2016

High Efficiency Heat Pump with Subcooling for Sanitary Hot Water Production Working with Propane

Miquel Pitarch-Mocholí
Instituto de Ingenieria Energética, Universitat Politècnica de València, Spain, mipimoc@upvnet.upv.es

Emilio Navarro-Peris
Instituto de Ingenieria Energética, Universitat Politècnica de València, Spain, enava@ter.upv.es

José Gonzalvez-Maciá
Instituto de Ingenieria Energética, Universitat Politècnica de València, Spain, jgonzalv@ter.upv.es

José Miguel Corberán
Instituto de Ingenieria Energética, Universitat Politècnica de València, Spain, corberan@iie.upv.es

High Efficiency Heat Pump with Subcooling for Sanitary Hot Water Production Working with Propane

Miquel PITARCH[1], Emilio NAVARRO-PERIS[1*], José GONZÁLVEZ-MACIÁ[1], José M. CORBERÁN[1]

[1]Universitat Politècnica de València, Institut d'Enginyeria Energètica,
Camí de Vera, s/n, València, 46022, Spain

* Corresponding Author (e-mail: emilio.navarro@iie.upv.es)

ABSTRACT

In the last decades the use of heat pumps are in increased due to its potential for high efficiency. They are considered an environmentally friendly technology, 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. For the specific case of sanitary hot water production, where the water temperature lift is considerably high, e.g. 50K, transcritical cycles working with CO_2 have focus most of the attention from the scientific community due to its high performance when working with high water temperature lift. The reason of this high performance, is the heat rejection process at gas cooler, which entails a high temperature glide in the refrigerant side, hence the temperature profile between the refrigerant and water side fits better for the transcritical CO_2 than the cases where a two phase process takes place. The temperature profile of the refrigerant in a subcritical system can be modified by adding subcooling in order to obtain a similar effect that the CO_2 transcritical cycles. In this way, the refrigerant temperature profile can be adapted to water temperature profile by means of subcooling, which will be higher for higher water temperature lift. Hence subcooling is a way to enhance the effectiveness of the condenser in a subcritical cycle. This paper presents the experimental results of a water to water heat pump for sanitary hot water production specially designed to work with large amount of subcooling, where the working fluid, is the natural refrigerant, propane. The results have shown a performance improvement up to 31% in heating COP, when working with subcooling compared with the same heat pump without subcooling.

1. INTRODUCTION

Most residential water heaters are equipped with conventional heaters generating heat by consuming fossil fuels or electricity. Heat pump water heating systems can supply more heat just with the same amount of energy input used for conventional heaters, (Kim et al., 2004). In this sense, an interesting alternative to the conventional Sanitary Hot Water (SHW) systems is the use of heat pump (HP) technologies, which is an application of growing interest nowadays. This potential for high efficiency is recognized by the European Directive 2009/28/CE, 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. As stated in Sarbu (2014), nowadays a new concept in the implementation of refrigeration systems is imposed, requiring tightly constructed configurations that work with refrigerants having a low TEWI (Total Equivalent Warming Impact), but keeping the performance as energetically efficient as possible. Natural refrigerants (carbon dioxide - CO_2 (R744), hydrocarbons (HCs), and ammonia - NH3 (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 SHW application has bring the attention of many researchers. This effort has been materialized in projects such as ECO-CUTE in Japan. Works like Rieberer et al., (1997), Nekså et al., (1998), Nekså (2002), and Cecchinato et al., (2005) has 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. These authors agree with the requirement of using stratified storage tanks to keep warm water separated from the cold water entering from the town, in order to achieve a good energy performance.

