Study of different subcooling control strategies in order to enhance the performance of a heat pump

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ABSTRACT

The performance of vapor-compression systems working with subcritical refrigerants varies with the degree of subcooling. There is an optimal subcooling that maximizes efficiency. However, it depends on the operating conditions and the control of the system needs to be adapted. Most of the works available in literature are able to operate in optimal conditions only at the design point or if a system is designed to be able to adapt its subcooling, only complex control algorithms which usually are difficult to set and time-costly, are used. This work focuses on the study of the main variables influencing the optimal subcooling and analyzes two different control methodologies from the theoretical point of view. Based on the theoretical study a final control strategy is selected and tested experimentally. The reliability, stability and robustness of the selected strategy is experimentally demonstrated for a wide set of operating conditions.

KEYWORDS

Heat pump, natural refrigerant, subcritical refrigerant, optimal subcooling, control

HIGHLIGHTS

- Subcooling enhances heat pump performance up to an optimal value
- Two methodologies to control the optimal subcooling are studied
- Optimal subcooling control based on the secondary fluid temperature lift
- Optimal subcooling control based on a temperature approach
- Stability of the temperature approach control is experimentally proved

NOMENCLATURE

SHW: Sanitary Hot Water

COP: Coefficient of Performance ($COP=QW^{-1}$), [-]

COP_norm: COP/COP_max for the same conditions

DT: Temperature difference [K]

Q: heating capacity [W]

W: compressor power [W]

Sh: superheat

Subscripts

a: at the outlet of the heat exchanger

b: at the dew point of the heat exchanger

w: water

in: inlet

out: outlet

norm: normalized

comp: Compressor

cond: Condenser

evap: evaporator

1.INTRODUCTION

The growth in the awareness to respect the environment is reflected worldwide by means of an increase in the targets and policies to enhance the improvement of the energy sector towards a more sustainable way. Higher performance technologies, heat recovery, natural working fluids and electricity generated by renewables contribute to the reduction in CO2 emissions.

Heat pumps accomplish most of the current requirements for the heating and cooling fields. Hence, the interest of this technology is growing and the improvement of their efficiency would have an impact on society.

In order to optimize the efficiency of vapor compression heat pumps, most of the works used to focus mainly on heat exchangers designs, compressor's improvement, expansion process or working fluid election but the effect of the degree of subcooling on the performance has not been systematically investigated until recently. Possibly, the reason was that in the most common heat pump configurations, the liquid receiver is placed right after the condenser forcing the refrigerant to exit the condenser under saturated conditions leading to zero subcooling. However, in configurations where it is possible to have subcooling some authors demonstrate an increase of performance linked to it. For instance, an improvement of 2% in the efficiency is found for a refrigeration case working with ammonia with 4.66K of subcooling in (Jensen and Skogestad, 2007a). Moreover, a theoretical study of the potential performance improvement by the application of subcooling in an air to air heat pump for various refrigerants was done in (Pottker and Hrnjak, 2015a) resulting, for instance, in an increase of 12% for R744. Later on, in (Pottker and Hrnjak, 2015b) they experimentally demonstrated the enhancement of the COP due to subcooling for an air conditioning system working with R134a for an inlet air temperature of 35°C at the condenser and 5.3kW of cooling capacity. Furthermore, in hot water production applications with high temperature lifts and suboolings around 44K, an improvement up to 30% is experimentally proved in (Pitarch et al., 2017b). In spite of this fact, for each operating condition and size, there is an optimal subcooling that maximizes the COP as it can be seen in (Pitarch et al., 2017c) where two different configurations (a dedicated extra heat echanger to subooling and the production of the subcooling in the condenser) are compared. The dependency of the optimal value with the working conditions is experimentally demonstrated in (Pitarch et al., 2017d) where improvements range from 7% up to 30% based 10K to 50K of secondary temperature lift. A theoretical study of the optimal subcooling for different refrigerants and heat exchangers design is performed in (Pitarch et al., 2017a) observing a similar behavior among the subcritical refrigerants studied and the optimal subcooling main dependence on the water temperature lift for a given system.

For predefined external conditions, the subcooling of the system is controlled mainly by the amount of effective charge circulating thorugh the system as it can be seen in (Fernando et al., 2004) for propane. This type of control has some similarities with the control and optimality of the pressure in the gas cooler for transcritical cycles being an example of this the work done in [7] by the application to CO2 and propane or in (Jensen and Skogestad, 2007b) for CO2 and ammonia. Although there is a vast literature on the thermodynamic analysis of closed refrigeration cycles, few authors discuss the control of the effective charge circulating through the systems. Some discussions are found in textbooks like (Stoecker, 1998) but these mainly deal with more practical aspects. Most of the available literature focuses on CO2 transcritical systems, heating and cooling applications (Sarkar et al., 2004), water heating applications (Stene, 2007), the design of heat exchangers (Chen and Gu, 2005) or the determination of the optimal pressure in the gas cooler (Kauf, 1999) are only a few examples of this.

For subcritical systems, the main control approaches, based on its capability of adaptation can be divided in:

- <u>Passive control:</u> The use of an optimal refrigerant charge based on the design point which is charged before of the system operation (Corberá et al., 2007). In this case, always that the system is not working at the design point, the external conditions vary or after a certain time, the efficiency of the system would not be the maximum.
- <u>Active control:</u>
    - The use of control systems based on equations found after testing the heat pump under a wide set of conditions and for a specific refrigerant. In this type, the optimal subcooling varies based on the programmed fitting and allows the system to work with near-optimal conditions (Bauck Jensen, 2008) even though it implies the test of the heat pump under the given application conditions which could be expensive.
    - The use of complex controlling methods such as extremum seeking control (Koeln and Alleyne, 2014) (Zhu et al., 2016) that are based on adaptative strategies able to bring the system operating at the highest efficiency. These systems are based on an input and complex controllers that need first to be tunned in order to determine all the values of the process (frequency, amplitude,

perturbations, phases…). Once the parameters are set, its operation starts from a point and tries to converge into the solution close to the maximum COP value. This process does not allow a real understanding of the problem as well as it is time costly (around 3h are needed only to set the installation close to the optimal COP for static conditions, as it is stated in (Hu et al., 2015)).

Nevertheless, in most cases, either the optimal degree of subcooling is unknown (and only is used a certain degree of it) or it is not specified (Fernando et al., 2004) and in works where it is used, the optimal conditions are not stated clearly (Pottker et al., 2012).

Only a few works can be found in this line. In [7] the optimal subcooling variation with the temperature lift at the secondary fluid is investigated. This dependece allows the control based on an equation for a given size of the system. Furthermore, the potential of the temperature approach as a control variable is analyzed in (Jensen and Skogestad, 2007b) and deeply studied in [7], where, from a theoretical point of view, it was found that the optimal subcooling in an infinite area condenser takes place when there are two pinch points at the condenser: one inside (dew point) and another at the condenser outlet (approach) and for finite heat exchangers, a similar value of those leads to nearly-optimal operation conditions. Hence, the control of the system based on the temperature approach appears to be a good methodology even though, up to the knowledge of the authors, it has not been experimentally done yet.

In (Pitarch et al., 2017a), the authors analyzed under a theoretical point of view, the effect of the optimal subcooling for different refrigerants and in (Pitarch et al., 2017d) the authors experimentally demonstrated the improvement due to the effect of working with optimal subcooling for each operating and external condition in a water heating application. However, the system was maintained in steady state using a manual control system.

Therefore, according to the above literature, the subcooling problem has been evaluated by a few authors within the heat pump sector. The improvement of the heat pump performance by the implementation of a certain degree of it as well as the existance of an optimal subcooling has been demonstrated in those works for different applications. Nevertheless, it is worth to notice that the optimal subcooling depends on the external operating conditions, the heat pump design and the refirgerant used. Thus, a commercialized heat pump requires not only the knowledge of the existence of an optimum subcooling but also, the integration of a feasible control system to allow working in these optimal subcooling based on external conditions. Some alternatives has been proposed theoretically but the estrategies experimentally tested in the literature definitively have some open issues in order to be able to implement them in commercial systems. The control is still considered an opened problem.

In this work, two different subcooling control strategies have been evaluated theoretically. The two strategies allow the system to work under optimal conditions. Based on this analysis and the comparison of them, an optimum control strategy compromising complexity-efficiency improvement has been selected. Furthermore, the strategy has been experimentally implemented and tested under a wide range of operating conditions. From the experimental tests, robustness and the characteristic times of the control response have been determined.

The work is organized as follows: first, a description of the system and a theoretical analysis, are presented. Second, the results of the study for the two control strategies considered are detailed investigated. Third, the selected control is experimentally implemented and its stability, presented. Finally, the main conclusions extracted from the work are exposed.

2- METHODOLOGY

2.1 Studied system

This study is based on a water to water heat pump that uses the HC propane (R290) as refrigerant, tap water to be heated until SHW temperatures as secondary fluid and the recuperation of the wasted heat water at temperatures close to the ambient as a heat source.

