Experimental study of a propane heat pump with high subcooling in the condenser for sanitary hot water production

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Abstract

Subcritical systems working with a subcooler have been proved as a good solution for sanitary hot water production, where the high water temperature lifts (usually from 10ºC to 60ºC) make it possible to produce a high degree of subcooling. This paper presents the experimental results obtained from a new heat pump prototype working with Propane (R290) as refrigerant, With the proposed configuration, the subcooling is made inside the condenser (separate subcooler heat exchanger is not used), and can be controlled independently at any point. The obtained results have shown that COP depends strongly on subcooling. In nominal condition (inlet/outlet water temperature at evaporator is 20ºC/15ºC and the water inlet/outlet temperature in the heat sink is 10ºC and 60ºC), the optimum subcooling is 42K with a heating COP of 5.35, which is about 25% higher than the same cycle working without subcooling.

Keywords: Propane, heat pumps, natural refrigerants, sanitary hot water, subcooling

NOMENCLATURE

BPHE: Brazed Plate Heat Exchanger

COP: Coefficient of Performance, [-]

Cp: Specific heat capacity [kJ kg^{-1} K^{-1}]

EU: European Union

EV: Expansion Valve

LR: Liquid Receiver

HX: Heat exchanger

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

$\dot{Q}$ *:* Capacity [kW]

Sc: Subcooling [K]

Sh: Superheat [K]

SHW: Sanitary Hot Water

SPF: Seasonal Performance Factor

T: Temperature [ºC]

Subscripts

condensing: Part of the condenser where condensing takes place

desuperheat: Part of the condenser where desuperheat takes place

h: Heating

liq: Liquid

ref: Refrigerant

sat,evap: Saturation at evaporator

w: Water

w,ci: Water condenser inlet

w,co: Water condenser outlet

w,cond: water through condenser

w,ei: Water evaporator inlet

w,sub: water through subcooler

1. INTRODUCTION

Heat pump water heating systems can supply more heat just with the same amount of energy input used for conventional heaters [1]. This potential for high efficiency is recognized by the European Directive [2], where a portion of the energy captured by a heat pump having an estimated average seasonal performance factor (SPF) higher than a reference value is considered as if it were obtained from renewable energy sources.

A heat pump needs a working fluid (refrigerant) in order to absorb heat from one area and reject it into another. The selected refrigerant must satisfy many requirements, like thermodynamic, safety and environmental aspects. Natural refrigerants (carbon dioxide - CO_2 (R744), hydrocarbons (HCs), and ammonia - NH_3 (R717)) are pointed out as harmless to the ozone layer, with no influence upon greenhouse effect or very less than traditional refrigerants.

Between the natural refrigerants, the use of CO_2 working in transcritical conditions for the sanitary hot water (SHW) application has brought the attention of many researchers, due to the advantage of R744 at high water temperature glides, which entails a high temperature glide in the refrigerant side too, improving the heat rejection process at gas cooler. This effort has been materialized in projects such as ECO-CUTE in Japan. Works like [4-7] have shown high efficiency of these cycles at high temperature lifts, as for instance in heating water from 10ºC to 60ºC or even higher temperatures, showing the transcritical CO_2 cycle as a viable alternative to the synthetic working fluids. [8] compared in a theoretical study the COP penalty of different heat pump systems (CO_2 cycle with different subcritical refrigerants working at zero subcooling) for SHW production. This study shows a higher COP for the CO_2 cycle for high water temperature lifts, but its performance has a high dependency with the water inlet temperature to the gas cooler. After a certain value of the inlet water temperature, COP is higher for the subcritical systems. Transcritical cycles also depends critically on the optimal control of cycle internal variables like the gas cooler pressure.

Although CO_2 transcritical cycles have proved a higher performance than subcritical systems working with zero subcooling to warm water with high temperatures lift, subcritical systems have also been used for the SHW application. This is the case of the commercial heat pump working with Propane Quantum [9], which warms up the water in sequences using low water temperatures lifts (around 5K), trying to increase the overall heating COP at the end of the process (warming water at typical temperatures of 60ºC). With this technique, the heat pump has a higher performance when the water inlet/outlet temperature is low, and it decreases as the water inlet/outlet temperature increases. The main disadvantage of this process is that the heat pump is not able to supply directly the SHW from the typical city water temperatures (around 10ºC).

It is common belief that subcritical systems working with high subcooling has a lower performance due to the area reduction for condensation. In fact, there are few studies addressing the proper use of subcooling. For the case of a non-natural fluid there are also some works concerning subcooling [7, 10, 11]. Cecchinato et al. [7] compares theoretically a CO_2 transcritical cycle with a R134a subcritical cycle working with subcooling. They pointed out that it is possible to increase the energy efficiency of the R134a cycle with an increase of subcooling. In this way, the results for SHW production are similar for both cycles in winter conditions, while CO_2 has a higher performance in summer. For the case of natural refrigerants, there are several studies carried out with Propane. This refrigerant is a good candidate for subcooling, not only due to its good environmental properties, but also due to thermodynamic ones. Propane has a high specific heat in liquid state compared to other refrigerants, like with R134a, so it profits from doing subcooling [12]. Another characteristic of Propane, is that it can work at high evaporating temperatures (critical temperature 96.74ºC), hence it is a good solution for waste heat recovery. [13] and [14], studied from the theoretical and experimental point of view the role of the charge in the R290 cycle, and pointed out that an optimum charge (and consequently a subcooling) exists for a given external condition. The works commented above reported a higher system efficiency working with certain subcooling, but always at low water temperature lift where usually the optimum subcooling is found between 5 K and 10 K.

