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Comparative analysis of two subcritical heat pump boosters using subcooling in order to increase the efficiency of systems with a high water temperature glide

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Abstract

This paper presents the experimental results obtained from two new heat pump prototypes for sanitary hot water production working with R290 as refrigerant and designed for heat recovery from water sources like sewage water or condensation loops (typical temperature condition between 10ºC to 30ºC). The heat pumps use subcooling in order to enhance the system efficiency. With the proposed strategy it is possible to match the water temperature glide with the refrigerant temperature in a more proper way (it should be taken into account that typically in this kind of system the temperature glide can vary between 10 K and 50 K). The obtained results have shown a high degree of improvement by making subcooling. COP more than 30% higher than the same cycle working without subcooling (Nominal point: inlet water temperature at evaporator is 20ºC and the water inlet/outlet temperature in the heat sink is 10ºC and 60ºC). In order to determine the advantages and disadvantages of each system configuration the performance of the two different prototypes are compared from the point of view of control, reliability, efficiency and cost.

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Keywords: Sanitary hot water, Heat pump, Subcritical, Natural refrigerant.

1. Introduction

An interesting energy efficient alternative to the conventional Sanitary Hot Water (SHW) systems (boilers) 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 [1], 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.

Since the first heat pump came up, scientist have been struggling in order to find a working fluid (refrigerant) that have to satisfy many requirements, like thermodynamic, safety and environmental aspects. In many of the studies, refrigerants like R22 and R134a were used, but nowadays a new concept in the implementation of refrigeration systems is imposed [2]. It requires 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.

The SHW application have a high water temperature lift. In systems with a high risk of legionella (large systems with recirculation circuits, hotels…), the hot water is required to be stored at 60ºC to prevent growth of these bacteria. The city water temperature depends on the region and the period of year. For instance, in Helsinki the city water temperature ranges from 4ºC to 11ºC, while in Athens ranges from 16ºC to 26ºC. For this conditions, many researchers have pointed out the natural refrigerant, CO_2, working in transcritical conditions as an efficient solution due to the high temperature glide in the refrigerant side. This effort has been materialized in projects such as ECO-CUTE in Japan [3]. Works like [4-7] has shown the high efficiency of these cycles at water temperature lifts even higher than 50K. Pitarch et al. [8] compared in a theoretical study the COP penalty of different heat pump systems (CO_2 cycle with different subcritical refrigerants working at subcooling zero) for SHW production when the water temperature at the HP inlet is increased (different water temperature lift). This study shows a higher COP for the CO_2 cycle at high water temperature lift, 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. In the last decade several authors have been studying the optimization of such a system [9-11].

Although subcritical systems (working with zero subcooling) have shown a lower performance for the high water temperature lift in the SHW application, they have been also used for this purpose. Is the case of the commercial heat pump working with Propane, Quantum [12], 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).

The possible benefits of making subcooling have concerned several researchers in the last decade. For instance, Justo Alonso and Stene [13] compares the theoretical calculated COP of a CO_2 transcritical cycle with two different systems working with propane, with and without subcooler, COP is 20% higher when CO_2 is used. Between the two Propane cycles they showed an increase of COP when working with subcooler respect to the one with no subcooling, although they do not say the degree of subcooling.

For a given external conditions, subcooling depends on the active charge of the system (this charge does not include the charge contained at reservoirs like liquid receiver). In this sense, some authors have studied indirectly the effect of a moderate subcooling in the system performance for low temperature lift of the secondary fluid (not for the SHW application), since they have studied the influence of charge on the heat pump performance in systems without a charge receiver [14-16]. From those studies is important to comment Corberan et al. [15,16], who studied from the theoretical and experimental point of view the role of the charge in the system, and pointed out that an optimum charge (and subcooling) exists for a given external condition. For the case of a non-natural fluid there are also some works concerning subcooling [17,18,7]. Cecchinato et al. [7] 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. Nevertheless, none of these publications studied the advantage of making subcooling in subcritical systems in order to take profit from the high water temperature lift in the SHW application (around 50 K). If a recommendation about subcooling is given, it is usually between 5 K and 10 K.

