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A propane water-to-water heat pump booster for sanitary hot water production: seasonal performance analysis of a new solution optimizing COP

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ABSTRACT

Electrical heat pumps for sanitary hot water production achieve a high performance with a good matching of water and refrigerant temperature profiles during the heat rejection stage, as it happens in CO_2 systems. This work considers the thermodynamic possibility to adapt the condenser pressure of a propane heat pump to maximize the COP, while producing sanitary hot water up to 60 ºC from a heat sink equal to 15 or 25 ºC. The performance of the heat pump is calculated through specific models which, in combination with a TRNSYS model of the whole system, allowed to assess its seasonal performance for a hotel in Strasbourg, also varying the control logic and the size of the storage tank. Results obtained led to the conclusion that, for achieving a high seasonal performance, the control logic of the tank has the largest influence.

Keywords: Propane, heat pump, waste heat recovery, energy optimization

Nomenclature

COP	Coefficient Of Performance [-]
DHW	Sanitary Hot Water [-]
EHP	Electric Heat Pump [-]

The original paper was presented on the 11th IIR Gustav Lorentzen Conference on Natural Refrigerants (GL2014), Aug.31 to Sept.2, 2014, Hangzhou, China".

$\dot{m}$ mass flow rate [$kg \cdot s^{-1}$]

$\dot{L}$ power consumption [W]

p pressure [Pa]

$\dot{Q}$ heat pump capacity [W]

SC SubCooling [K]

T temperature [ºC]

t time [h]

V volume [m^3]

YLPF1 yearly performance factor considering the heat pump consumption [-]

YLPF2 yearly performance factor considering the heat pump consumption, and the consumption of the circulation pump that sends water to the evaporator and condenser loops [-]

YLPF3 yearly performance factor considering the heat pump consumption, the consumption of the circulation pump that sends water to the evaporator and condenser loop, and the consumption of the circulation pump that sends water to the user [-]

Subscripts

comp compressor

cond condenser

evap evaporator

out outlet

overall refers to the overall efficiency

pump refers to the water circulation pumps

suppnet refers to the water supply from the net

tank refers to the storage tank

top refers to the top of the storage tank

user refers to the user of the sanitary hot water

Greek symbols

Δ difference

ρ density [$kg \cdot m^{-3}$]

η efficiency [%]

1. INTRODUCTION

Electrical heat pumps (EHP) technology has been proved to be a high efficient solution in the energy conversion for the production of heat. This is recognized in some countries, such as Italy (see ItalianLaw (2011)), where a portion of the energy used by a heat pump having a seasonal COP higher than a reference value is considered as if it were obtained from renewable energy sources. The use of electrical heat pumps for sanitary hot water production is spread in developed countries, and the concerns related to the energy savings are of primary importance. The recent 812/2013 regulation from the European Union (EU 812/2013) provides guidelines for the energy labeling of systems for sanitary hot water (DHW) production, acknowledging that DHW usage is a "significant share of the total energy demand in the Union" and that the "scope for reducing the energy consumption is significant".

The selection of working fluids and the design of EHP systems are of primary importance to achieve energy savings, respecting the environment. As stated in Sarbu (2014), nowadays a new concept in the implementation of refrigeration systems is imposed, requiring tightly constructed configurations that work with refrigerants having a low GWP (Global Warming Potential), but keeping the performance as energetically efficient as possible. In this direction, the use of natural fluids could be an interesting option and the impact of their adoption on the energy performance of EHPs is the objective of an ongoing FP7 European Project, Next Generation of Heat Pumps working with Natural fluids, NxtHPG, (Corberan and Montagud (2014)).

Concerning the use of natural fluids in heat pumps applications for sanitary hot water production, the adoption of carbon dioxide (CO_2) has been proposed in some seminal works by Rieberer et al. (1997) and by Neksa et al. (1998); also the advantage of its use has been proved, with respect to standard solutions, by Cecchinato et al (2005).

These works showed that systems working with CO_2 trans-critical cycles can achieve a good performance, once high temperature water production with large temperature lift is required. In those cases, the thermodynamic behaviour of the CO_2 trans-critical cycles related to the heat rejection process at variable temperatures is profitably used to achieve a good matching among the fluids, as showed in Fig. 1.

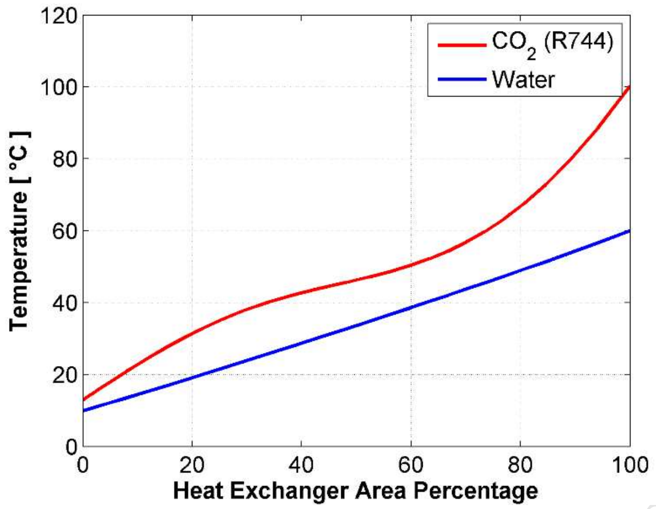


Figure 1. Example of CO_2 and water temperature profiles in a gas cooler of a CO_2 heat pump for the production of sanitary hot water at 60 °C, working under optimized conditions.

In addition to CO_2, also another natural fluid, propane, can be used for sanitary hot water production. A report by IEA (IEA, Annex 32) shows the seasonal performance of a propane heat pump for the combined space heating and sanitary hot water production in a Norwegian passive house. The same system was compared to different solutions in another report by Justo Alonso and Stene (2010). For the operating conditions considered, the authors concluded that COP is 20% higher when CO_2 is used.

In the present work, a propane system able to adapt its condensing pressure to optimize the COP as a function of the boundary conditions, is considered. In particular, for a fixed outlet temperature of 60ºC for the sanitary hot water, also different condensing pressures can be fixed (with a variable sub-cooling allowed by the system configuration) for a given boundary condition at the evaporator and condenser inlet.