For the specific application of SHW production, Justo Alonso and Stene [15] compares the theoretical COP of a CO_2 transcritical cycle with two different systems working with Propane, with and without subcooler. An increase of the COP was shown for the Propane cycle working with subcooler with respect to the one with no subcooling, although the degree of subcooling is not mentioned. [16] studied from the theoretical point of view the performance of a heat pump to warm water for a hotel. They pointed out that an optimum condensing pressure exist (which is related to the degree of subcooling) for a given external condition.

There are many works studding the potential of using a heat pump coupled with a heat source in order to increase the efficiency of the systems. Some studies couple the heat pump with a solar panel souce [17],[18],

[19] or to couple the heat pump with an additional condensation loop [20]. Nevertheless, it is more difficult to find any study about heat pumps optimized to work under these conditions.

Recently, Pitarch et al. [21], presented the experimental results of a propane water-to-water heat pump booster prototype for SHW production. The prototype has a separate heat exchanger to produce subcooling (the subcooler), so the condenser area is mainly used for condensing. In this prototype, subcooling cannot be controlled. The degree of subcooling depends on the external conditions and subcooler size (subcooler could be bypassed to have zero subcooling). Improvements of about 31% in the heating COP were reported when using the subcooler (nominal point: inlet/outlet water temperature at evaporator is 20ºC/15ºC and the water inlet/outlet temperature in the heat sink is 10ºC and 60ºC), compared to the cycle working with zero subcooling, the subcooling obtained at this point is 44K.

This paper presents the experimental results of a heat pump prototype able to modify the degree of subcooling at any external condition by controlling the active refrigerant charge on the system. Instead of producing subcooling in a separate heat exchanger as Pitarch et al. [21], this prototype produces subcooling in the condenser. Hence the dedicated area to condensate will depend on the degree of subcooling. A new question arises when producing subcooling in the condenser due to the trade-off between increasing capacity and increasing the condensing temperature: Which is the optimum subcooling for a given condition? This paper analyzes the heat pump performance (heating COP) as a function of subcooling at several external conditions.

2. HEAT PUMP PROTOTYPE

This prototype was designed and built in order to study the effect of subcooling when it is performed in the condenser, for the SHW production in the application of heat waste recovery. The system is able to produce high degrees of subcooling, trying to exploit the advantage of the low inlet water temperature, the used refrigerant is the natural fluid propane. The waste heat could come from any available source of energy, such as sewage water or a condensation loop with temperatures between 10ºC and 35ºC. This heat pump produces sanitary hot water at 60ºC, while different water inlet temperatures to the condenser are considered (10ºC to 50ºC). The system has been designed to obtain around 50 kW in the nominal point, i.e. 20ºC/15ºC at the water inlet/outlet evaporator and producing sanitary hot water at 60ºC from an inlet temperature of 10ºC.

2.1 Heat Pump refrigerant cycle

Figure 1 shows the scheme of the water-to-water heat pump. Besides to the essential heat pump components (evaporator, compressor, condenser and expansion valve), this heat pump prototype includes a liquid receiver and a throttling valve. Both components are placed between the condenser and the expansion valve, being the throttling valve right after the condenser and downstream the liquid receiver. Liquid receivers are normally charge reservoirs used to accommodate the changes in the active refrigerant charge due to changes in the operating mode or changes at external conditions. In this case, the liquid receiver has the functions of accommodating the changes in the active charge due to variations on the degree of subcooling at the condenser but also to guarantee saturated liquid refrigerant at the outlet of the throttling valve. The throttling valve is the control active component that allows setting the subcooling at the condenser independently from the external conditions.– One should notice that in practice, the throttling valve is an electronic expansion valve, but since the refrigerant is in liquid saturated state at its outlet, it will be referred as a throttling valve.

The pressure at the throttling valve inlet (point 3) will depend on the heat transfer process at the condenser, and the pressure at the liquid receiver (point 4) will depend on the opening of the throttling valve. The liquid receiver ensures that the refrigerant leaves the throttling valve in liquid saturated state (point 4), which corresponds to the saturation temperature at the liquid receiver pressure. Therefore, since the throttling valve outlet is constrained to be on the saturation liquid line, the refrigerant stream at the condenser outlet (point 3) must be subcooled. The pressure drop at the throttling valve will determine the subcooling produced at the condenser:

- Throttling valve totally open: The liquid receiver and the condenser will be at similar pressure. This produces low degrees of subcooling.
- As the throttling valve closes, the pressure drop increases and the temperature of point 3 decreases, hence increasing subcooling.

Therefore, by increasing the pressure drop at the throttling valve, the refrigerant charge migrates from the liquid receiver to the condenser, which is flooded with refrigerant liquid and producing more subcooling. The temperature of point 3 can decrease up to a limit, which is determined by the water inlet temperature to the condenser and the heat transfer taking place on it. Beyond this point, subcooling can only increase if the condensing saturation temperature is increased.

The expansion valve controls the superheat at the compressor inlet. One should notice, that the pressure drop at the expansion valve will not only depend on the condensing and evaporating conditions, but also on the degree of subcooling. The pressure drop between condenser and evaporator is shared by the expansion and throttling valve, the higher the subcooling the higher the pressure drop at the throttling valve, and consequently the lower the pressure drop at the expansion valve. When producing high degrees of subcooling (up to 50 K), most of the pressure drop between the condenser and evaporator is done at the throttling valve, and the pressure drop required at the expansion valve is minimum. With this configuration, and the right components design, it is possible to produce high degrees of subcooling in order to take profit of the high water temperature lift in the SHW application and improve performance.