In addition to CO2, also another natural fluid, Propane, has been investigated by several researchers for the SHW application. A report by IEA (IEA, Annex 32) shows the seasonal performance of a Propane Heat Pump for the combined space heating and SHW production in a Norwegian passive house. In another report, Justo Alonso and Stene (2010) compares the theoretical calculated COP of a CO2 transcritical cycle with two different systems working with propane, with and without subcooler. They concluded that COP is 20% higher when CO2 is used, 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. The two different systems working with propane studied by Justo Alonso and Stene (2010) were one with subcooling zero, and the other with a subcooler. They showed an increase of COP for the Propane cycle working with subcooler respect to the one with no subcooling, although they do not say the degree of subcooling. Tammaro et al., (2015) studied from the theoretical point of view with a heat pump model the effect of the condenser pressure on the COP for the Propane cycle, and pointed out that an optimum condenser pressure exists for a given refrigerant charge.

Up to the knowledge of the authors, no experimental results of subcritical systems working with subcooling for the SHW production are published, but from the few theoretical works there are enough evidence to believe that this kind of cycles can work with high efficiency for this application. For instance, Cecchinato et al., (2005) compares theoretically a CO_2 transcritical cycle with 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. In this sense, Propane is a good candidate, 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 takes profit from doing subcooling. Another characteristic of propane, is that it can work at high evaporating temperatures, hence it is a good solution for the waste heat recovery, Schmid (2009) pointed out that 15% of the thermal energy provided to the building is lost, unused, via the sewage system.

The scope of the present paper is to evaluate the performance of a Propane water-to-water heat pump prototype for SHW production, in the application of heat recovery from any water source, for instance from sewage water or a condensation loop. The prototype is able to overcome high subcooling in order to take profit of the high water temperature lift in the SHW application. First, the refrigerant cycle in order to achieve the needed subcooling and the experimental layout used in the laboratory are presented. Finally, the experimental results for COP, heating capacity and other important parameters at different temperatures water source at the condenser and evaporator are shown.

Corberan et al., (2015), showed the preliminary results for this Propane water-to-water prototype. In these preliminary results, the performance obtained by the Propane cycle working with high subcooling is comparable to the one obtained by the existing CO2 heat pumps in the market, showing the high potential for this kind of cycles for the SHW application.

2. HEAT PUMP PROTOTYPE

This prototype has been designed and built in order to study a heat pump booster for waste heat recovery trying to exploit the advantage of the low inlet water temperature to produce subcooling and improve COP. 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, which temperatures usually goes between 10 and 35 ºC. This heat pump produces sanitary hot water at 60ºC and it is tested at different condenser water inlet temperatures in order to study the influence of the water temperature lift with the performance of the heat pump, from 10 to 55ºC. The system has been designed to obtain around 50 kW in the nominal point, i.e. 20ºC at the water inlet 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 prototype with subcooler. A liquid receiver located right after the condenser ensures that (at steady state conditions) the refrigerant leaves the condenser in liquid saturated state (point 3), the liquid receiver is big enough to fulfil this condition for all test conditions. For continuity the refrigerant leaves the liquid receiver at the saturation condition, at the condenser saturation temperature. Thereafter, the refrigerant is subcooled in a heat exchanger (HX) specially designed for this reason (subcooler). In the water side, it passes first through the subcooler, where it is preheated before passing through the condenser, where the water reaches the target temperature for the SHW application (usually 60ºC).

The subcooler has been selected in order to obtain a low difference temperature between the water inlet and refrigerant outlet. Therefore, the maximum possible subcooling for this cycle is obtained at each condition, where subcooling will depend on the water inlet temperature. With this configuration, it is possible to obtain a great subcooling for high water temperature lift, which improve the heat rejection process.

Inlet water temperatures at the subcooler 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, then water temperature at the inlet of subcooler can reach 55ºC. For this reason, experiments with water inlet temperatures to the subcooler ranging from 10ºC to 55ºC have been done. This variation in the inlet water temperature implies a big variation on the water mass flow rate (Producing hot water at 60ºC). The higher the water mass flow rate, the higher the pressure drop at the components. Subcooler has lower cross sectional area than condenser, since it has been designed for refrigerant liquid, producing higher pressure drops in the water side. In order to reduce consumption from the auxiliary components such as the water pump, part of the water mass flow is bypassed at the subcooler, keeping the pressure drop in the water side lower than 0.4 bar. The partial bypass is only necessary for high inlet water temperature (more than 50ºC), and is controlled by a three-way valve.