For instance, Fig. 2 shows the comparison between a reference operating condition (dashed lines) without sub-cooling at the exit of the condenser and the one that allows to maximize the COP, keeping constant the remaining parameters (continuous lines): the latter situation allows a better matching between the temperature profiles at the condenser, increasing the COP.

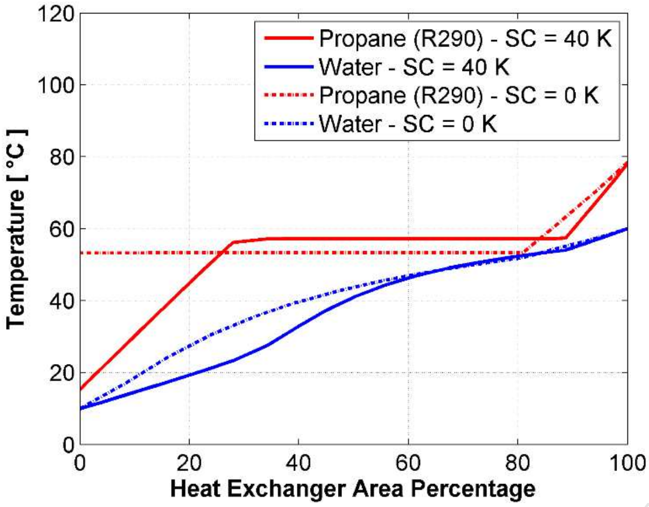


Figure 2. Examples of propane and water temperature profiles in a condenser of a propane heat pump for the production of sanitary hot water at 60 °C, working under different pressures at the condenser.

Fig. 3a reports examples of a parametric analysis carried out in the vapor compression software package IMST-ART (Corberan and Gonzalvez-Macia (2009)) for the heat pump considered in the present work. It shows the evolution of the COP at varying the condensing pressure for different temperatures of the water at the condenser inlet (keeping constant the remaining parameters, for the source temperature equal to 20 ºC). Considering the sole effect of the variable pressure at the condenser, there is a maximum, with an increase of the COP up to 4%, for the operating conditions considered here.

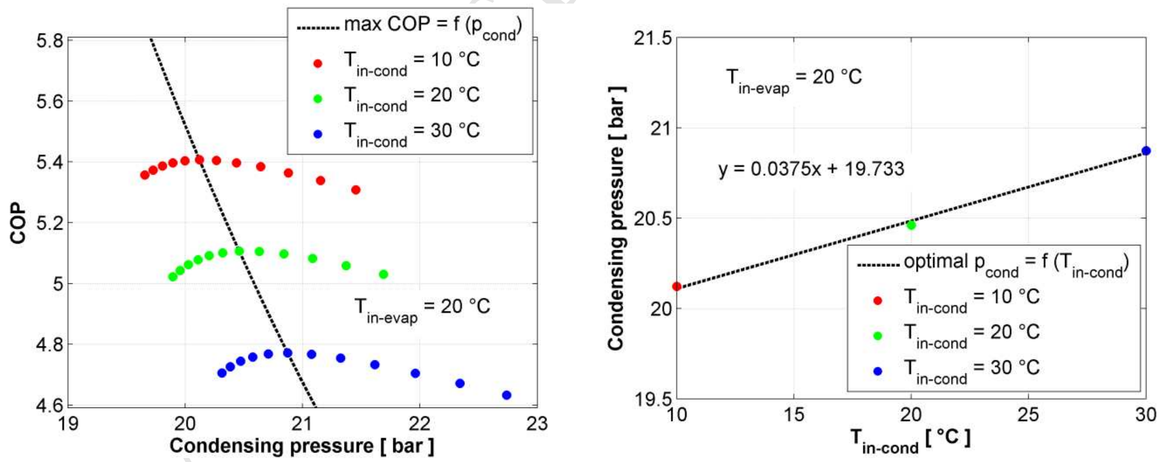


Figure 3. a) COP evolution as a function of the condensing pressure at different water inlet temperatures at the condenser. b) Optimal condensing pressure as a function of the water inlet temperature in the condenser (in all cases, water outlet temperature at the condenser is of 60 °C and source temperature 20 ºC).

It is possible to relate the optimal condensing pressure to the water temperatures at the condenser inlet, as reported in Fig. 3b for a fixed temperature at the evaporator inlet, and to imagine that the system is able to control the pressure to the optimal value, also while the boundary conditions vary.

As it is known, the seasonal performance of a heat pump not only depends on the heat pump design itself but also on the real operating conditions over the year, that are definitely affected by the load profile of the end-user and the temperatures of the secondary fluid at the inlet of both the evaporator and the condenser. In Fernandez et al. (2010), a performance evaluation of air-to-water heat pump working with CO2 for sanitary hot water production was conducted to investigate the effect of ambient temperature and the water temperature entering the gas cooler on the overall coefficient of performance (COP) during full tank heating tests in three scenarios typical of residential water heating: initial tank heating without demand, tank reheating after water usage, and tank reheating after standby losses. Results led to the conclusion that the overall COP was maximized at higher ambient temperatures and at lower hot water temperatures entering the gas cooler, being the COP of the reheating process reduced in about 30-40% than that corresponding to initial tank heating.

The scope of the present paper is to evaluate the performance of a propane water-to-water heat pump system producing sanitary hot water for a specific end-user type. The end-user considered here is a hotel in an average climate (Strasbourg, according to EU standard reference climates), whose load profile has been reported all over the year, being the knowledge of the real sanitary hot water demand the most important point for a correct evaluation of the performance of such a system, as shown in Agudelo-Vera et al. (2013).

As the hotel is part of a commercial center, the system considered is a water-to-water heat pump booster from an intermediate temperature level with respect to the ambient, in order to increase the system performance with a benefit from low grade waste heat recovery as reported by Cipolla and Maglionico (2014) for sewage water, or by Aynur (2010) for neutral loops.

With respect to the condenser side, the presence of a tank in the system is considered, which would influence the inlet temperature, allowing to store heat. The effect of the tank and of the control of the system are studied parametrically, to assess their influence on the system seasonal performance.

The present study was carried-out through simulations of the whole system in TRNSYS. The EHP, heat storage and hydraulics were modeled using experimental data for a daily load profile on the demand side; and the performance of the EHP adopting the new concept of optimized pressure at the condenser has been correlated to significant parameters, as in Corberan et al. (2011a), based on the vapor compression software package IMST-ART.