Figure 1 shows the scheme of the heat pump studied in this work and its P-h diagram.

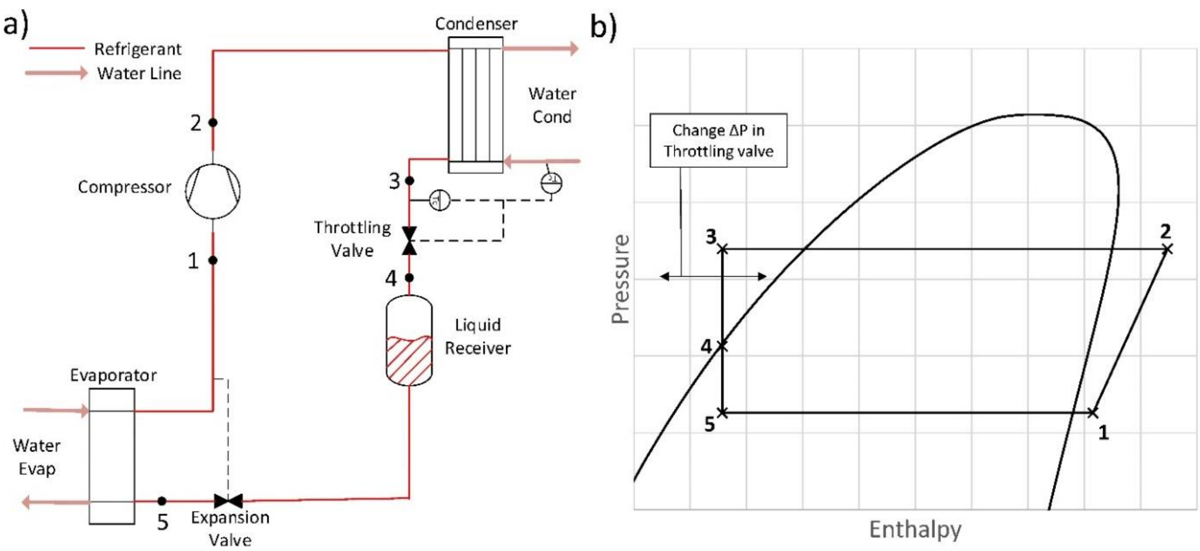

Figure 1: Water-to-water heat pump with subcooling controlled by a throttling valve. a) Scheme b) P-h diagram(Miquel Pitarch i Mocholí, 2017)

The main characteristics of this system, besides the typical components of a heat pump (condenser, evaporator, compressor and expansion valve(EV)), are the existence of a throttling valve (TV) and a liquid receiver placed respectively in series at the outlet of the condenser. In this configuration, the liquid receiver has mainly two functions: to accommodate the variation of the active refrigerant charge due to the modification of external/operation conditions and to ensure saturated liquid conditions of the refrigerant at the outlet of the throttling valve. Thus, it is possible to have a variable subcooling (for which a part of the condenser area is used) and to control it using the throttling valve

The proposed control mechanism for the subcooling will influence the control of the expansion valve (EV). This valve must ensure the desired degree of superheat. However, its pressure drop will not only depend on the condenser and evaporator pressures, but also on the degree of subcooling (pressure drop in the throttling valve). As it can be seen in Figure 2b, as the subcooling increases, the point 4 reduces its pressure and the expansion valve only introduces a small drop pressure (from 4-5) and vice versa.

For more details about the system, please refer to the doctoral thesis (Miquel Pitarch i Mocholí, 2017).

2.2 Theoretical analysis

In this work, two different control strategies based on the optimal subcooling are evaluated in deep. The first considered strategy is the implementation of a relation between the subcooling and of the secondary temperature lift as it was pointed to be directly related in (Pitarch et al., 2017a). The second strategy consists of the control by means of the temperature approach as it was theoretically proposed in (Pitarch et al., 2017a) and (Jensen and Skogestad, 2007b).

In order to understand better the main variables used in this work, Figure 2 has been included. Figure 2a shows the temperature profile of the secondary fluid (water) and the refrigerant working with three different subcoolings around the optimal point for finite heat exchangers. It can be seen that as the subcooling increases (reducing the temperature temperature approach, DTa), the temperature difference at the dew point (DTb), increases. Figure 2b shows the temperature differences (DTa and DTb) and the heating COP as a function of the subcooling.

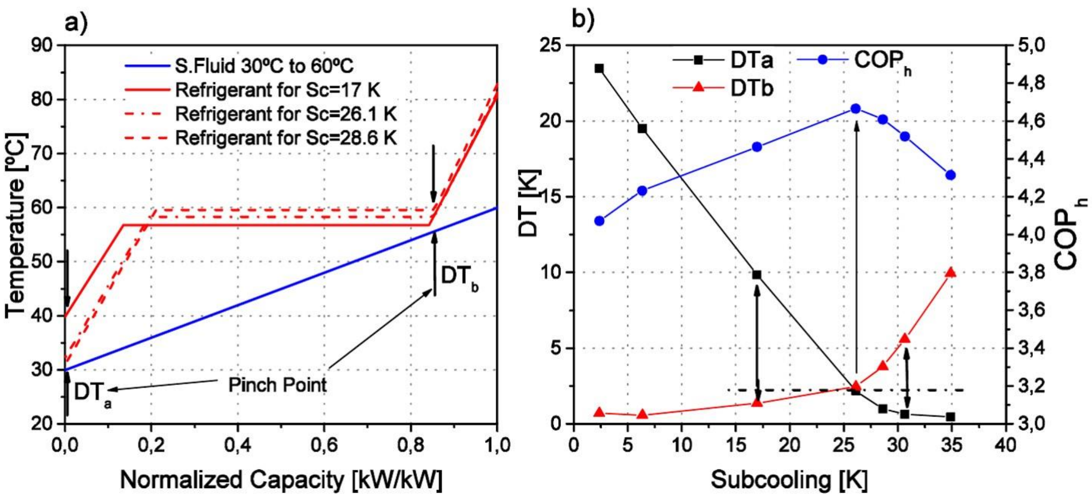

Figure 2: (a) Temperature profile vs. normalized capacity: Refrigerant R290 with different subcooling and the secondary fluid going from 30°C to 60°C (finite heat exchange area) (b) Pinch points values with different subcoolings.

For infinite heat exchangers, the optimal condition was theoretically found when two pinch points exist and they equal to zero, that is DTa=DTb=0. However, it was shown that although the difference was not too large under other operating conditions and finite heat exchangers, DTa≈DTb≠0 but not further investigations were done.

The second control strategy is based on this hypothesis. Hence, a deep analysis of the pinch points (DTa and DTb variation) in the optimal point for different operating conditions in finite heat exchangers has been done.

A model of the heat pump described in the previous section has been built using the IMSTart software (José Miguel Corberán, 2009). The model has been validated with experimental data accounting for a total of 61 points under a wide range of water temperature conditions collected in Table 1.

Table 1: Experimental water temperature range considered in the validation of the model

SHW condenser inlet temperature	10,30,50°C
SHW condenser outlet temperature	60°C
Water evaporator inlet temperature	10,20,30°C

Figure 3 represents the validation of the model for DTa, DTb and the heating COP. The experimental results are represented by dots, triangles, and crosses and the model results in lines for a water inlet temperatures of (a) 10°C (evaporating temperature 0°C), (b) 20°C and (c) 30°C and different water inlet temperatures at the condenser. The first number in the legend corresponds to the water inlet temperature at the evaporator, the second number to the water inlet temperature at the condenser and the third number, to the water temperature at the outlet of the condenser.

