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Optimal design and operation of a multi-variable heat pump system for sanitary hot water production

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ABSTRACT

According to the European Commission, buildings are nowadays responsible for the 40 % of the energy consumption and 36 % of CO_2 emissions in Europe, corresponding the gross of the energy consumption to the air-conditioning and sanitary hot water (SHW) production systems. Within the introduction of the Near Zero Energy Buildings (NZEB) concept, the percentage of the energy consumption of SHW production systems is expected to dramatically increase. Therefore, in order to reduce the energy usage for SHW production in the residential sector, it will be key the use of highly energy efficient technologies as well as good design, installation procedures, and operation strategies carried out in the facilities.

This research work is framed in a H2020 European project titled Geot€ch 'GEOthermal Technology for economic Cooling and Heating' whose aim is to develop a multi-variable heat pump system solution for heating, cooling and SHW production, making the best use of hybrid HP and control technologies.

The present work is focused on optimizing the design and operation of such a multi-variable HP system when it works for SHW production. In this context, it is not only important to maximise the HP efficiency but also to minimize the system efficiency losses that appear when coupling the HP to the storage tank. In order to fulfil with the European normative EN-1717:2000 regarding the general requirements of devices to prevent pollution of potable water by backflow, the coupling of the HP with the storage tank must be indirect.

An integrated system model in TRNSYS has been created in order to analyse three different options of indirect coupling: (i) coil heat exchanger inside the storage tank, (ii) external brazed plate heat exchanger and (iii) double wall condenser at the HP. The aim of this work is not only to select the optimal type of HP coupling but also to optimize the system operation for three representative climates existing around Europe. Results conclude that up to 30% of the system energy efficiency can be spoiled either by not selecting the optimal type of coupling or not making the system work under optimal operating conditions.

Keywords: *multi-variable heat pump system, sanitary hot water production, energy optimization, TRNSYS.*

1. Introduction

According to the European Commission, the residential sector is currently responsible for 40 % of the energy consumption and 36 % of CO_2 emissions in Europe [1]. Particularly, in the case of the energy consumption in the EU households, the Heating Ventilation and Air Conditioning (HVAC) system corresponds to the main consumer with a 65 % of the total use. Whereas the consumption associated to Sanitary Hot Water Production (SHW) accounts only for the 14.5 % of the gross energy consumption [2].

The concept of Near Zero Energy Building (NZEB) was introduced for the first time by the EU in the Energy Performance Directive (EPBD) 2010, with the aim of decreasing the HVAC associated energy consumption as much as possible. In this context, it is expected that the introduction of the concept of NZEB will make the current HVAC percentage of energy consumption highly decrease. Consequently, the SHW production will play a key role in the reduction of the energy consumption in the residential sector.

In order to reduce the energy consumption related to the SHW production, it is not possible to act on the demand without running the risk of not satisfying the user comfort. In this way, the use of highly energy efficient technologies and the reduction of the energy losses of the whole system are the key factors to achieve an energy reduction. The reduction of the system energy losses can be achieved by means of a proper sizing of the different system components as well as a correct design of the facility, regarding the interconnection between the different components and the integrated system operation. Moreover, the replacement of the current systems for SHW production, gas (55 %) and electricity (20 %) [3], that are poor efficiency technologies with high CO_2 associated emissions, plays a key role in Europe`s objective of decarbonisation.

Accordingly, the use of Heat Pump (HP) technology for SHW production is continuously increasing. In this context, the Air Source Heat Pump (ASHP) corresponds to the fastest growing HP segment across Europe, according to [4], since this technology has a very high energy efficiency, well beyond the conventional systems with gas and electricity and it is recognized by the European Directive (ED) 2009/28/CE as a renewable energy resource [5]. Nevertheless, when considering the global efficiency of the system it is important to analyze not only the HP but also the system as a whole, including the interconnection of the different components and the integrated system operation. Therefore, in order to optimize the system energy performance, not only the heat pump must be efficient but also the design and operation of the integrated system.

As a part of the system, it is important to remark the importance of the thermal energy storage (TES) in the system global efficiency, as highlighted in [6, 7]. Several studies point out the importance of a good stratification in the storage tank and analyze its influence on the system energy efficiency [7–9]. It should also be stressed the importance of making a proper control and hydraulic integration of the different components, as stated in [8, 10, 11], where the effect of the hydraulic integration of the HP and the TES as well as different control strategies over the system are assessed. These strategies for an optimal design and operation of the system get more complex when the number of variables in the system increase. Additionally, in the case of SHW production, and according to the European Normative EN-1717:2000, the coupling between the HP and the TES must be indirect to prevent pollution of potable water by backflow in case of a refrigerant leakage. This indirect connection affects significantly the energy performance of the facility reducing the energy efficiency up to 30%.