2. MODEL OF THE SYSTEM

The global model of the system has been developed in TRNSYS. It is composed by a basic heat pump cycle, the neutral loop, the thermal storage, the user and the hydraulics. In the following, a detailed description of the model for the heat pump and the rest of the system components is provided.

2.1. HEAT PUMP MODELLING: PERFORMANCE CORRELATIONS

The prototype EHP is a water-to-water model working with propane (R290). Its nominal heating capacity is of 49 kW when heating water from 10 to 60°C at the condenser and working with water entering the evaporator at 20°C. Table 1 reports its main characteristics.

Refrigerant	**Secondary fluids**	**Condenser** (Type and UA)	**Evaporator** (Type and UA)	**Scroll Compressor** (Displacement and rpm)
Propane (R290)	Water	Plates 1400 $W\cdot K^{-1}$	Plates 9700$W\cdot K^{-1}$	170 cm^3 2900 rpm

Table 1.Heat pump main characteristics.

The heat pump is modelled in the vapor compression software package IMST-ART as a standalone system. The IMST-ART heat pump model incorporates the key elements of the vapor compression circuit: evaporator, condenser, compressor, expansion valve and connecting pipe work. Using IMST-ART, the heat pump model was constructed on a component by component basis, considering the information of the catalogue data provided by the component manufacturers. IMST-ART software has been experimentally validated in several studies like the ones presented in Corberan et al. (2008a and 2011b) and Gonzalvez-Macia et al. (2007). Results obtained demonstrated maximum error bands of less than ±4% for a wider range of operating conditions where other water-to-water heat pump systems were modelled.

The EHP has the objective of heating water up to 60°C ($T_{out-cond}$ = 60°C) and can work with variable water temperatures at the inlet of the condenser and evaporator. In a physical system, in fact, the water inlet temperature at the condenser would be the temperature of the water coming from the bottom of a storage tank (or from the city supply network) and the one at the evaporator would depend on the source of energy being used (sewage water, neutral loop etc.). Since inlet and outlet water temperatures at the condenser are given, the matching of the water temperature profile with the propane temperature profile will depend on the water mass flow rate.

Looking at Fig.3, it can be concluded that an optimal working condition can be determined, in terms of condensing pressure that yields maximum Coefficient Of Performance (COP) during operation, this being defined as in Eq. (1):

$$COP = \frac{\dot{Q}_{cond}}{\dot{L}_{comp}} \tag{1}$$

where $\dot{Q}_{cond}$ is the heating capacity in kW and $\dot{L}_{comp}$ is the electrical power of the compressor, in kW.

On the other hand, and in order to consider different source temperatures at the evaporator, each optimal condensing pressure value is determined (in steady state conditions) at 4 different water inlet temperatures in the evaporator (10, 15, 20, 25 °C) coupled to 5 different water inlet temperatures at the condenser (10, 20, 30, 40, 50 °C) for a total of 20 working conditions.

In the hypothesis of the EHP always working at the optimal condensing pressure value, a map of performance is then obtained as a function of the inlet temperatures of the secondary fluid at the evaporator and at the condenser, so that the cooling capacity $\dot{Q}_{evap}$ in kW, other performance indicators and water mass flow rates at the evaporator and at the condenser, are correlated as shown in Eq. (2):

$$\dot{Q}_{evap} = p_{00} + p_{10}x + p_{01}y + p_{20}x^2 + p_{11}xy + p_{02}y^2 + p_{30}x^3 + p_{21}x^2y + p_{12}xy^2 + p_{03}y^3 \tag{2}$$

where x is the water inlet temperature in the evaporator expressed in °C (from now on $T_{in-evap}$) and y is the water inlet temperature at the condenser expressed in °C (from now on $T_{in-cond}$). These polynomial expressions were programmed in TRNSYS creating a new TRNSYS type for the heat pump. This new type receives as inputs the working conditions (temperature and flow rate) of the secondary fluids entering the condenser and the evaporator, and provides as outputs the performance parameters of the heat pump. Table 2 reports the values of the coefficients p_{mn} obtained for the main performance indicators, whereas Fig.4 shows the performance maps obtained by fitting data using Matlab (Matlab release R2012a).

	p_{00}	p_{10}	p_{01}	p_{20} $(\cdot 10^3)$	p_{11} $(\cdot 10^3)$	p_{02} $(\cdot 10^3)$	p_{30} $(\cdot 10^3)$	p_{21} $(\cdot 10^5)$	p_{12} $(\cdot 10^5)$	p_{03} $(\cdot 10^5)$
$\dot{Q}_{cond}$	28.31	0.923	0.003203	10.18	-5.192	-4.274	0.07276	2.702	-4.875	4.375
$\dot{Q}_{evap}$	21.01	0.8745	-0.03173	7.95	-4.379	-3.624	0.0861	0.8559	-4.4782	3.529
$\dot{L}_{comp}$	7.296	0.04863	0.03493	2.217	-0.8029	-0.654	-0.01292	1.831	-0.09822	0.8507

Table 2. Coefficients p_{mn} correlating the main performance indicators.

