Optimal design and operation of a central domestic hot water heat pump system for a group of dwellings employing low temperature waste heat as a source

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Abstract

In this work, a study of an energy recovery system from a low-grade temperature source based on heat pumps for domestic hot water is done. The main components of the system are a pre-heating heat exchanger, an optimized heat pump for domestic hot water production, and a variable-volume storage tank. A model has been developed in TRNSYS to analyse the best configuration and control strategy of the system in order to satisfy the profile demands of 10, 20, and 30 multifamily houses, which are considered as a representative target market for this type of application. From this analysis, the influence of the design/sizing parameters on the system CO_2 emissions has been obtained and a design criterium for their minimization has been supplied. Finally, a sensitivity analysis based on different net and heat source temperatures has been done in order to estimate the generalizability of the proposed solution. The obtained results show that this kind of system, with the proper design, sizing, and operation, offers potential CO_2 emissions reductions by a factor of almost five compared to a conventional gas boiler system but a bad system selection could reduce this potential benefit up to 25%.

NOMENCLATURE

10H: ten multifamily houses

20H: twenty multifamily houses

30H: thirty multifamily houses

DHW: domestic hot water

Tei: water inlet temperature at the evaporator [ºC]

Teo: water outlet temperature at the evaporator [ºC]

Tci: water inlet temperature at the condenser [ºC]

Tco: water outlet temperature at the condenser [ºC]

Tnet: water mains/net temperature [ºC]

Tsource: water heat recovery (district heating) temperature [ºC]

T_{st}: stored water temperature

Thot: water temperature under system conditions [ºC]

Tuser: water temperature supplied to the user [ºC]

Tdemanded: demanded water temperature [ºC]

Qcond: heat pump heating capacity [kW]

COP_{hp}: heat pump coefficient of performance [–]

COP_{sys}: system coefficient of performance, [–]

α: control level rate [–]

Tset: stored water control temperature

Subscripts

ST: storage tank

HP: heat pump

SHP: water-to-water high-efficiency heat-recovery heat pump optimized for water heating applications (up to 90 ºC)

HE: heat exchanger

1. INTRODUCTION

As corporate, national, and international goals move towards reducing carbon footprints and the effects of global warming, renewables and low-carbon solutions will replace current fossil fuel alternatives within the energy sector [1].

The building sector accounts for 21% of the total energy consumption, and almost 15% of that percentage is related to water heating for domestic hot water (DHW) (with a significant increase of that share being projected as room heating loads decrease) [2]. In addition, according to the European Environment Agency (EEA), the residential sector is responsible for 11.5% of total CO_2 emissions. Thus, improving water heating production within this sector is becoming unavoidable for a sustainable future, overall, taking into account that the employment of more efficient technologies and the introduction of passive houses mean that DHW becomes the most important thermal need of a building.

Most residential water heaters are equipped with conventional boilers that use electricity or fossil fuels. In spite of their simplicity and affordability, they are neither energetically nor environmentally desirable. Compared to these solutions, Heat Pumps (HPs) are seen as a potential alternative for water heating [3] as: they have demonstrated high efficiency, can operate in standalone form, avoid the need for a back-up technology and are the best solution in order to recover waste energy or use heat from a low temperature district heating network. Therefore, they will constitute in the following years a key system in order to reduce the current emissions associated to this type of application.

HPs have been widely used within many heating and cooling applications. However, most applications are characterized by low secondary fluid temperature lifts. In applications where high secondary fluid temperature lifts are required, an efficient alternative is the use of transcritical cycles capable to operate with high temperature lifts [4]-[7]. Nevertheless, these heat pumps are not so flexible and show a poor efficiency when the temperature lift is reduced. Another more “traditional” solution using subcritical systems consist of using a progressive water heating processes of the water contained in a tank [8]. This alternative has the inconvenient of not being able to produce water at the temperature required by the demand conditions directly and from the efficiency point of view is not so optimal but in the other side this kind of systems are less expensive.

Recently, researchers have been working on the optimization of subcritical HPs for that kind of applications and HPs with coefficients of performance (COPs) similar that those found with transcritical cycles at high secondary fluid temperature lift have been obtained. This is possible thanks to the applications of high (optimal) degrees of subcooling [9], [10] and the proper redesign of the HP. This kind of HP has been called a "Subcooled *H*eat *Pump*" (SHP).

In addition, DHW production have a high variability in profile demand along the day. This characteristic change significantly the sizing criteria of heat pumps for this application compared to the criteria followed for HP in heating and cooling systems where a constant demand profile is common.

In that sense, one of the most difficult tasks linked to the sizing of DHW installations is the characterization of a representative demand profile, especially within the building sector, where DHW can vary significantly from hour-to-hour and high peaks are expected not only during the day but also during different days of the week and through the year. The geographical situation, social and economic factors, type of building, and number of users, among others, are some of the factors that condition the DHW demand. The efficiency of the system is highly dependent on the DHW profile. Thus, having a representative load profile is crucial in order to achieve high user satisfaction and a proper and efficient sizing of the rest of the components. [11] review several approaches used in order to estimate representative DHW load profiles.

Once this representative DHW load profiles are defined, and as a consequence of their variability, one-step further deals with the integration of the HP with a water storage tank (ST). This point becomes essential in order to avoid oversizing of the system and allows the decoupling of DHW consumption from DHW production [12]. However, it will add irreversibility's to the system as the temperature water production in the heat pump must be higher in order to compensate the ST thermal losses [13], cost and control complexity.

One step further is related to the integration of the HPs within a recovery system in order to satisfy the DHW demand. In that sense, a holistic approach is essential as optimize the system performance means more than optimize its components individually. Most of the works found in the literature dealing with DHW production [14] - [20] from heat recovery uses a specific system configuration but it is difficult to find general information about the sensitivity of the obtained results to the selected configuration (size, control strategy, system topology…).

This paper analyses heat recovery systems for DHW production in the residential sector in order to supply gidelines to define the optimum system configuration and estimate which could be the penalty in the efficiency obtained from a bad design. The basic components of the system are the new designed SHP working with propane, a heat-recovery heat exchanger (HE) and a variable-volume storage tank (ST). The selection of the components has been done in order to maximize the efficiency of the application. In addition, the sizing of the components and the definition of the operating control strategy in order to minimize the global energy consumption of the system are investigated.

The results presented shows the influence of the proper sizing and the control strategy in the final system performance and they can yield valuable information for the next years, guiding the proper system design when the substitution of common DHW production technologies by HPs takes place in order to meet the requirements of new policies aimed towards a decarbonized energy sector. In this point it should be noted that the obtained differences could be higher if a conventional subcritical heat pump were used.

The main characteristics of the considered system are:

- <u>The DHW load demand</u>: it is based on stochastic models and has been obtained with DHWcalc software and validated with the profile obtained with SynPro [21] [22]. Different profiles considering 10, 20, and 30 multifamily houses are included. A yearly one-minute step profile is used.
- <u>Heat Source</u>: sufficient availability of the heat source is assumed (as it is the case in district heating or sewage water).
- <u>Direct heat recovery system</u>: Heat exchanger preheat initially the tap water.
- <u>Heat pump:</u> the SHP is able to modulate the subcooling in order to maximize the efficiency as a function of the condenser secondary fluid temperature lift. SHP have performances comparable to transcritical HPs at high condenser water temperature lift while it maintains common high performances of subcritical systems when operating under low secondary temperature lifts.

- Storage tank: variable volume ST has been employed, that is, a fully-mixed tank with all the water stored at the same temperature. This type of tank reduces the losses compared to stratified tank.

2. Developed Model

This section describes the model developed in TRNSYS [23] in order to perform the analysis of this work. The section is divided in three parts, in the first one the description of the external model conditions is done, in the second one the type created for SHP model integrated in TRNSYS is explained and finally, the developed system model is described in detail.

As external conditions, the model will assume that the water from the net will be at 10ºC in almost all the simulations, the availability of heat source water will be considered as enough and therefore, the

2.1. Heat source and heat sink characterization

The study considers enough availability of the heat source. This situation could correspond to low-temperature district heating or heat recovery from sewage water. The temperature of this water service could be from 40ºC in the first case up to temperatures slightly higher than the ambient like sewage water heat source. Based on that, the design temperature has been set to 20 ºC that could be considered a critical temperature for these applications.

Regarding the heat sink, the DHWcalc tool [21] has been used to generate the profiles.

In order to validate the generated profiles a one-minute resolution DHW load profile facilitated by the developers of the software SynPro [22] was used. This profile was based on 20 apartments with an average occupancy of 1.95 persons/house (39 people in total) and the agreement between both profiles was good.

The considered annual average energy consumption is 576 kWh per person, which is based on a total hot water consumption (at 45 ºC) of 54.1 litres per person per day and a net water temperature equal to 10 ºC.

Four categories of tapping are considered. The mean flow rates and the volumes (durations) per tapping are based on VDI 2067 [24] and have been set as follows :

- Hand-washing/cleaning: this is a small fraction which groups all the water mass flow rate usages of around 3 lpm with a duration of up to 5 min.
- Shower: this category includes medium flow rate tapping (9 lpm) and durations of up to 10 min
- Bath: includes consumption with medium hot water flow rates (9 lpm) but long durations (more than 25 min).
- Cooking: this category refers to low water mass flow rates and medium durations (15 min).