This research work is focused on optimizing the design and operation of an ASHP system for SHW production. The system analyzed is a multi-variable system where, not only the frequency of the HP compressor can vary but also the secondary fluid mass flow rate at both sides of the heat pump (variable fan frequency in the evaporator, and variable water circulation pump frequency in the condenser). On one hand, the optimal indirect connection between the HP and the TES will be determined. For this purpose, three different options of indirect connection are considered: a coil heat exchanger inside the tank, an external braze plate heat exchanger (BPHE) and a double wall condenser (DWC) at the heat pump. On the other hand, the optimal operating points of each of the different components of the system (compressor, fan and circulation pump(s)) will be determined for each type of indirect connection. The assessment will be performed for the three representative climates in Europe: Athens (warm climate conditions), Strasbourg (average climate conditions) and Helsinki (colder climate conditions), according to the Energy labelling normative [12].

2. Methodology

The study presented in this research work was performed using the transient system simulation tool TRNSYS [13]. TRNSYS was used to analyze the energy performance of three different types of indirect connection: coil case (CC), external heat exchanger case (EHXC) and double wall condenser case (DWCC). Additionally, a model for the direct connection case (HP directly connected to the TES, where the secondary fluid of the condenser is the water stored in the tank) was created, and it corresponds to the reference case or base case (BC). The integrated system models developed are shown in Figure *1*.

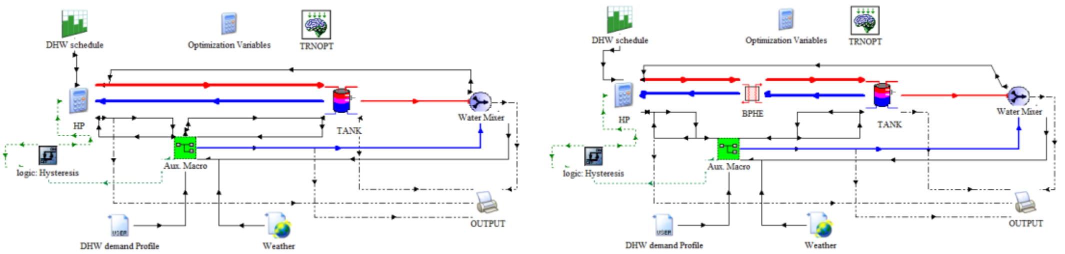

Figure 1. TRNSYS integrated system model layout. (a) BC, CC and DWCC models; (b) EHXC model [14].

The main difference between the models consists of the type of coupling between the HP and the TES, whilst the control strategy and the components are common. According to the EN 1717:2000, the connection between the HP and the TES has to be indirect, in order to prevent the potential contamination of the drinkable water due to a refrigerant leakage. Figure *2* shows a schematic diagram of the different types of indirect connection analyzed.