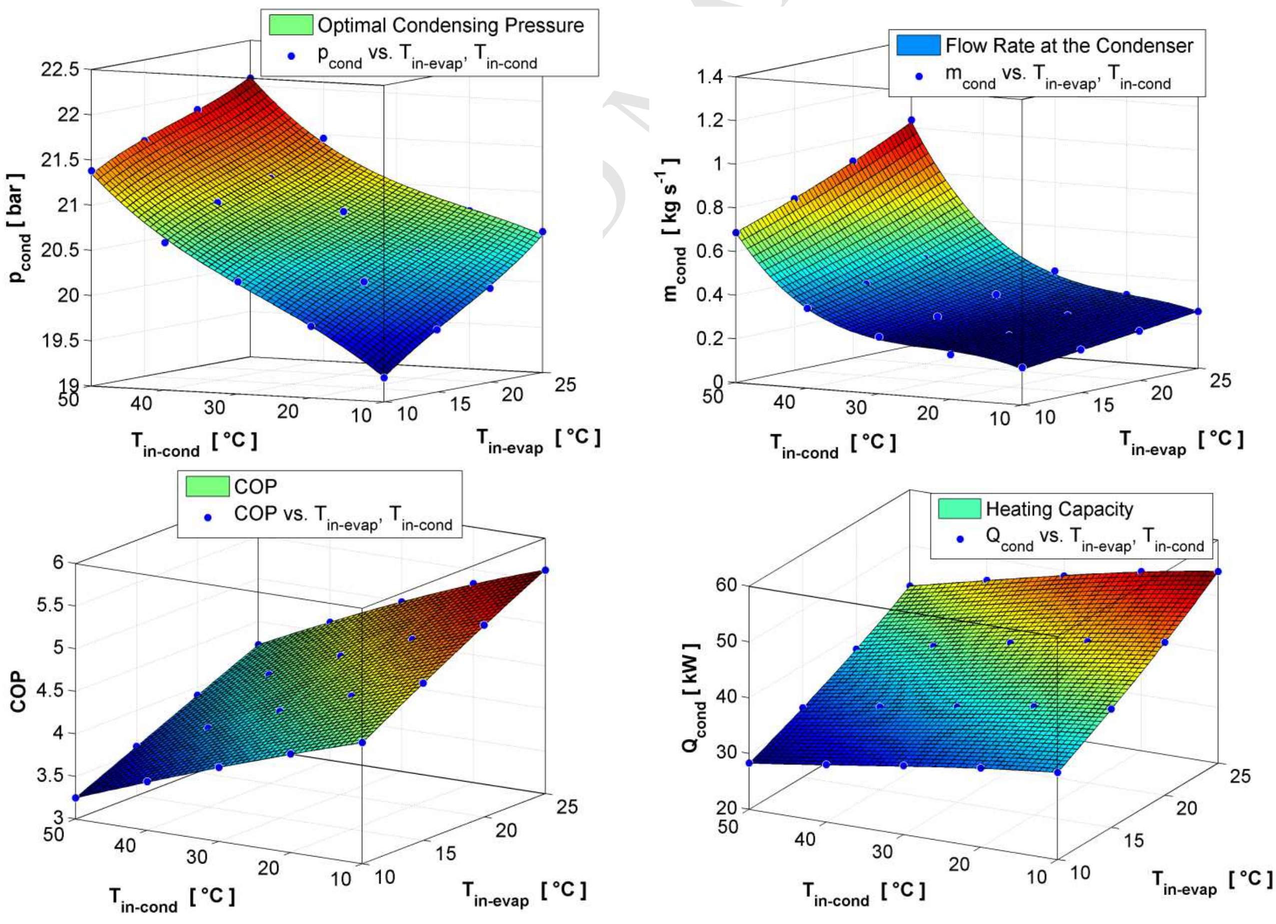


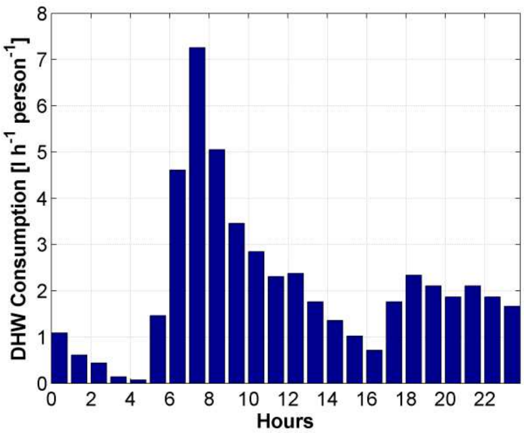
Figure 4. Performance maps of the heat pump in terms of (clockwise from top-left): a) optimal condensing pressure, b) mass flow rate of water at the condenser, c) COP and d) heating capacity, as a function of the inlet temperatures of water at the evaporator and the condenser.

2.2. END-USER DESCRIPTION

To assess the performance of the heat pump, the case study of a hotel inside a commercial centre is considered. A temperature of 15 ºC was considered as the source temperature, which would correspond to the sewage water of the hotel. The daily load profile of DHW consumption, per person, is reported in Fig. 5, taken from Ashrae Handbook (2012). The location of the end-user is Strasbourg, France, which is one of the reference for average climates in Europe according to the European Regulation EU-812/2013 (the other two being Athens, Greece, representing warmer regions, and Helsinki, Finland, representing colder regions).

It is worth noting that: according to Energy Saving Trust (2008), within the UK, climate differences within regions and seasonal variability have a negligible effect on the consumption of DHW when compared to that of occupancy; according to Blokker et al. (2011), in a hotel the dominant variable determining the DHW consumption is the number of rooms.

During the year, the occupancy has been estimated to vary between 80% and 120% of the base level, 100%, assumed to be of 250 people. To simulate this variation, the level of occupancy is tied linearly to the monthly average ambient temperature of Strasbourg in such a way that, on the coldest month of the year, the occupancy is maximum; and on the hottest month, the number of people is minimum as shown in Fig. 5b. Climate data are taken from the Meteonorm (Meteonorm: Global Climatological Database and Software) database, used in TRNSYS 16 (Klein et al. (2006)).



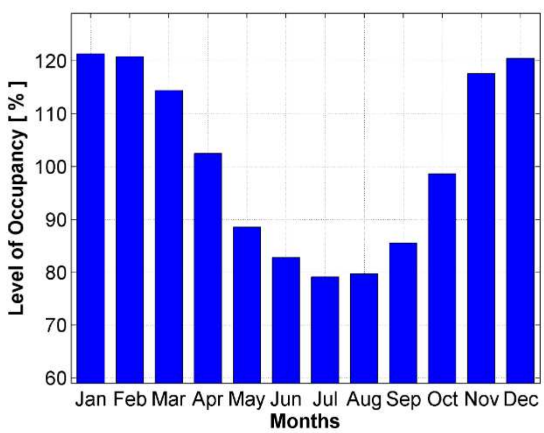


Figure 5.a) Daily load profile of sanitary hot water (DHW) consumption, in litres per person per hour. b) Level of occupancy in the hotel during the year.

Table 3 summarizes some data about the load profile used.

	Daily average hourly consumption	Peak hourly consumption (7to8)	8to20h average hourly consumption	8to20h consumption percentage
Base level occupancy (250 people)	523 l·h^{-1}	1814 l·h^{-1}	565 l·h^{-1}	53.92% of total

Table 3.End-user DHW consumption main data.

2.3. SYSTEM COMPONENTS MODELLING

A sketch of the system model as used in TRNSYS for quasi-steady state simulations is shown in Fig. 6.