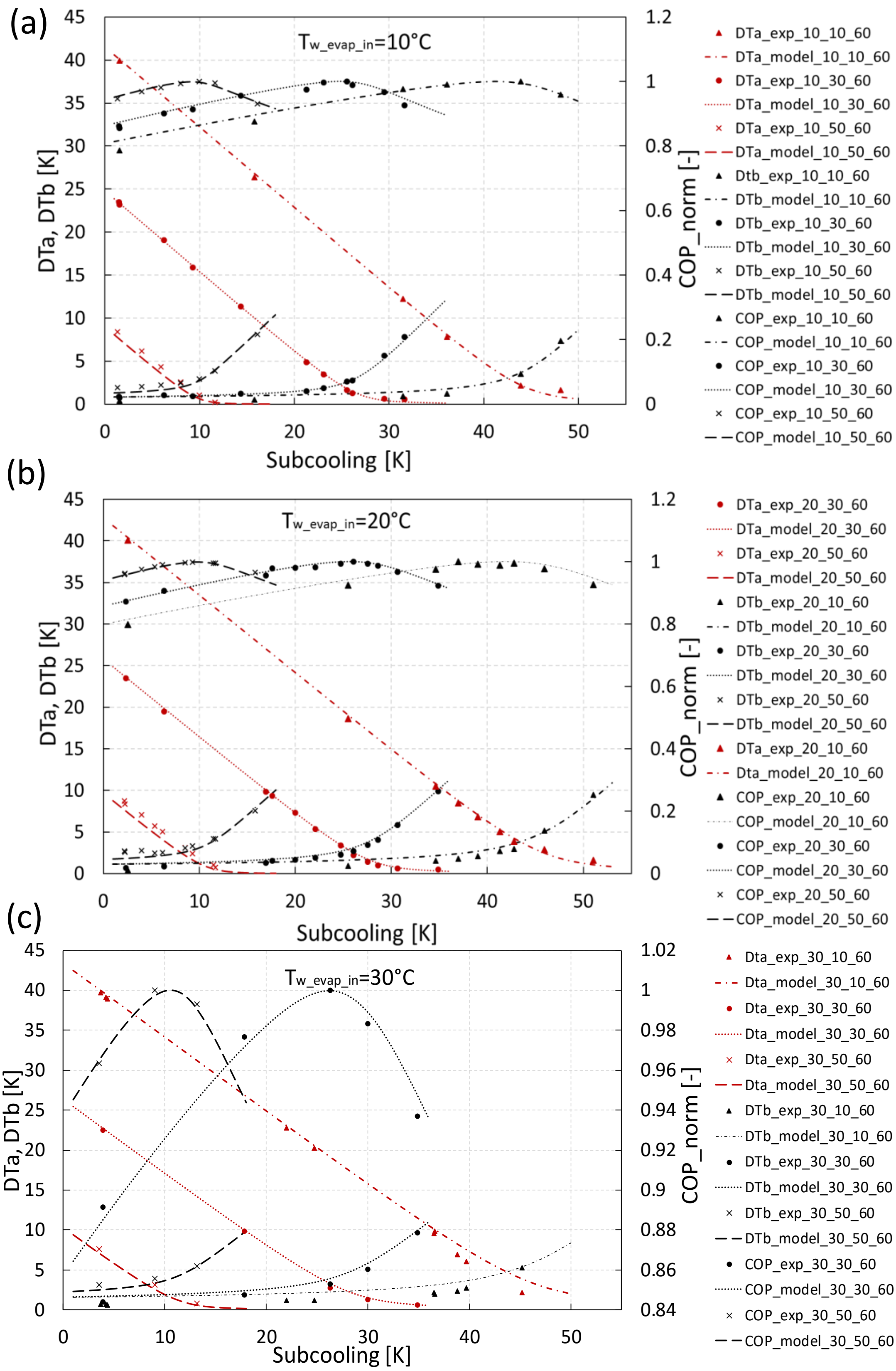

Figure 3: Model and experimental results of the normalized COP, DTa and DTb (a) for water

inlet temperature of 10°C (b) for water inlet temperature of 20°C and (c) for water inlet temperature of 30°C

According to the previous figures, it can be seen that the model captures the behavior of the real heat pump in all the conditions with an error lower than 4%.

Once the model has been validated, the number of points simulated around the maximum COP has been extended in order to be able to analyze with more detail the values of DTa and DTb around the optimal point (maximum COP). Since the study focuses mainly on the condenser, to avoid disturbances on the interpretation of the results, the efficiency of the compressor has been considered constant in all the cases.

Table 2 collects the water temperature ranges considered in this work. As it can be seen, with the mix of those temperatures, most of heating water applications range are covered.

Table 2: Water temperatures range considered in this work.

SHW condenser inlet temperature	10-50°C
SHW condenser outlet temperature	40-90°C
Water evaporator inlet temperature	10-30°C

3-RESULTS

3.1-Optimal Subcooling control based on a linear approach.

According to previous works, the subcooling can be theoretically controlled by means of a linear fitting function of the water temperature lift (Pitarch et al., 2017a). Thus, this possibility is evaluated for the considered points.

Figure 4 represents the secondary temperature lift (the difference between the water temperature at the outlet and the water temperature at the inlet of the condenser) and the optimal subcooling.

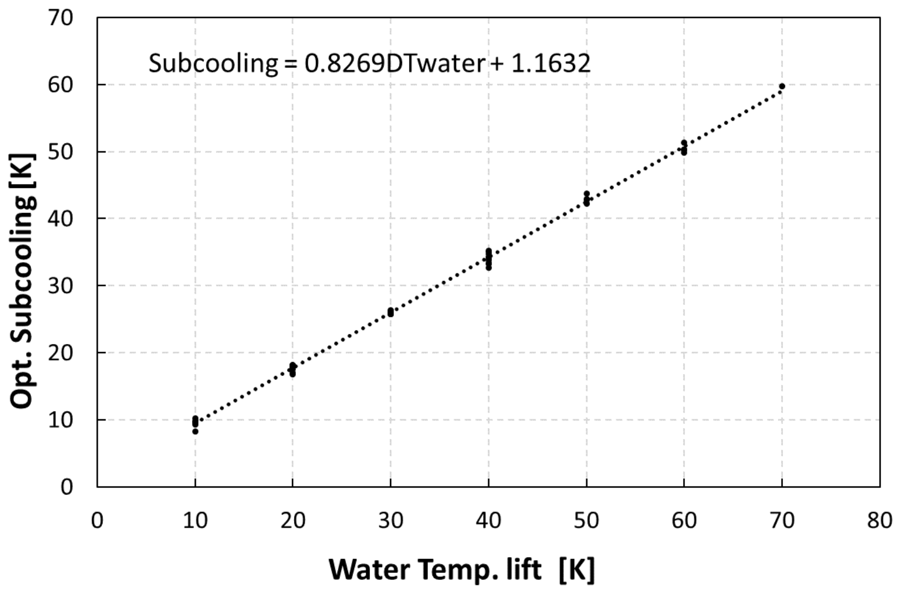

Figure 4: Optimal subcooling function of the water temperature lift for Propane. Linear fitting.

From the figure is deduced that a linear fitting function of the water temperature lift would lead to values very close to the optimal subcooling for each condition being the error lower than 1% in all the cases.

Figure 5 represents the COP calculated with the linear fitting and the optimal COP calculated with the software IMST-ART previously validated.

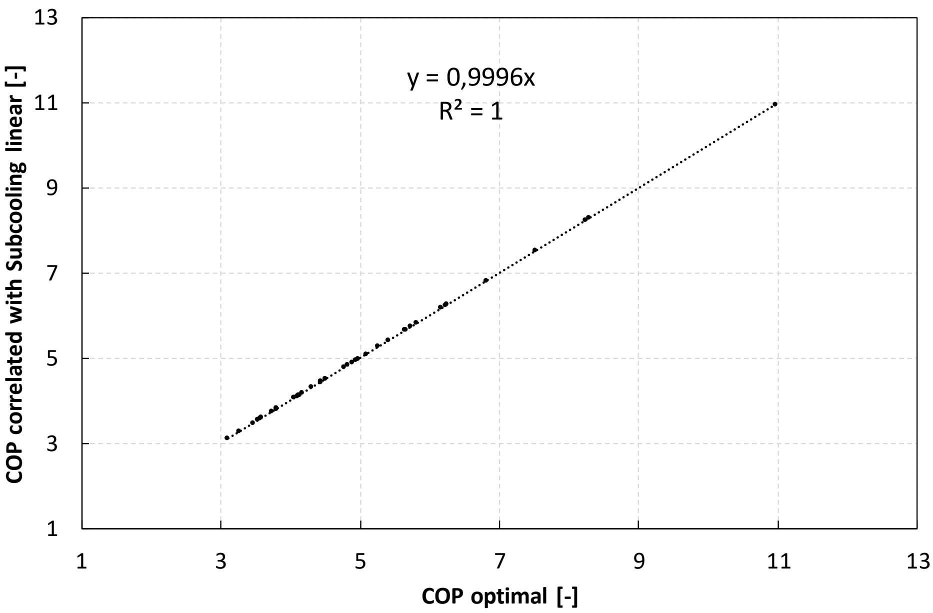

Figure 5: COP deviation from the optimal with the subcooling linear control

According to the figure, if a control of the subcooling is done by means of a linear fitting function of the secondary temperature lift (DTw), the deviation would be negligible (maximum of 0.04%) in all the cases. Nevertheless, this type of control must be adapted and the parameters adjusted based on the refrigerant used. In addition, the error incurred by using it may differ from one to another refrigerant. In fact, for refrigerants such R32, which critical point is closer to the condensing pressure, this control would result in higher errors than for instance, propane or R1234yf [5]. In spite of it, for a given size of the heat pump components, the parameters adjustment for the linear fitting control could be found, for most refrigerants, by the testing only 6/7 points.

Another interesting consideration when using a control methodology is represented by the analysis of the maximum possible deviation (accuracy of the methodology used) in order to obtain a certain maximum decrease of the efficiency. In this case, for each DTw (water temperature lifts) considered, the most critical external conditions from the studied points have been chosen. That is, for each DTw, the point that had the sharpest COP variation with subcooling. Thereafter, if 2% is chosen as the maximum desired decrease of the optimal COP at any condition, different deviation percentages from the optimal subcooling are allowed. For all the considered points and DTw, the minimum allowed deviation is around 10%.

