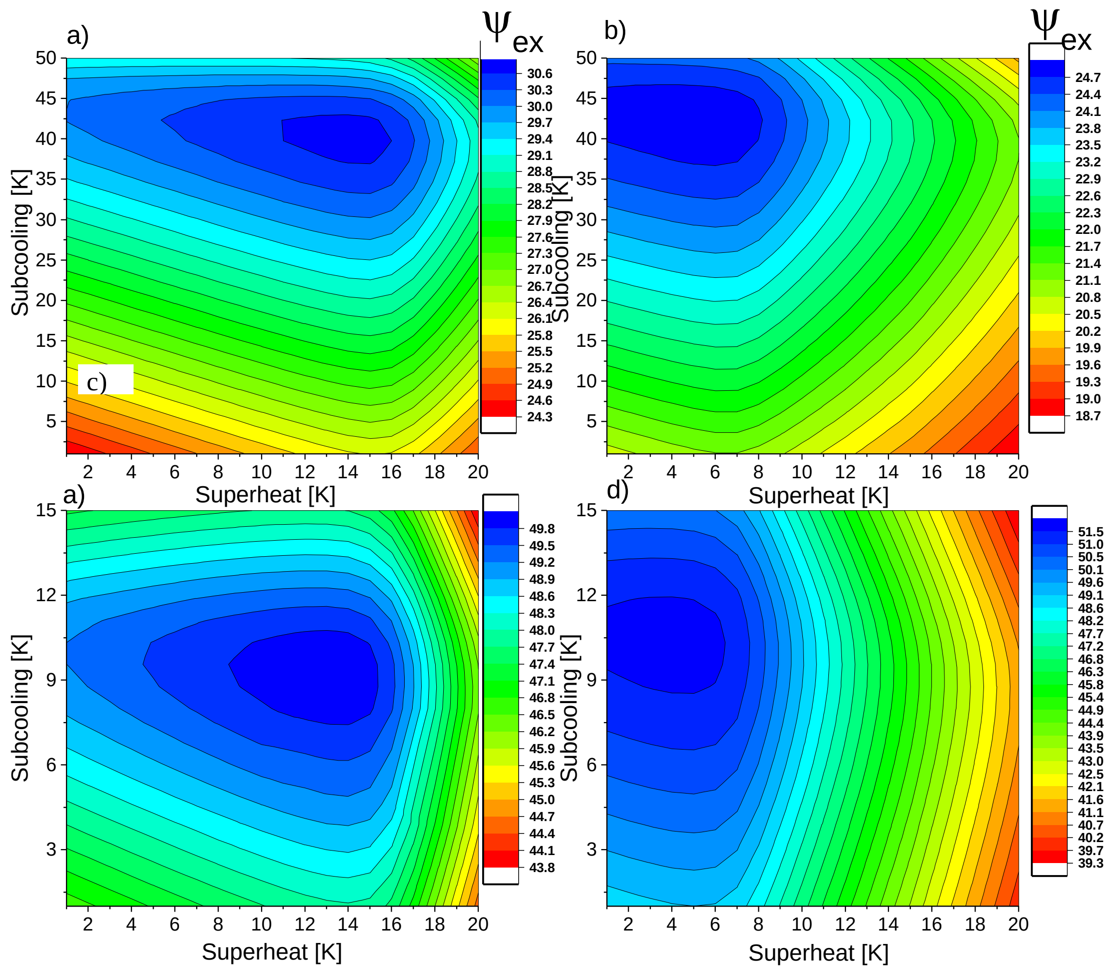

Figure 8. Heat pump exergy efficiency as a function of subcooling and superheat a) Tw,e from 20 ºC to 5 ºC and Tw,c from 10 ºC to 60 ºC; b) Tw,e from 20 ºC to 15 ºC and Tw,c from 10 ºC to 60

ºC; c) Tw,e from 20 ºC to 5 ºC and Tw,c from 50 ºC to 60 ºC; d) Tw,e from 20 ºC to 15 ºC and Tw,c from 50 ºC to 60 ºC

From Figure 8, it is seen that optimum subcooling depends on temperature lift of the heat source and heat sink, nowadays there are some authors that are focusing in developing models in order to predict the charge of the system and based on that be able to predict the subcooling (Bahman et al., 2018). With the approach shown here, that kind of models can be used in the other way around, the prediction of the charge of the system corresponding to the optimum subcooling which provides the optimum COP for a given temperature lift in the heat source.

