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Heat Pump control comparison of a conventional superheat system over a subcooling control system for Domestic Hot Water production

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ABSTRACT

The water heating consumption of the average European dwelling accounts at this moment for 14.5 %, according to the European Commission, whereas heating and cooling percentage account for 64.7 %. The European Union, with the introduction of the Nearly Zero Energy Buildings (NZEB) concept, intends to decrease in a great amount the total energy consumption of the residential sector acting over the percentage associated to heating. This will have an effect on the water heating consumption, increasing its consumption percentage.

Therefore, in order to reduce the residential sector consumption and more concretely the percentage associated to water heating consumption, it will be key the use of highly energy efficient technologies since it is not possible to reduce the water heating energy consumption by acting on the demand without the risk of not satisfying the user comfort. In that sense, heat pumps are a technology of increasing interest. It is considered as renewable when its SPF is above 2.5, its production does not depend on external factors (as sunlight) and will play a key role in the cities of the future since it has the capability of acting as a booster from heat distribution networks.

At this moment, the heat pumps available in the market for domestic hot water production using subcritical cycles are cost-effective but its design does not differ significantly from the used for heating applications.

In the frame of the European project NEXTHPG, a heat pump system specifically designed for domestic hot water production with a variable subcooling control was developed. In this work, a comparison between the energy consumption at the system level of this prototype with a commercial water-to-water heat pump for domestic hot water production is performed.

The results show a big difference in terms of energy efficiency between both cases. The variable subcooling case results in a 26.68 % positive energy efficiency difference regarding the conventional case when compared in their maximum efficiency point. The sizing of the heat pump-storage tank duality plays an important role since it could lead to an energy efficiency loss of almost 17 %. These results show clearly the advantages of the innovative heat pump model in front of the conventional heat pumps used for domestic hot water production.

Keywords: *domestic hot water, heat pump, energy efficiency, optimization, subcooling control*

1. Introduction

At the present time, in Europe the energy consumption percentage associated to Domestic Hot Water (DHW) does not represent a big percentage (14.5 %), compared with the one associated to Heating Ventilation and Air conditioning (HVAC), which is quite higher (65 %) [1]. But, with the concept of Near Zero Energy Buildings (NZEB) this percentage will highly increase in the near future since the percentage associated to heating will be highly reduced and thus the energy consumption of the DHW production will play a key role. Heat Pump (HP) technologies are for DHW application an interesting alternative to the conventional systems e.g. boilers due to its considerable higher efficiency. Another fact that makes HP so attractive is that they can work with low-global-warming-potential refrigerants such as propane and $CO_{2;}$ as well as their capability to be integrated with renewable energies, as i.e. photovoltaic or solar thermal panels.

There is no study where subcooling was considered a design parameter but one research paper [2]. The propane water-to-water heat pump booster developed in the NEXTHPG project has a specific variable subcooling control unlike the conventional HP control through superheat. A significant improvement in performance is observed compared with a propane cycle without subcooling, either in the theoretical and experimental analysis that can be consulted in [2–4]. The COP in the nominal point was quite high and competitive [3]. Thus, the following objective consists on analyzing its operation coupled with a real system in an installation.

In this way, the aim of this work is to compare the conventional control strategy with the innovative one developed under the project NEXTHPG concerning a common system design [5] and called as Subcooled Heat Pump (SHP). Therefore, the conventional control strategy of a constant temperature lift in the condenser is compared with the innovative one concerning a variable subcooling control with a variable temperature lift in the secondary of the condenser. Moreover, the proper sizing of both, SHP and Thermal Energy Storage (TES) system, are studied for both control strategies analyzed from the point of view of a global energy performance indicator.

This document is split in four main parts: introduction, methodology, results and discussion and conclusions. In the first section, a brief summary of the issue and state of art is stated. In the second section the model and the process are commented. Within this section the reader can understand how the model was created in TRNSYS, the assumptions and requirements imposed for the optimization and fully understand the analysis of the results. A final section, the results and discussion, where the results are shown in graphics and tables to present it clear to the reader. Finally, in the conclusions section a brief summary of the work and the most important ideas are gathered.