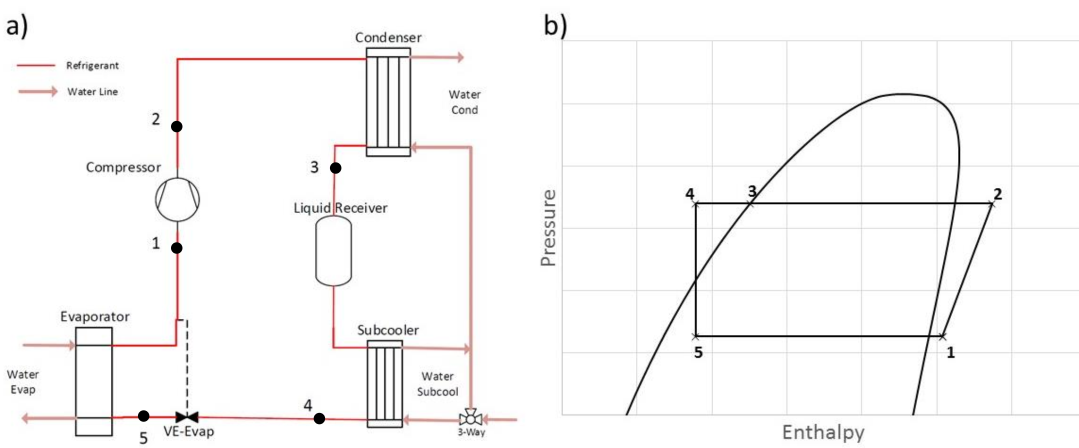

Figure 1: Heat Pump subcooler in series with condenser a) Scheme, b) P-h diagram.

2.2 Heat Pump Design

The different components have been designed in order to reach the high subcooling at the separate heat exchanger. Since refrigerant density is higher at subcooler than at condenser, subcooler size can be optimized for refrigerant liquid, so it has an appropriate refrigerant velocity for heat transfer. In this way, subcooler has less plates and smaller plate pitch than the condenser.

The liquid receiver has a volume of 7 liters to ensure the compensation of refrigerant volume variations between the different measurements conditions. The total charge of the system is 6.4 kg of Propane. One should notice that the total charge of the system and LR volume could be further reduce if the operating range is narrowed down, since this volume was selected in order to fulfill very different conditions.

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

Table1: Components

Component	Type	Size
Compressor	Scroll (2900 rpm)	170.2 cm^3
Condenser	Brazed Plate HX Counterflow	3.5 m^2
Subcooler	Brazed Plate HX Counterflow	0.87 m^2
Evaporator	Brazed Plate HX Counterflow	6 m^2
Liquid Receiver	-	7 l
Expansion Valve	Electronic EV	5 – 60 kW

3. TEST CAMPAIGN

3.1 Experimental Setup

Figure 2 shows the test rig, which allows to test water-to-water heat pumps with a heating capacity up to 70 kW. Between the dashed lines it is the unit to be tested, where points 1&2 are the inlet/outlet for the heat sink (demand side), and 3&4 are the inlet/outlet for the heat source (waste heat side). The test rig is able to keep to a constant value the water temperature at these points. In order to ensure a steady state behavior, all the measured points have been checked to lie under the limits marked by the norm UNE-EN 14511-3.