Figure 6 represents the variation of normalized COP with the subcooling for the most critical conditions of each DTw. Black points represent the optimal subcooling obtained with the equation of Figure 4. Red crosses represent the normalized COP when a subcooling 10% lower than the optimal is applied. Green triangles represent the normalized COP when a subcooling 10% higher than the optimal is applied. Yellow dotted line represents the 2% loss of COP from the optimal.

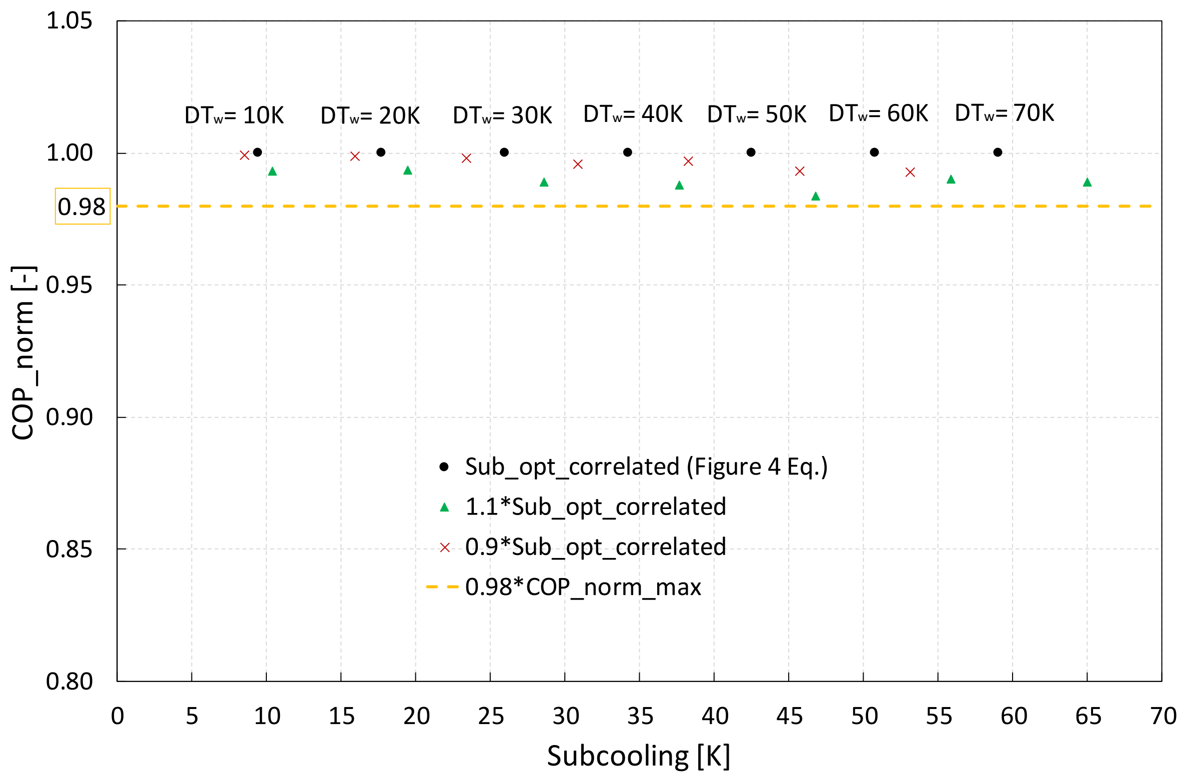

Figure 6: Optimal subcooling variation with the normalized COP based on the water temperature lift for different evaporating temperatures.

Two main conslusions can be derived from Figure 6:

- Deviations from the optimal subcooling until 10% would lead to near-optimal conditions by only decreasing in 2% the optimal COP in the worst cases. Thus, simplicity of the control approach could be a more valuable quality than precision.
- Smaller subcoolings than the optimal are preferable than higher.

3.2 Optimal Subcooling control based on the temperature approach

In (Jensen and Skogestad, 2007b) the temperature approach at the condenser was proposed and statistically studied as a good candidate to control the subcooling. However, the reason of this choice is not analyzed in deep. In (Pitarch et al., 2017a), a theoretical analysis of this variable based on infinite heat exchanger was done. The authors found that the optimal subcooling (maximum COP) takes place when two pinch points (one at the outlet of the condenser and another at the dew point) exist and they equal to zero. Due to the impossibility of the zero pinch point value in real heat exchangers, the values of DTa and DTb (nomenclature according to Figure 2) were cursorily investigated concluding that the optimal condition is found when two pinch points exist, that is DTa≈DTb. Finally, it was assumed DTa=DTb as a good approach but not further details are given.

In this part, a detailed study of the relationship between the optimal subcooling and the temperature differences DTa and DTb values for a real heat exchanger (finite), is done. First, the representation of the DTa, DTb, COP and subcooling values for all the considered is presented.

Figure 7 shows the variation of the temperature approach (DTa), the temperature at the dew point (DTb) and the normalized COP for different water temperature lifts (DTw) and the water inlet temperature at the evaporator of (a) 10°C, (b) 20°C and (c) 30°C. The vertical lines with the arrow indicate the normalized COP correspondent to DTa=DTb.

For the three cases (a), (b) and (c), the curve represented above is the variation of the normalized COP with subcooling for different DTw while the curves represented below belong to DTa and DTb variations with subcooling. Specifically, DTa decreases as subcooling increases and DTb increases with the increase of subcooling. The yellow horizontal line corresponds to the minimum normalized COP obtained under the assumption of the equality DTa=DTb considering all the points and DTw. The COP normalized when DTa=DTb is marked with the line ended in an arrow for each case.

Dotted lines, solid lines or dash lines represent different conditions at the condenser secondary fluid with the particularity that they are grouped by temperature lifts (for instance, in Figure 7a, a water inlet evaporator temperature of 10ºC is given for DTw=10K, solid line corresponds to the 30ºC of water inlet temperature at the condenser and 40ºC as water outlet temperature at the condenser while the dash line belongs to 10ºC (evaporator water inlet temperature), 50ºC of water inlet temperature at the condenser and 60ºC of water outlet temperature). It is worth it to notice that the important variable to consider is the difference between the inlet/outlet conditions of the secondary fluid at the condenser (DTw) rather than the values.