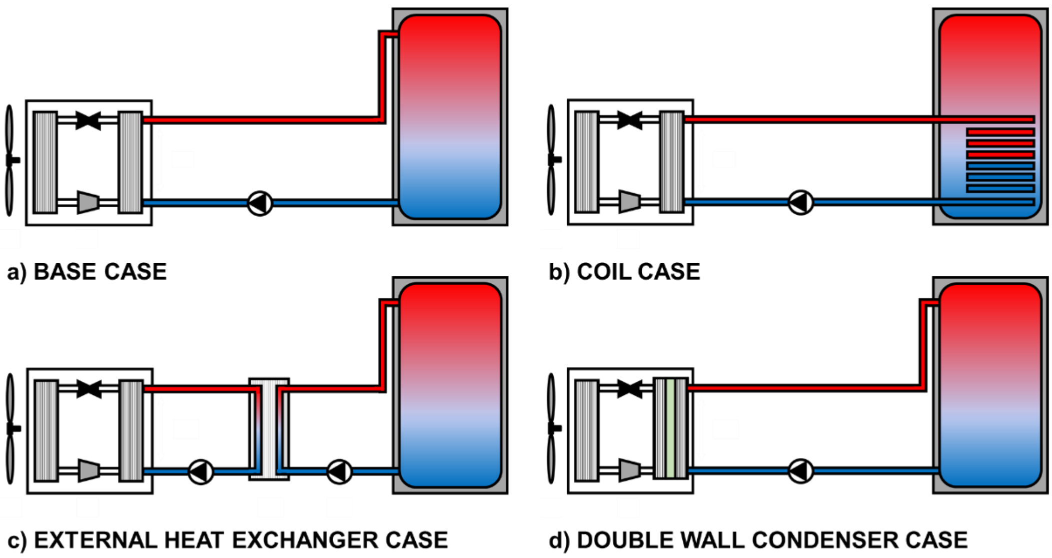

Figure 2. Sketch of the differences between the cases analyzed: a) Base Case (BC), b) Coil Case (CC), c) External Heat Exchanger Case (EHXC) and d) Double Wall Condenser Case (DWCC) [15].

As it can be observed, the BC does not fulfill with this normative, since the water circulating through the condenser is exactly the same as the one entering the storage tank. The CC complies with the normative since the water in contact with the condenser is not in direct contact with the tank because the heat exchange process takes place in a coil heat exchanger inside the tank. The EHXC includes a BPHE in which the heat exchange process takes place between the water coming from the condenser of the HP and the water from the storage tank. Finally, the DWCC makes possible the indirect connection by means of a DWC in the HP, which introduces an air gap in the condenser between the refrigerant channel and the drinkable water channel.

In order to make a fair comparison between the different cases, it is important to guarantee that the thermal energy stored in the tank is exactly the same in each simulation. Thus, a control strategy to handle the average temperature in the storage tank is programmed in the model. The control temperature selected is 45 °C, but the conclusions are of general application independently of the value of this control temperature. Regarding the legionella potential problem, the tank will be treated periodically with chemicals.

Finally, it should be pointed out that the control only allows the HP to work for SHW production during the period comprised between 05 am to 08 am because the heating and cooling mode are priority working modes during the day. This strategy allows the system to work in heating or cooling mode during the day also being able to satisfy the SHW demand.

2.1. Integrated system model

In the model used for the analysis, shown in Figure *1*, the main components of the system can be identified: the HP, the TES system, the BPHE and the DWC.

The HP was experimentally tested and fully characterized in the laboratory (more details are presented in [16, 17]), its performance correlations were developed and it was implemented as a black box in the integrated system model in TRNSYS. It comprises the model of an innovative Dual Source Heat Pump (DSHP) developed in the Geot€ch European project. The HP works with R32 as refrigerant with a nominal capacity of 8 kW and it is a variable capacity HP (inverter compressor) that can provide heating, cooling and SHW. Additionally, the HP is equipped with a variable capacity fan as well as a variable circulation pump, which are included in the model. In the present work, the analyzed case corresponds to the DSHP for SHW production and only working with air as source.

The TES system was simulated using the TRNSYS type 60, as it is the most detailed model according to TRNSYS [13], it has been widely used in literature, and experimentally validated in [18]. A commercial model of 200 liters was chosen, concretely WOLF SE-2 model with the possibility of integrating a coil inside. The volume is the usual one for the demand considered, a single-family of 3 people, which was calculated with *DHWcalc* software [19].

The BPHE, which is also integrated in the EHXC case, is the model B16 from SWEP. The model was correlated, UA-Value and pressure drop, as a function of the number of plates and mass flow rates in order to couple the model of the BPHE chosen in the integrated system model. The reason was to obtain in this way the optimal number of plates of the BPHE in the system. The considerations and correlations are presented in next section.

In order to include the DWC in the model, it was necessary to consider a fouling factor (β), provided by the manufacturer [20], that represents the inefficiency introduced in the condenser when including the air gap. A new thermal conductivity (k') for the plate was obtained in order to simulate the air gap introduced in the DWC departing from the fouling factor, as shown in Equation (1). Considering the new thermal conductivity, it was necessary to obtain new HP performance correlations since it introduces an inefficiency penalty in the model. IMST-ART software [21] was used to obtain the new correlations, which were updated in the DWCC modelled in TRNSYS.

$$\frac{e}{k}+\beta=\frac{e}{k'}\rightarrow k'=\frac{e}{\frac{e}{k}+\beta} \tag{1}$$