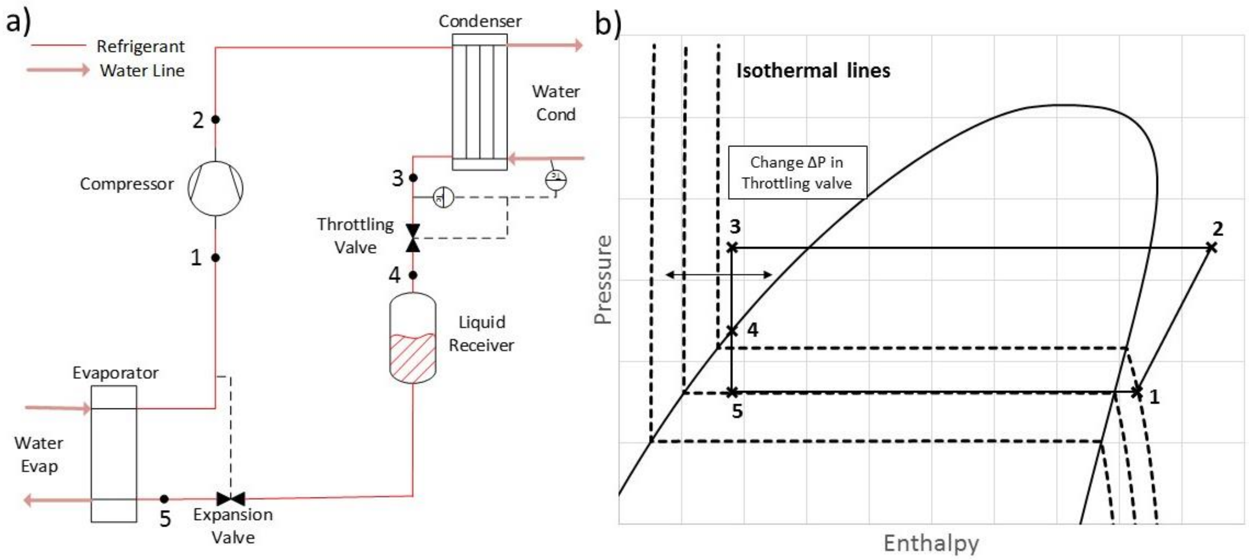

Figure 1: Heat Pump subcooling controlled by a throttling valve a) Scheme, b) P-h diagram.

2.2 Heat Pump Design

Application with a high degree of subcooling, has some peculiarities that must be taken into account when designing the system. The condenser was selected in order to produce high degrees of subcooling (part of the condenser is used for subcooling and other part for condensing) without a significant increase in the condensing pressure. Figure 2 shows a theoretical representation of the water and refrigerant temperature profile in the condenser as a function of the normalized heating capacity. A slight increase in the saturation condensing temperature can be seen when a high degree of subcooling is produced.

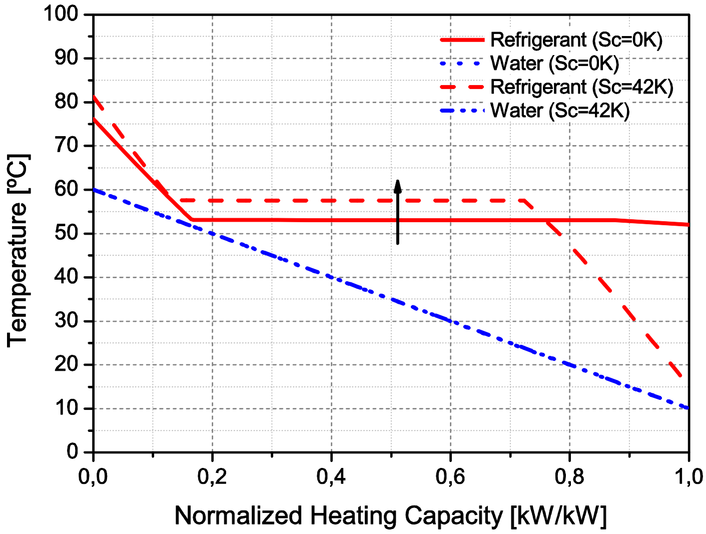

Figure 2: Temperature profiles of propane and water in the condenser vs. the normalized heating capacity without and with 42K subcooling

The evaporator has to be able to work with low inlet refrigerant quality. The heat pump has been designed to obtain around 47 kW in the nominal point. The nominal point has an inlet water temperature to the evaporator point ($T_{w,ei}$) of 20ºC and a water inlet temperature to the condenser ($T_{w,ci}$) of 10ºC, which corresponds to typical values for waste heat recovery and SHW applications.

In order to reach the required pressure drop at the expansion valve when a high degree of subcooling is being produced, a second expansion valve was installed in parallel with the main one, so it could be used to increase the cross sectional area in those points where a low pressure drop is needed.

The liquid receiver (LR) has to ensure the compensation of refrigerant active charge variations. These variations could be produced by: changes in external conditions (water temperatures), or internal conditions (degree of subcooling). The liquid receiver volume is 7 liters, this LR volume was selected in order to fulfill very variable conditions. One should notice that the LR volume could be further reduce if the operating range is narrowed down.

Table 1 shows the characteristics of the different components of the propane cycle.