3.4 Optimal operation for different applications

From all this work, the optimum superheat and subcooling of a single stage vapor compression system is obtained as a function of the external conditions (temperature lift of the source and sink).

From the heat pump design, these results are useful from the point of view of defining the setting of the subcooling and superheat of a vapor compression system depending on the application field.

Table 6: Optimal subcooling and superheat for different HP applications

Application	Heat source Tin-Tout [ºC]	Heat sink Tin-Tout [ºC]	Heat source DT [K]	Optimal superheat [K]	Heat sink temp.lift [K]	Optimal subcooling [K]
DHW production	15 - 10	10 - 60	5	~ 7 (min)	50	43
Dryer	47 - 40	40 - 75	7	~ 7 (min)	35	31
Heating	15 - 10	40 - 45	5	~ 7 (min)	5	8
Heat recovery	35 - 5	10 - 60	30	28	50	43
Heat recovery	20 - 5	10 - 60	15	16	50	43

As a rule of thumb, it is described that heat pumps for heating and cooling have an optimum subcooling between 5K and 10K and the superheat should be maintained as low as possible in order to optimize the performance of the systems. (~7 K due to temperature measure complexity of satured conditions and the safety of the compressor) (Pottker and Hrnjak, 2015). While these rules are close to optimal values for applications where the temperature lifts at the heat source or sink are small (typical HP applications until recent years such as heating applications), they are only valid for the specific application in which they were obtained. In fact, the results presented in the previous sections shows that the optimum subcooling depends mainly on the temperature lift of the secondary fluid in the condenser and that the superheat depends mainly on the temperature lift of the secondary fluid of the evaporator.

Table 6 shows some application examples of heat pump systems with different temperature lift in the secondary fluids and supply the optimum subcooling and superheat for them.

4. CONCLUSIONS

A heat pump has been analyzed component by component (exergy analysis) as a function of subcooling and superheat for several external conditions at the condenser and evaporator.

The main conclusions extracted from the subcooling analysis are:

- The components that contribute most to the heat pump exergy efficiency improvement with subcooling are the expansion valve and the condenser. The most influent one will depend on the external conditions in the condenser.
- For low secondary fluid inlet temperatures and high temperature lifts of the secondary fluid at the condenser, the components that account with higher irreversibility are the condenser and the compressor. Hence, if looking for improvements, these components are the first to look at. For instance, it could be discussed if a refrigerant mixture with a high temperature glide and working with subcooling could improve performance in these cases.
- For high secondary fluid inlet temperatures and low temperature lifts of the secondary fluid at the condenser, the components that account with higher irreversibility are the expansion valve and the compressor. In this case, it could be discussed the use of a liquid-suction heat exchanger or doing mechanical subcooling to improve the efficiency in the expansion valve.

Regarding to the superheat study, there is an optimal superheat, and it depends mainly on the secondary fluid temperature lift in the evaporator as subcooling does in the condenser. Other relevant results are:

- Working at the optimal superheat, the efficiency of the heat pump for the condition 20 ºC to 5 ºC in the evaporator (optimal superheat 14 K) increases 3% compared to the case without superheat. This improvement is lower than the one observed for subcooling for the same temperature lift.
- The compressor is the most influent component in the observed improvement of the heat pump performance with superheat.
- Below the optimal superheat, the relative irreversibility of each component is not affected significantly by superheat or by the secondary fluid temperature lift at the evaporator. Hence, the irreversibility distribution in the components will depend mainly on the conditions at the condenser.

Therefore, based on the temperature lift of the working sources and sinks of a heat pump, it seems necessary to design heat pumps able not only to control subcooling, which has already proved to have a high influence on the heat pump performance, but also able to control the degree of superheat, which can push a bit higher the heat pump performance. As it is show in the heat pump system presented in this paper, subcooling and superheat are the two variables that can be controlled at the will of the engineer in a heat pump of single stage cycle working between a heat source and a heat sink. The rest of the parameters of the system like condensing temperature, evaporating temperature and the like usually are more fixed once the compressor and the heat exchangers are selected.