2. Methodology

Two integrated system model were created using TRNSYS transient simulation tool [6], one for the conventional case and another one for the variable subcooling control case. In *Figure 1* the TRNSYS integrated system model for the variable subcooling control case is shown, which looks alike the constant temperature control lift case.

a) b)

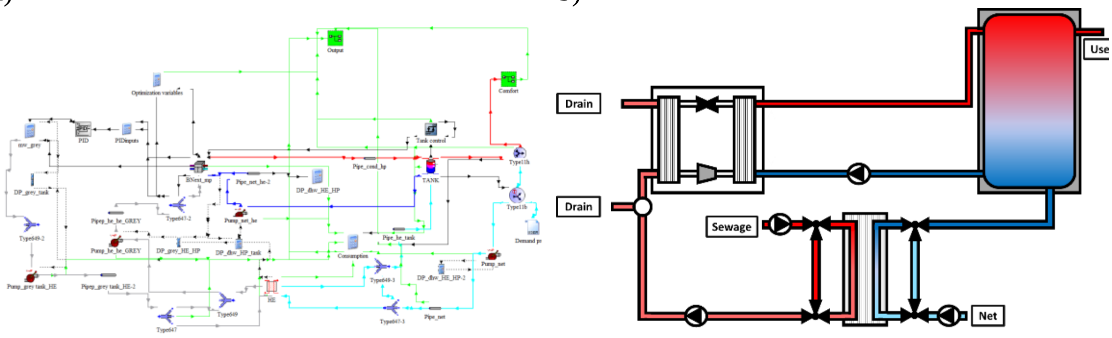

Figure 1. Illustrative description of the model: a) TRNSYS integrated system model and b) sketch of the model.

In the model, see Figure 1, can be identified the main components of the system: the SHP, the TES system and the heat exchanger. The SHP outlet is connected to the inlet of the tank and the inlet of the SHP is also connected to the tank. The water from the net is preheated in the heat exchanger before entering to the tank. The hot stream of the heat exchanger comes from the sewage water that serves as secondary fluid in the evaporator after the heat exchange process. The control of the system consists of a hysteresis controller that depends on the temperature of the node selected in the TES, which in this case was the fifth node, and the set-point temperature. The set-point temperature in the tank is fixed to 60 ºC while the user demand is considered at 45 ºC, reason why a mixer valve is included in the model. The user demand corresponds to a 20 multi-family houses of 1.98 people per house and has been obtained with DHWcalc software [7], considering one year period.

The SHP has been experimentally tested and fully characterized in the laboratory. It is a 47 kW nominal capacity SHP with a nominal COP of 5.71 working with R290 (Propane) as refrigerant. The SHP has been implemented as black box in an own created TRNSYS type and operates as the real SHP prototype of the NEXTHPG project since the correlations of the experimental model where introduced in the TRNSYS type created [8]. The temperature lift in the evaporator of the SHP is fixed to 4.5 K since an exhaustive study has been performed in order to determine the optimal temperature lift for the best SHP performance. The tank is modeled as a stratified storage tank considering 15 equal volume segments called nodes using the TRNSYS type 60. The height of the inlets and outlets are placed alike a commercial tank: inlet to the TES from the SHP at 90 % of the total height, outlet from the TES to the SHP at 10 %, outlet from the TES to the user at 95 % and finally inlet from the net at 5 %. Finally, the heat exchanger is used to preheat the cold-water inlet in the tank with the sewage water considering an unlimited availability. The heat exchanger operation is only considered when the SHP is switched on and there is DHW demand from the user, reason why the bypasses in the heat exchanger were added.

Four circulation pumps are introduced in the model, two connecting sewage-heat exchanger-evaporator, another connecting the heat exchanger with the TES system and another one connecting the TES system and the SHP condenser.

The optimization variables considered for the study concern the sizing of the SHP and the TES system. Thus, one of these variables corresponds to the scale SHP and the other to the volume of the tank. The scale SHP considered varies from 0.1 to 1 whereas the range of volumes considered is 0.1-2 m^3.