Component	Type	Size
Compressor	Scroll (2900 rpm)	29.6 m^3h^{-1}
Condenser	BPHE Counter-flow	3.5 m^2
Evaporator	BPHE Counter-flow	6 m^2
Liquid Receiver	-	7 l
Throttling Valve	Electronic EV	5 – 60 kW
Expansion valve	Electronic EV	5 – 60 kW
	Electronic EV	5 – 60 kW

Table1. Components of the heat pump prototype

3. TEST CAMPAIGN

3.1 Experimental setup

Figure 3 shows the test rig, which allows testing water-to-water heat pumps with a heating capacity up to 70 kW. The test rig used to measure this heat pump prototype is the same as the one used by Pitarch et al. [21]. For a better explanation of the test rig and the type of sensors use to measure the heat pump, the reader is referred to [21].

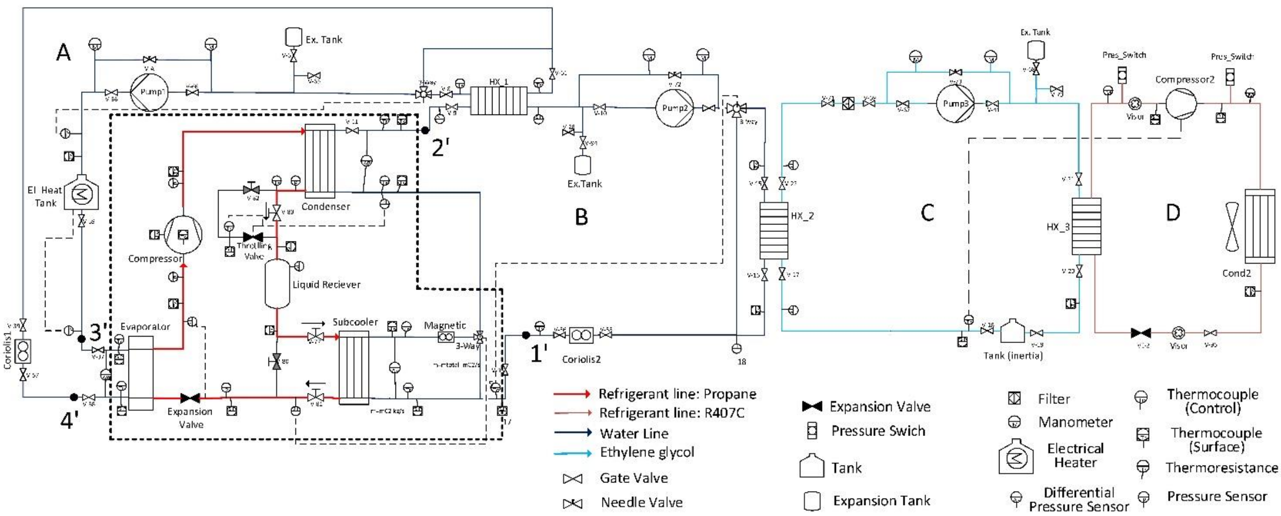

Figure 3: Overview scheme of the Test Rig with sensors

Regarding to the security issues related with the use of Propane, the laboratory is equipped with gas sensors and an alarm system able to detect a Propane leakage and start with a security routine. If commercialized, these heat pump will be installed in a ventilated place outdoor.

3.2 Performed Test

The boundary conditions, such as outlet water temperature, are defined by the kind of application. In the SHW application, 60ºC was selected due to legionella legislation in EU countries. The inlet water temperature at the condenser depends on the city water temperature, which usually ranges between 10ºC to 30ºC depending on location and period of the year. But it also depends on the water tank connection and sizing, making possible to have higher inlet water temperatures, for instance, when recovering heat losses at the tank in periods of inactivity. In this sense, the heat pump was tested at inlet water temperature to the condenser ranging from 10ºC to 50ºC.

In the evaporator, the inlet water temperatures ranged from 10ºC to 30ºC, which corresponds to the waste heat recovery application. The water mass flow through the evaporator was adjusted in order to obtain a 5 K water temperature decrease at the nominal point, i.e. from 20ºC to 15ºC. The water mass flow rate adjusted in the nominal point was kept constant for the rest of test points (around 7000 kg h^{-1}), this procedure is described in the European Standard EN 14825 [22]. In the refrigerant side, superheat was kept constant to 10K for all measured points and subcooling was varied for each external condition in order to study its effect on COP and heating capacity.

Once all the target parameters were reached and the heat pump is working in steady state condition, the acquisition data record data every 10 seconds during 30 minutes. Table 2 contains the measurement points of the test matrix. Each external condition (water temperatures) has been tested at different subcooling values, making a total number of measured points of 58. The COP and heating capacity were calculated at each point according to [22]. These calculations include the auxiliary consumption of the water pumps as it is indicated in the previous standard.

Water in Evaporator Temperature [ºC]	Water in Condenser Temperature [ºC]	Water out Condenser Temperature [ºC]	Refrigerant subcooling range [K]
10	10	60	From 1 to 48
	30		From 1 to 32
	50		From 1 to 17
20	10		From 1 to 52
	30		From 1 to 35
	50		From 1 to 17
30	10		From 4 to 45
	30		From 4 to 35
	50		From 3 to 14

Table 2: Test matrix with a total number of 58 measured points.