2.2. Performance indicators and optimization variables

In order to assess the energy performance of the different cases, it was decided to use the Seasonal Performance Factor (SPF) definition from the European project SEPEMO-build *"SEasonal PErformance factor and Monitoring for heat pump systems in the building sector (SEPEMO-Build)"* [22]. Particularly, the SPF4 shown in Equation (2) was selected, as it evaluates the system global efficiency.

$$SPF_4=\frac{Q_{W\_hp}+Q_{W\_bu}}{E_{B\_pump}+E_{S\_fan}+E_{W_{hp}}+E_{W\_bu}} \tag{2}$$

It should be pointed out that in order to check that user comfort is met at all times, a restriction to the model was imposed. The objective consists of guaranteeing always a minimum temperature of 35 °C supplied to the user. In order to control this restriction, the comfort indicator presented in Equation (3) was programmed in the model, limiting its value to 95 % during the daily working hours (8 am to 9 pm).

$$Comfort_{factor}=\frac{\int \dot{m}_{Tsup\geq 35}}{\int \dot{m}_{Total}} \quad if\ 8\,am<t<9\,pm \tag{3}$$

Regarding the optimization of the system operation, the variables considered are the following: frequency of the compressor (fc), frequency of the fan (ff) and temperature difference at the secondary fluid in the condenser (dTc). Additionally, in the case of the EHXC, also the number of plates (Np) of the BPHE and the ratio between the secondary and primary flow in the BPHE ($\dot{m}$2-BPHE to the tank and $\dot{m}$1-Condenser to the BPHE), must be taken into account. Figure *3* shows the range of values considered for the optimization process of each case.

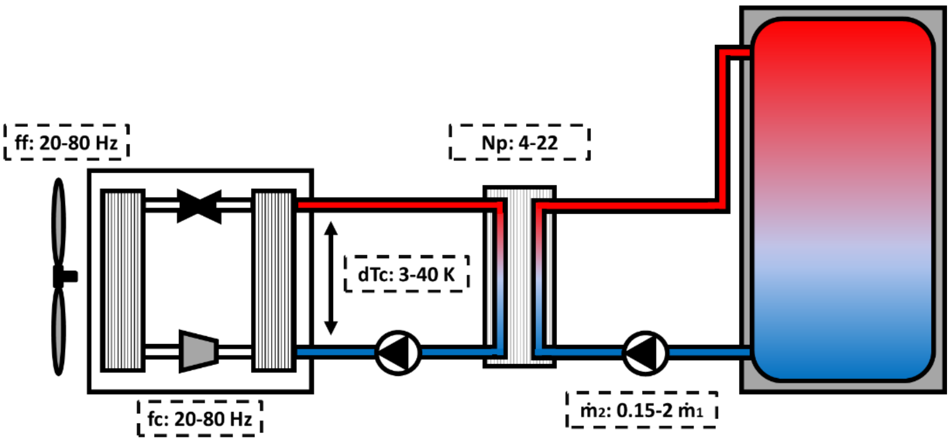

Figure 3. Optimization variables and values considered for the assessment [15].

2.3. Optimization methodology

The optimization study was performed by using TRNSYS optimization tool that is called TRNOPT [13]. TRNOPT results were validated with a sensitivity study. The sensitivity study performed consists on varying one of the three variables within the optimization range: ff (20-50-

80) Hz, fc (20-35-50-65-80) Hz, and dTc (3-5-10-25-40) K, while keeping the rest of the variables at a nominal point. The nominal point considered was: ff=50 Hz, fc=50 Hz and dTc=5 K. Figure *4* shows, for each type of indirect connection considered, an example of the SPF4 results obtained when only varying the fan frequency for the case of Helsinki in summer conditions. Additionally, Figure *4* shows the optimization point resulting from TRNOPT tool for each type of indirect connection. As it can be observed, the optimization point resulting from TRNOPT optimization tool coincides with the maximum of the SPF4 curve located at a value of 20 Hz fan frequency, presenting a slightly higher SPF4 value as expected. This sensitivity analysis was performed for all the variables and locations considered validating thus the use of TRNOPT tool for carrying out the optimization of the multi-variable system.