2.1. Performance indicators and comfort requirements

The performance indicator is an own defined indicator, SPF_{user}, used for the evaluation of the whole system, since the objective consist on analyzing the system global performance concerning the energy consumption the user is paying for and the useful energy that it receives from the system. The SPF_{user}, similar to the Seasonal performance factor (SPF) defined in "SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector" (SEPEMO-Build) is a ratio that assess the global energy performance of the system [9]. The mentioned indicator is defined in Equation (1).

$$SPF_{user} = \frac{Q_{user}}{W_{compressor} + W_{pump1\_sewage} + W_{pump2\_sewage} + W_{pump3\_net} + W_{pump4\_HPtoTES}} \quad (1)$$

Where Q_{user} is the useful heat that the user receives, calculated as the energy contained in the water flow exiting the mixer valve to the user at 45ºC. The quotient of the formula corresponds to the total energy consumption of the facility, being W the electrical consumption of the component i.e. SHP or pumps.

Three different requirements need to be fulfilled in order to consider the case in the optimization: annual percentage of discomfort, hourly discomfort and SHP starts. The objective by fixing these restrictions consists on one hand guaranteeing the user comfort and the appropriate operation of the SHP. Regarding the user comfort two restrictions have to be fulfilled. First the annual percentage of discomfort, that has to be below 5 % and it is calculated according to Equation (2). Second, the hourly discomfort whose maximum value is fixed in 30 min/year for each hour of the day (00:00-24:00), which means a maximum value of almost 5 seconds per day out of comfort. There is considered discomfort when the water temperature provided to the user is below 45 ºC, as shown in Equation (2). Finally, the restriction of SHP starts is settled in a maximum of 9 starts per hour.

$$Annual_{discomfort} = \frac{\sum m_{user\ (T<45°C)}}{\sum m_{user}} < 0.05 \quad (2)$$

There were some cases discarded also from the study due to the results showed. It occurs that for big volumes of the TES and high values of scale SHP, the SHP never reaches stable conditions concerning the nominal mass flow that should circulate through the secondary circuit because the time that it remains switched on is too low. Nevertheless, this cases show efficiency performance drops (SPF_{user}).

2.2. Considerations and assumptions

Type 60 was chosen to simulate the TES. It has been widely used and experimentally validated in research publications, such as [10], and is defined in TRNSYS user manual and mathematical reference as *"the most detailed tank model available in the standard TRNSYS library"* [6]. The aspect ratio, the relation between the height and the diameter, has been considered fixed with a value of 4 as being a value considered acceptable and common in tanks as well as advised in literature for proper stratification [11]. Finally and regarding the TES model, a de-stratification-conductivity value equal to 2 W/mK was used; calculated according to Equation (3) and following the recommendations of TRNSYS mathematical reference [6]. A complete explanation of the modelling of this value can be found in [12, 13].

$$k_{destratification}(W/mK) = k_{tank-wall} \cdot \frac{A_{tank}}{A_{fluid}} \quad (3)$$

In this case study, it is assumed that the availability of sewage water is unlimited and always at 30ºC that corresponds with an application of district heating for example. Limited availability of sewage water, in order to consider other applications, will be a future work case. Other considerations are the temperature drop in the condenser, constant through a PID controller at 4.5K and the initial conditions of the tank, supposed to be full and at 50ºC.

The TES and the SHP are coupled through a direct connection. The direct connection serves as reference as the most efficient case, and in order to consider any other indirect connection option the system global efficiency percentage reduction values obtained in [14] should be applied. These values are of general application. An indirect connection is required by the EN 1717:2000 due to the risk of putting the refrigerant in potential contact with the potable water.

3. Results and discussion

In the following, the results for the comparison of the two cases proposed regarding the performance indicator chosen, SPF_{user}, are presented (see *Figure 2* and *Figure 3*). The results are presented for the optimization variables considered, the scale SHP and the volume of the tank, as above explained. It should be noted that not all the cases are included in the plot, since there are some that not comply with the restrictions yet commented and some others that make the SHP to not reach the nominal conditions, as above explained.