In recent studies, Pitarch et al. [19] presented the experimental results of a Propane water-to-water heat pump prototype for SHW production, in the application of heat recovery from any water source which is an application that recently have received a considerable attention [20]. The prototype was able to produce high subcooling in order to take profit of the high water temperature lift in the SHW application. The subcooling was made in a separate heat exchanger (subcooler). The results showed a significant improvement of performance compared with the Propane cycle working without subcooling, especially in the high water temperature lift. In the nominal point, with a subcooling of 44K and 50K water temperature lift, the degree of improvement is 31%. The COP in the nominal point was 5.61, which is quite competitive with the CO_2 systems for SHW production. In other study, Pitarch et al. [21] used a different heap pump design in order to produce subcooling. For this prototype the control strategy used was totally different because of all the subcooling was produced at the condenser. By means of an additional throttling valve, the active charge on the system can be controlled at any point, and thereby the subcooling. The experimental results clearly showed an optimum subcooling (active charge) for each external condition (water temperatures). Unfortunately, a direct comparison between both alternatives could not be done as the size of the condenser area was different for each heat pump.

The motivation of the present paper is to fairly compare the two HP designs presented by [19] and [21]. In order to do that, a model was developed using the commercial software IMST-ART [22]. This model was validated with the experimental results from Pitarch et al. [21] (subcooling at the condenser). Then, the condenser of the model were increased to have the same heat exchange area than the HP design with a separate subcooler. The results obtained with this model approximation, can be compared with the experimental results obtained by Pitarch et al. [19].

2. Heat pump designs

The prototypes presented by [19] and [21] are a water-to-water HP for SHW production, in the application of heat recovery from any water source working with the natural refrigerant Propane. In order to refer to each HP prototype from this point the following nomenclature will be used:

- SMS heat pump: The Subcooling is made in a separate heat exchanger, the subcooler
- SMC heat pump: The subcooling is made at the condenser (Active charge control)

Both heat pump designs use the same evaporator, compressor, and condenser, with the only difference that in one case a subcooler is connected in series with the condenser. The system was 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 Refrigerant cycle: Subcooler in series with condenser (SMS heat pump)

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) and serves as a charge reservoir. 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 degree of subcooling will depend on the external conditions and the condenser size.

One should notice that the refrigerant density of the subcooled refrigerant is higher than in the two-phase state. Thereby, if a separate heat exchanger is used to produce subcooling, subcooler size can be optimized for refrigerant liquid, so it has an appropriate refrigerant velocity for heat transfer. In this way, subcooler requires smaller plate pitch than at the condenser.

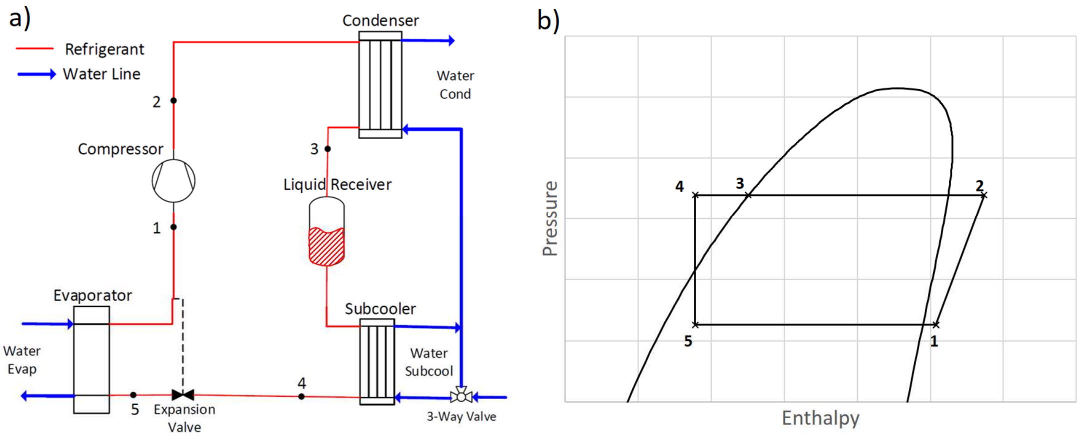

Figure 1: Heat Pump subcooler in series with condenser a) Scheme, b) P-h diagram.