The test rig consists of four loops:

a) The water loop for the heat source (Evaporator). Simulates the heat recovery from a water source.

b) The water loop for the heat sink (Condenser). Simulates the SHW production.

c) The water/glycol loop.

d) The chiller. Works with R410A

The temperature of the water at the evaporator it is recovered through a heat exchanger that interacts with the sink heat loop and an electrical heater adds extra heat if it is necessary. Therefore, the temperature of the water at the outlet of condenser it is cooled in part due to the recovery heat exchanger with the evaporating water loop. The rest of heat that needs to be rejected in order to reestablish the water inlet temperature at the subcooler is pumped out to the ambient with a chiller, which is connected to the water condensing loop through a water/glycol loop. Components such as 3-way valves or pumps with variable frequency drive are controlled by means of PIDs in order to reach the target inlet and outlet water temperatures.

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.

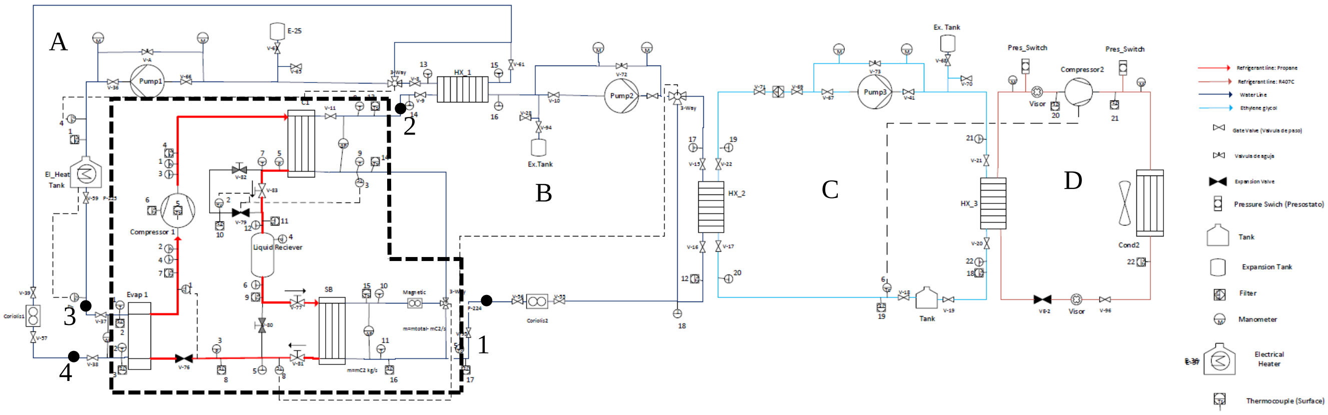

Figure 2: Overview scheme of the Test Rig with sensors

The capacities of the heat pump are measured in the water side in order to measure it as accurate as possible, six thermoresistances have been located at inlet/outlet of heat exchangers directly in contact with the water. To monitor and measure temperature in other points, a total number of 27 T-type thermocouples are used. The water mass flow through evaporator and condenser is measured with Coriolis mass flow meters, for control reasons a magnetic mass flow meter is measuring the water mass flow through subcooler, which in most of the cases is the same as in the condenser. For the pressure probes at the refrigerant side there are 3 high accuracy Rosemount sensors. In the water side there are 3 differential pressure sensors to measure the pressure drop in the heat exchangers. With this measurement and according to the European standard 14511-3, the auxiliary consumption of the water pumps is calculated. In order to control the system and measure those key parameters to evaluate the performance of the heat pump, all the sensors are connected to a data acquisition system “Agilent 34970A”, where all parameters are monitored.

Table 2 shows the main sensors with the relative and absolute uncertainty intrinsic to the sensor. The total uncertainties of the main measured parameters can be seen in the annexed material Annex A, which includes the intrinsic uncertainty of the sensors, the acquisition data and standard deviation of the mean value, which is calculated from the measured sequence during 30 minutes. For the calculated parameters such as heating COP, the equation of propagating error has been used (Coleman and Steele).