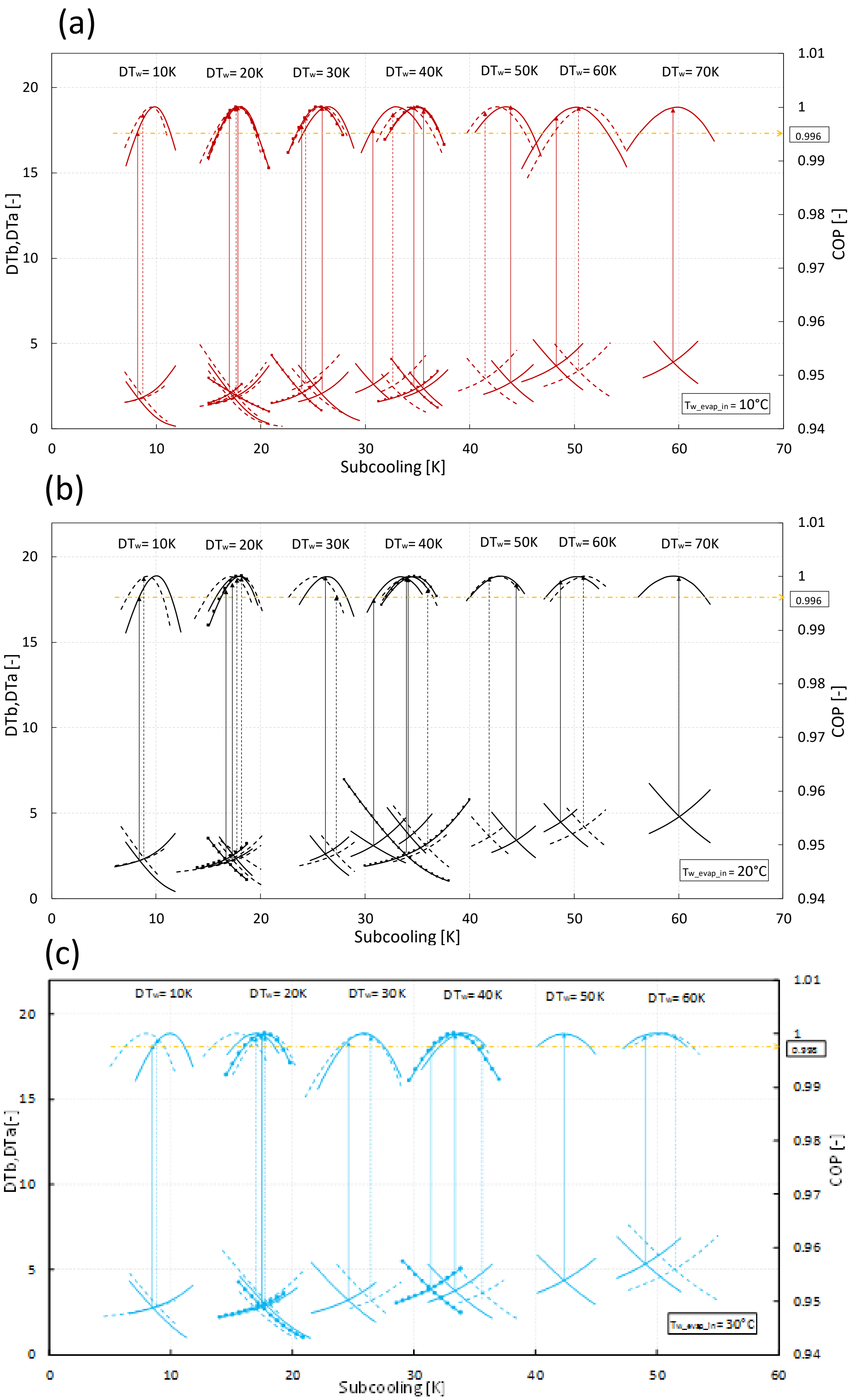

Figure 7: DTa, DTb and normalized COP variation with the subcooling for different water

temperature lifts and the water temperature at the inlet of the evaporator of (a) 10°C (b) 20°C and (c) 30°C. Vertical lines with the arrow indicate the normalized COP correspondent to DTa=DTb. Yellow horizontal line corresponds to the minimum normalized COP obtained under if DTa=DTb.

As it can be seen in Figure 7, the approximation of considering in real cases (finite heat exchangers) the equality of both terms would deviate from the optimum the results only 0.4% in the worst cases. Therefore, the theoretical result found by (Pitarch et al., 2017a) in which the optimal subcooling takes place when DTa=DTb for infinite heat exchangers can be applied for finite heat exchangers for a wide range of working conditions not compromising the optimal efficiency significantly. Thus, for a given heat exchanger size, the assumption that the optimal subcooling occurs when the temperature approach and the temperature at the dew point would be equal could in practice be considered as a general design criteria, at least for a heat exchanger that has been designed considering the subcooling area and a temperature approach lower than 5K, further studies are required in order to extend this behavior to more general situations.

Figure 10 represents, the variation of the normalized COP with the temperature approach (DTa) for some of the analyzed points. Only a few points are represented in order to see the figure clearly. It should be noticed that the worst cases (the cases with higher dispersion) are represented. The first number of the legend corresponds to the water inlet temperature at the evaporator (the evaporating temperature is 10K lower than this value), the second to the water inlet temperarture at the condenser and the third, to the water outlet temperature at the condenser. Red lines correspond to points with 10°C of water temperature at the inlet of the condenser, black lines to 20°C and blue lines to 30°C. Yelow line is the average DTa and the yelow dotted lines correspond to the maximum deviation from optimal conditions if the average value is assumed in all the cases.

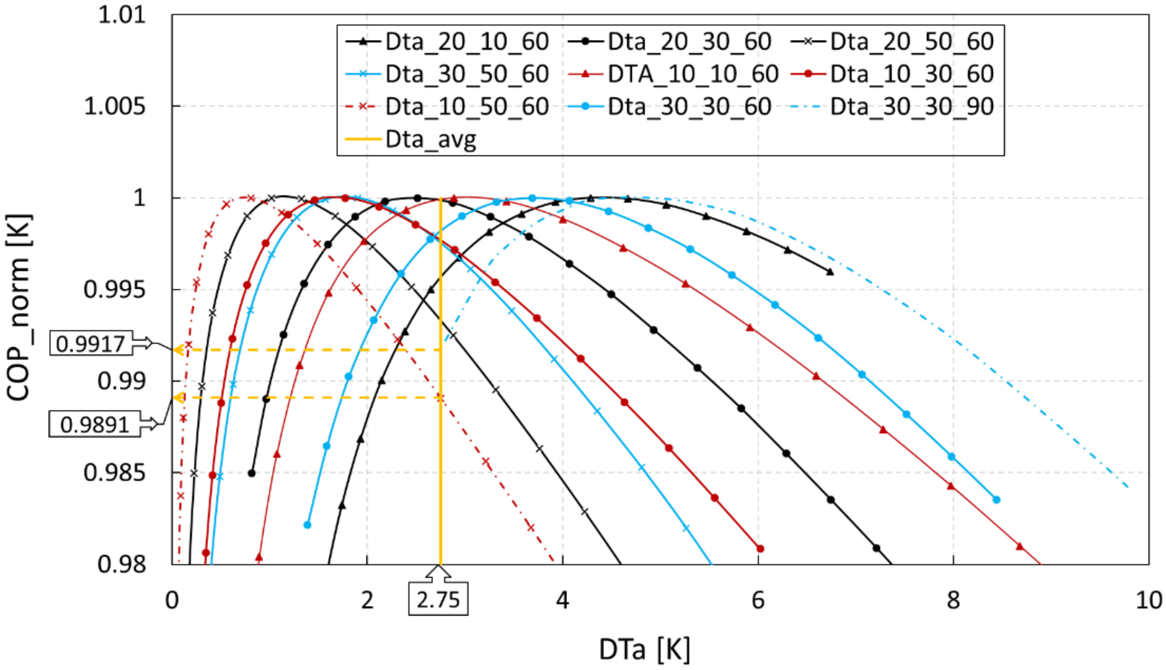

Figure 8: Temperature approach that corresponds to the optimal subcooling variation with the normalized COP based on the water temperature lift for different evaporating temperatures. The first number of the legend corresponds to the water inlet temperature at the evaporator (the evaporating temperature is 10K lower than this value), the second to the water inlet temperarture at the condenser and the third, to the water outlet temperature at the condenser. Red lines correspond to points with 10°C of water temperature at the inlet of the condenser, black lines to 20°C and blue lines to 30°C. Yelow line is the average DTa and the yelow dotted lines correspond to the maximum deviation from optimal conditions if the average value is assumed in all the cases.

As it can bee seen in the Figure, the optimal values of DTa differ as maximum approximately in 3K (from ~1.5K to ~4.5K) for the considered range of external conditions. Furthermore, the maximum performance decrease that would incur the consideration of a constant temperature approach for all the cases would be slightly higher than 1%. In this case, for any considered value of DTa from 2-4, the deviation from the optimal COP would be lower than 2%, what demonstrates that for DHW water applications, the control of the subcooling by means of the temperature approach seems to be reliable.

In this work, a constant value of the temperature approach (equal to the average value of all the analayzed points, that is, equal to 2.75K) has been considered as a control setting point for the installation regardeless the external conditions.

Figure 9 represents the COP calculated with the constant temperature approach (DTa) of 2.75K and the theoretical optimal COP calculated.

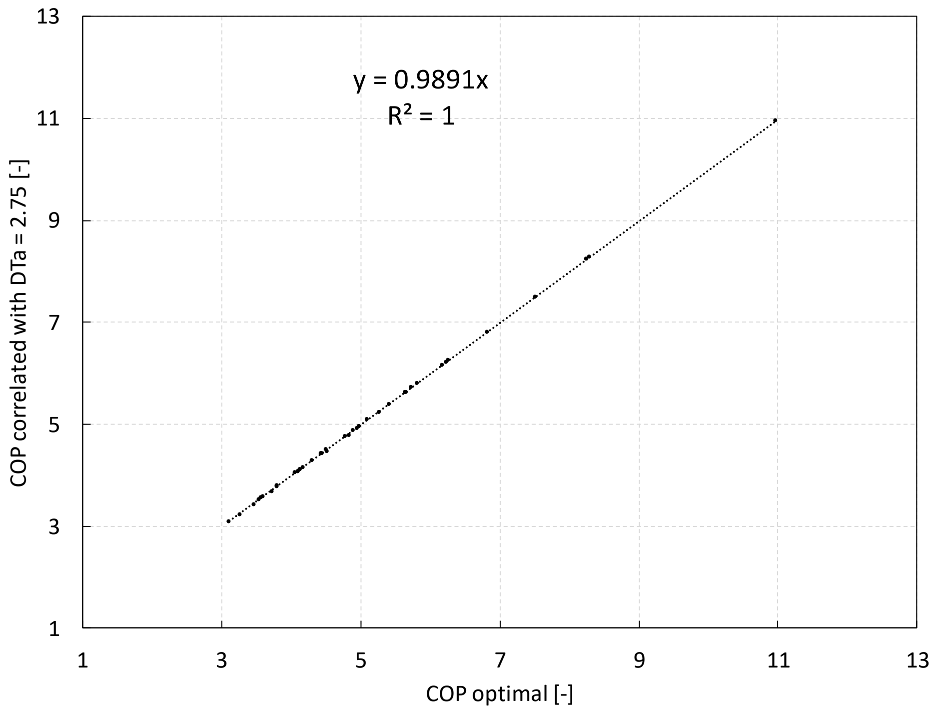

Figure 9: COP deviation from the optimal with the constant temperature approach control.