2.2 Refrigerant cycle: Subcooling made in the condenser (SMC heat pump)

Figure 2, shows the scheme of the water-to-water heat pump to make subcooling in the condenser. The evaporator, compressor, condenser and liquid receiver are the same than in the previous case, but for this case the subcooler was removed from the system. A throttling valve is located between the condenser and the liquid receiver, it introduces an active control of refrigerant charge in the system, and hence it can be used to set the subcooling at the condenser independently from the external conditions. In a system with a fixed area for heat exchange in the high pressure side, this feature allows the condenser to dedicate the optimum needed area for subcooling at each condition, while the SMS system has a fixed heat exchange area for subcooling.

Liquid receivers are normally charge reservoirs used as passive elements to accommodate the changes in the active refrigerant charge due to changes in the operating mode or changes at the external conditions. In this case, the main functions will be to accommodate the changes in the active charge due to variations on the degree of subcooling at the condenser and to assure saturated liquid conditions at the outlet of the throttling valve.

The pressure at the inlet of throttling valve (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: 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.

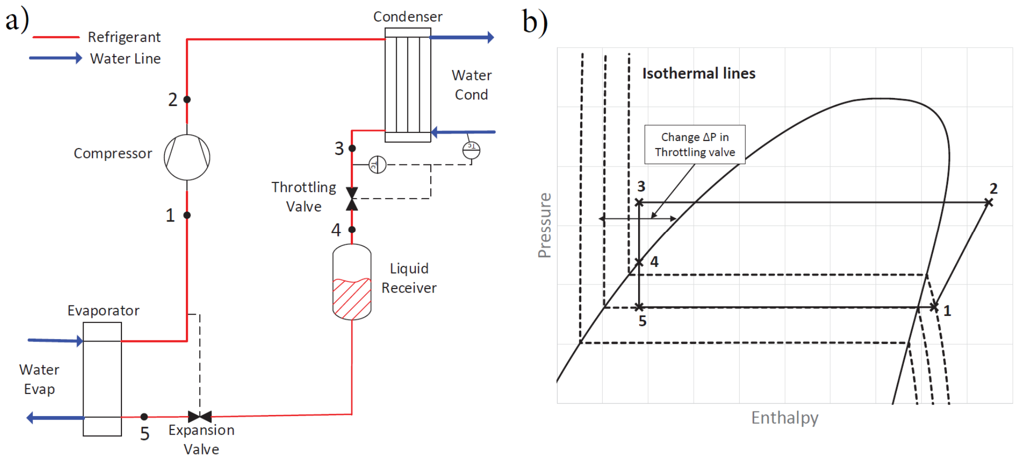

Figure 2: Heat Pump subcooling controlled by a throttling valve a) Scheme, b) P-h diagram.

3. Numerical Model

For the prediction of the heat pump performance, the commercial IMST-ART software were used [22]. It is a heat pump modelling software which main features are fast calculation and accurate evaluation of the refrigerant units. The model incorporates a number of sub-models representing the different parts of the heat pump: compressor, condenser, evaporator and expansion valve, the definition of these components are based on the geometric characteristics of them and for compressors the common information supplied by manufacturers. The calculations are based on steady state conditions. For more detailed description of the model see [23].

Table 1 shows the main components of both systems. The compressor is scroll type and it is modelled by the AHRI coefficients provided by the manufacturer. The heat exchangers (HXs) are modelled by introducing the type of HX (Plate), the size (height and width), the plate pitch, the number of plates, the HX material, etc. The expansion valve is modelled as an isenthalpic pressure drop in order to fulfil the desired superheat, in this case 10K. The throttling valve cannot be modelled, but the subcooling can be introduced as a parameter, so in practice the HP making subcooling in the condenser can be modelled with the IMST-ART software.