ACKNOWLEDGEMENTS

Part of the results of this study were developed in the mainframe of the FP7 European project 'Next Generation of Heat Pumps working with Natural fluids' (NxtHPG). Part of the work presented was carried by Miquel Pitarch-Mocholí with the financial support of the Phd scholarship from the

Universitat Politècnica de València. Part of the work presented was carried by Estefanía Hervás Blasco with the financial support of a PhD scholarship from the Spanish government SFPI1500X074478XV0. The authors would like to acknowledge the Spanish 'MINISTERIO DE ECONOMIA Y COMPETITIVIDAD', through the project ref-ENE2014-53311-C2-1-P-AR "Aprovechamiento del calor residual a baja temperatura mediante bombas de calor para la produccion de agua caliente" for the given support.

REFERENCES

Ajuka, L., Odunfa, M., Ohunakin, O., Oyewola, M., Oyewola, M., 2017. Energy and exergy analysis of vapour compression refrigeration system using selected eco-friendly hydrocarbon refrigerants enhanced with tio2-nanoparticle. Int. J. Eng. Technol. 6, 91. doi:10.14419/ijet.v6i4.7099

Arat, H., Arslan, O., 2017. Exergoeconomic analysis of district heating system boosted by the geothermal heat pump. Energy 119, 1159–1170. doi:10.1016/j.enery.2016.11.073

Bahman, A., Ziviani, D., Groll, E., 2018. Development and Validation of a Mechanistic Vapor-Compression Cycle Model.

Cecchinato, L., Corradi, M., Minetto, S., 2010. A critical approach to the determination of optimal heat rejection pressure in transcritical systems. Appl. Therm. Eng. 30, 1812–1823. doi:10.1016/j.applthermaleng.2010.04.015

Chen, Y., Gu, J., 2005. The optimum high pressure for CO2 transcritical refrigeration systems with internal heat exchangers. Int. J. Refrig. 28, 1238–1249. doi:10.1016/j.ijrefrig.2005.08.009

Choi, J., Kim, Y., 2004. Influence of the expansion device on the performance of a heat pump using R407C under a range of charging conditions. Int. J. Refrig. 27, 378–384. doi:10.1016/j.ijrefrig.2003.12.002

Corberán, J.M., Martínez, I.O., Gonzálvez, J., 2007. Charge optimisation study of a reversible water-to-water propane heat pump Etude sur l'optimisation de la charge d'une pompe à chaleur ré versible eau-eau au propane. doi:10.1016/j.ijrefrig.2007.12.011

Corberan, J.M., Gonzalvez, J., Montes, P., Blasco, R., , 2002. Purdue 'ART' A Computer Code To Assist The Design Of Refrigeration and A/C Equipment R9-4 “.

Corberán, J.M., Fernández de Cordoba, P., Gonzalvez, J., Alias, F., 2001. Semiexplicit method for wall temperature linked equations (sewtle): a general finite-volume technique for the calculation of complex heat exchangers. Numer. Heat Transf. Part B Fundam. 40, 37–59. doi:10.1080/104077901300233596

Di Somma, M., Yan, B., Bianco, N., Graditi, G., Luh, P.B., Mongibello, L., Naso, V., 2017. Multi-objective design optimization of distributed energy systems through cost and exergy assessments. Appl. Energy 204, 1299–1316. doi:10.1016/j.apenergy.2017.03.105

Dinçer, I., Rosen, M. (Marc A.., 2012. Exergy : Energy, Environment and Sustainable Development. Elsevier Science.

Fernando, P., Palm, B., Lundqvist, P., Granryd, E., 2004. Propane heat pump with low refrigerant charge: design and laboratory tests. Int. J. Refrig. 27, 761–773. doi:10.1016/j.ijrefrig.2004.06.012

Hasan, A.A., Goswami, D.Y., Vijayaraghavan, S., 2002. First and second law analysis of a new power and refrigeration thermodynamic cycle using a solar heat source. Sol. Energy 73, 385–393. doi:10.1016/S0038-092X(02)00113-5

Hervas-Blasco, E., Pitarch, M., Navarro-Peris, E., Corberán, J.M., 2017. Optimal sizing of a heat pump booster for sanitary hot water production to maximize benefit for the substitution of gas boilers. Energy 127. doi:10.1016/j.energy.2017.03.131

Hjerkinn, T., 2007. Analysis of Heat Pump Water Heater Systems for Low-Energy Block of Flats. Master thesis at the Norwegian University of Science and Technology (NTNU), Dept. of Energy and Process Engineering. EPT-M-2007-24.