4. RESULTS AND DISCUSSION

Figure 4a), c) and e) show the heating COP as a function of subcooling for different external conditions ($T_{w,ei}$ and $T_{w,ci}$). It can be seen, for a fixed external condition: $T_{w,ei}$ and $T_{w,ci}$, that there is an optimal subcooling where the COP is maximized. The optimum subcooling can vary significantly depending on the inlet water temperature at the condenser, the refrigerant temperature at the condenser outlet is limited by the inlet water tempeature. The lower the inlet water temperature of the condenser, the higher the optimum subcooling. As an example, the optimum COP for the nominal point ($T_{w,ei}$=20ºC; $T_{w,ci}$=10ºC) is 5.35, and corresponds to a subcooling of about 42 K. This maximum COP is around 25% higher than the COP corresponding to the same external water temperatures, but with the lowest subcooling (around 2 K). In this case, it is clear the advantage taken from the low inlet water temperature ($T_{w,ci}$) to produce subcooling and improve COP. If a point with a higher $T_{w,ci}$ is taken, the improvement is less significant, for instance for $T_{w,ci}$=50ºC, the degree of improvement is less than 7% when going from the minimum to the optimal subcooling.

Table 3 shows the degree of improvement when $T_{w,ci}$ goes from 50ºC to 10ºC for different water inlet temperatures at the evaporator and for the minimum and optimum subcooling. At the minimum subcooling (around 2 K), the improvement is directly related with the decrease of the condensing pressure when $T_{w,ci}$ goes from 50ºC to 10ºC, this improvement is higher for higher $T_{w,ei}$. At the optimum subcooling, there is an improvement related with the decrease of the condensing pressure and another with the increase of the optimal subcooling. The optimum subcooling for $T_{w,ci}$=10ºC is higher than for $T_{w,ci}$=50ºC. At the optimum subcooling, the degree of improvement does not follow the same trend as the minimum subcooling with $T_{w,ei}$, since the degree of improvement is lower at $T_{w,ei}$=30ºC.

Water inlet temp. Evaporator $T_{w,ei}$ [ºC]	**Change in water in Condenser $T_{w,ci}$ [ºC]**	**COP improvement at Min. Subcooling**	**COP improvement at Optimal Subcooling**
10		12 %	36 %
20	From 50ºC to 10ºC	16 %	39 %
30		18 %	35 %

Table 3: Heating COP improvement when Tw,ci goes from 50ºC to 10ºC.

As expected, if the inlet condition at the condenser is fixed ($T_{w,ci}$), it can be seen an increase of COP as the $T_{w,ei}$ increases. For instance, for the condition $T_{w,ci}$=10ºC and the optimum subcooling, COP increases about

24% when passing from 10ºC to 20ºC at the water inlet temperature of the evaporator. This COP increase is directly related to the increase of the evaporating pressure.

Figure 4b), d) and f) show the heating capacity as a function of the degree of subcooling. The heating capacity does not have a maximum value for the optimum subcooling (maximum COP). Instead, in most of the measured points, a direct relation with subcooling can be seen: as subcooling increases the heating capacity increases. This relation is linear up to the optimum subcooling, from this point, heating capacity increases, but in a lower degree. In the linear part, the heating capacity does not depend on the inlet water conditions at the condenser ($T_{w,ci}$), but it depends on the degree of subcooling and the evaporator conditions ($T_{w,ei}$). One should notice, that even if heating capacity does not depend on $T_{w,ci}$ for the same subcooling and $T_{w,ei}$, lower $T_{w,ci}$ have higher potential to produce subcooling, hence the heating capacity at the optimum subcooling is higher for lower $T_{w,ci}$. There is an external condition, which does not have the same trend than the others, this condition is: $T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC. At this point, the heating capacity decreases for subcooling higher than the optimum.