The probability of each event (tapping) considers socio-economic factors and is based on data presented in [22].

Table 1 collects the main characteristics of the draw-offs considered in the profile used as the base of this work.

Table 1: Draw-off types and characteristics based on [24]

Type of draw-off	Temperature [ºC]	Mean flow [lpm]	Probability [%]	Duration [min]	Standard deviation σ [l/h]
Hand-washing/cleaning	45	3	45	5	2
Shower	45	9	17	10	2
Bath	45	9	5	25	2
Cooking	45	3	33	15	2

DHWcalc tool applies probability distribution based on a Gaussian distribution to generate the profiles of the different days. The weekend–weekday variation is set to 120% and the seasonal variations are obtained with a sinusoidal function with a maximum in the day 45 and a sine amplitude equal to 10%.

Figure 1 represents the DHW demand load at 45 ºC for 20 multifamily houses based on the above. The demand is shown under a daily scale instead of the minute scale used due to resolution reasons.

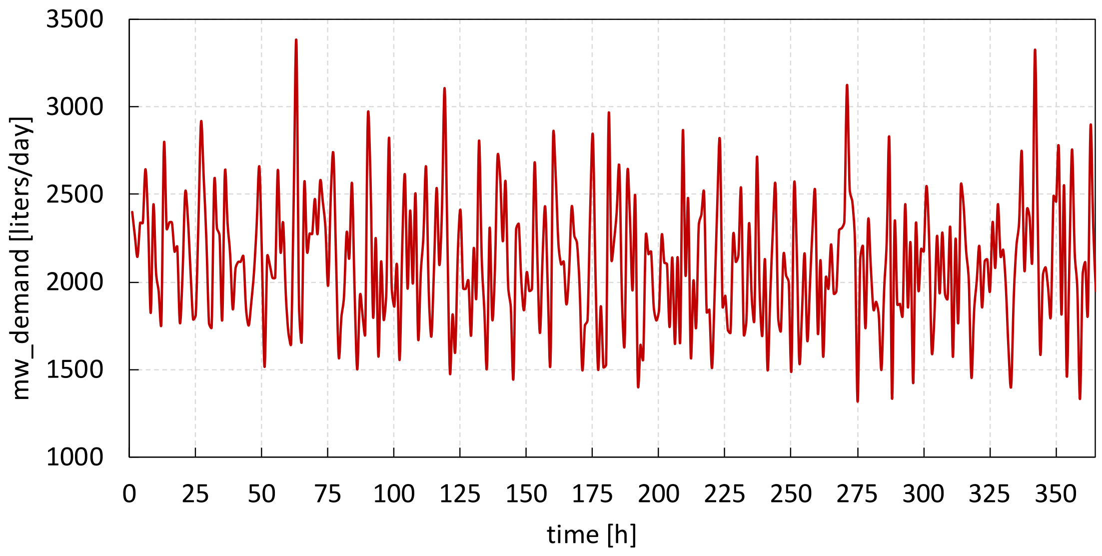

Figure 1: DHWcalc generated daily profile for 20 multifamily-house DHW consumption at 45 °C and Tmains of 10 °C

The work presented considers 20 multifamily houses (20H) as the base case. Additionally, a sensitivity analysis of the number of houses, types of draw-offs, and the timeframe used in the generation of the profiles is included.

Table 2 collects the main characteristics of the DHW profiles used in this study for hot water consumption at 45 ºC and a mains temperature of 10 ºC. These cases corresponds to the following profiles:

a. Reference case (20H)
b. Demand for 10 houses (10H)
c. Demand for 30 houses (30H).
d. Demand for 20 houses during 10 consecutive years (20HY)
e. Demand determined by the software default draw-off values and the same average consumption as 20H (20Hdef).
f. Application of the solution for one third of the 20H demand, which could happen on holidays (20HL).

Table 2: Main DHW load characteristics of the profiles used in the study [24]

DHW load profile	20H	10H	30H	20HY	20HL	20Hdef
Annual energy demand [kWh]	31283.2	15641.6	46924.8	31283.2	10427.7	31283.2

Daily DHW demand at 45 ºC [l/day]	2110	1055	3165	2110	703.33	2110
Instant peak at 45 ºC [l/min]	36.1	24.08	33.05	(max. 50.5, eighth year)	12.03	48.68
Profile timeframe	1 year	1 year	1 year	10 years	1 year	1 year

2.2 Model of the SHP

The considered SHP used in the study is a high-efficiency heat-recovery water-to-water HP working with propane for water heating applications (up to 90 ºC). The SHP is composed of the typical HP components including a liquid receiver placed between the evaporator and the compressor (ensuring zero-superheat conditions). The expansion valve controls the subcooling, which is optimized based on external conditions. Thanks to the variable subcooling application, this SHP is capable of operating under high COPs for high and low temperature lifts (see figure 2) [25]. This characteristic is especially relevant for energy recovery applications where a HE is placed before the HP condenser and the water inlet temperature to the heat pump can change significantly.

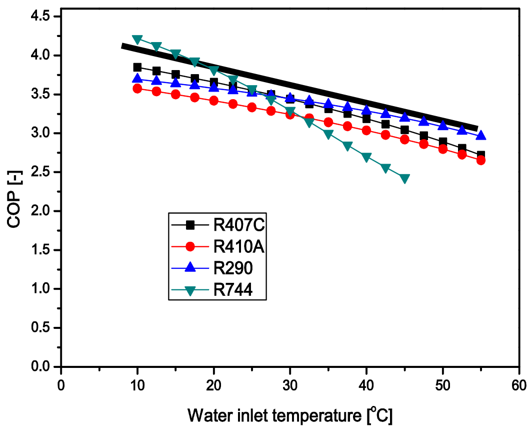

Figure 2: Typical HP performances as a function of the refrigerant used with variation of the inlet water temperature for water heating at 60 °C. The heat pumps working in subcritical conditions works with constant subcooling of 5 K. The line without marks correspond to the expected performance of the subcooled heat pump.

A new TRNSYS type has been implemented in order to describe properly the behaviour of this heat pump. To build the model, the following procedure has been followed:

1. A prototype of the SHP has been experimentally tested in more than 50 different experimental conditions [26].
2. IMST-ART software [27] has been used in order to develop a SHP model.
3. The developed model has been validated with the experimental campaign.
4. Once the model has been validated, it has been used as a virtual lab in order to simulate 3569 operating conditions of the heat pump.
5. A correlation has been obtained from these 3569 cases. This correlation is the one used in the developed TRNSYS type.

Table 3 collects the range of temperatures of the secondary fluid (sink and source) included in the study. Notice that only the feasible cases have been considered (for instance, the outlet water temperature at the evaporator is always lower than the inlet water temperature at the evaporator). Furthermore, superheat is fixed to zero and the optimal subcooling is calculated as a linear function of the water temperature lift at the condenser according to [28].

Table 3: External conditions simulated in IMST-ART to obtain the type of HP for TRNSYS

	Temperature Range [ºC]
Evaporator water inlet temperature	5–45
Evaporator water outlet temperature	2–42
Condenser water inlet temperature	5–60
Condenser water outlet temperature	40–90

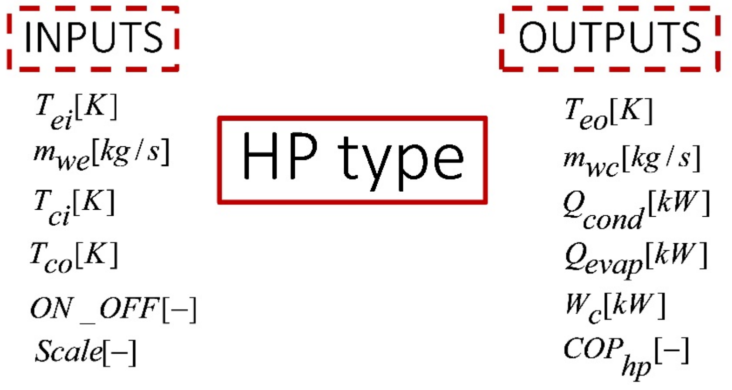

Figure 3: HP type inputs and outputs

Figure 3 shows the inputs and outputs of the HP model. T_{ei} is the water inlet temperature at the evaporator (the district heating/wasted temperature after the HE), m_{we} is the water mass flow rate through the evaporator (the district heating/wasted mass flow rate), T_{ci} is the water inlet temperature at the condenser (the mains temperature after the HE), T_{co} is the water outlet temperature at the condenser (the temperature that goes to the ST), and *Control* is the ON/OFF signal given from the operation control of the installation (further detailed in the control section).

T_{eo} is the evaporator water outlet temperature, m_{wc} is the hot water mass flow rate produced by the HP, Q_{cond} is the heating capacity of the HP, Q_{evap} is the heat recuperated by the HP in the evaporator, and W_c is the electric compressor consumption.

Scale is set as a parameter and represents the heating capacity size of the HP. It is the parameter used in the optimization.