As it can be observed in the figure, if a control of the subcooling is done by means of a constant value in the temperature approach, the maximum deviation is 1.09% in all the cases.

Therefore, given condenser size and refrigerant, a good methodology in order to control the optimal subcooling would be to set a constant temperature approach at the outlet of the condenser independent of the external operating conditions. This methodology has the advantages of being simple to implement and accurate in order to operate under near-optimal conditions.

From the performed analysis, two subcooling control methodologies have been theoretically studied. Based on the results, the next part of the work will be focused on implementing the control of the subcooling from the temperature approach in a test bench. This potential strategy has not been evaluated in a real system up to the knowledge of the authors. Thus, some effects not considered on the previous theoretical analysis like the characteristic times of this control, the robustness of the strategy, the control coupling effects between subcooling and superheat and other possible factors that could appear in real systems is tested in the following part.

3.3 Stability control

In the configuration of Figure 1, the control of the superheat is done by means of the expansion valve (EV) while the throttling valve (TV) is in charge of the subcooling control.

Figure 10 is the control scheme used in the prototype experimentally tested.

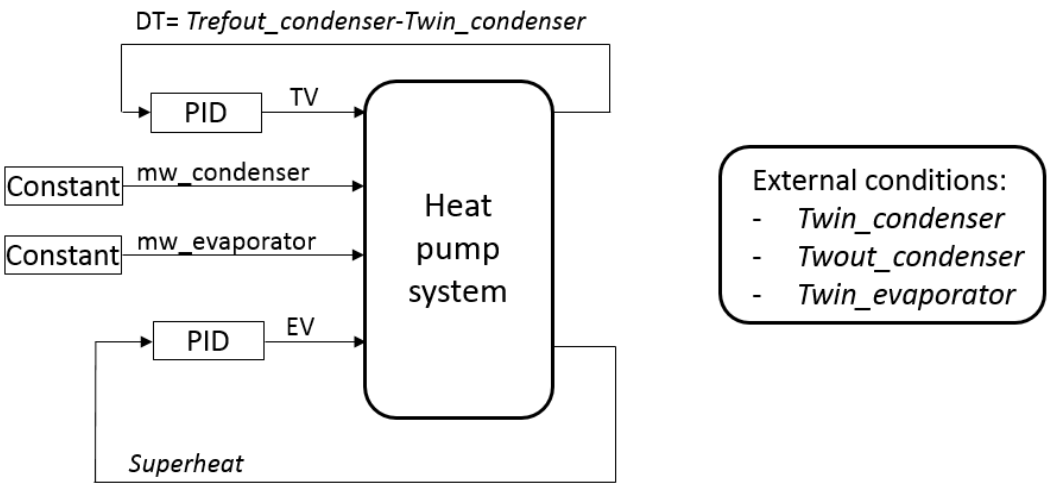

Figure 10: Heat pump control scheme

The testing conditions are attained by a certain water mass flow in the evaporator and the condenser which is defined before the experimental measurement. The control of the superheat is already automatically done. Thus, this section focuses on the control of the subcooling. In this case, the control is done by means of a PID (KS 90-1 PID from GmbH (PMA Prozeß- und Maschinen-Automation GmbH, n.d.)) that regulates the opening of the TV in Figure 1 in order to maintain a set point of DT. Only two temperature sensors were needed (one at the refrigerant condenser outlet (input 1, x1) and another at the water condenser inlet (input 2, x2)), in this case two thermocouples were used, and the PID controls the difference between both temperatures, DT (process value x1-x2). The three PID signals are connected as voltage signals (0-10V) and the control algorithm is so called "*on/off controller or signaler with one output*". The method of controller operation is inverse based on the process value.

The operating procedure of the PID is based on the achievement of a setting point in the temperature approach and the correspondent actuation over the throttling valve. That is, based on Figure 1, if the throttling valve is completely opened, the refrigerant would exit the condenser under saturated conditions (cero subcooling). In this condition, the maximum temperature approach is given, DT=liquid saturated refrigerant temperature at the condenser – water inlet temperature at the condenser. Thereafter, for a certain degree of subcooling, a lower value of the temperature approach is required and the PID would make the valve closing certain percentage until the setting point is reached which would correspond to a certain degree of subcooling. For example, if one fixes the temperature approach to 3 as setting point and the water inlet temperature is 10ºC, the PID would close the valve until the refrigerant exits the condenser at 13ºC (difference between the refrigerant temperature at the outlet of the condenser and the water inlet temperature at the condenser is 3) which is directly translated to the existence of a certain degree of subcooling.

Finally, the other extreme would be to set the temperature approach to a very low value closed to zero. That is, maximum subcooling. In this case, the PID will be closing the valve until the approach is minimized which, in the example, implies the refrigerant to exit the condenser at almost 10ºC. The refrigerant must be in subcooled conditions at the inlet of the TV in order to allow a reliable control.

To proceed with the evaluation, the stability and robustness of the system, the installation was set to equilibrium conditions for a point that was experimentally tested before (but manually controlled). The equilibrium point characteristics are collected in

Table 3.

Table 3: Heat pump conditions for the PID tunning

SHW condenser inlet temperature	30°C

SHW condenser outlet temperature	60°C
Water evaporator inlet temperature	20°C
Water evaporator outlet temperature	15°C
Superheat	10K
Subcooling	12K

Once the installation was stable, the tuning of the PID was done following the heuristic method proposed by Ziegler-Nichols (Ziegler and Nichols, 1995). The critical gain for closed loops methodology was used to firstly adjust the PID parameters. First, the integral and derivative terms were set to zero. Second, the proportional gain (which value was originally zero) was incremented in order to find the critical value that results in an output with consistent oscillation. Afterwards, based on the critical proportional term and the oscillation period, the integral, the derivative and the proportional values were calculated according to the proposed table for a classic PID by Ziegler-Nichols. The critic proportional value that leads to stable oscillations was found to be 160 with an oscillation period of 90 seconds. From those values and following the methodology, the rest of the values used in the process were set: the proportional term to 360, the integral term to 180 and derivative time set to 45.

After, in order to verify the capability and stability of the PID under other points, the modification of the main conditions and the analysis of the PID response, was done.

Figure 11 represents the variation of the main temperatures involved, the superheat, the subcooling and the temperature approach with the time. In this case, different values of DTa (temperature approach), that is, the same point as in the first phase but different subcooling values, are tested. The subcooling is represented in blue. The temperature approach (variable to control, DTa), in black. The inlet water temperature is represented in purple and the superheat in green.

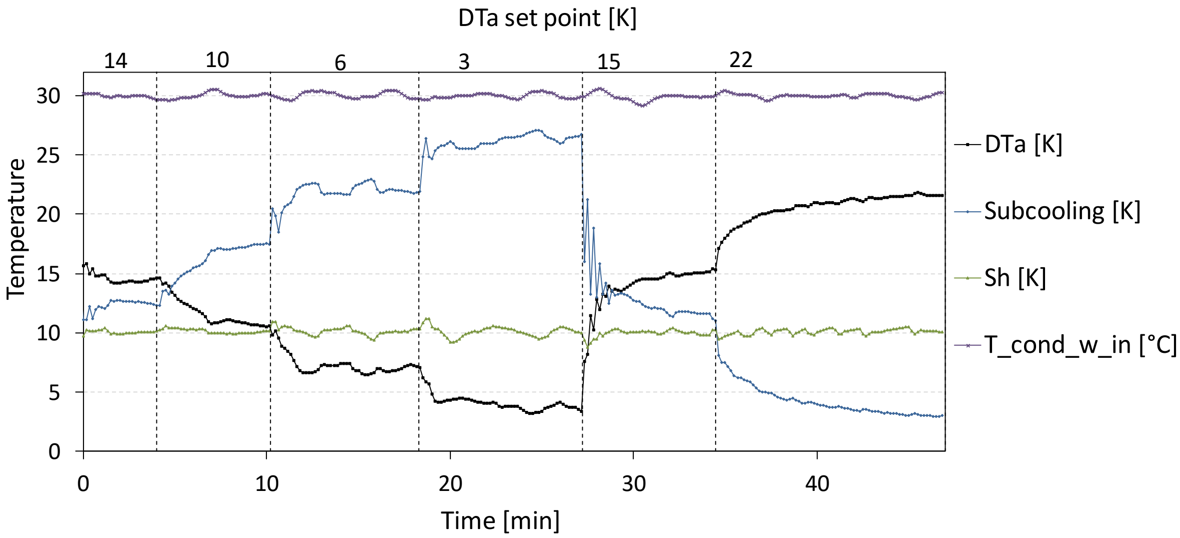

Figure 11: Verification of the PID response for different DTa setting values from a stable system condition (Twcondin=30°C, Twevapin=20°C).