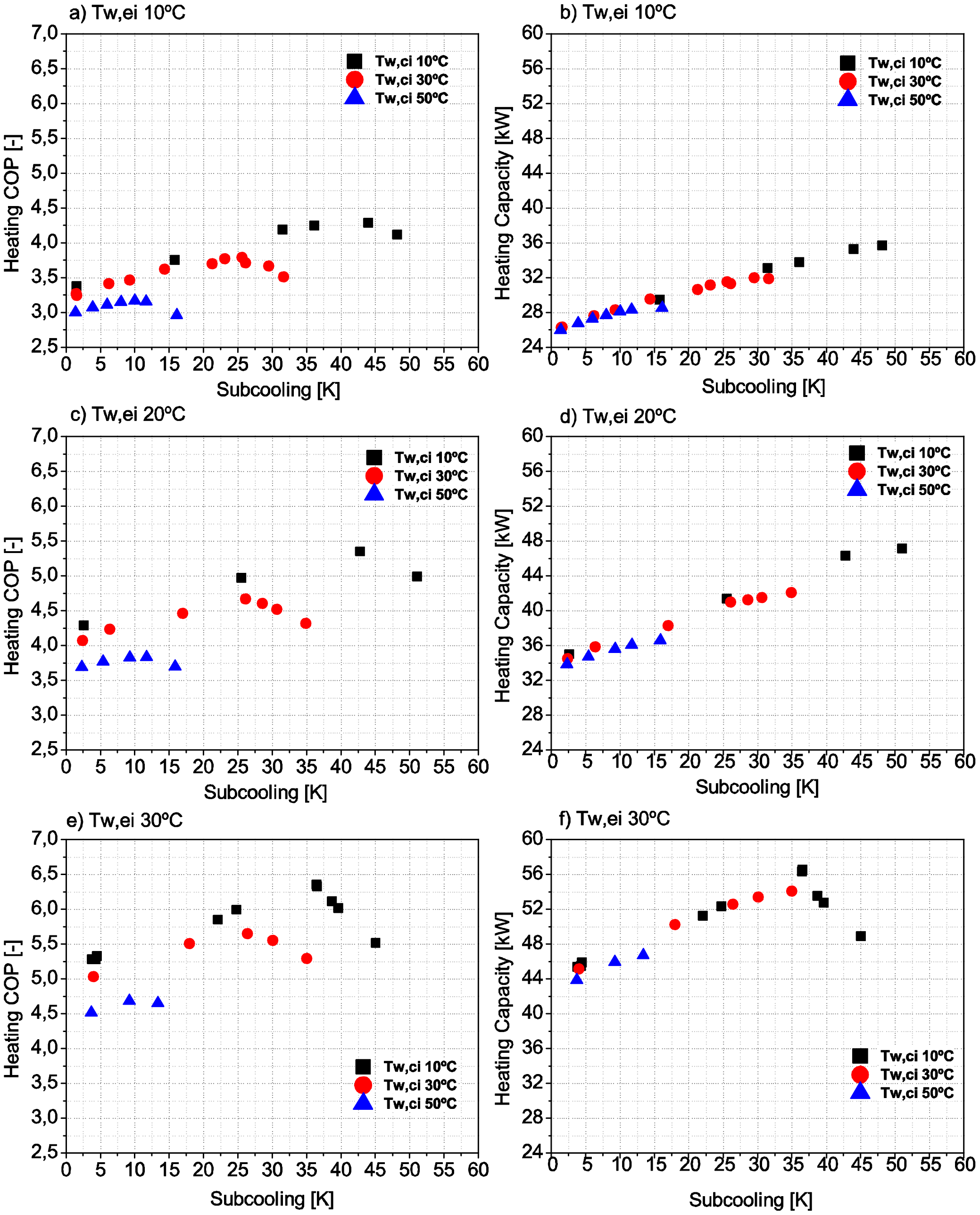

Figure 4: COP heating vs. subcooling (Maximum uncertainty ±0.08) : a) Tw,ei = 10ºC c) Tw,ei = 20ºC e) Tw,ei = 30ºC. Heating capacity vs. subcooling (Maximum uncertainty ±0.05 kW): b) Tw,ei = 10ºC d) Tw,ei = 20ºC f) Tw,ei = 30ºC. (Tw,co = 60ºC).

Figure 5a) shows the subcooling and refrigerant temperature at the outlet of condenser for each external condition working at the optimum point (maximum COP) as a function of $T_{w,ci}$. In both cases, for the subcooling and the refrigerant outlet temperature, there is a clear linearity relation with $T_{w,ci}$. $T_{w,ei}$ does not affect significantly to these values. The refrigerant temperature at the condenser outlet is close to the corresponding $T_{w,ci}$ (about 2ºC higher). This means that the condenser is able to produce a high subcooling with a low temperature approach at the optimum point. As observed before, the point corresponding to the condition: $T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC, does not have the same trend as the rest of the points.

Figure 5b) shows the saturation pressure at the condenser as a function of the subcooling. For each external condition, the condensing pressure increases with the subcooling. Two different zones can be seen in each condition:

1) At low subcooling: the condensing pressure increase slowly with subcooling,
2) After a certain value of the subcooling the condensing pressure starts to increase at a higher rate.

This point of inflection has a low dependency on $T_{w,ei}$, but the higher the $T_{w,ci}$, the higher the subcooling at which occurs the point of inflection. The optimum subcooling for each condition lies in the point of inflection, where the condensing pressure starts to increase at a higher rate. Once again, the condition: $T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC, has a different behavior, since right after the optimum subcooling, the condensing temperature decreases slightly, starting to increase again at higher subcooling.

Subcooling is measured as the temperature difference between the condensing saturation temperature and the refrigerant temperature at the condenser outlet. Its minimum value depends on $T_{w,ci}$. At low subcooling (relative to the optimum), the refrigerant temperature is still far from $T_{w,ci}$, so subcooling can be increased by cooling down the refrigerant at the condenser outlet without increasing significantly the condensing pressure. When the refrigerant outlet at the condenser is closer to $T_{w,ci}$, the increase of subcooling is mainly due to an increase in the condensing pressure (point of inflection). This behavior is closely related with COP and heating capacity trends.

For a fixed refrigerant mass flow rate, the heating capacity will only depend on subcooling (Sc) at the liquid refrigerant (linear relationship between heating capacity and subcooling)

$$\dot{Q}_h = \dot{Q}_{desuperheat} + \dot{Q}_{condensing} + \dot{m}_{ref} \cdot Cp_{ref,liq} \cdot Sc \quad (1)$$

On the right of the optimum point, the heating capacity does not depend only on the subcooling, but also on the condensing pressure.

Regarding the heating COP with near zero subcooling, most of the area of the condenser is used for condensing. As the subcooling increases, the area for condensing decreases and more space of the condenser is used for subcooling, increasing the condensing pressure at the same time. Therefore, the optimum COP is a compromise between the improvement due to an increase in the enthalpy change at the condenser with subcooling, and the degradation due to the decrease of condensation area (increase of condensing pressure).

It should be noted that the subcooling at which the condensing pressure starts to increase (at a higher rate-point of inflection), will depend on the condenser size. For instance, for a smaller condenser, a higher temperature approach between the inlet water temperature and the outlet refrigerant temperature would be expected (now it is about 2ºC). This would give a lower optimum subcooling, and lower heat pump performance would be obtained.