The obtained correlations for the cooling capacity is given by Eq. 1:

$$Q_{evap\_corr} = f(T_{eo}, T_{ei}, T_{ci}, T_{co}) = c_0 + c_1 T_{ei} + c_2 T_{eo} + c_3 T_{co} + c_4 T_{ci} + c_5 T_{ei}^2 + c_6 T_{eo}^2 + c_7 T_{ci}^2 + c_8 T_{ei} T_{eo} + c_9 T_{ei} T_{ci} + c_{10} T_{eo} T_{co} + c_{11} T_{eo} T_{ci} + c_{12} T_{co} T_{ci} \quad (1)$$

Nevertheless, from a practical point of view, it would be more practical to write down the equation (1) in terms of the evaporator water mass flow rate instead of the evaporator outlet temperature. In order to do that, the outlet evaporator temperature can be expressed according Eq. 2.

$$T_{eo} = T_{ei} - \frac{Q_{evap}}{m_{wevap} cp} \qquad (2)$$

And solving using an computer algebraic system “CAS” Eq. 3 is obtained:

$$Q_{evap\_corr} = f(m_{we}, T_{ei}, T_{ci}, T_{co}) =$$
$$= \frac{1}{c_6} \cdot \left[\sqrt{\left(\frac{-m_{we} \cdot cp}{2} \right) \cdot (A1 \cdot cp - 4 \cdot A2 - 4A3)} - \left(m_{we} \cdot cp + c_{10} T_{co} + c_{11} T_{ci} + c_2 + 2 \cdot (c_6 + {}^{c_8}\!/_2) \cdot T_{ei} \right) \right] \qquad (3)$$

where:

$$A1 = \left(m_{we}^2 \cdot cp^2 + 2 \cdot m_{we} \cdot \left(c_{10} T_{co} + c_{11} T_{ci} + c_2 + 2 \cdot (c_6 + {}^{c_8}\!/_2 \right) \cdot T_{ei} \right)$$

$$A2 = \left(\begin{array}{l} c_0 c_6 + c_1 c_6 T_{ei} - {}^1\!/_4 \cdot (c_{10}^2 T_{co}^2 + 2c_{10} (c_{11} T_{ci} + c_2 + c_8 T_{ei}) \cdot T_{co} + \\ + c_{11}^2 T_{ci}^2 + 2c_{11} (c_2 + c_8 T_{ei}) \cdot T_{ci}) \end{array} \right)$$

$$A3 = \left(\begin{array}{l} (c_{12} c_6 T_{ci} T_{co} - {}^1\!/_4 \cdot (c_2^2 + 2c_2 c_8 T_{ei} - 4 \cdot (c_3 c_6 T_{co} + c_4 c_6 T_{ci} + \\ + c_5 c_6 T_{ei}^2 + c_6 (c_7 T_{ci} + c_9 T_{ei}) \cdot T_{ci} - \frac{c_8^2 T_{ei}^2}{4})) \end{array} \right)$$

c0 = 273.4477, c1 = –7.406263d-0, c2 = –4.687551, c3 = 0.2076619, c4 = 2.646866, c5 = –2.310626e-03, c6 = 1.010544e-02, c7 = –1.053072e-03, c8 = 9.182375e-03, c9 = 1.296127e-03, c10 = –1.655369e-03, c11 = –7.495019e-03 and c12 = 6.849771e-04 and Only the relevant terms in order to reproduce the heat pump behaviour have been maintained.

Thereafter, the heating capacity is obtained from the correlation in Eq. 4.

$$Q_{cond\_corr} = f(T_{eo}, T_{ei}, T_{ci}, T_{co}) = c_0 + c_1 T_{ei} + c_2 T_{eo} + c_3 T_{ci} + c_4 T_{ei}^2 + c_5 T_{eo}^2 + c_6 T_{ci}^2 + c_7 T_{ei} T_{eo}$$
$$+ c_8 T_{ei} T_{ci} + c_9 T_{eo} T_{co} + c_{10} T_{eo} T_{ci} + c_{11} T_{co} T_{ci}$$

where *c0 = 3.2379e02, c1 = –7.439879e-01, c2 = –4.946912, c3 = 0.2076619, c4 = 2.645119, c5 = 9.859616e-03, c6 = –9.507318-04, c7 = 9.589339-03, c8 = –1.344549e-03, c9 = –5.224201e-03, c10 = –7.669214e-03, c11 = 7.365229e-03.*

The HP hot water mass flow rate capacity is obtained from Eq. 5:

$$m_{wcond} = \frac{Q_{cond\_corr}}{cp \cdot (T_{CO} - T_{Ci})} \cdot Scale \qquad (5)$$

The compressor consumption is calculated from Eq. 6:

$$W_{c\_corr} = f(T_{eo}, T_{ci}, T_{co}) = c_0 + c_1 \cdot (T_{co} - T_{ci}) + c_2 T_{eo} + c_3 T_{co} + c_4 T_{eo}^2 + c_5 T_{co}^2 + c_6 T_{co}^3 + c_7 T_{co} T_{eo}^2 + c_8 T_{eo} T_{co}^2$$

(6).

co=-2.345e+02;c1=-1.320e-02;c2= 3.663;c3=-1.035;c4=-1.284e-02; c5=2.801e-03;c6=6.055e-06;c7=3.717e-05;c8=-3.018e-05

The HP performance (COP_{hp}) calculation is done directly from the above correlations according to Eq. 7.

$$COP_{corr} = Q_{cond\_corr} / W_{c\_corr} \qquad (7)$$

Summing up, the heat pump is described by the equations (2), (3) (5) (6) and (7) .

Finally, the real values of the variables are calculated by applying the scale factor as shown in Eqs. 8 to 11.

$$Q_{evap} = Q_{evap\_corr} \cdot Scale \qquad (8)$$

$$Q_{cond} = Q_{cond\_corr} \cdot Scale \qquad (9)$$

$$W_c = W_{c\_corr} \cdot Scale \qquad (10)$$

$$COP_{hp} = COP_{corr} \qquad (11)$$

Calculations of the type shows deviations lower than 4% in almost all the cases.

2.3 Model Description

The system under analysis is presented in figure 4, the scheme shows a direct heat recovery from the net through a heat exchanger followed by a heat pump in charge of suppling the rest of the required energy.

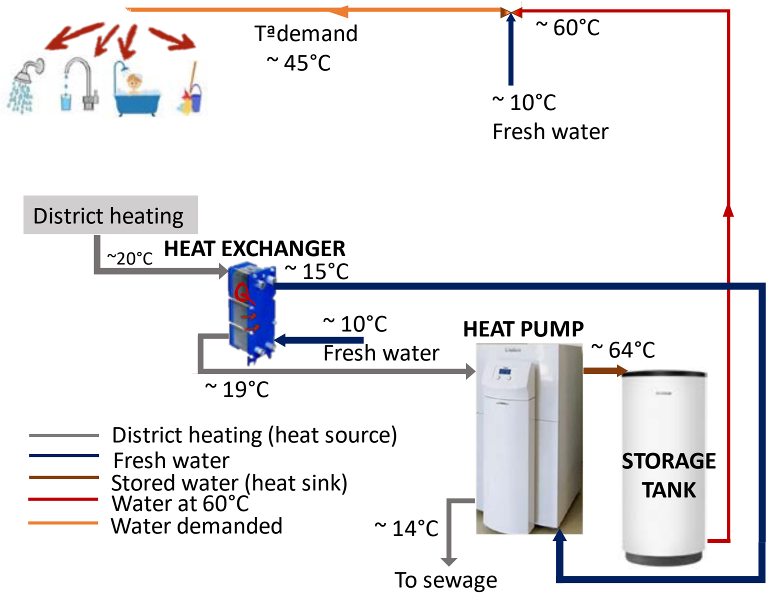

Figure 4: Scheme of the system under analysis with the approximate values of the temperatures.

The main components of the system are:

- Heat exchanger with an efficiency of 0.75 (type 5b). This heat exchanger will allow a first energy recuperation

- The SHP model presented previously: The water evaporator mass flow rate was defined in order to have a water temperature difference in the evaporator of 4.5ºC. Thi heat pump with the characteristic curve show in figure 2 is especially indicated when a net water pre-heater is installed and that temperature could change significantly depending on external parametes

- Storage tank of variable volume (type 39): with an aspect ratio of H/D=4 and a heat loss coefficient of 0.8 W/m2K (based on Spanish regulation). The storage tank assumes an ambient temperature of 20ºC. The storage tank has been selected in order to maximize the efficiency of the system, therefore, contrary to the more common stratified tanks, this tank only have one inlet and outlet and maintain a uniform temperature inside of it.

- Water pumps (Type 742): used with an efficiency of 0.3. Only the pressure drop of the heat exchangers were considered in order to evaluate their consumption.

In order to define the load profile, the stochastic model described previously has been used. The service water temperature was considered at 45ºC and the productions was fixed based on the target of having a minimum storage water temperature of 60ºC (legionella regulation restriction for this type of systems [29]).

The simulations uses a time step of 1 minute (as a consequence of the profile characteristics longer time steps could due to not size properly the system) and include 1 year simulation period.