From the figure can be seen the direct relation of the subcooling and the controlled variable, DTa (both curves have the same shape but opposite trend). First, DTa was set to 15K (from the equilibrium initial conditions), second, it was changed to 10K and the system was able to reach the setting in less than 5min, afterwards, the setting value of DTa was modified to 6K, 3K, 15K and 22K. In all the cases the control converged and the set point was attained in less than 3 min.

Furthermore, since this control could affect the control of the expansion valve, the superheat was an important value to look at. From Figrue 11 can be seen that the value of the superheat remains practically constant to the setting value of 10°C which means that the control of the DTa variable is not affecting the control of the superheat.

At this point, it has been verified that when the system is at steady state conditions, it is able to automatically attain the DTa setting value. Another important test would be the verification of this capability from a starting up of the system, the time for the system to reach stable conditions and the response of this control with the modification of external conditions.

Figure 12 represents the starting up of the heat pump until it reaches the DTa setting value of 4 and evaluates the response of the main parameters to variations at the condenser water inlet temperature (from 30°C to 40°C).

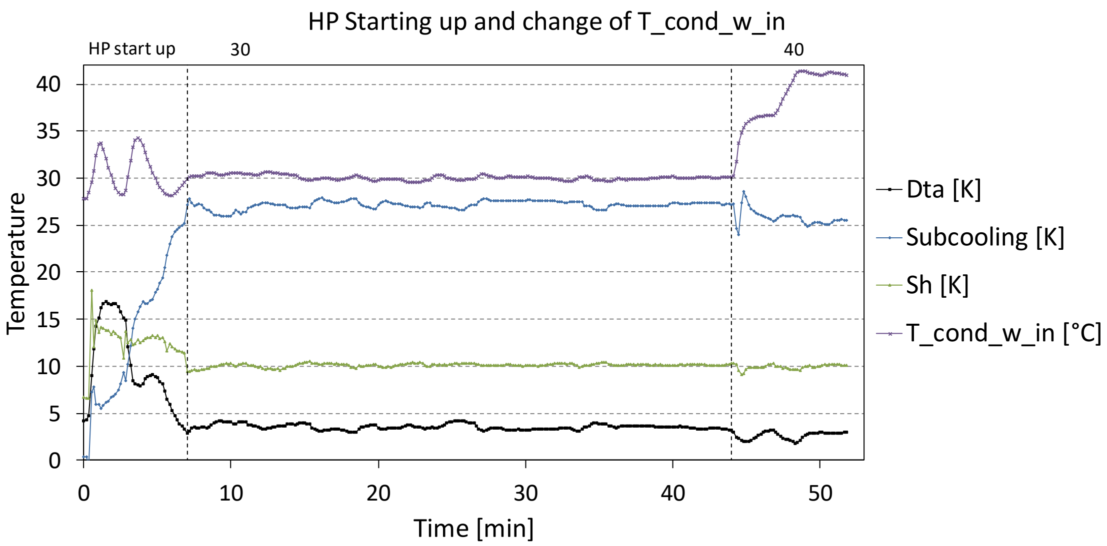

Figure 12: Heat pump starting up, control of DTa=4 and increase of the water inlet temperature at the condenser with the time.

According to the figure, the heat pump starting up approximately lasts 8 minutes. After this time, the temperatures are stable. Then, even though in real applications, the external conditions remain practically constant with time, a sudden water temperature at the inlet of the condenser of 10°C was applied considering it as worst condition to test the control. Based on the Figure, the system could absorbed this modification and returned to stable conditions in less than 5 minutes.

Figure 13 is the continuation of Figure 12. It shows the response of the external condition temperatures, the subcooling and superheat with time. In this case, a decrease of the water inlet temperature at the condenser until 20°C has been tested. Thereafter, in order to obtain real conditions of the installation, the secondary mass flow has been decreased to reach 60°C at the outlet of the condenser. In the secondary axis, the outlet water temperature is represented.

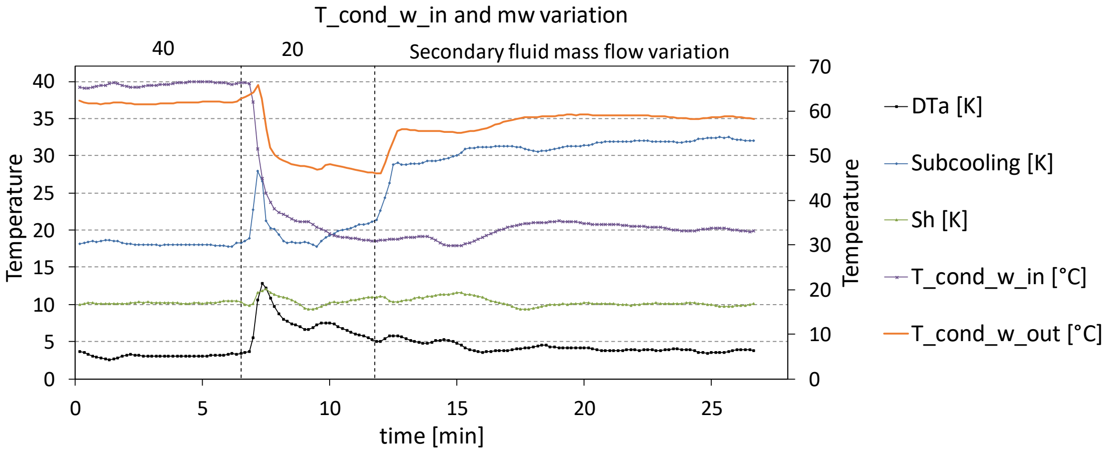

Figure 13: DTa=4 control, decrease of the water inlet temperature at the condenser and variation of the secondary mass flow in order to have the water temperature at the outlet of the evaporator at 60°C.

According to the figure, a step decrease of 20°C in the water inlet temperature can be absorbed by the installation in less than 5 minutes which is further conservative compared to real behavior of the installation. Afterwards, the system is capable of stabilizing and a subsequent reduction on the water mass flow rate is able to be captured in less than 2 minutes from when the system remains stable.

Figure 14 is the continuation of Figure 13 and evaluates the evolution of the system after the modification of the external conditions in order to have tested the range of operation in this type of applications. First, the water inlet temperature at the condenser is stable an equals to 20°C (from Figure 13), after 3 minutes, this temperature is decreased until 10°C. Thereafter the evaporating temperature is decreased in 10°C and finally, the secondary water mass flow rate at the condenser is adapted in order to obtain water at 60°C at the outlet of the condenser following the same reasoning than in Figure 13.

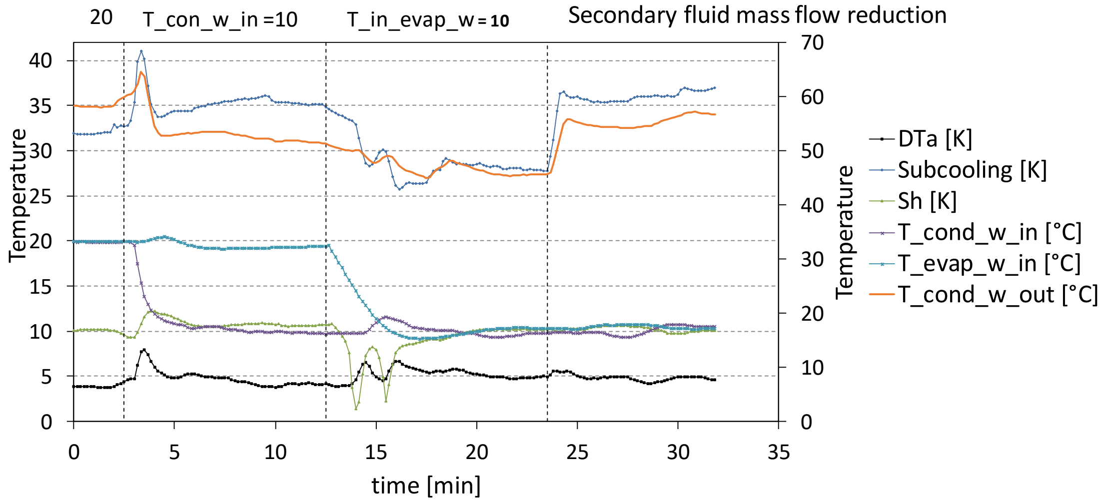

Figure 14: DTa=5 control, decrease of the water inlet temperature at the condenser, decrease of the water inlet temperature at the evaporator and reduction of the water mass flow rate.