Table1: Components of the HP system

Component	Type	Size
Compressor	Scroll (2900 rpm)	29.6 m3h-1
Condenser	BPHE Counterflow	3.5 m2
Subcooler	BPHE Counterflow	0.87 m2
Evaporator	BPHE Counterflow	6 m2
Liquid Receiver	-	7 l
Expansion Valve	Electronic EV	5 – 60 kW
Throttling Valve	Electronic EV	5 – 60 kW

The heating COP calculation does not take into account the consumption of the auxiliary components unless it is indicated.

3.1 Model validation: Subcooling made in the condenser (SMC)

For the validation of the SMC heat pump model, the experimental results from Pitarch et al. [21] were used. Figure 3 shows this comparison for COP and condensation pressure. The results are presented as a function of the subcooling and for different water inlet temperatures to the condenser (Tw,ci). The inlet water temperature to the evaporator (Tw,ei) is fixed to 20ºC and the water mass flow rate to the evaporator is fixed to 7000kgh-1.

There is a good agreement between the experimental and theoretical values for COP, COP dependency with the subcooling is captured by model. COP increases with subcooling up to a maximum, and then starts to decrease and the optimum subcooling is reproduced with a deviation lower than 2 K). The maximum discrepancies are found at high water inlet temperatures to the condenser (Tw,ci=50ºC), being less than 4%.

Regarding to the condensing pressure, the theoretical results also match with the experimental ones. At low subcooling, it increases slowly with subcooling, and after a certain value of the subcooling the condensing pressure starts to increase at a higher rate. This point of inflection is where the maximum COP takes place. As in the heating COP, the greatest discrepancies are found at high water inlet temperatures to the condenser (Tw,ci=50ºC), being less than 2%.

Table 2 shows the heating COP, the heating capacity and the compressor consumption for the model and the experimental values working at the optimum subcooling. It can be seen a great agreement for the three variables, COP discrepancies are below the 1%, for Tw,ci 10ºC and 30ºC, while the heating capacity and the compressor consumption are all below 2% of discrepancies.

3.2 Model adjustment to the heat pump with subcooler (SMSL)

Comparing the experimental results obtained by Pitarch et al. [19] and [21], the SMS heat pump configuration (with subcooler) has a slightly higher performance than the SMC heat pump configuration (subcooling at condenser). For instance, at the nominal point (Tw,ei=20ºC and warming water from 10ºC to 60ºC), the SMS heat pump gave a heating COP of 5.61, while the COP for SMC was 5.35.

Nevertheless, the experimental results obtained with the separate subcooler configuration cannot be fairly compared with the SMC heat pump configuration, since the first one has 25% more of heat exchange area on the high pressure side.

Table 2: SMC mode validation at the optimum subcooling

	COPh SMC exp.	Qh [kW] SMC exp.	Wc [kW] SMC exp.	COPh SMC model	Qh [kW] SMC model	Wc [kW] SMC model
Tw,ci = 10ºC	5.44	46.31	8.51	5.43	45.57	8.39
Tw,ci = 30ºC	4.75	40.98	8.64	4.79	41.55	8.68
Tw,ci = 50ºC	3.91	35.96	9.21	4.03	36.54	9.06