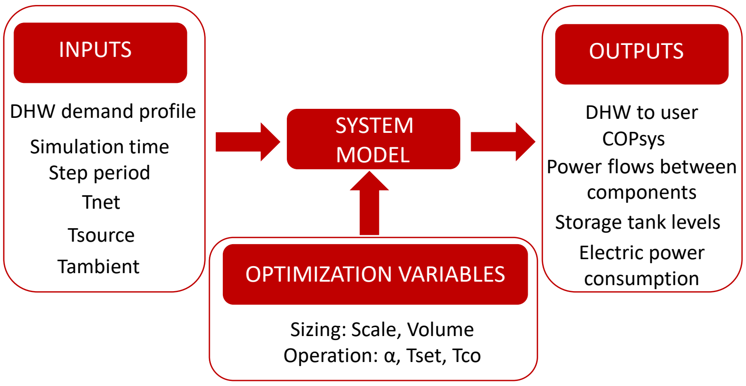

Figure 5 Structure of the model

Fig.5 shows a scheme of the structure of the used model showing the input, the output and the optimization variables. In that scheme, scale is the size of the heat pump, volume is the size of the tank, alfa is the tank level when the heat pump switches on, Tset is the water tank temperature and Tco is the condenser outlet temperature.

COPsys is the COP of the whole system and is defined as:

$$COP_{system} = Energy/Electric = \int_{t=0}^{t=simulationtime} (Q_{ST\_out})/(W_{c\_corr} + \sum W_{pumps}) \quad (12)$$

One should notice that the "useful energy" introduced by the system is a result of the heat exchanged in the pre-heating HE and the heating capacity of the HP after the ST losses.

Regarding the control algorithm, the main source of irreversibility of the system arises from the addition of a ST. Thus, the control of the installation is based on the minimization of the temperature and the time that the water is stored in the ST required to satisfy the demand. As the temperature of the stored water increases, so does the irreversibility. The minimum possible temperature of stored water considered in this work is 60 ºC. Hence, the control set temperature is T_{set} = 60 ºC.

In order to maintain a temperature of at least 60 ºC at the ST, the outlet temperature of the condenser in the HP needs to be some degrees higher. Specifically, based on the simulation results, the minimum outlet temperature to achieve this requirement stably is 64 ºC. Thus, from all the simulations, the water outlet temperature at the condenser is set to 64 ºC.

The minimum time for which the water is stored is dependent on both the volume of the ST and the level of the water inside the tank. Dynamic control based on the water level inside the tank is required and both the volume and the level of the tank are parameters to be optimized. This control parameter is called α and expresses the control level as a percentage of the volume according to Eq. (1).

$$Control\_level = \alpha \cdot Volume \tag{1}$$

where *Volume* is the capacity (size) of the tank. The determination of the optimal α is also an objective of this work.

A maximum ST level is also required to avoid overproduction; the maximum level set is based on the production capacity of the HP in one-time step as indicated in Eq. 2.

$$\max\_level = Volume - mw\_cond \cdot step \tag{2}$$

where *mw_cond* is the HP water mass flow rate production for a determined scale and *step* the simulation time step.

Finally, to preserve the durability of the HP, a maximum of nine starts within the same hour is recommended by manufacturers and has been considered in the control. This feature is programmed in the HP type. Figure 6 summarized the control algorithm followed.

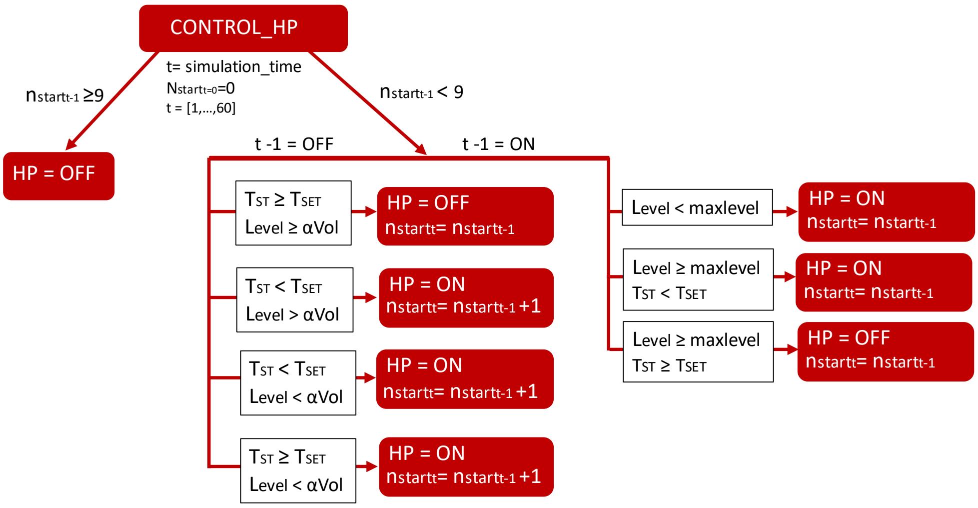

Figure 6 Control algorithm implemented for the system

The used comfort criteria are based in two conditions, satisfy the demand 99% of the time and do not allow more than one-minute shortage at the same hour daily. The second condition is added in order to consider the user satisfaction characteristics of this type of hot water demand.

The optimization of CO2 emissions has been performed for a system considering all these constrains.

3. PERFORMED STUDIES

The analysis of the variation in the CO_2 emissions derived from different sizing criteria of this heat recovery systems and to supply design guidelines about this kind of systems are the main outcome of this work. To this aim, the study considers four different points:

a) Optimization of the system for the nominal conditions.
b) Sensitivity analysis of the obtained solution with the external conditions

c) Analysis of the influence of each system component (SHP, Heat Exchanger and storage)
d) Comparison with other technologies

These points are described below.

(a) Optimization of the system for the nominal conditions.

The optimization of the HP size (*Scale*), the volume of the ST, and the control level (α) is carried out in order to minimize the CO_2 emissions for the reference conditions (see table 4) has been done. The calculatation of the CO_2 emissions has been made using the conversion coefficients from [30].

Table 4. Considered reference conditions.

DHW demand profile	20H/10H/30H
Tnet	10ºC
Tsource	20ºC
Tambient	20ºC

Constant values of the mains and district heating temperatures have been used as reference conditions.
The values chosen are selected from a conservative point of view, with 10 ºC as the net water temperature and 20 ºC as the heat source water temperature.
Due to the excessively time-consuming process resulting from long simulation periods and small-time steps, a one-year simulation timeframe with a time step of one minute was considered as a good compromise in terms of the accuracy/time ratio. As will be pointed out later on, larger time steps could lead to oversizing.
In order to find the optimum value of the differen parameters and obtain information about the influence of each one a set of parametric studies were performed. Figure 7 summarizes the simulations performed in the parametric studies done for each demand size: 10H, 20H, and 30H.

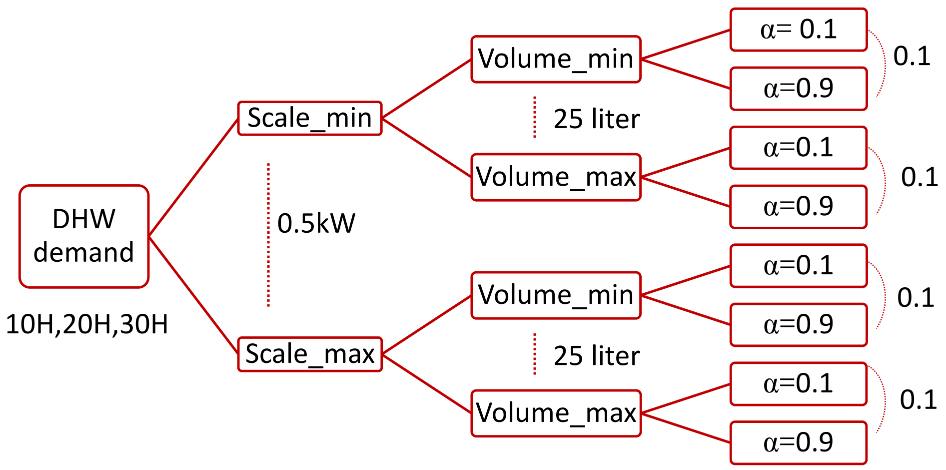

Figure 7. Parametric studies for each demand size. 0.5 kW scale increments, 25 litre ST size increments, and increments of 0.1 in α.

The different parametric study performed are:

- *Scale* variation range: it is expressed in terms of heating capacity (Qcond) and goes from the minimum size that meets the discomfort requirements to the size that satisfies the demand with an operating time of a minimum of 1.5 hours per day (in 0.5-kW steps).
- *Volume* variation range: for each heating capacity, the maximum and minimum ST sizes that meet discomfort levels are investigated (in 25-liter steps).
- *A* variation range: for each ST size, the variation of the control level in terms of the volume percentage, α, goes from 0.1 to 0.9 (in steps of 0.1).

The results presented in the next section have been obtained from more than 12000 simulations.

(b) Sensitivity analysis of the obtained solution with the external conditions

To verify the generality of the results obtained, a sensitivity analysis for one of the optimal solutions shown in the 20H case has been done. The sensitivity analysis has included:

- Sensitivity to the external conditions: the objective is to validate the solution with different net and wasted-heat water temperatures; hence the conclusions of the work developed in the previous analysis could be extended to different locations and conditions.
  a. Tnet variation: from 5 to 25 ºC
  b. Twasted/district variation: 10–35 ºC
- Sensitivity to other DHW profiles: The target of this analysis is to analyse the validity of the solution when different peaks take place for the same average consumption

(20H). This part of the study allows understanding the importance of the type of profile chosen.

c. For 20H draw-off and input characteristics of 10 consecutive years, (20HY).