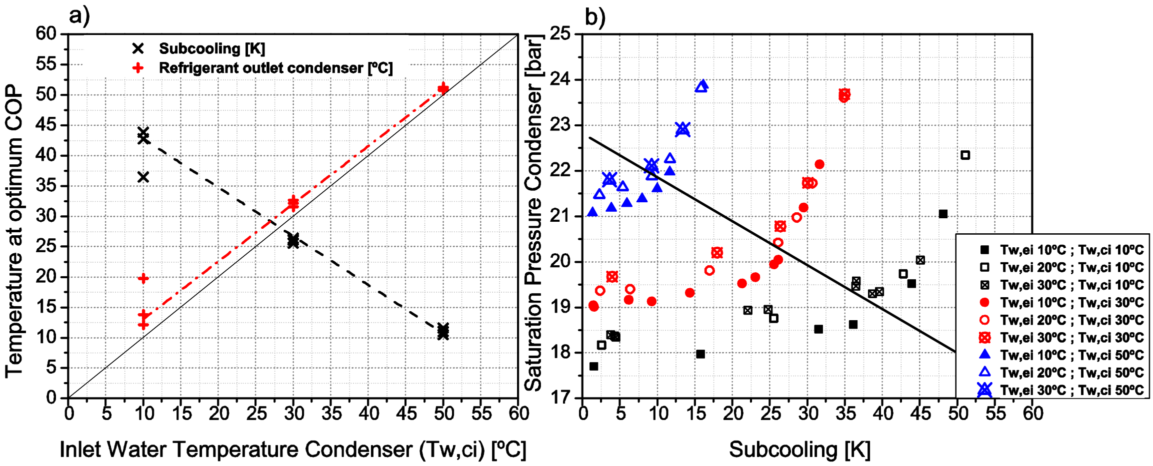

Figure 5: a) Subcooling and refrigerant outlet temperature at the condenser vs. $T_{w,ci}$ at the optimum COP point b) Condensing pressure vs. subcooling ($T_{w,co}$ = 60ºC).

Figure 6 shows the pressure-enthalpy diagram for the cycle working at the condition: a) $T_{w,ei}$=30ºC ; $T_{w,ci}$=30ºC and b) $T_{w,ei}$=30ºC ; $T_{w,ci}$=10ºC. Condition b) was out of the trend showed by the rest of the points: different optimum subcooling for the same $T_{w,ci}$, a decrease of the heating capacity after the optimum subcooling. In both cases, a) and b), the point represented in the pressure-enthalpy diagram, is where the outlet refrigerant temperature is close to the $T_{w,ci}$. The observed behavior in each point is explained as follows:

a) $T_{w,ei}$=30ºC and $T_{w,ci}$=30ºC: The temperature of refrigerant at the condenser outlet (point 3) is limited by the water inlet temperature. As seen previously, the refrigerant reaches temperatures close to the water temperature $T_{w,ci}$, without increasing the pressure at condenser significantly. In the optimum subcooling, there is a temperature approach of 2ºC between refrigerant and water. The subcooling is controlled by the throttling valve, which produces a pressure drop between point 3 and 4. The expansion valve gives the needed pressure drop (point 4 to 5) to obtain 10 K of superheat at the compressor inlet. The refrigerant temperature at the evaporator outlet (point 1) is limited by the water temperature at the evaporator inlet ($T_{w,ei}$=30ºC). If the refrigerant temperature at point 1 is close to $T_{w,ei}$, the evaporating temperature will be around 20ºC ($T_{sat,evap} = T_{ref,1} - Sh$).

b) $T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC: The refrigerant temperature at point 3 is limited by $T_{w,ci}$=10ºC. In this condition the pressure drop needed in the throttling valve is higher compared to the a) case. In order to obtain 10 K of superheat, the evaporating temperature should be around 20ºC (same evaporating condition than before, $T_{w,ei}$=30ºC). But the pressure at point 4, at the liquid receiver, is already lower than the one corresponding to the evaporating temperature of 20ºC, and the expansion valve will introduce an extra pressure drop. From a certain degree of subcooling, the evaporating pressure is not controlled by the expansion valve anymore. This results in higher superheats and lower performance, even if the high subcooling is reached with an insignificant increase in the condensing pressure.

This behavior explains the different trend observed in the point with high water temperatures at the evaporator inlet and low water temperatures at the condenser inlet, $T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC.