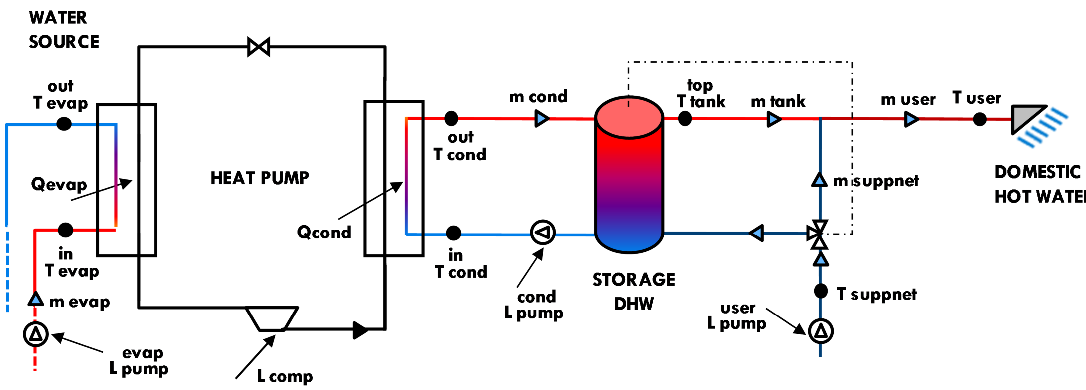


Figure 6. System model layout.

Although DHW is heated up to 60°C by the EHP for legionnaires' disease concerns, the tempering valve (Type 11b in the stock TRNSYS 16 library) is meant to deliver water at 50°C to the user (T_{user}= 50°C) by mixing hot water coming from the top of the tank, $\dot{m}_{tank}$at its temperature $T_{top-tank}$with an appropriate amount of reintegration water $\dot{m}_{suppnet}$taken from the supply network ($T_{suppnet}$=15°C throughout the year), determined as shown in Eq. (3).

$$\dot{m}_{suppnet} = \frac{(\dot{m}_{user}T_{user} - \dot{m}_{tank}T_{top-tank})}{T_{suppnet}} \tag{3}$$

The vertical, cylindrical tank (Type 60c in the stock TRNSYS 16 library) is insulated (heat transfer coefficient to the environment U_{loss} = 0.8 W·m^{-2}·K^{-1}) and kept in a closed environment, that remains at T_{env}= 20°C throughout the year. It has a height-over-diameter ratio equal to 4 to help stratification (Castell et.al (2010)). The tank volume was calculated using Eq. (4):

$$V(m^3) = \sum_{t=0}^{t=24h} DHW(t) \tag{4}$$

where $DHW(t)$ is the hourly sanitary hot water demand. Considering the daily demand profile and the base level of occupancy (250 people), the volume of the tank results in 12.5 m^3 approximately. However, in an effort to keep the bottom colder, the volume of the tank considered in the simulations is a 20% greater than that of the base level daily consumption, finally resulting in 15 m^3. The tank has 2 inlets and 2 outlets as shown in Fig.7. The inlet from the heat pump condenser is on top, so as the outlet towards the user. The inlet from the supply network is located at the bottom, so as the outlet going to the condenser. The tank has a total height of 6.73 m, which for simulation purposes in TRNSYS, is equivalent to the situation of having 3 tanks of 2.243m of height each connected in series, as the difference existing in the mixing effect of buoyancy inside the volume of each of the three tanks connected in series, compared to that of a bigger single tank can be considered as a secondary effect that results negligible for the annual SPF performance evaluation for the specific case studies considered in the present work. As shown in Fig.7, 15 isothermal nodes, equally spaced, are considered from top (node 1) to bottom (node 15) of the tank.

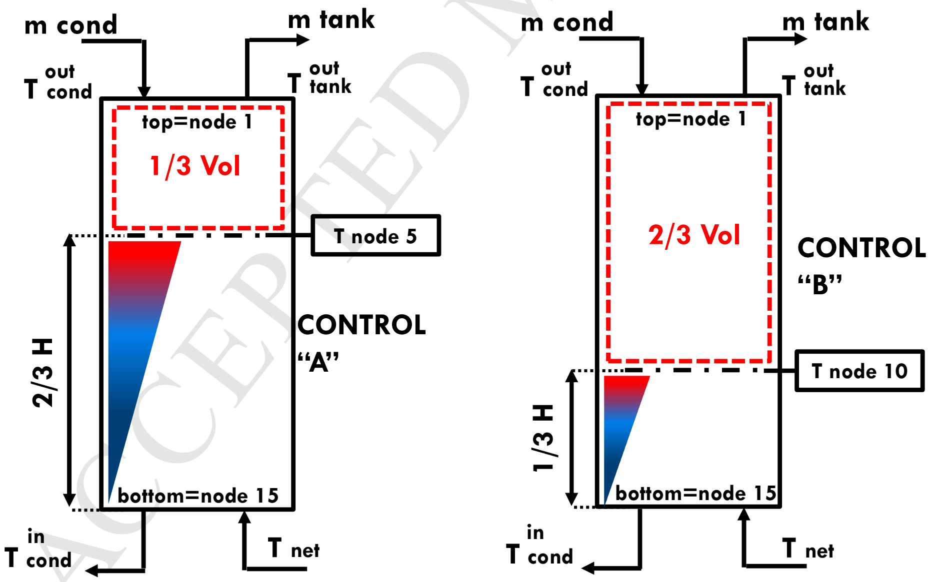


Figure 7. Tank layout for different types of control: standard control (CONTROL A) and "night&day" control (CONTROL B).