(c) Analysis of the influence of each system component (SHP, Heat Exchanger and storage)

In order to analyse the impact on the energy consumption derived from the characteristics of the DHW production application and the components included in the system, the following simulations for the reference case (20H) has been performed:

- Reference HP case: calculation of the annual COP_{hp} and associated CO_2 emissions when the system is composed of the HP alone and the DHW load profile is constant in each time step.
- HP + HE: calculation of the annual COP_{sys} and associated CO_2 emissions when the system is composed of a HP with a HE keeping a constant demand profile. This case can be considered as the ideal case from the energy recovery point of view. (System with less losses).
- HP+HE+ST: calculation of the annual COP_{sys} and associated CO_2 emissions for the system with the HP, the HE and the This case can be considered as the real system as there is a variability in the demand in order to motivate the inclusion of a ST.

(d) Comparison with other technology

Finally, for the reference conditions at 20H solution, an annual comparison study with other technologies in terms of CO_2 emissions is included.

Four systems are considered:

- HP: Only the HP and the ST, with COP_{hp} = 5.74 (Table 6)
- HP + HE: the optimal system composed of HP+ST+HE
- NG Boiler: natural gas boiler with an efficiency of 0.92
- NGB + Solar: considers 50% of production from solar and 50% from a natural gas boiler with an efficiency of 0.92

CO_2 emissions associated with each type of source are chosen according to the Spanish conversion rates: 0.331 $kgCO_2/kWh$ for an electric source and 0.252 $kgCO_2/kWh$ for natural gas [30].

4. RESULTS

Although the analysis has been done based on a one-minute time step and yearly simulation time, due to the amount of data, the results are presented using the hourly time step. The temperature results are the integral of the minute-temperatures each hour, while the mass flow and energy hourly results are the sum of the respective variable for each minute within the hour.

Figure and Figure show an example of the outputs obtained in each simulation. In this example, a 20H demand, a HP heating capacity Qcond of 7.56 kW, a volume of 400 liters, and a control parameter α equal to 0.8 (control level 0.8*400 = 320 litres), have been used.

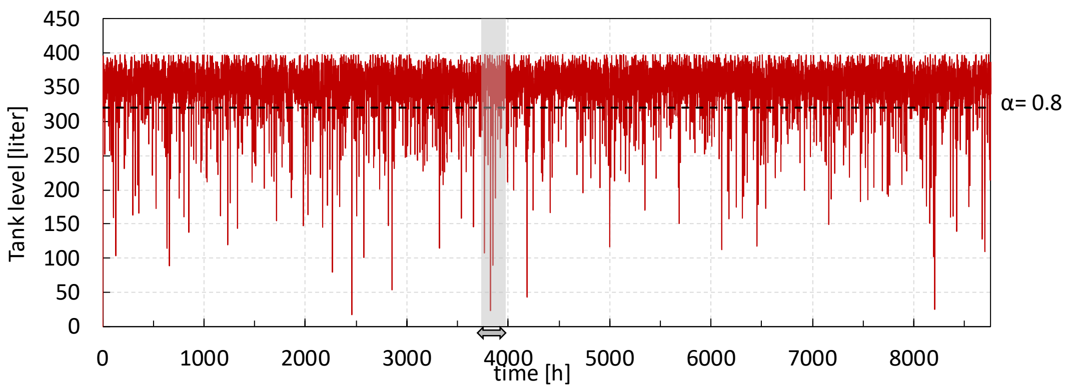

Figure 8. Hourly average tank level for 20H in a complete year, Qcond= 7.56 kW, α = 0.8, and volume = 225 litres

Figure gives a general view of the hourly results. However, the use of a yearly scale still makes the analysis difficult. Hence, a three-day period, represented in grey (from hour 3796–3868), is used for the next figures.

Figure 9 considers some of the most important outcomes of the model for a three-day period. Figure 9(a) represents the water mass flow rate in the tank (in red), where mw_cond is the water mass flow rate at the outlet of the condenser (inlet of the ST), blue is used for the water mass flow rate going out from the ST (mw_st), and dotted columns represent the water mass flow rate load required at the tank temperature (mw_hot). Figure (b) shows the hourly average level of the tank in red and the hourly average temperature in blue, while Figure 9(c) depicts the hourly energy supplied to the user (black dotted lines) and demanded by the user (red line) within the three days.

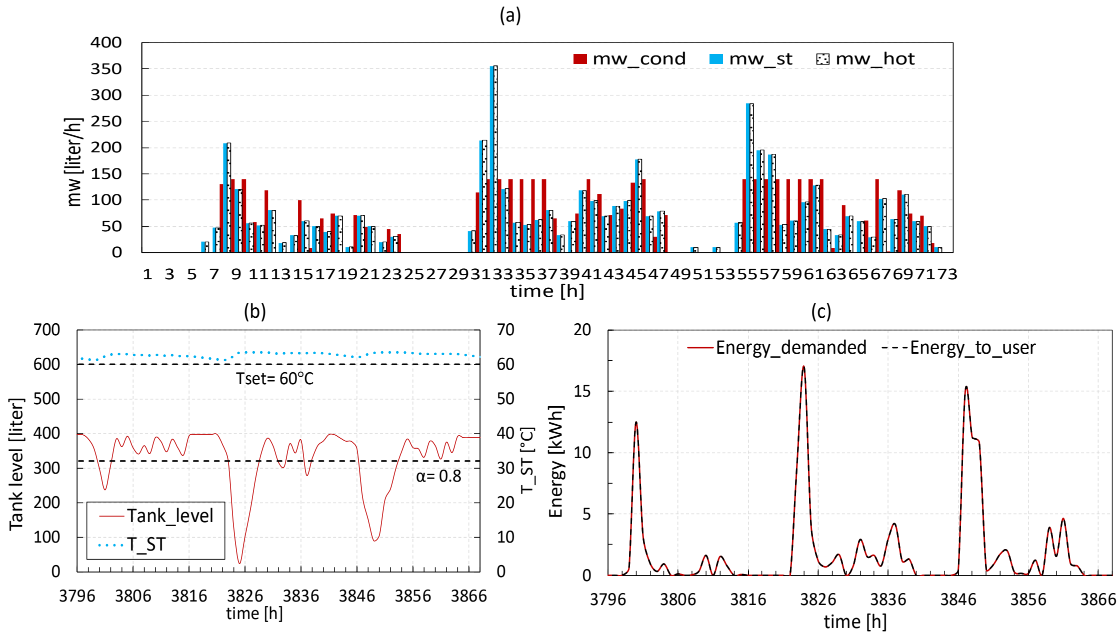

Figure 9. Hourly results for 20H during two days, Qcond= 7.56 kW, α = 0.8, and volume = 225 litres. (a) Water mass flow rate production (mw_cond), ST outlet water mass flow rate (mw_st), and water mass flow rate load required at the tank temperature (mw_hot); (b) average tank level and temperature; and (c) energy supply to the use and energy demanded by the user.

According to Figure 9 (b), the tank temperature never falls below 60 ºC and the ST is not oversized as in some periods it is full and in others it is almost empty while remaining capable of supplying the required energy in time thanks to the α and temperature controls that manage the HP production time. As can be seen in figure 9 (c), the energy supplied fulfils the requirements for the chosen sizing values and controls. The most convenient solution from the energy point of view is to produce as closely as possible to the demand, minimizing the time during which the water remains stored in the tank. Figure 9 also shows that only a few HP operating periods are required to maintain the level and temperature of the tank at the control values.

(a) Optimization of the system for the nominal conditions.

Figure 10 and Figure 11 give an example of the results obtained with the different parametric studies performed with each demand size in order to optimize the system parameters. In this case, a 20H load profile has been used. Figure 10 shows the annual CO_2 emissions associated with the system and Figure the respective HP operating hours for the possible solutions (combinations HP-ST that meet comfort standards). The parametric studies are based on different sizes in 0.5-kW steps, but Figure 10 and Figure 11 outline only the three most representative ones: (a) the minimum HP size, (b) the optimal HP size, and (c) the maximum HP size.

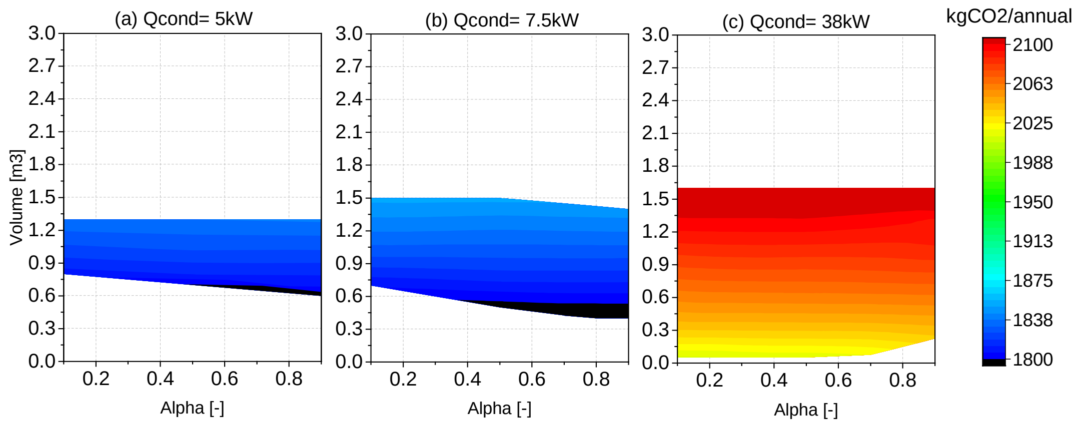

Figure 10. Annual CO_2 emissions associated with the system as a function of the ST size (volume) and control level (α) for (a) the minimum HP size, (b) the optimal HP size, and (c) the maximum HP size. Only the solutions that meet the discomfort standards are represented.