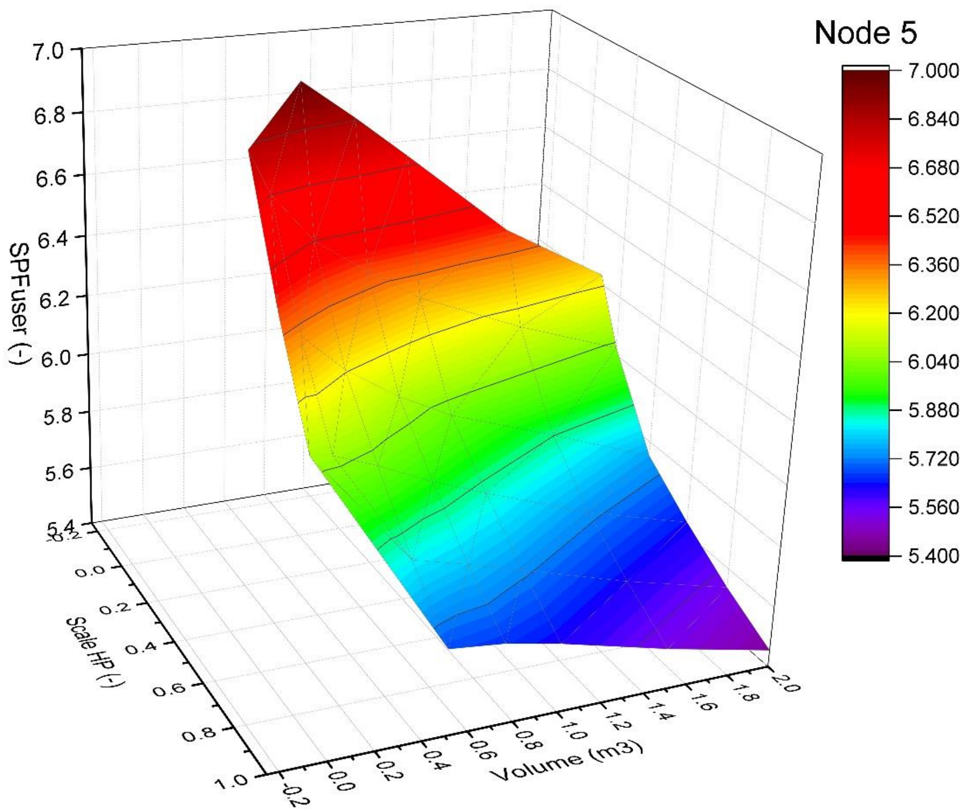

Figure 2. Results for the subcooling control case with the temperature control in the fifth node of the tank.

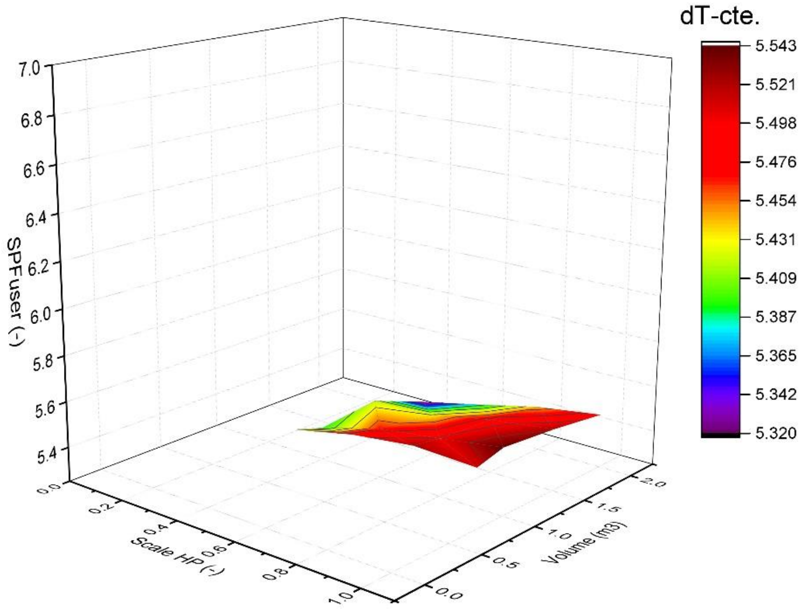

Figure 3. Results for the constant temperature lift case.

Looking at *Figure 2* and *Figure 3* the difference in SPF_{user} is the most remarkable aspect. The case with a variable subcooling presents a considerably higher value compared with the case of a constant temperature lift of 5 K. It is absolutely due to the fact that the SHP presents a higher SHP efficiency (COP). Additionally, the second case needs more time to heat up the water at the demand temperature of 60 ºC since when using a constant temperature lift, the water must be heated in steps of 5 K hence the working time increases substantially and for reaching the demand temperature the lowest temperature in the tank will be of 55 ºC. Whereas the variable temperature lift in the variable subcooling case allows the system to faster charge the tank at the demand temperature thanks to the higher temperature lift.