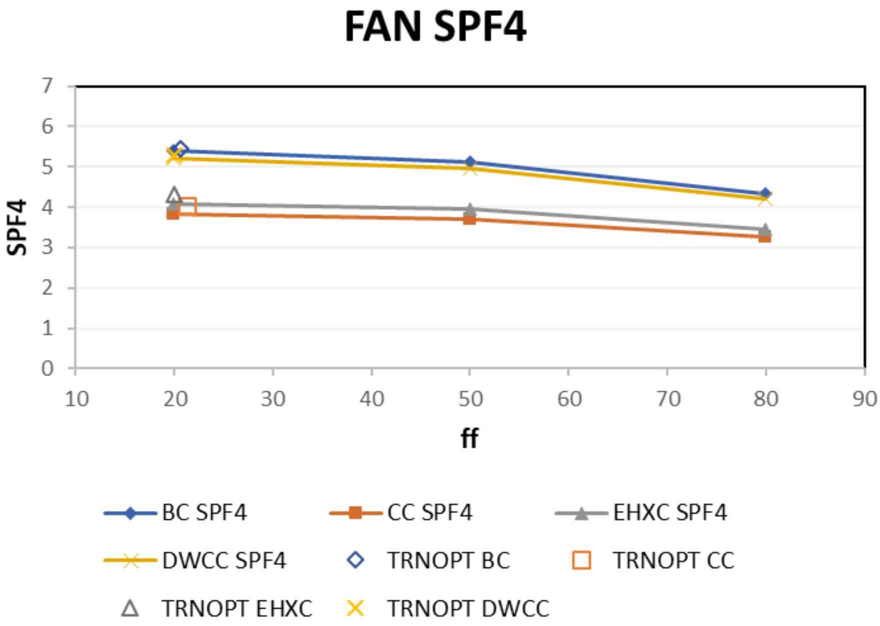

Figure 4. SPF4 results for the variation of the fan frequency and optimization point resulting from TRNOPT tool.

2.4. Considerations and assumptions

Equation (4) corresponds to the correlation of the UA-Value of the BPHE introduced to the integrated system model, as a function of the number and area of plates and the mass flow rates circulating through it. In a similar way, Equation (5) introduces the pressure drop correlation of the BPHE.

$$UA = \frac{C_1 \cdot N_p \cdot A_p}{\left(2 \cdot \dot{m}_1 / (N_p - 2)\right)^{-C_2} + \left(2 \cdot \dot{m}_2 / N_p\right)^{-C_2}} \tag{4}$$

$$\mathrm{dP}_1 = \mathrm{C}_1 \cdot \left(\frac{m_1}{(N_p - 2)/2}\right)^2 \qquad \mathrm{dP}_2 = \mathrm{C}_1 \cdot \left(\frac{m_1}{(N_p)/2}\right)^2 \tag{5}$$

As previously shown in Figure *5*, the number of plates was limited to a maximum value of 22, which corresponds to that value in which the relative improvement of the SPF4 with the number of plates is lower than 1.5 %, as shown in Figure *5*. It should be pointed out that the extra-consumption of the circulation pumps introduced when increasing the number of plates is taken into account in the pumps consumption (see equation 4 and 5).

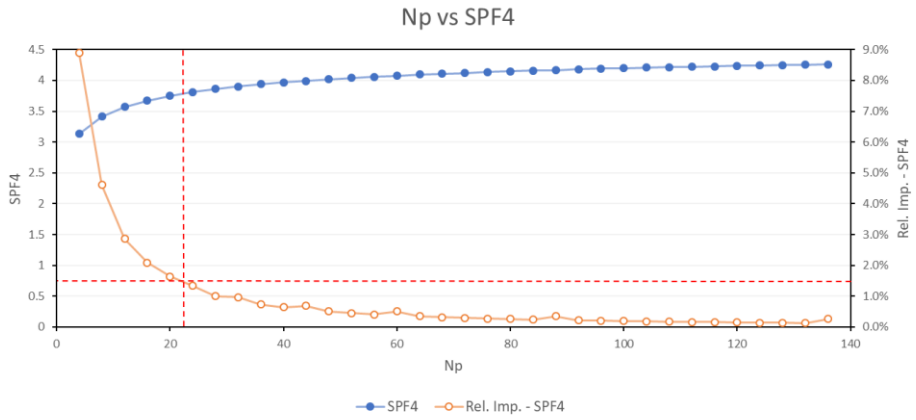

Figure 5. Relative improvement of SPF4 with the number of plates.

The assessment of the system was performed for summer and winter period. For the analysis, one typical day was selected for summer and another one for winter, for each of the locations studied (Athens-Strasbourg-Helsinki). The typical day was selected in a way that the mean temperature of the day is similar to that of the season considered and the temperature evolution of the day followed a typical smooth sinusoidal evolution. The typical days selected are presented in Table 1. In the table, also the tap temperatures considered are presented for each of the locations.