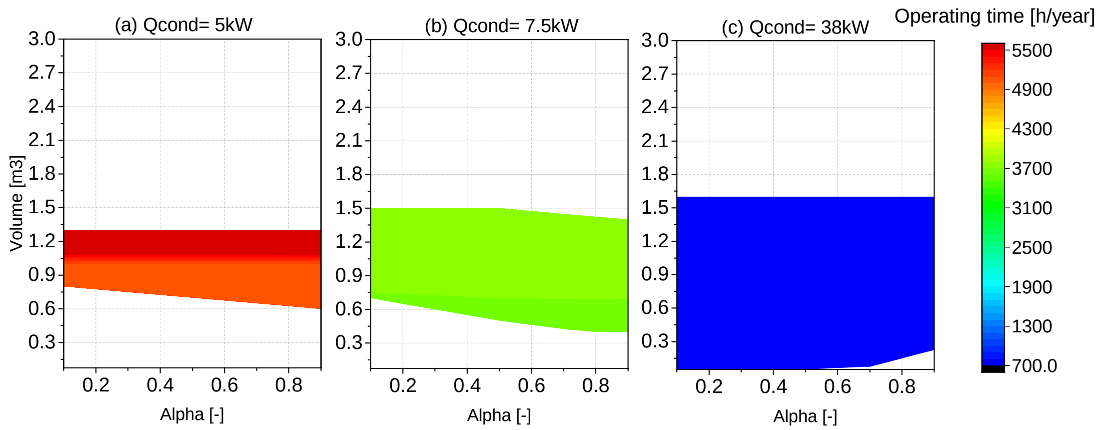

Figure 11. Annual HP operating hours function of the ST size (volume) and control level (α) for (a) the minimum HP size, (b) the optimal HP size, and (c) the maximum HP size. Only the solutions that meet the discomfort standards are represented.

From Figure 10 and Figure 11, several comments about the system behaviour can be done:

- Not all the combination sizes are possible solutions. In fact, there is a maximum ST size (depending on the HP size) capable of meeting the discomfort requirements. The region of possible solutions becomes larger as the HP size increases.

- The operating time depends mainly on the HP size. The ST size has a greater influence with small heating capacities. High tank volumes lead to higher operating times.
- Combinations of small HP-ST sizes lead to lower CO_2 emissions. Among the solutions, the best combinations from an energy point of view have the lowest ST size and high α but a longer operating production time. The reason for that is based in two points: a) the smaller the ST the smaller the losses associated with it,b) when the heat pump is too large it produces too much water that must be stored in the tank more time than the required one.
- For a given HP size, CO_2 emissions increase linearly with the ST size. An increase of around 5% from the minimum volume to the maximum volume takes place.
- Higher HP sizes lead to significantly higher CO_2 emissions (even though fewer operating hours are required). When the operating time increases from 1.5 h/day to 12 h/day, the CO2 emissions decreases by 15%.
- As a consequence of using a small time-scale and a variable volume ST, the ST sizes obtained are small. In fact, an ST of less than 400 litres for a 20H demand is obtained in optimal cases; this is smaller than the common ST used in DHW applications. For this type of systems where the peak demands could last only 5 minutes, a time steps in the simulations of one hour could change significantly the obtained solutions.
- With high ST volumes, the control of the ST level (α) become less important. In these cases, the HP operating control is driven mainly by the ST temperature.
- With small ST volumes, the dominant control is α and it widens the solution region compared to when no control level is used, especially with small HP sizes (optimum). For instance, in (b), the minimum volume without the control level is 700 litres, while an ST tank 57% smaller (400 litres) is possible when high water level controls are applied.

Similar results are obtained with other demand sizes and in all the parametric simulations.

Following the approach shown in Figure 10 and Figure 11 but taking into account only the optimal control (α value), the binomial analysis of the ST-HP size is done for 10H, 20H, and 30H load demands.

Figure 12, Figure 13, show the CO_2 emissions per house and the annual operating time, respectively, for the solution maps (in terms of HP-ST size) and the optimal α value for 10H, 20H, and 30H. To distinguish the influence of the demand size, the same scale has been used in each figure. Nevertheless, a magnified view of 10H has been considered in some cases.

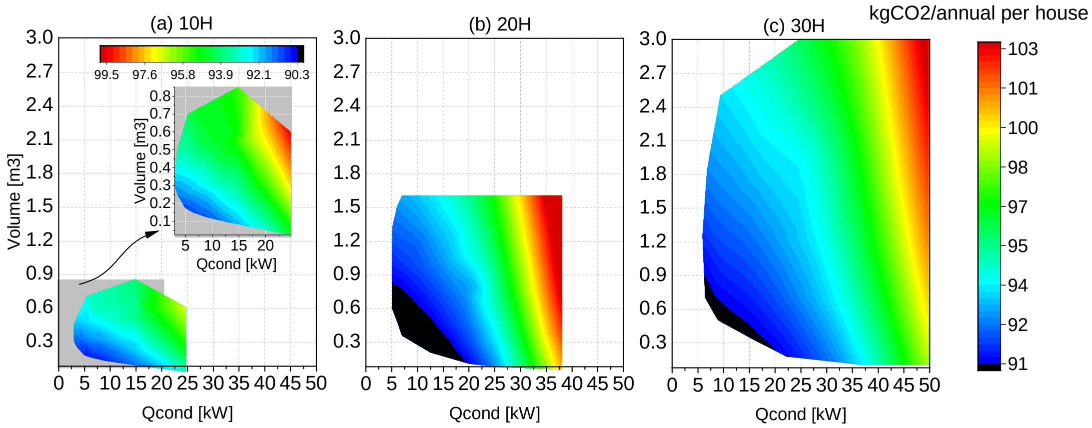

Figure 12. CO_2 annual emissions per house for each HP-ST size and (a) 10H demand, (b) 20H demand, and (c) 30H demand

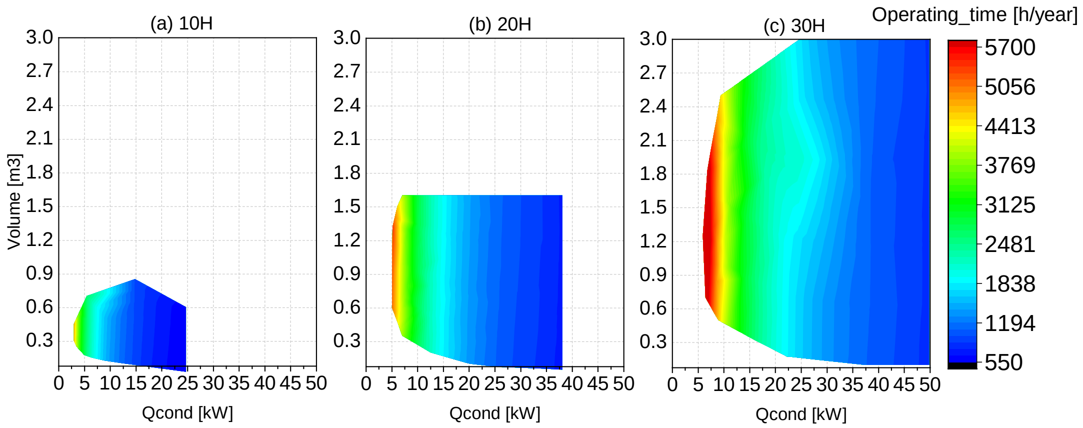

Figure 13. Annual operating time for each HP-ST size and (a) 10H demand, (b) 20H demand, and (c) 30H demand

The most remarkable conclusions from Figure 12 are:

- Higher aggregated demands (more houses served by the same system) lead to lower CO_2 emissions per house. This could be explained considering that for the same hot water demand, the demand profile requiring less energy to satisfy it is the constant demand profile, therefore more houses are served by the same system the profile demand will be more homogeneous and therefore the energy required per dwell will be smaller.

- There is a small optimal region of HP-ST size combinations that lead to similar annual emissions for each demand type. The minimum emissions correspond to a line of solutions with the smallest ST and HP sizes in each case.
- For a given demand, CO_2 emissions can vary up to 20% from the optimal solution to the worst case.

From Figure 13, the following conclusions can be drawn:

- A bigger solutions map leads to a wide range of operating times, from 1.5 h/day to almost 16 h/day in the 30H case, while with a smaller demand, where fewer combinations are possible, the range is from around 7 h/day to 2 h/day.
- Optimal solutions appear with high to medium operating time production. These solutions implies less heat losses in the storage tank.

The COPsys is in all the cases, is higher than 5.