Inokuty, H., 1928. Graphical method of finding compression pressure of CO2 refrigerating machine for maximum coefficient of performance, in: The Fifth International Congress of Refrigeration, Rome. pp. 185–192.

Itard, L.C.., Machielsen, C.H.., 1994. Considerations when modelling compression/resorption heat pumps. Int. J. Refrig. 17, 453–460. doi:10.1016/0140-7007(94)90005-1

Jolly, P.G., Tso, C.P., Chia, P.K., Wong, Y.W., 2000. Intelligent Control to Reduce Superheat Hunting and Optimize Evaporator Performance in Container Refrigeration. HVAC&R Res. 6, 243–255. doi:10.1080/10789669.2000.10391261

Koeln, J.P., Alleyne, A.G., 2014. Optimal subcooling in vapor compression systems via extremum seeking control: Theory and experiments. Int. J. Refrig. 43, 14–25. doi:10.1016/j.ijrefrig.2014.03.012

Pitarch, M., Hervas-Blasco, E., Navarro-Peris, E., Gonzálvez-Maciá, J., Corberán, J.M., 2017. Evaluation of optimal subcooling in subcritical heat pump systems. Int. J. Refrig. 78. doi:10.1016/j.ijrefrig.2017.03.015

Pitarch, M., Navarro-Peris, E., Gonzálvez-Maciá, J., Corberán, J.M., 2018. Experimental study of a heat pump with high subcooling in the condenser for sanitary hot water production. Sci. Technol. Built Environ. 24, 105–114. doi:10.1080/23744731.2017.1333366

Pottker, G., Hrnjak, P., 2015. Effect of the condenser subcooling on the performance of vapor compression systems. Int. J. Refrig. 50, 156–164. doi:10.1016/j.ijrefrig.2014.11.003

Redón, A., Navarro-Peris, E., Pitarch, M., Gonzálvez-Macia, J., Corberán, J.M., 2014. Analysis and optimization of subcritical two-stage vapor injection heat pump systems. Appl. Energy 124, 231–240. doi:10.1016/j.apenergy.2014.02.066

Sarkar, J., Bhattacharyya, S., Gopal, M.R., 2004. Optimization of a transcritical CO2 heat pump cycle for simultaneous cooling and heating applications. Int. J. Refrig. 27, 830–838. doi:10.1016/j.ijrefrig.2004.03.006

Selbaş, R., Kızılkan, Ö., Şencan, A., 2006. Thermoeconomic optimization of subcooled and superheated vapor compression refrigeration cycle. Energy 31, 2108–2128.

doi:10.1016/j.energy.2005.10.015

Valencia, G., Beltrán, J., Romero, O., Cabrera, J., 2017. Comparative Evaluation of Different Refrigerants on a Vapor Compression Refrigeration System via Exergetic Performance Coefficient Criterion. Contemp. Eng. Sci. 10. doi:10.12988/ces.2017.7763

van de Bor, D.M., Infante Ferreira, C.A., 2013. Quick selection of industrial heat pump types including the impact of thermodynamic losses. Energy 53, 312–322. doi:10.1016/j.energy.2013.02.065

Xu, L., Hrnjak, P.J., 2014. Potential of controlling subcooling in residential air conditioning system. Purdue conference. Conference paper 1465.

Yang, J.L., Ma, Y.T., Li, M.X., Guan, H.Q., 2005. Exergy analysis of transcritical carbon dioxide refrigeration cycle with an expander. Energy 30, 1162–1175. doi:10.1016/j.energy.2004.08.007

Yang, M.-H., Yeh, R.-H., 2015. Performance and exergy destruction analyses of optimal subcooling for vapor-compression refrigeration systems. Int. J. Heat Mass Transf. 87, 1–10. doi:10.1016/j.ijheatmasstransfer.2015.03.085

Yataganbaba, A., Kilicarslan, A., Kurtbaş, İ., 2015. Exergy analysis of R1234yf and R1234ze as R134a replacements in a two evaporator vapour compression refrigeration system. Int. J. Refrig. 60, 26–37. doi:10.1016/j.ijrefrig.2015.08.010

Yumrutaş, R., Kunduz, M., Kanoğlu, M., 2002. Exergy analysis of vapor compression refrigeration systems. Exergy, An Int. J. 2, 266–272. doi:10.1016/S1164-0235(02)00079-1

LIST OF FIGURES