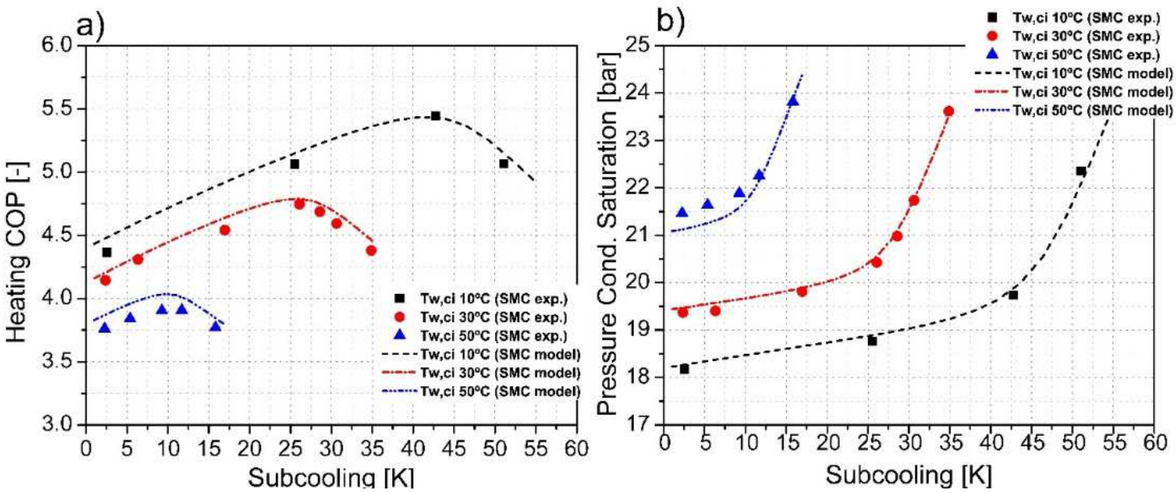

Figure 3: Model validation. a) Heating COP, b) Condensing pressure

In order to define a fair comparison between both systems the condenser of the SMC model was enlarged in order to have the same heat exchange area than the condenser and the subcooler of the SMS HP prototype together. In this sense, it’s the results of both systems can be compared directly.

In order to have an equal heat transfer area in the high pressure side without changing significantly the heat transfer coefficient the height of the condenser was increased. The other alternative, increasing the number of plates, affects the refrigerant velocity, and hence the heat transfer coefficient. Therefore, the condenser height will pass from 0.476 m to 0.591 m, a 25% longer (25% more area). The area of the modified condenser will be 4.37 m2, the same than the condenser and the subcooler together. For the sake of clarity, the SMC model with the enlarged condenser will be referenced as SMCL, the Subcooling Made in Subcooler with Larger condenser. One should notice that even if the subcooler area is added to the condenser, the plate pitch of the heat exchange area dedicated to the subcooling in the SMS system is different to the SMCL system.

4. Results

4.1 High efficient subcritical heat pump alternatives to produce SHW

Figure 4a) shows the heating COP as a function of the subcooling for the SMC and SMCL models (different condenser height) and the experimental results for the SMS HP configuration. COP increased for the whole range of subcooling for the SMCL model (condenser height 0.59m). At low and high subcooling, the COP improvement is moderate, for instance, in the case of Tw,ci=50ºC at subcooling higher than 12 K, COP is practically the same for both condenser heights. The maximum COP improvement is found at the optimum subcooling.

As reported by Pitarch et al. [21], if the SMS configuration is compared with the SMC with less heat exchange area, the SMS system has a greater performance. This situation changes if the system is equipped with a larger condenser with an equivalent heat exchange area to the SMS (the SMCL model). In this case, the performance of SMCL is still below the results for SMS at Tw,ci=10ºC, but at higher Tw,ci, SMCL have higher performance than the SMS for water inlet temperatures of 30ºC and 50ºC.

Figure 4b) shows the condensing pressure as a function of the subcooling. Condensing pressure is reduced for the whole subcooling range for the enlarged condenser model (SMCL). This decrease is more significant at the optimum and low subcooling. At Tw,ci=50ºC, the condensing pressure at the optimum subcooling decreased below the corresponding experimental pressure for the SMS HP design. At lower Tw,ci (30ºC and 10ºC), the condensing pressure at the optimum subcooling is still higher than the corresponding experimental pressure for the SMS HP design.

Table 3 shows the heating COP (COPh), the heating capacity (Qh), the heat corresponding to the subcooling (Qsub), the heat corresponding to condensing + desuperheat (Qcond), and the compressor consumption. These parameters are shown for the SMS HP prototype (experimental) and the results for the SMCL model (height of 0.59 m) working at the optimum subcooling. The heating capacity is slightly higher at the SMCL model, while the compressor consumption depends on the point.