Variable speed circulation pumps (Type 742, TESS Libraries) of constant overall efficiency $\eta_{overall}$= 0.5 and no heat losses to the water are employed. Pressure drops Δp in kPa are a function of internal diameter and length of the pipes: the diameter is of 38.1·mm (1.5 inches) on each closed loop circuit; pipe length is of

100 m on each loop except the one connecting the heat pump and the tank, which is of 50 m (the heat pump working with propane has to be located on the roof for safety reasons (see Corberan et.al (2008b)) whereas the tank could be in the basement of a hypothetical user building). A 30% increase to pressure drop is applied to keep into account concentrated pressure drops due to valves, turns and other accessories. The density of water ρ_{water} is considered constant and equal to 1000 kg·m^{-3} and the consumption of each pump in kW is calculated as shown in Eq. (5).

$$\dot{L}_{pump} = \frac{\dot{m}_{water}\Delta p}{(\rho_{water}\eta_{overall})} \tag{5}$$

The heat pump is turned on or off by means of a differential controller (Type 2b in the stock TRNSYS 16 library). Two different kinds of control are identified as shown in Fig.7: base case control (“CONTROL A”), where the control sensor is located at 2/3 of the height of the tank (node 5) corresponding to a standard control, and the so-called “night&day” control logic (“CONTROL B”), which aims at increasing the working time during the night, while reducing it during the day to take advantage of the lower electrical energy prices at night. In this “night&day” control logic, the position of the control sensor is different for the night and the day. During the night, it is located closer to the bottom at 1/3 of the height of the tank (node 10); and during the day, it is located closer to the top of the tank, at 2/3 of the height (node 5). For the base case, “CONTROL A” is considered. The controller set temperature is of 45°C, measured at 2/3 of the height of the tank, which corresponds to the isothermal node 5 (see Fig. 7). The deadband is set to 10K, so that the heat pump turns off when the measured temperature (node 5) crosses 50°C and turns on when this same temperature falls below 40°C. This means that, at all times, 1/3 of the tank volume (5 m^3) located at the top of the tank (from node 1 to node 5) is at the delivery temperature (T_{user} = 50°C) or above stored if the EHP is not working, which ensures satisfying the user demand for at least three hours (at the base occupancy level of 250 people).

3. PARAMETRIC ANALYSIS AND RESULTS

The performance of the heat pump is not only affected by the temperature level of the available energy source (evaporator inlet temperature) but also by the rest of the system characteristics, such as the volume of the tank and the share of tank that is decided to be kept hot at the delivery temperature, which will also depend on the coupling to the demand (control logic). Therefore, a parametric analysis is conducted, with inputs summarized in Table 4.

	#1: Water inlet temperature at the evaporator	**#2: Share of the tank volume heated at T > 50°C (control logic)**	**#3: Volume of the tank**
Value: A	15 °C (sewage)	1/3	15 m^3 (H/D = 4)
Value: B	25 °C (neutral loop)	1/3 from 8 to 20h, otherwise 2/3	20 m^3 (H/D = 4)

Table 4.Parametric analysis input data.

There are three main parameters identified: Parameter #1 which corresponds to the water source temperature (15ºC for sewage water (CASE A); and 25ºC (CASE B) from a condensing loop existing in the commercial centre where the hotel is located). Parameter #2 which stands for the two different types of control logics identified for the EHP; and Parameter #3 which considers the influence of the tank volume in the system performance.

Regarding Parameter #2 in Table 4, the two kinds of control previously explained are considered: “CONTROL A” and “CONTROL B”. The main difference between both types of control is that, in the case of ”CONTROL A”, the heat pump will be switched on until 1/3 of the tank volume (5 m^3) has a temperature above 50ºC independently of the time of the day; whereas in “CONTROL B”, the heat pump will be switched on until 2/3 of the tank volume (10 m^3) reaches a temperature higher than 50ºC during the night. For the calculation of the electricity cost, a rate of 0.11 €/kWh has been considered during the day, and 0.06 €/kWh at night for the commercial centre. Of course these prices could vary from one country to another and for different applications, but if the difference between the electricity cost for the day and the one for the night is approximately the same, the obtained results in terms of electricity cost savings would be equal to the ones presented in this paper.

The performance of the system can be evaluated by calculating the yearly performance factor (YPF) as shown in Eqs. (6), (7) and (8):

$$YPF1 = \frac{\int_0^{8760} \dot{Q}_{cond}\,dt}{\int_0^{8760} \left(\dot{L}_{comp} + \dot{L}_{cond,pump}\right) dt} \quad (6)$$

$$YPF2 = \frac{\int_0^{8760} \dot{Q}_{cond}\,dt}{\int_0^{8760} \left(\dot{L}_{comp} + \dot{L}_{cond,pump} + \dot{L}_{evap,pump}\right) dt} \quad (7)$$

$$YPF3 = \frac{\int_0^{8760} \dot{Q}_{cond}\,dt}{\int_0^{8760} (\dot{L}_{comp} + \dot{L}_{cond,pump} + \dot{L}_{evap,pump} + \dot{L}_{user,pump})\,dt} \tag{8}$$

where time is expressed in hours, all factors are in $kJ \cdot h^{-1}$ and $\dot{L}_{cond,pump}$, $\dot{L}_{evap,pump}$, $\dot{L}_{user,pump}$ correspond to the electrical consumption of the water circulation pumps on the condenser side, evaporator side and user side respectively.

In all simulations, time step is of 30 s and temperature of water delivered to the user is verified never to fall below 49°C. Figs. 8, 9 and 10 show some days of simulations for the Reference Case (AAA in Table 4). In order to better analyse the performance of the system, several critical points have been identified (points 0–4) in these figures.

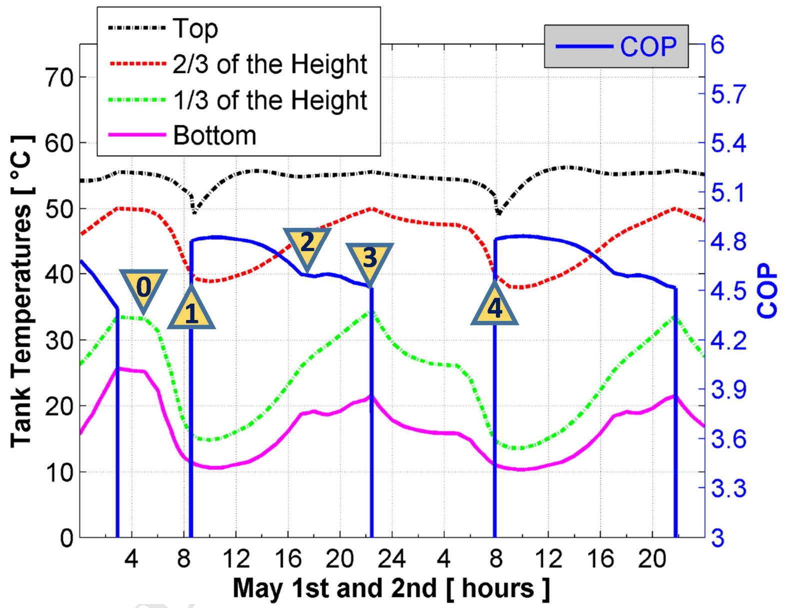


Figure 8. Temperatures along the height of the tank and COP during 2 days of simulations. Case AAA (Reference case).