Based on the figure, the diminution of the water temperature at the inlet of the condenser generates a transitory peak variation in almost all the variables but the system is stable in less than 3 minutes. It is important to understand that this transitory is not merely a consequence of the implemented control but of the equilibrium in the system after a variation of the conditions at the condenser and that this event would not happen in real conditions since the water inlet temperature used to be very stable across the time. Once the values are stable, a reduction of the water temperature at the inlet of the evaporator was done. This variation was interesting in order to see how the expansion valve could react in the control of the superheat as well as to extend the study under more conditions, for instance within a colder climate. Two transitory peaks were observed (green line) but again, the system is able to stabilize in around 3 minutes. Finally, the adjustment of the water mass flow rate in order to obtain 60°C in the water condenser outlet temperature was done and the system, stabilized very fast.

After the tuning of the PID and analysis of the response under different conditions, the heat pump was tested for some of the points at their optimum that were measured before the application of the automatic control. In this case, the PID used had only a unitary precision and the value chosen for the measures was 3 (the closest value to the average, 2.75). The tests were performed from the heat pump completely turned off.

Figure 15 represents the evolution of the DTa with the time for a DTa setting value of 3 from the starting up of the system for different conditions measured before. (a) water temperature at the inlet of the condenser of 10°C, 60°C at the outlet of the condenser, 20°C at the inlet of the evaporator and 15°C at the outlet of the evaporator, (b) water temperature at the inlet of the condenser of 30°C, 60°C at the outlet of the condenser, 20°C at the inlet of the evaporator and 15°C at the outlet of the evaporator and (c) water temperature at the inlet of the condenser of 50°C, 60°C at the outlet of the condenser, 20°C at the inlet of the evaporator and a 15°C at the outlet of the evaporator.

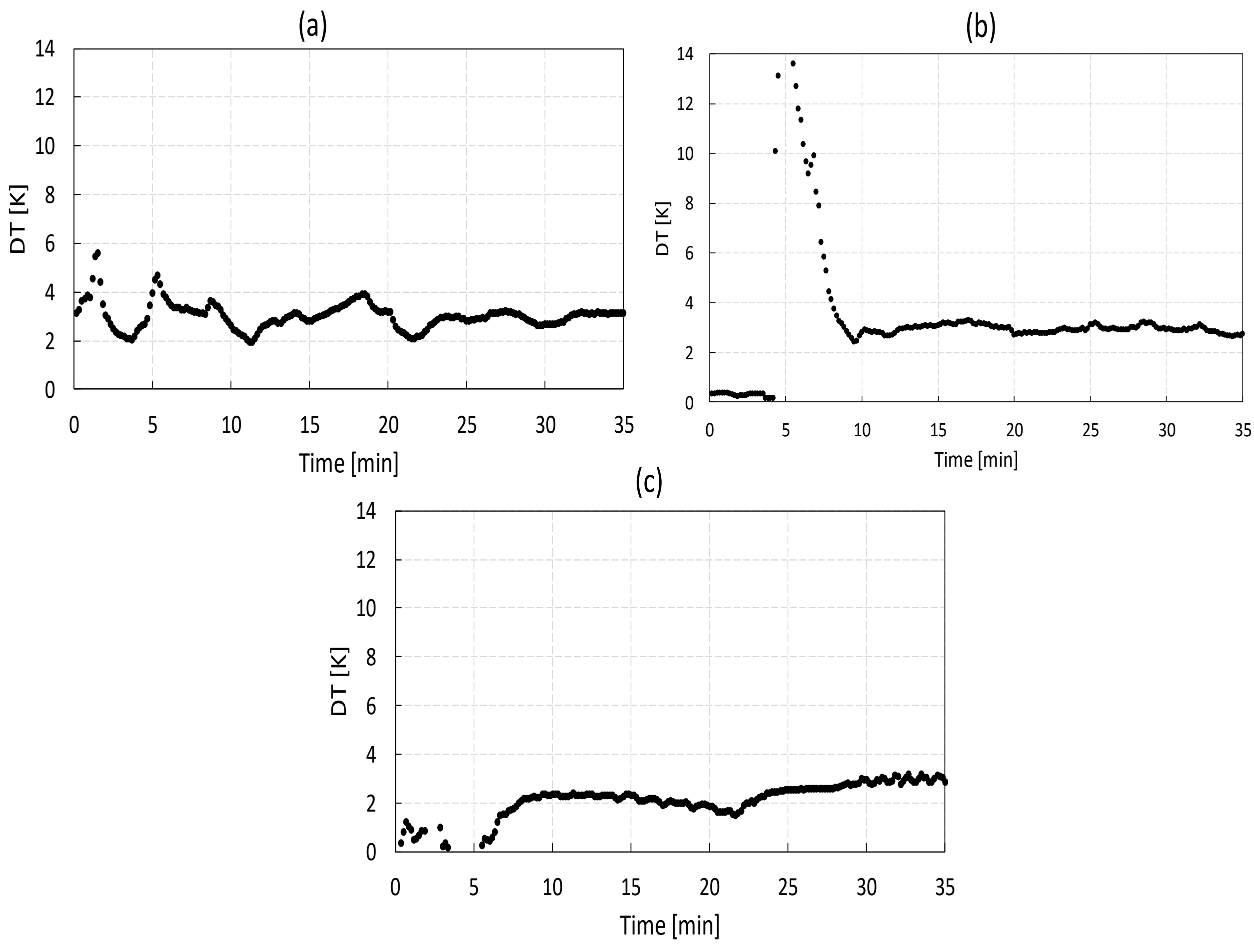

Figure 15: DTa variation (set to 3) with the time for (a) Twinevap=20°C, Twoutevap=15°C, Twincond=10°C and Twoutcond=60°C. (b) Twinevap=20°C, Twoutevap=15°C,

Twincond=30°C and Twoutcond=60°C (c) Twinevap=20°C, Twoutevap=15°C, Twincond=50°C and Twoutcond=60°C.

According to figure 15a, the system is able to control the temperature approach after approximately 20 minutes from the starting up of the heat pump. Notice that the transitory is not due to the control itself but about the stabilization of the entire system after the starting up.

From Figure 15b can be inferred that the system is stable after 10 minutes from a completely stopped mode. Afterwards, the control of DTa remains within the limits for the rest of the test.

As it can be observed in Figure 15c, the system is also able to reach the equilibrium state in the setting point in a bit more than 20 minutes.

Finally, Table 4 collects the experimental results measured before the control of the subcooling and with the control of the subcooling by setting a constant value of 3 in DTa.

Table 4: Experimental values measured without the subcooling control and with the control set to DTa=3 and considering the variation of the compressor efficiency.

Water inlet temp. Evaporator °C	Water temperature lift at the condenser °C	Optimal subcooling [K]	COP heating max [-]	DTa at the maximum subcooling [K]	Subcooling for DTa=3K control [K]	COP heating for DTa=3K control [-]
20	10-60	42.77	5.46	4.77	41.45	5.46
	30-60	26.1	4.66	2.57	24.67	4.65
	50-60	9.27	3.91	2.38	8.34	3.91

From the table can be seen that even if the optimal DTa experiments a variation of 2.4K among the three measures, the control of the system through a constant value of DT=3K independently from external conditions, leads to negligible differences on the COP heating. Thus, indicating that the curve of the temperature approach (DTa) remains flat with respect the COP during a few degrees and bearing out the reliability of the subcooling control by means of the temperature approach.

CONCLUSIONS

Subcooling has demonstrated to play an important role in the heat pump efficiency. In fact, an increase of the COP up to 30% can be possible with the proper control of it for a certain conditions. However, only a few works analyze this paramenter and from the experimental point of view only complex algorithms have been implemented in order to control subcooling and operate under the optimal value. In this work, based on previous experimental and theoretical works, two potential subcooling control strategies have been evaluated under a theoretic point of view.

- Linear fitting equation: the optimal subcooling dependence mainly on the temperature lift of the secondary fluid was investigated. An approximation made by this relationship was found and the use of it leads to deviations from the optimum COP lower than 0.04% in all the cases. As a drawback, the paramenter of the obtained equation depend on the external conditions and the obtained errors could be higher for other refrigerants like R32.
- Constant value of the approach (DTa): The temperature approach influence on the optimality of the subcooling for finite heat exchangers was studied. Results show that the optimal subcooling does not occurs when DTa=DTb=0 as in the infinite heat exchanger area case but for the analyzed cases it is true that the optimum subcooling verifies DTa≈DTb≠0. If it is assumed that the optimum is found in the equality condition (DTa=DTb), the deviations from the maximum COP are lower than 1.09% in all the cases. Thus, the assumption of considering the equality seems to be reasonable as a general rule for a given condenser size and refrigerant and temperature approaches lower than 5K.

Both strategies seem to be appropriated for the control of the subcooling. However, due to simple implementation, not compromising accuracy and applicability independent of the refrigerant, the control by means of the temperature approach was selected and the stability of it was experimentally proved for the considered system and conditions. The time from shut off to stable behavior of the system is around 20 minutes and the deviation from the optimal COP is lower than 0.2% for the measured points.

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20°C and blue lines to 30°C. Yelow line is the average DTa and the yelow dotted lines correspond to the maximum deviation from optimal conditions if the average value is assumed in all the cases.

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Figure 10: Heat pump control scheme

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