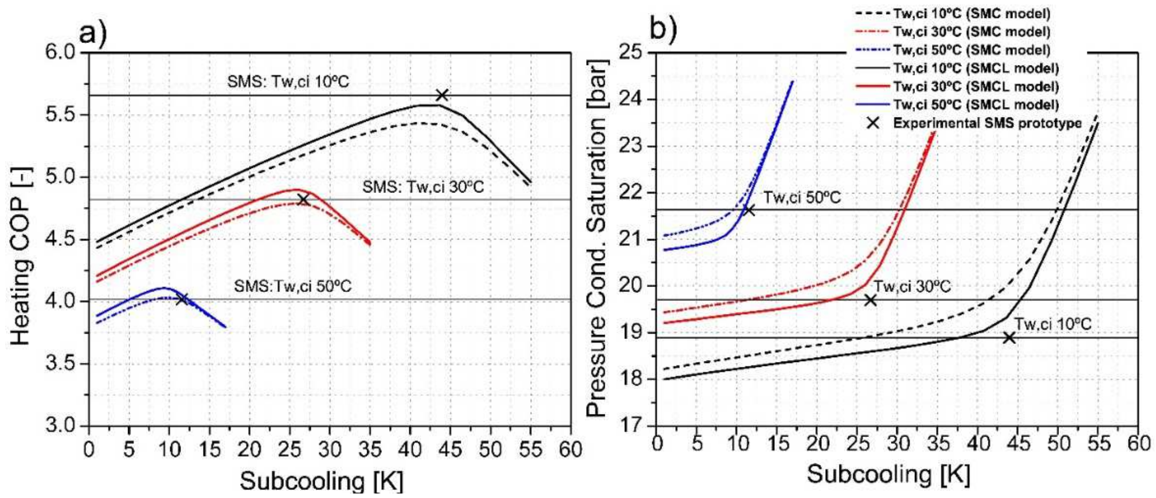

Figure 4: Results for two condenser height. a) Heating COP, b) Condensing pressure

Table 3: Experimental SMS and SMCL model comparison at the optimum subcooling

	COPh	Qh [kW]	Qsub [kW]	Qcond [kW]	Wc [kW]	COPh Model	Qh,SMCL [kW]	Qsub,SMCL [kW]	Qcond,SMCL [kW]	WcSMCL [kW]
Tw,ci=10ºC	5.66	45.72	12.07	33.65	8.08	5.58	46.15	12.47	33.68	8.26
Tw,ci=30ºC	4.82	40.07	7.30	32.77	8.31	4.91	41.76	8.03	33.73	8.52
Tw,ci=50ºC	4.00	35.95	3.26	32.68	8.98	4.11	36.53	3.15	33.38	8.89

Figure 5 shows the condenser area dedicated for subcooling as a function of subcooling for different water inlet temperatures to the condenser (Tw,ci) for the SMCL and the fixed area for the SMS. The area for subcooling depends strongly on the Tw,ci. For instance, the dedicated area to produce 15 K of subcooling goes from 5% to 65% for Tw,ci=10ºC and Tw,ci=50ºC respectively. When the system works at the optimal subcooling the dedicated area to subcool is lower as Tw,ci increases, for instance, the dedicated area at the optimum subcooling for Tw,ci=10ºC is 47%, while it is 24% when Tw,ci=50ºC. Therefore, the heat exchange area dedicated to produce subcooling at the optimum point will change depending on external parameters, such as Tw,ci. This might be one of the reasons in order to explain the performance of SMCL compared with the SMS system, which improves at higher Tw,ci. It is observed that for all the cases the area dedicated for subcooling in the SMS system is lower than in the SMCL system. This is probably derived from the fact that the subcooler of the SMS system was designed to only subcool the refrigerant, and the plate pitch of that heat exchanger is thinner than for the SMCL, this fact allows a higher heat transfer coefficient and hence lower heat transfer area for this part of the cycle.