Table 2: Sensors and their uncertainty

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		

3.2 Performed Test

The boundary conditions are defined by the kind of application. In this case for SHW, it has been selected production at 60ºC. The inlet water temperature to the subcooler ranges from 10 to 55ºC due to the different specifications it can have depending on the water tank selection, size and type (stratified). For those points where pressure drop at the subcooler (water side) is higher than 0.4 bar, it is partially bypassed, usually for inlet water temperatures higher than 50ºC.

In the evaporator, the inlet water temperatures ranges from 10ºC to 25ºC. The water mass flow through the evaporator is 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 is kept constant for the rest of test points (around 7000 kg/h), this procedure is described in the European Standard EN 14825. In the refrigerant side, superheat is kept constant to 10K for all measured points.

Once all the target parameters are reached, the acquisition data record data every 10 seconds during 30 minutes in order to ensure a stable condition. Table 3 contain the measured points.

Table 3: Test matrix. Total number of points 18. *Water mass flow ratio between subcooler and condenser

Water in Evaporator Temperature [ºC]	Water in Subcooler Temperature [ºC]	Water out Condenser Temperature [ºC]	Mass flow ratio*: $m_{w,sub}/m_{w,cond}$
10	10	60	1.00
	20		1.00
	30		1.00
	40		1.00
	50		0.83
	55		0.45
20	10		1.00
	20		1.00
	30		1.00
	40		1.00
	50		0.67
	55		0.30
25	10		1.00
	20		1.00
	30		1.00
	40		1.00
	50		0.63
	55		0.29

4. RESULTS

Figure 3 shows the heating COP and heating capacity accounting with auxiliary consumption. For a given inlet water temperature at the evaporator (Tw,ei) it can be seen a linear relationship between the water inlet temperature at the subcooler (Tw,ci) and the heating COP, which decreases as the Tw,ci increase. For the nominal water inlet temperature at the evaporator (20ºC), heating COP decreases about 35% when passing from an inlet water temperature at the subcooler of 10ºC to 55ºC. This linearity is also observed for a given water inlet temperature at the subcooler and changing the water temperature at the evaporator. In this case, heating COP increase as the temperature at the inlet evaporator increase, which is directly related with the increase of the evaporating pressure. When it passes from 10 to 25ºC, COP increases about 38% (Tw,ci = 10ºC). The maximum measured heating COP is around 6.15, while at the nominal point (Tw,ei=20ºC ; Tw,ci=10ºC) COP is 5.61.

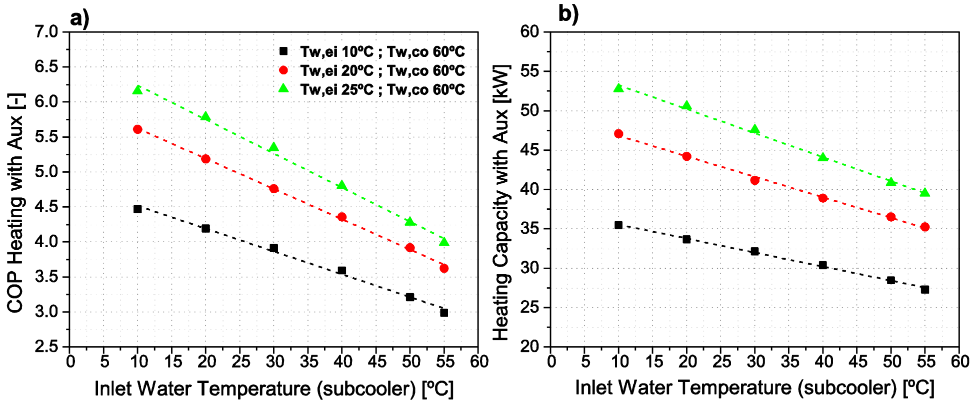

Figure 3: a) COP heating with auxiliary, b) Heating capacity with auxiliary vs. inlet water temperature to subcooler for different water inlet temperature to evaporator

Regarding to heating capacity, it has a similar behavior than the heating COP. The maximum measured capacity is about 52.8 kW, and at nominal conditions is 47.1 kW. For the nominal water inlet temperature at the evaporator, 20ºC, heating capacity decreases about 25% when passing from an inlet water temperature at the subcooler of 10ºC to 55ºC. As in the heating COP, capacity increase as the temperature at the inlet evaporator increase, which is directly related with the increase of the evaporating pressure. When it passes from 10 to 25ºC, capacity increases about 49% (Tw,ci = 10ºC).