As it can be observed, at point 1, the heat pump switches on when the temperature sensor located at 2/3 of the height of the tank (node 5) falls below 40ºC. Not much earlier than this, at point 0, temperature on top of the tank is shown to decrease sharply, due to a strong demand on the user side (between 7 and 8h, as shown in Fig. 9). A bigger inertia of the tank (its volume) will increase the delay between the peak of the demand and the beginning of a working cycle of the heat pump. In other words, this peak of the load profile is what triggers the EHP on most of the time.

As it can be observed, at the beginning of each on cycle (point 1), the COP of the EHP is higher (around 4.8) as the bottom of the tank is colder. From that moment, the heat pump starts heating the water in the tank, making the COP decrease as the temperature at the bottom of the tank increases, reaching a value around 4.5 at the end of the cycle (point 3). This remarks how maintaining a good stratification of temperatures along the height of the tank improves performance. When the water temperature of the control sensor located at 2/3 of the height of the tank reaches the value of 50 ºC, the heat pump switches off (point 3). Then, the temperature evolution of the water in the tank will only depend on the evolution of the water flow rate at the demand side as well as the heat losses to the ambient which can be considered negligible.

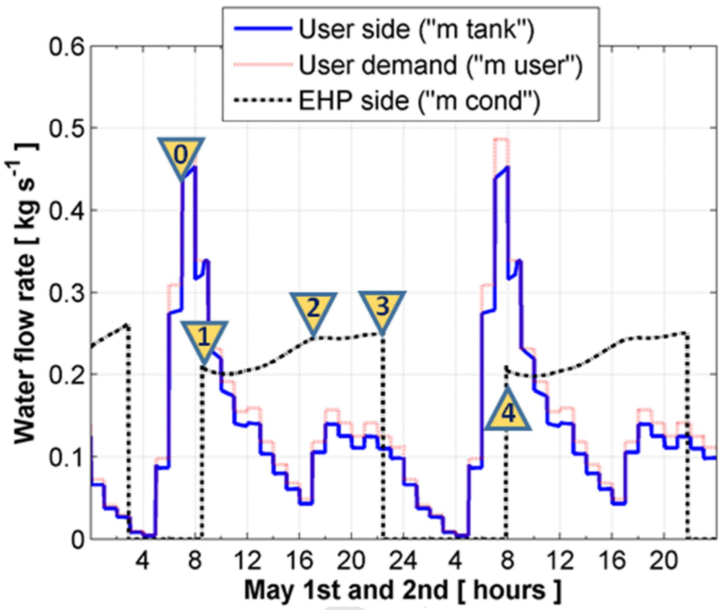


Figure 9. Evolution of the water flow rates entering and exiting the tank during 2 days of simulations. Case AAA (Reference case).

Fig. 9 shows, for the same days of simulation, the evolution of the water flow rates entering and exiting the tank, on the EHP side ("m cond") and on the user side ("m tank"), as well as the user demand ("m user"). It should be pointed out that, the water flow rate that is extracted from the top of the tank ("m tank"), is a result of the energy balance presented in Eq.3. This is why it sometimes takes slightly lower values than the water sent to the user ("m user"), in those cases where the temperature at the top of the tank is higher than the desired water temperature to be sent to the user (50ºC). Nevertheless, it is directly related to the user demand, so the higher the sanitary hot water demand, the higher will be "m tank", and vice-versa.

As it can be observed, when the heat pump is switched off, the water flow rate entering and exiting the heat pump ("m cond") is zero; so, the greater the sanitary hot water demand ("m tank"), the sharper will be the decrease in the temperature in the tank (from point 0 to point 1). Analogously, it can be observed that the evolution of the temperature in the tank is flatter when the user demand is lower (from point 3 to point 4).

On the other hand, when the heat pump is working, the evolution of the temperature in the tank will be a result of the energy balance between the water flow rate circulating in the EHP ("m cond") and the one that is circulating on the user side ("m tank"). As it can be observed, when the user demand decreases (from point 1 to point 2), the water temperature in the tank increases, and so does the water flow rate entering the EHP ("m cond"); whereas at point 2, the user demand gets slightly higher values and remains approximately constant until reaching the end of the cycle (point 3), making the trend in the evolution of the water temperature inside the tank be approximately flat. Finally, when the water temperature of the control sensor (2/3 of the height) falls below 40ºC (point 4), the cycle recommences.

Fig. 10 shows the evolution of the performance of the heat pump during the same days.

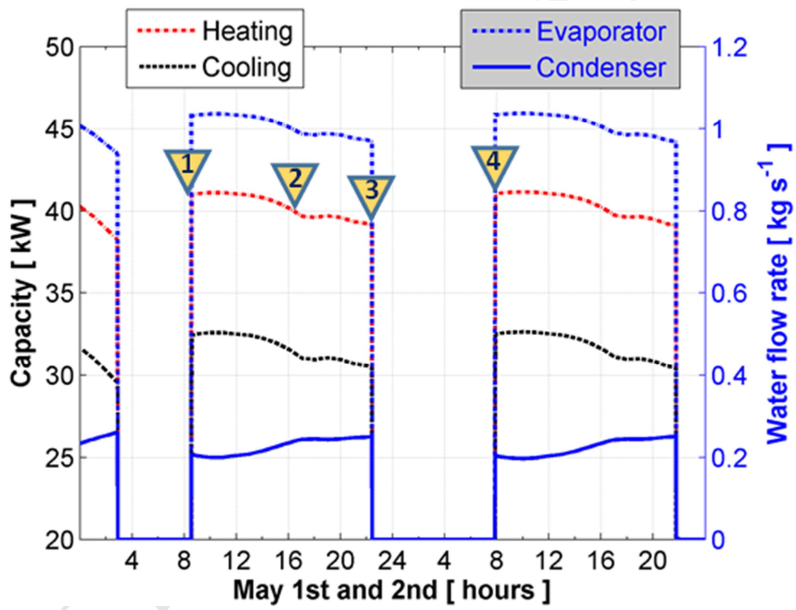


Fig.10 Heating, Cooling capacity and water flow rates during 2 days of simulations. Case AAA (Reference case).