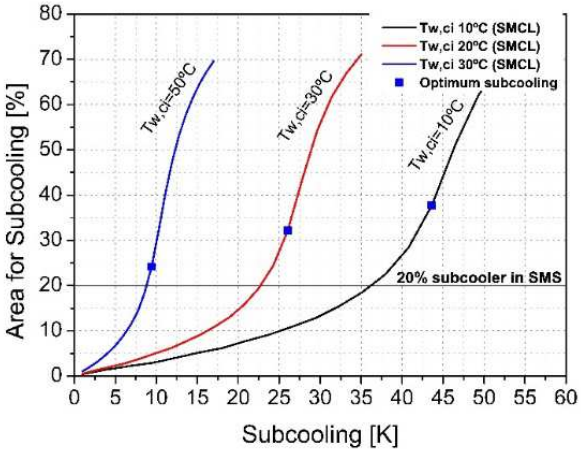

Figure 5: % of area dedicated for subcooling as a function of subcooling and Tw,ci (SMCL and the fixed area for SMS)

Figure 6 shows the heating COP and the condensing pressure as a function of subcooling and the condenser size (height). The inlet and outlet water temperature to the condenser are fixed at 30ºC and 60ºC respectively. For a constant condenser size, COP increases with subcooling up to a maximum value, the optimum subcooling is around 25 K for all condenser size. Condenser size has different effects depending on the degree of subcooling:

1) At low subcooling, COP slightly increases with condenser size. Beyond a certain size, the increase on performance is insignificant.

2) At the optimum subcooling, COP increases for the whole studied condenser size range. The degree of improvement is higher at smaller sizes.

3) At high subcooling, COP slightly increases at small condenser size, but after a certain point, condenser size has no effect on COP.

Regarding to the condensing pressure, it decreases with condenser size for low and optimal subcooling, but at high subcooling the condensing pressure has no effect with condenser size.

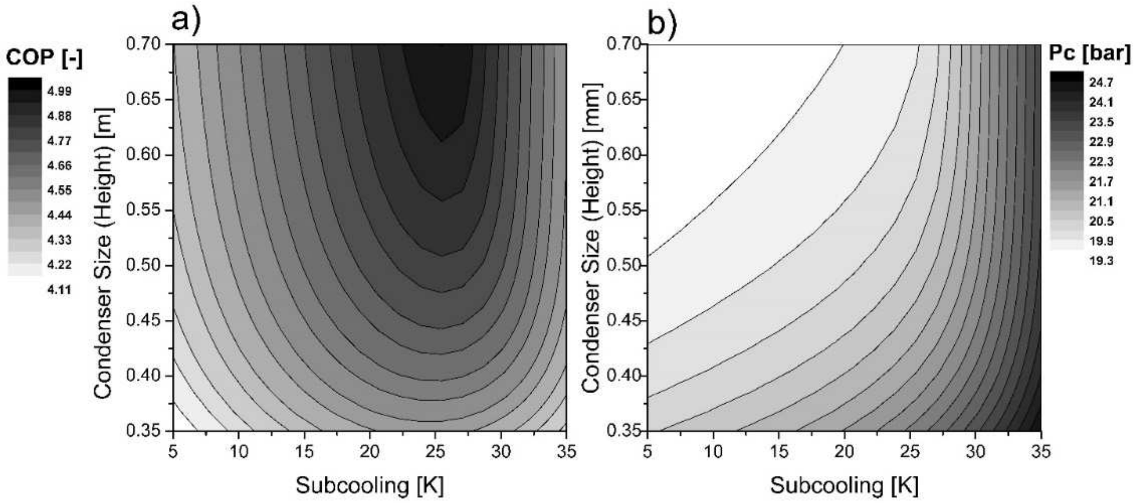

Figure 6: a) Heating COP and b) Condensing pressure as a function of subcooling and condenser size (Tw,ci=30ºC)

Figure 7a) shows the condenser dedicated area as a function of condenser size for: de-superheat, two-phase and subcooling. The dedicated area for de-superheat and subcooling slightly decreases with condenser size, about 2% when going from a condenser height of 0.35 m to 0.7 m. On the other hand, the dedicated area for the two-phase flow increases 4%. The percentage of dedicated area for subcooling is decreasing with size, although the total area increases. Figure 7b) shows the refrigerant and water temperature profile in the condenser for two different condenser size, 0.35 m and 0.70 m. It can be seen a better temperature match between refrigerant and water at the bigger condenser.