Figure 4a) compares the results for the heating COP accounting with auxiliary consumption for the cycle working with subcooling against the results obtained with a cycle working without subcooling, and the same boundary conditions (Tw,ei=20ºC). The highest improvement is about 31%, which corresponds to the highest water temperature lift (Tw,ci=10ºC), with a subcooling of 43.9K. As the inlet water temperature to the subcooler increase, the COP difference between subcooled and non-subcooled cycle decreases, having an improvement about 6.8% for a water temperature lift of 5K (Tw,ci=55ºC), which has a subcooling of 8.6K. Therefore, the highest improvement produced by adding subcooling is at high water temperature lift, since it is here where subcritical cycles working without subcooling has the worst water/refrigerant temperature match, and by adding subcooling makes a great difference in the heat transfer process. One should remember that in this prototype, subcooling is made in a separate heat exchanger from the condenser, so condenser is exclusively used for condensing in all measured points.

Figure 4b) shows the comparison for condensing saturation temperature for the cycle working with subcooling against the results obtained with a cycle working without subcooling, and the same boundary conditions (Tw,ei=20ºC). The condensing temperature is higher for the cycle working with subcooling, being this difference higher at high water temperatures lift, around 2ºC more in the saturation temperature. For low water temperature lift (Tw,ci>50ºC), the condensing temperature in both cases is quite similar. This behavior is related with the amount of subcooling, because, even though condenser is not being used for subcooling, the water is pre-heated at the subcooler before entering to the condenser, so the inlet water temperature at condenser is higher in than for the cycle working without subcooling. Finally, it can be conclude that the benefits obtained from producing subcooling are higher than the COP degradation due to the increase of condensing temperature.

Performance decreases as the water inlet temperature to the subcooler increases due to two reasons:

- Reduction of subcooling
- Increase of the condensing pressure

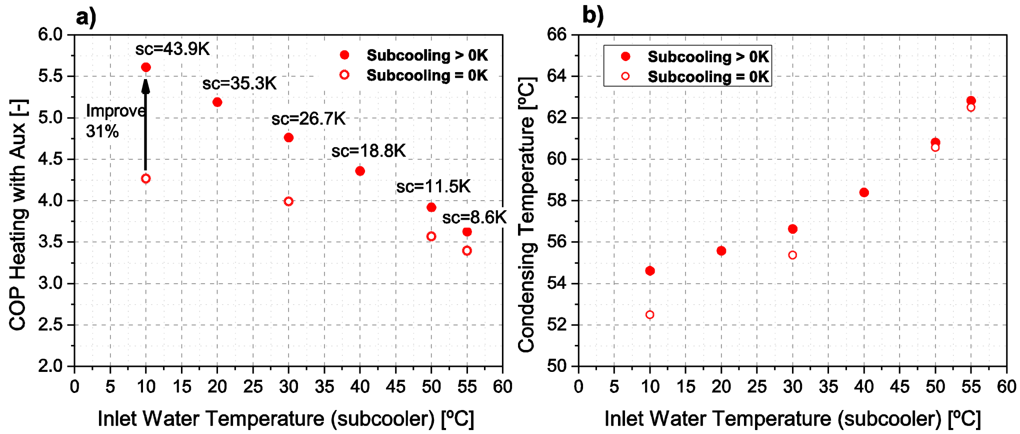

Figure 4: a) COP heating with auxiliary, b) Heating capacity with auxiliary vs. inlet water temperature to subcooler for different water inlet temperature to evaporator

Figure 5a) shows the subcooling depending on the inlet water temperature at the subcooler. It can be seen a linear dependency, where the evaporating temperature has an insignificant influence. Subcooling is directly related with the refrigerant outlet temperature at the subcooler, which is quite close to the water inlet temperature. This means that the subcooler is able to produce the maximum subcooling at all conditions, even when subcooler is partially bypassed in the water side (Tw,ci>50ºC). Subcooling goes from 44 K at the lowest water inlet temperature to 9 K at an inlet water temperature of 55ºC.