It is shown in Fig.10 how flow rates of water at the condenser and the evaporator vary in order to achieve the optimal condensing pressure for each set of boundary conditions (water inlet temperature at the condenser, which is the temperature at the bottom of the tank, water inlet temperature at the evaporator fixed at 15°C), and subsequently, how heating and cooling capacities evolve. Thus, at point 1, the heating capacity of the heat pump gets greater values, as the temperature difference between the inlet water temperature at the condenser (bottom temperature of the tank) and the desired outlet water temperature at the condenser (fixed at 60ºC) gets greater values. At this point, and according to the heat pump performance correlations, the lower the temperature at the bottom of the tank, the higher the heating capacity; analogously, the water flow rate will get lower values at the beginning of the cycle (point 1).

Then, as the water inlet temperature in the condenser (bottom of the tank) increases, the heat pump capacity slightly decreases, and the water flow rate entering the condenser gets higher values. It should be pointed out that the evolution of the cooling capacity is the same as that of the heating capacity, as it is a result of the variation of the optimal condensing pressure for each value of the water temperature entering the condenser. The same happens with the evolution of the water flow rate at the evaporator, which follows the evolution of the cooling capacity, for a given temperature difference in the evaporator of 5K.

The results of the simulations are summarized in Tables 5a and 5b, depending on the value of the temperature level of the available energy source (Parameter #1) since the higher the temperature of the source, the better the performance of the EHP, being the increase in the YPF1 a 14.66% higher (from AAA to BAA), when changing the type of source from sewage water (15ºC) to condensing loop (25ºC).

	YPF1	**YPF2**	**YPF3**	**% ON time (8-20h)**	**% ON time (20-8h)**	**L_{comp} [kWh] ($\cdot 10^{-6}$)**	**Average tank bottom T[°C]**	**Electricity Cost savings (%)**
Case: AAA	4.98	4.94	4.93	88.08	34.29	46.70	11.28	(Reference case at T_{source}=15ºC)
ABA	4.89	4.85	4.84	25.97	99.21	47.98	12.76	23%
AAB	4.97	4.93	4.92	98.42	24.24	46.83	11.65	-5%
ABB	4.91	4.87	4.87	24.62	100.00	47.72	12.27	24%

Table 5a. Parametric analysis results, cases with A value (15ºC) for the water inlet temperature at the evaporator.

	YPF1	**YPF2**	**YPF3**	**% ON time (8-20h)**	**% ON time (20-8h)**	**L_{comp} [kWh] ($\cdot 10^{-6}$)**	**Average tank bottom T[°C]**	**Electricity Cost savings (%)**
Case: BAA	5.71	5.61	5.60	60.63	34.79	41.52	12.72	(Reference case at T_{source}=25ºC)
BBA	5.60	5.50	5.50	27.88	69.47	42.51	15.21	15%
BAB	5.74	5.64	5.63	79.56	14.22	40.76	11.57	-8%
BBB	5.61	5.51	5.51	5.36	91.13	42.12	13.67	30%

Table 5b. Parametric analysis results, cases with B (25ºC) value for the water inlet temperature at the evaporator

It is shown that this specific implementation of the “night&day” control logic (Parameter#2=B) is always detrimental to the EHP performance, particularly if the source is at a higher temperature due to the higher heating capacity of the EHP in this case, which implies a better capacity to "follow" the demand and a lesser need for storage. This decrease in performance gets values from 1.8% (from AAA to ABA) and 1.96% (from BAA to BBA), up to 2.32% in the worst of the cases (from BAB to BBB).The higher value for the tank volume (Parameter#3=B) does not practically affect the performance of the system, presenting a slightly decrease of -0.2% from AAA to AAB, and being practically negligible when the “night&day” logic is applied (from BBA to BBB). With this logic, in fact, the longer working cycles during the night heat up the bottom of the tank to a higher temperature, as it can be observed in case BBA where there is an increase of 2.5K in the average water temperature at the bottom of the tank with respect to the reference case BAA. This effect is shown to be lower in the case of a larger tank (case BBB).

Taking a look at the electricity costs savings, it can be observed that, as it was expected, “night&day” logic is favourable, being the maximum electricity cost savings around 23% (from AAA to ABA), and being practically the same for a higher volume of the tank (from ABA to ABB). However, if the “night&day” logic is not applied, the electricity costs are a bit higher, around 5%, the higher the tank volume (from AAA to AAB). So, it can be concluded, that considering a higher tank volume is only beneficial in the case of considering the “night&day” logic. This is especially relevant when considering a higher source temperature. As it can be observed in table 5b, up to 30% electricity cost savings can be obtained when increasing the volume of the tank and considering the “night&day” logic (from BAA to BBB). However, it should be stressed that a bigger tank would imply a higher installation cost. Thus, when building a new installation, an economic analysis should be done in order to select the optimal solution, for a given sanitary hot water demand profile.

4. CONCLUSIONS

In this paper, a water-to-water EHP prototype for sanitary hot water production using propane as a refrigerant has been coupled to a storage tank and an hourly demand profile for a specific end-user (a hotel in Strasbourg, France). Performance of the EHP has been modelled in IMST-ART and assessed through a parametric analysis, with the temperature of the source, the control logic and the volume of the storage tank as parameters. The results showed that the performance of the EHP, although satisfactory in all cases, depends on decisions taken on the whole system: the “night&day” control logic proved to have a marginal, detrimental effect to the energy performance, but a significantly positive effect in the electricity cost savings, for the given electricity rates (electricity cost at night almost half of the electricity cost during the day); a larger tank resulted beneficial only when the “night&day” control logic was adopted. It is then suggested that, in order to estimate the performance of an EHP water heater, there is a need to consider a specific user with its own characteristics, also choosing appropriate control strategies. Future work will also have to include a comparison between the propane solution and a CO_2 heat pump working at the same boundary conditions.

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A water-water R290 HP booster for Sanitary Hot Water (DHW) production was optimized >The assessment of the system performance was carried out in a complete TRNSYS model >A DHW profile for a hotel in Strasbourg was considered> A parametric analysis was carried out for the system design and control parameters >Control logic of the tank has the largest influence in the performance of the system.

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