Another remarkable aspect that can be extracted from the Figures consists on the number of cases that overcame the imposed restrictions. As it can be seen, the case for a constant temperature lift shown in *Figure 3* has a considerably lower number of cases that comply with the three restrictions compared with the subcooling variable case shown in *Figure 2*. Most of the cases that not fulfill with the necessary requirements of the second case are the ones for a small-scale SHP value. Since small scale SHP values need more time to heat the needed volume storage tank and are not able to rapidly charge the tank after a peak demand before the next demand peak and thus overcome the comfort requirements mainly. Accordingly, and theoretically, when increasing the temperature lift to 10 K or 20 K more cases will comply with the requirements and the SPF_{user} will increase, but this will be analyzed in further studies.

Regarding the results for the optimization variables, a common tendency is observed in both cases. First, it is observed a decreasing tendency of the SPF_{user} with the volume of the tank independently of the scale SHP considered. Secondly, a decreasing tendency of the SPF_{user} is observed with the increasing of the scale SHP. Both cases, the constant temperature lift case and the variable subcooling case coincide in the latter affirmations.

The decreasing tendency with the volume of the tank is due to the increasing of the thermal losses and stratification losses with the increase of the volume of the tank. But this is mainly due to the fact that with a bigger tank volume a higher energy needs to be supplied by the SHP to reach the temperature demand and thus the working hours of the SHP increase and with it the energy consumption of the system.

Next, in *Table 1* are presented the optimal values for the tank volume for each scale SHP along with the SPF_{user} resulting for both cases, constant temperature lift and variable subcooling case.

Scale HP	**Constant temperature lift**		**Variable subcooling**	
	Volume (m^3)	**SPFuser**	**Volume (m^3)**	**SPFuser**
0.1	-	-	0.75	6.85
0.2	-	-	0.3	6.68
0.4	0.75	5.44	0.2	6.41
0.6	0.75	5.49	0.2	6.08
0.8	0.75	5.52	0.5	5.83
1	0.75	5.51	0.5	5.68

Table 1. Optimal volume and SPF_{user} for each scale HP of each of the cases analyzed.

The results for the optimal tank volume and SPF_{user} introduced in Table 1, are accordingly with the tendencies observed and above explained. The best SPF_{user} result correspond to the minimum tank volume that fulfill with the requirements for each of the values of scale SHP analyzed. So, the lower the tank volume the lower the energy consumption of the system because it needs less energy to charge the tank at the demanded temperature. It is also important to remark that the proper sizing of the SHP for the application plays a key role in the system efficiency optimization and could lead to efficiency losses from 2.15 % to almost 20 % regarding the variable subcooling case and around 1 % for the constant temperature lift.

Finally, it has to be remarked the higher efficiency that the variable subcooling case presents over the conventional case which accounts for 26.68 % comparing the optimal cases. In case of comparing the optimal case for each scale SHP, the variable subcooling case is always a minimum value of 3.09 % over the conventional one and a maximum value of 17.83 % over. With this results the advantages of a variable subcooling control over the conventional is clarified.

4. Conclusions

In this work, a comparison between two different ways of producing DHW through a SHP system are compared. On one hand the conventional way with a constant temperature lift in the condenser and in the other hand the innovative way introduced through a variable subcooling control that allows the SHP to work with a variable temperature lift in the condenser. Both cases are compared in terms of system global efficiency (SPF_{user}) for different sizes of SHP and tank volume, with the objective of finding the optimal sizing of both SHP and tank.

The results show a big difference in energy efficiency between both cases, being the variable subcooling case much more energy efficient than the conventional case. Concretely, 26.68 % difference in the optimal point of both cases. Regarding the sizing of the SHP and tank it is advisable to proper size the SHP-tank duality since the losses in energy efficiency for the subcooling case could lead to a maximum of almost 17 % from 2.15 % considering only the optimal values for each pair of scale SHP and tank volume. Finally, the optimal values are obtained for the lowest tank volume that complies with the restrictions of comfort and SHP starts imposed.

The results presented show clearly the advantages of a variable temperature lift in the secondary of the condenser and thus the potential of the innovative variable subcooling control over the conventional control strategies. Also, it is also remarkable the importance of the appropriate sizing of the duality of the size of the SHP and tank and the penalization of oversizing the SHP or the tank volume.

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