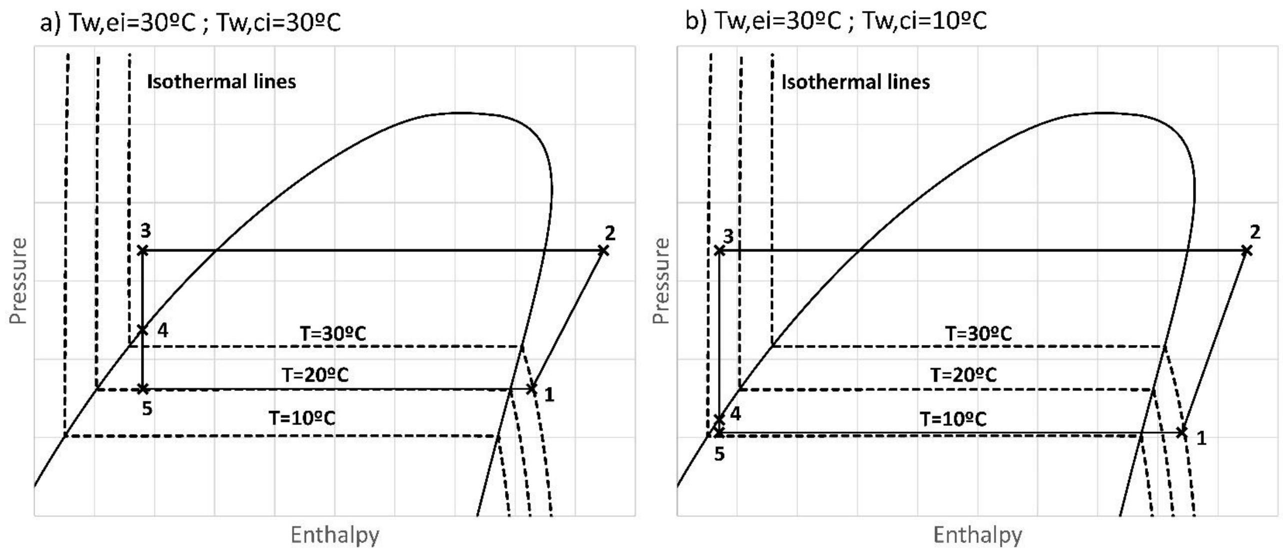

Figure 6: Pressure-Enthalpy diagram a) Tw,ei = 30ºC ; Tw,ci = 30ºC b) Tw,ei = 30ºC ; Tw,ci = 10ºC

Figure 7 shows the refrigerant quality at the inlet of the evaporator as a function of $T_{w,ci}$ and subcooling. A high dependency on the subcooling can be observed. For instance for the condition with $T_{w,ei}$=10ºC and $T_{w,ci}$=10ºC, the refrigerant quality goes from 0.36 to 0.08 when going from the minimum to maximum subcooling. The lowest refrigerant quality is found at high subcooling and high evaporating conditions. This is near zero, which means that expansion valves have a low pressure drop. These high variations in the inlet quality could lead to high variation on the refrigerant mass contained in the evaporator, which needs to be taken into account in the design process of such a system. The system has been working stable in all conditions, even at low refrigerant inlet quality at the evaporator.

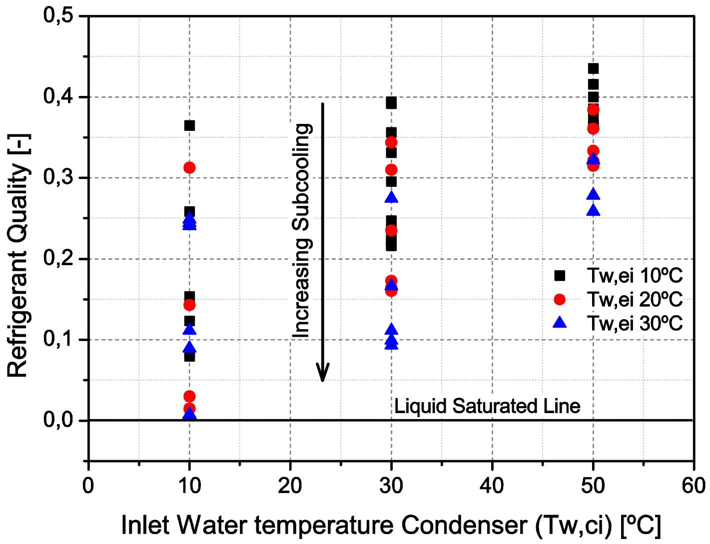

Figure 7: Refrigerant quality at evaporator inlet

Therefore, a subcritical cycle with Propane has demonstrate to have a good performance for sanitary hot water production when it is working with a high degree of subcooling produced in the condenser. The results obtained with this prototype are close to the ones obtained by [21], the heat pump making subcooling in a separate heat exchanger (subcooler). For instance, in the nominal point, [21] reported a COP of 5.61, while the present heat pump working in its optimum subcooling (around 42K) has a COP of 5.35. These results have shown that the new proposed design is a feasible alternative to the previous one with one less heat exchanger. Nevertheless, it is difficult to point out one solution as the best one, since both prototypes do not have the same heat exchange area. Pitarch et al. 2016 has an extra heat exchanger (subcooler), what gives a 25% more area than the used in the present work (only the condenser).

For more information about the measured points, like water mass flow rate and other important parameters it is referred to annex A.

5. CONCLUSIONS

The conclusions drawn from the present study are:

- Subcooling can be varied independently from the external conditions (water temperatures).
- An optimum subcooling exist, being a compromise between the improvement due to an increase in the enthalpy change at the condenser with subcooling, and the degradation due to the decrease of condensation area (increase of condensing pressure).
- The performance of this subcritical propane heat pump cycle can be improved by about 25% by having subcooling at the optimal conditions (42 K) when it works at high water temperature lift (50 K).
- For a lower water temperature lift (10 K), the rate of improvement due to subcooling is lower, approximately 7%.
- The heating COP at the optimum subcooling is about 36% higher when the inlet water temperature at the condenser ($T_{w,ci}$) goes from 50ºC to 10ºC (nominal conditions).
- The heating capacity increases with subcooling for the whole measured range. Only the point "$T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC" has a decrease in heating capacity after the optimum subcooling.
- For a given subcooling, there are no significant differences in the heating capacity with the inlet water temperature at the condenser, $T_{w,ci}$.
- The condensing saturation pressure increase slowly with subcooling until the "optimum" subcooling is reached. From this point the condensing pressure increases at a higher rate.
- At point "$T_{w,ei}$=30ºC and $T_{w,ci}$=10ºC" is not possible to work at high subcooling without decreasing the evaporating temperatures, which leads to a higher superheats and lower performance.
- Low refrigerant quality can be found at evaporator inlet, when working at high evaporating pressure and high subcooling.

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LIST OF FIGURES