One particularity of this application with high subcooling and high evaporating temperatures, is the low refrigerant quality at the evaporator inlet, or even subcooled liquid. Figure 5b) shows the inlet refrigerant quality at the evaporator, where negative values mean subcooled refrigerant. It has a linear dependency with the inlet water temperature at the subcooler, which is related with subcooling, higher subcooling leads to lower refrigerant quality. The refrigerant quality also depends on the evaporator conditions, higher inlet water temperatures at the evaporator leads to lower refrigerant qualities. It can be seen that for the lowest quality values, refrigerant is at subcooled state at the evaporator inlet, and, hence at the expansion valve outlet. On the other hand, the highest refrigerant quality is about 0.4. These big 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 systems.

The system has been working stable at all conditions, even at low refrigerant inlet quality at the evaporator. Nevertheless, further studies with higher evaporating temperatures, and hence with lower refrigerant quality needs to be taken into account.

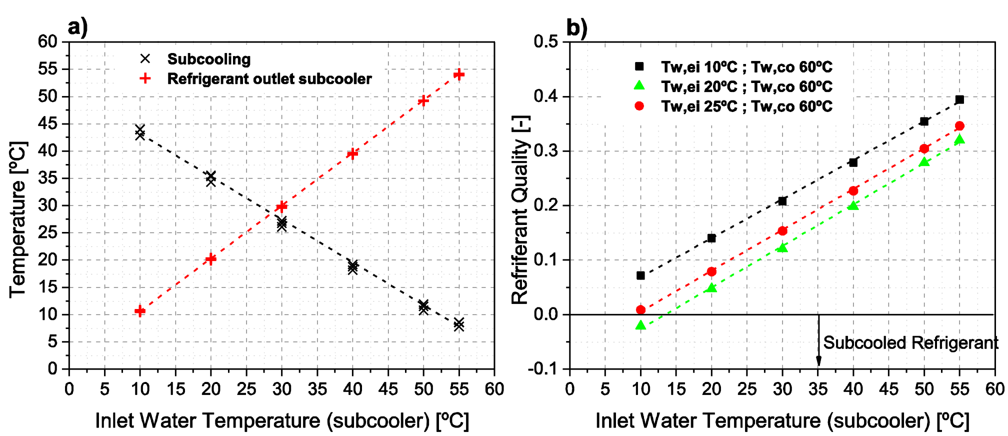

Figure 5: a) Subcooling and refrigerant temperature at subcooler outlet, b) Refrigerant quality at inlet evaporator vs. inlet water temperature to subcooler for different water inlet temperature to evaporator

5. CONCLUSIONS

Transcritical systems working with CO_2 are normally used for SHW production due to its good performance at the high temperature lift. This paper presents the experimental results of a water-to-water heat pump booster prototype for SHW production using Propane as a refrigerant, and using subcooling in order to enhance the heat transfer process in a subcritical cycle. The conclusions drawn from the present study are:

- Heating COP is improved by adding subcooling in the Propane cycle. The benefits obtained from producing subcooling are higher than the COP degradation due to the increase of condensing temperature.
- Performance increase due to subcooling at the nominal point is about 31%. Subcooling about 44 K, $T_{w,ei}$=20ºC and producing hot water at 60ºC from water at 10ºC.
- Performance increase due to subcooling at the nominal point is about 6.8%. Subcooling about 8.6 K. $T_{w,ei}$=20ºC and producing hot water at 60ºC from water at 55ºC.
- COP heating and heating capacity decrease linearly with the inlet water temperature at the subcooler.
- At the nominal point, COP and heating capacity with auxiliary consumption are 5.61 and 47.1 kW respectively. The maximum measured values are 6.15 and 52.8 kW for COP and capacity respectively.
- Subcooling depends mostly with the water inlet temperature to the subcooler, and not with the inlet water temperature to evaporator.
- The maximum subcooling is reached when refrigerant temperature at the subcooler outlet gets closer to the water inlet temperature ($T_{w,ci}$).
- Low refrigerant quality or even subcooled liquid can be find at evaporator inlet with points working at elevated evaporating pressure and high subcooling.

A subcritical cycle with Propane has demonstrate to have a good performance for sanitary hot water production when working with subcooling.

REFERENCES

Cecchinato, L., Corradi, M., Fornasieri, E., Zamboni, L., 2005. Carbon dioxide as refrigerant for tap water heat pumps: a comparison with the traditional solution. *Int. J. Refrigeration* 28(8), 1250-1258.

Coleman, H.W., Steele, W.G., 1999. Experimentation and Uncertainty Analysis for Engineers (2nd ed.)John Wiley & Sons, Inc., New York

Corberán, J. M., Gonzálvez-Maciá, J., Navarro-Peris, E., Pitarch-Mocholí, M., and López-Navarro, A., 2015 Subcooling control : a way to enhance the performance of condensers for hot water production with a high water temperature glide. *In 24th IIR Conference of Refrigeration - Yokohama*.

ECO-CUTE project, http://www.r744.com/assets/link/enEX_ecocute.pdf, (05 of February of 2015)

European Directive 2009/28/EC Of The European Parliament And Of The Council. eur-lex.europa.eu

Kim, M., Kim, M. S. & Chung, J. D. Transient thermal behavior of a water heater system driven by a heat pump. Int. J. Refrig. 27, 415–421 (2004).

IEA Heat Pump Programme Annex 32, Systems concept. Integrated water-to-water propane heat pump installed in a passive house in Southern Norway. www.annex32.net/pdf/ Final_reports/System_Concepts_IEA_HPP_Annex32.pdf.

Justo Alonso, M., Stene, J., 2010. IEA Heat Pump Programme Annex 32. Umbrella Report, System Solutions, Design Guidelines. Prototype System and Field Testing.

Nekså, P., Rekstad, H., Zakeri, G.R., Schiefloe, P.A., 1998. CO2-heat pump water heater: characteristics, system design and experimental results. Int. J. Refrigeration 21(3), 172-179.

Nekså, P., 2002. CO2 heat pump systems. Int. J. Refrigeration 25 (4), 421-427

Rieberer, R., Kasper, G., Halozan, J., 1997. CO2-a Chance for once through Heat Pump Heaters, CO2 Technology in Refrigeration, Heat Pumps and Air Conditioning Systems. IEA Heat Pump Centre, Trondheim, Norway.

Sarbu, I., 2014. A review on substitution strategy of non-ecological refrigerants from vapour compression-based refrigeration, air-conditioning and heat pump systems. Int. J. Refrigeration 46, 123-141.

Schmid, F., 2009. Sewage water: interesting heat source for heat pumps and chillers. In: Energy-engineer FH, Swiss Energy Agency for Infrastructure Plants. Zürich, Switzerland.

Tammaro, M., Montagud, C., Corberán, J.M., Mauro A.W., Mastrullo, R., 2015. A propane water-to-water heat pump booster for sanitary hot water production: Seasonal performance analysis of a new solution optimizing COP. Int. J. Refrigeration 51, 59-69

Acknowledgements

This work has been developed in the 7 framework program of the European Union by the project Next Generation of Heat Pump Technologies (NEXTGHP) grant agreement 307169. The authors give thanks for the given support. 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.