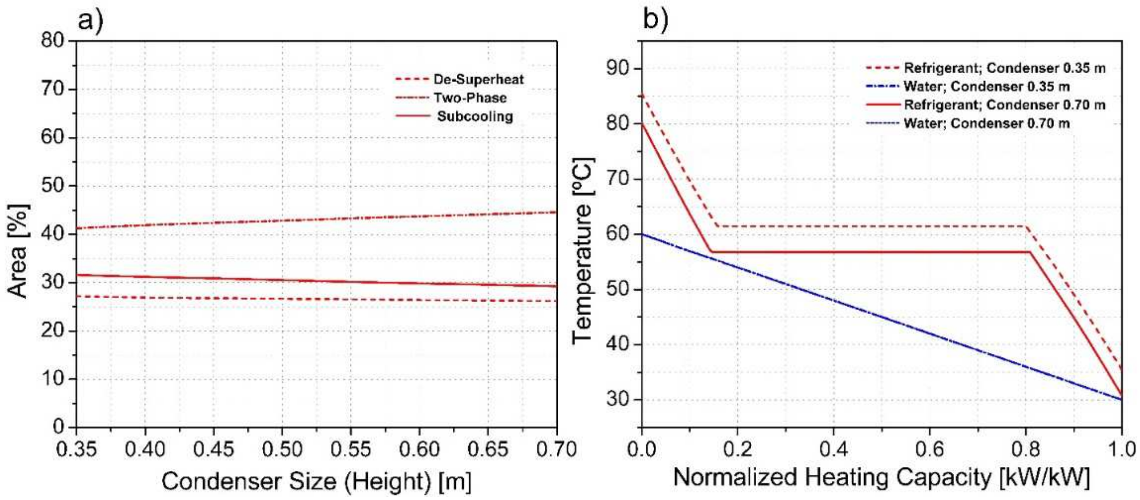

Figure 7: a) Condenser dedicated area as a function of condenser size b) Refrigerant and water temperature profile in the condenser (Tw,ci=30ºC)

From all these results, it can be concluded that, depending on the working conditions, the SMC design could obtain a higher performance than the SMS due to the ability to adjust the area dedicated for subcooling at any point. Nevertheless, the SMC design has other disadvantage that needs to be considered. SMC is not able to work in the optimum condition when the refrigerant outlet temperature at the condenser (Tw,ci) is lower than the evaporating temperature (for further information see Pitarch et al. [21]). This can be the case when the system is working at high evaporating temperatures (working point with a Tw,ei=30ºC), In that cases, the evaporation temperature of the system could be lower than the values obtained with the SMS system reducing the efficiency. There are several technics in order to avoid the behavior described above. The high water temperatures at the evaporator inlet could be used to directly warm the low city water temperatures previous entering at the condenser in that way the subcooling operation of the system is reduce and the water temperature at the inlet of the evaporator is reduced. Another possibility would be to place the liquid receiver at the evaporator outlet and control the subcooling directly with the expansion valve. In this way, the evaporating pressure would not be limited by the refrigerant temperature at the condenser outlet. However, the stability of this system configuration needs to be proved.

Comparing these results with catalogue data of other R744 heat pump dedicated to sanitary hot water production (Q-ton from Mitsubishi) it can be seen that the subcooling approach could have improvements from 5% for water temperature lift of 50 K up to 15% for water temperature lift of 30K.

4. Conclusions

This paper presents a comparison between the different HP designs to produce SHW using natural refrigerants. The study includes two subcritical systems designed specifically to work at high water temperature lift. The main conclusions are:

- The SMCL system is able to optimize the heat exchanger area in a better way.
- The SMCL has shown a significant reduction of the COP working at high subcooling and high evaporating temperatures.
- SMLC has one less heat exchanger, but an additional electronic valve and a more sophisticated control algorithm than the SMS system. In order to choose between both systems, cost criteria should be the most important factor. From the point of view of energy efficiency, both systems have shown very competitive and close COP with only minor differences for some specific working conditions.
- The effect of the sizing of the condenser of the SMLC configuration was analysed. The obtained results have shown that in order to be able to reach the higher COPs obtained in this work, dimensioning the condenser properly is critical.

The obtained results show that the proposed systems constitute very efficient solutions that could increase the efficiency of the systems available for the production of sanitary hot water.

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