Figure 14 collects the optimal HP-ST sizes for each demand size presented in figure 12, the curves presented in the figure represent the different combinations of HP-ST operating with a similar efficiency in order to satisfy a given DHW demand. In addition, the required annual operating hours are also represented as a function of the optimal binomial sizes. Green colour is used for 30H, blue for 20H, and orange for 10H. Dotted lines indicate the operating time and continuous lines represent ST sizes.

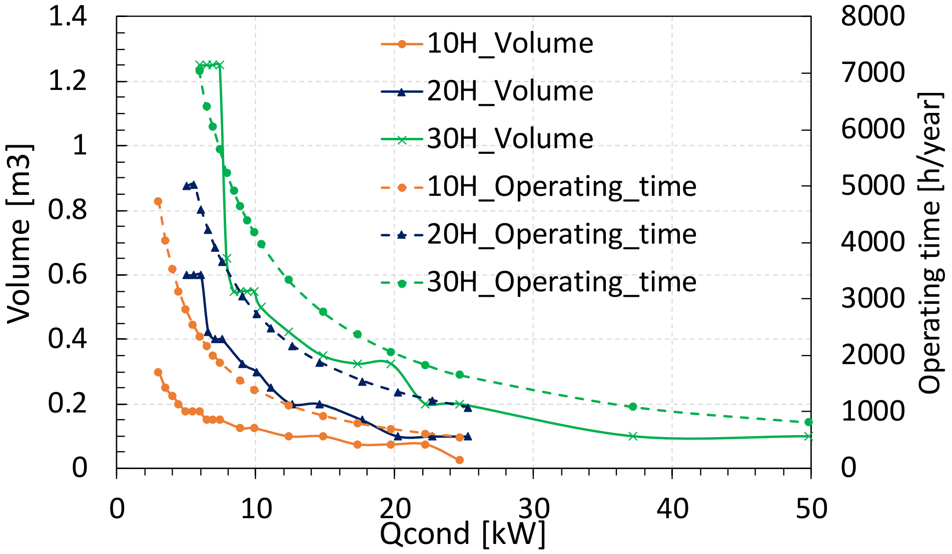

Figure 14. Optimal HP-ST size combination and its associated annual operating time for demands of 10H, 20H, and 30H.

Figure 14 is useful for sizing an HP system for DHW production with the characteristics described in this work. According to the figure, with small HP sizes, there is a minimum ST size required, while as the HP size increases, lower ST volumes are possible. In addition, increasing the heating capacity of the HP does not always lead to lower ST volumes (due to discomfort requirements).

(b) Sensitivity analysis of the obtained solution with the external conditions

A sensitivity analysis of the system considering different heat source temperatures (district heating temperatures) and water mains is included for the case represented in Figure and Figure . This is one of the best ST-HP size combinations in the 20H case (minimum HP-ST size among the optimal solutions).

Figure 15 represents the performance of the system for 20H, with the optimal α in each case, scale equal to 0.15, and volume equal to 400 litres. The green line indicates a net temperature of 15 ºC, the red line 10 ºC, and the black line 5 ºC.

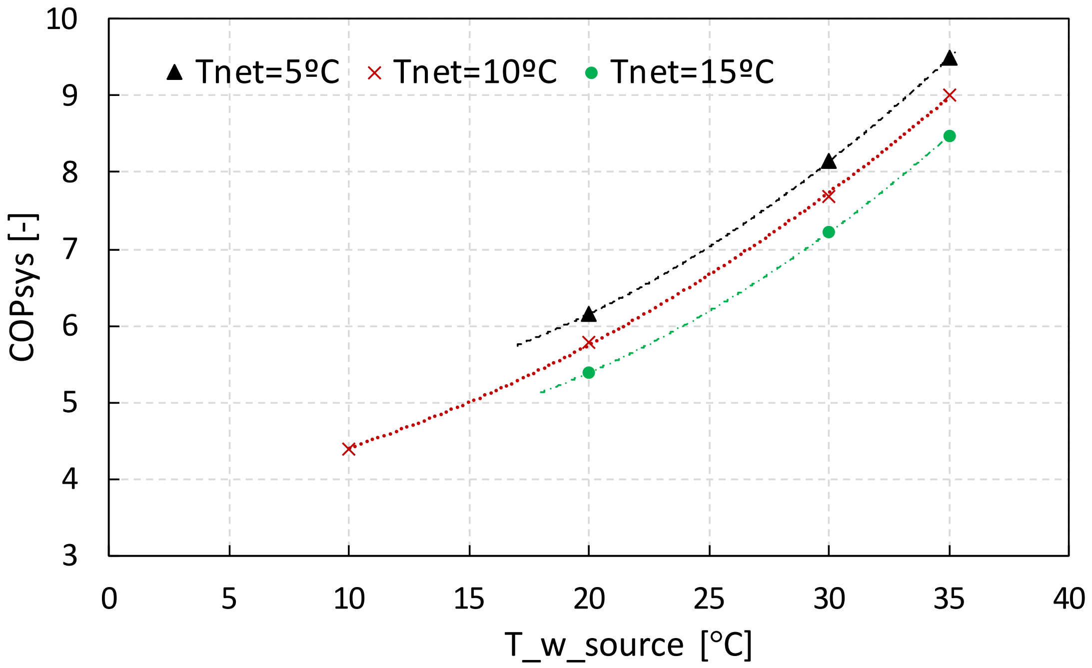

Figure 15. Performance of the system for 20H, with Qcond= 7.5 kW, α = 0.8, and volume = 225 litres for different water source and mains temperatures

The figure 15 shows that the COPsys increases as the temperature inlet temperature of the heat sink is reduced (maintaining the outlet temperature of the heat sink constant), this is because this procedure is equivalent to effectively reduce the mean temperature of the heat sink. The COPsys also increases and the temperature of the the heat source increases. The whole study presented in this work is based on one of the most critical conditions: a net water temperature of 10 ºC and water heat source temperature of 20 ºC. However, in most heat recovery applications, these conditions may underestimate the potentiality of a HP system.

From that way, if the heat source temperature changes to 35 ºC, which could be a representative temperature for low temperature district heating applications, COPsys can increase up to 9.

Table 5 collects the main results of the sensitivity study for the optimal solution obtained for demand of 20H working in different Tnet and Tsource conditions. Design conditions are highlighted in bold. The system capacity supply indicates the capacity of the system based on the user requirements, that is, the percentage of energy that the system is able to supply.

In all combinations, the systems can provide the required energy with an annual discomfort level below 0.6%. Thus, the proposed solution could be valid under a wide range of external conditions without compromising the level of satisfaction. The most influential variable is the temperature of the heat source.

Table 5 also shows that the adaptation of the HP-ST system to different external conditions can be done with tank control volume parameter α_opt. Favourable conditions lead to lower values of control levels and the opposite. Hence, if a system is designed following this type of approach, the optimal level of control would be adapted based on the external conditions.

Table 5. Main results of the system for 20H and different net and water source temperatures for scale = 0.15 and volume = 400 litres.

Tnet [°C]	Tsource [°C]	α_opt [–]	Qcond [kW]	Annual Elec. Consump. [kWh]	CO_2 emissions [$kgCO_2$]	Operating time [h]	COPhp [–]	System capacity supply [%]
5	20	0.8	7.54	5079.39	1681.27	3448.15	5.15	99.95%
5	30	0.4	8.92	3838.36	1270.49	2457.41	5.76	99.94%
5	35	0.3	9.7	3291.14	1089.36	2047.18	6.1	99.98%
10	10	0.8	6.23	7085.68	2345.36	5137	4.54	99.66%
10	**20**	**0.8**	**7.56**	**5408.68**	**1790.27**	**3659.03**	**5.14**	**99.95%**
10	30	0.5	8.95	4063.54	1345.03	2592.3	5.76	99.98%
10	35	0.4	9.74	3473.65	1149.78	2153.45	6.1	99.97%
15	20	0.9	7.58	5802.68	1920.69	3911.73	5.14	99.92 %
15	30	0.5	8.99	4329.09	1432.93	2751.5	5.76	99.93%
15	35	0.4	9.77	3687.13	1220.44	2277.73	6.09	99.95%

Finally, in order to check the generality of the obtained solution, the same system solution has been used in order to satisfy an input load profile generated with DHWcalc using the same conditions but a simulation period of 10 consecutive years, with the same average consumption. In all cases, the system was able to serve the energy demand with the required satisfaction and similar CO_2 emissions associated with the production. Thus, it can be stablished that the obtained solution is independent of the particular random profile generated.

(c) Analysis of the influence of each system component (SHP, Heat Exchanger and storage)

In this section, an analysis of the influence of the different components in order to satisfy the required total energy demand is performed. Three different systems able to satisfy a 20H demand load profile of DHW have been analysed. These systems are: a standalone HP, a HP+HE and HP+HE+ST. The DHW profile is considered as constant except for the case including the ST, where the 20H profile generated with DHWcalc has been used. One should notice that the heating capacity includes the heat from the HP and the pre-heater HE when it is present.

Table 6. System analysis. Results for HP, HP+HE, HP+HE (30 °C) and HP+HE+ST

System	HP	HP + HE	HP+HE+ST
Tsource [ºC]	20	20	20
Annual energy demand [kWh]	31283.2	31283.2	31283.2
Average required heating power [kW]	3.57	3.57	3.57
Tco [ºC]	60	60	64
HE capacity [kW]	0	0.357	0.357
HP heating capacity, Qcond [kW]	3.57	3.21	3.5
COPhp	5.74	5.39	5.14
COPsys	5.71	5.95	5.2

Table 6 presents the results of the analysed cases, according to it, the addition of a pre-heating recovery HE improves the COPsys by almost 5% even though the COP of the HP decreases by around 6%. In fact, 10% of the heating energy required comes from the HE, this value that could be higher (for higher heat source temperatures) shows the relevance of this component. This is a consequence of the second law of thermodynamics, when the temperature allows the heat exchange, always is better to recover heat directly than to use a heat pump for that.Therefore as a rule of thum, in this type of systems, first recover energy with a HE and then pump the rest of the energy.

When a DHW non-uniform profile is used, an ST tank is required. This component increases the irreversibility significantly. Higher condenser outlet water temperatures (64 ºC) are required and heat losses take place in the ST. In fact, the COPsys of the HP+HE+ST system is 12% lower than that of the HP+HE system for the same external temperatures in the analysed cases.

In order to understand the cases analysed in Table 6, Figure shows a more detailed comparison of the results of the standalone HP system and the HP+HE+ST system. Since a one-year simulation with a one-minute step scale does not allow to visualize the results graphically, only the results for one day are shown. Figure (a) represents the production and hot water requirements for the reference case, HP. Figure 16 (b) shows the hot water production in red, the consumption as dotted columns, and the ST outlet water mass flow in grey. Figure 16(c) represents the water temperature at the inlet and inside the ST in the case of Figure 16(b), and Figure (d) represents the water level of the tank during the day in the case of Figure (b).

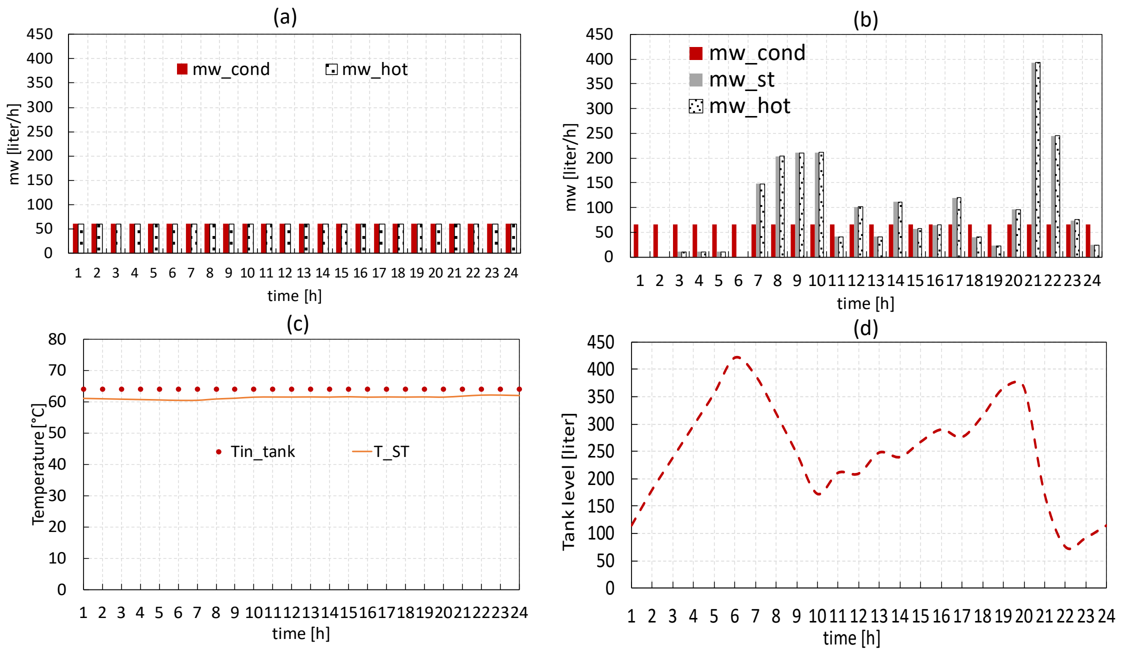

Figure 16. Results on an hourly base for one day of the year: (a) water mass flow rates for the HP case, (b) water mass flow rates for the HP+HE+ST case, (c) ST temperatures for the HP+HE+ST case, (d) ST level for the HP+HE+ST case

From the obtained results, it is worth pointing out that the variability of the DHW load profile results in a significant COP loss of the system; that is, the characteristics of the profile condition are the most influential parameter in the efficiency of the system. This result is mentioned in [31], where the authors analysed the efficiency of a system based on different profile shapes, numbers of peaks, and distances between peaks in addition to operating schedules. From that study, they concluded that the imposition of, for instance, “night” production leads to a 20% loss of efficiency and the position and the number of peaks have significant impacts on the final solution from the sizing of the system and form the efficiency point of view.

Therefore, the optimization of these systems could be addressed from a different perspective: instead of optimizing systems for a determined type of user, the system and the user habits could be optimized for the maximum energy efficiency.

(d) **Comparison with other technology**

Table 7 contains the electric consumption and the annual CO_2 emissions associated with four different systems taking into account the temperatures of the design conditions (Tnet = 10 ºC and Twsource = 20 ºC and an annual energy demand of 31283.2 kWh).

Table 7: Annual CO_2 emissions associated with different DHW production systems for 20H with Tnet = 10 °C and Twsource = 20 °C

Annual electricity consumption [kWh]	HP *	5450
	HP + HE **	5341.5
	Gas boiler	34003.5
	Gas boiler + 50% Solar	17001.7
Associated CO_2 emissions [$kgCO_2$]	HP	2014.5
	HP + HE	1768.0
	Gas boiler	8568.9
	Gas boiler + 50% Solar	4284.4

* COP_{hp} = 5.74

** COP_{hp} = 5.14, with 87.76% of the energy supplied by the HP and 12.24% by the HE

Figure 17 shows the annual CO_2 emissions of the considered systems and 20 multifamily houses and highlights the potentiality of HP systems for DHW production. CO_2 emissions could be reduced by a factor of up to 4.5 through the substitution of gas boilers by HP systems.

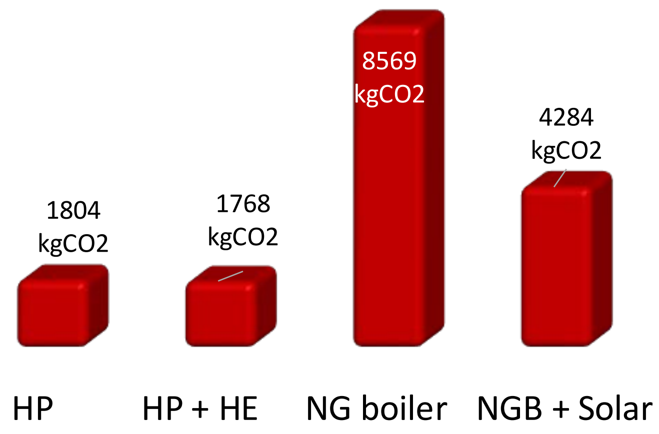

Figure 17. Annual CO_2 emissions associated with HP, HP+HE, NG boiler, and NGB + solar systems for 20H DHW production.

5. CONCLUSIONS

This work analyses a system to produce DHW from the recovery of low-grade temperature sources using HPs. The study analyses the influence of the proper sizing and the operation strategy of an HP-ST system in order to satisfy DHW production within the residential sector based on the minimization of the associated CO_2 emissions for a given demand profile. In addition, it could supply an indicative value about the minimum energy required to satisfy the DHW demand with current technologies.

In order to maxime the efficiency of the system, it is composed of a pre-recuperator heat exchanger installed before the heat pump, an innovative heat pump with a special dependence of the COP as a function of the condenser water inlet temperature (the SHP) and a storage tank of variable volume in which the temperature is almost constant along all the tank and no stratification is produced.

The main conclusions obtained from this work have been:

- A set of HP-ST sizes combinations with similar associated CO_2 emissions exists nevertheless a significant penalty in the energy consumption, up to 20%, can be obtained from going outside this set of solutions.

- Small sizes of both components (HP+ST) are preferred, allowing to minimize the time that the water is stored in the tank.
- The net water temperature and the temperature of the water heat source do not affect critically to the design of the components.
- The water pre-heating heat exchanger significantly enhances the performance of the system, lowering annual CO_2 emissions by around 15% on average.
- The need for the ST is justified in order to satisfy the variable demand curve of this type of application. It introduces a significant reduction in the annual system performance. In order to minimize that source of losses a variable volume tank with no stratification has been used in all the study. Using that type of tank, 12% of system efficiency reduction compared with no tank use (constant demand profile) has been obtained.
- More centralized DHW production system (more houses connected to the same system) leads to a flatter hot water profile demands and lower annual emissions per house.
- Derived from the previous point, it should be pointed out that for the same DHW demand, the energy consumption could change up to 12% depending on the used profile. Therefore, solutions to minimize the environmental impact associated with DHW production should take that point into account and should also imply the need for education and adoption of some habits in DHW use by the user.

Finally, it should be commented that the obtained results are independent of other factors like energy policies or prices that could be added in a second level analysis. This point gives generality to this study.

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