Table 1. Typical day and tap temperature values.

	ATHENS	STRASBOURG	HELSINKI
Typical day summer	30th July	21st June	30th June
Typical day Winter	3rd February	4th March	29th December
Mean temperature	26.09	17.94	14.92
Mean temperature	26.09	17.93	14.93
Mean temperature	9.65	1.67	-6.77
Mean temperature	9.68	1.68	-6.82
Ttap summer (C)	19.62	13.40	10.46
Ttap winter (C)	10.62	5.29	5.11

3. Optimal design and operation results

This section presents the results for the optimal operation points of the fan (ff), compressor (fc), temperature difference in the condenser (dTc), flow rate through the secondary circuit of the BPHE ($\dot{m}_2$) and the number of plates (N_p) of the BPHE; as well as the SPF4 results. In Figure *6* the SPF4 and optimal operation point results for Helsinki winter case are presented. Exactly the same analysis was performed for the rest of the locations and seasons. Results are shown in Table *2*.

The SPF4 results in summer are higher than the ones in winter, as well as the ones for the warmer climate (Athens) are better than that for the colder climate (Helsinki). The highest SPF4 corresponds always to the BC followed by DWCC with only a difference of 3.66 % in average. While CC presents a difference of 27.10 % in average compared to the BC. Finally the EHXC has a 5.41 % better SPF4 than the CC. Therefore, it can be concluded that the DWCC is the optimal indirect connection case taking into consideration the system global efficiency. Further details can be found in [15] in which a techno-economic assessment is also presented.

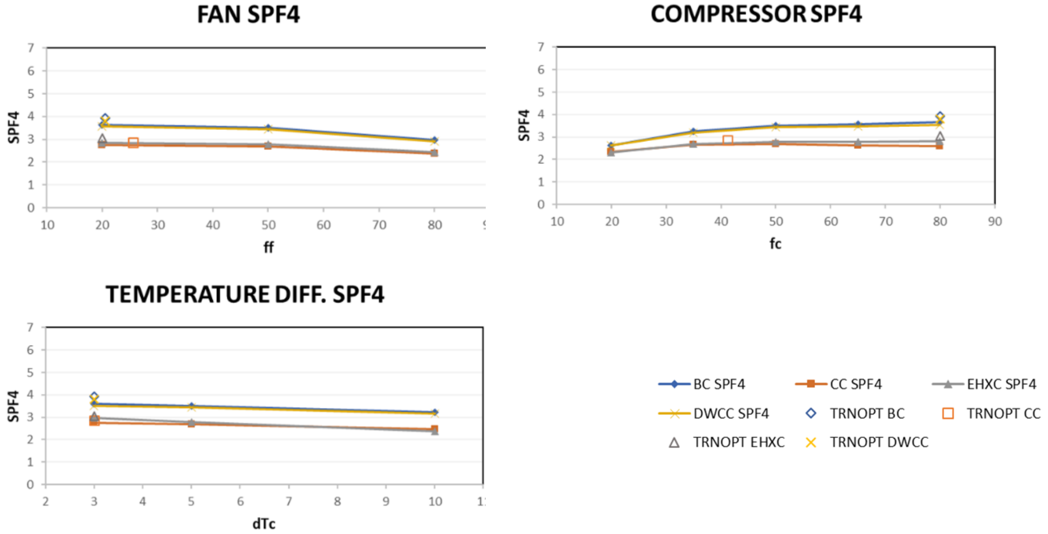

Figure 6. SPF4 and optimal operation point results for Helsinki winter case.

Table 2 shows the results for the three representative climates in Europe: warmer (Athens), average (Strasbourg) and colder (Helsinki) The optimal operation points are obtained for each of the different indirect connection cases assessed: BC, CC, EHXC and DWCC both for the winter and summer period. Additionally, the SPF4 results are shown for each configuration and location considered. The optimal type of indirect connection for each case is highlighted in bold.

Regarding the optimal operation points, results show that the fan frequency tends to the minimum values (ff=20-25 Hz); the temperature difference in the condenser also tends to the minimum values (dTc=3-3.5 K), while the number of plates in the condenser tends to the maximum value (Np=22). The mass flow rate in the secondary circuit of the EHXC tends to a range of 0.74-0.91 times the mass flow rate in the primary circuit; and finally, the compressor frequency shows a different trend depending on the period and climate analyzed. As it can be seen in , the compressor frequency tends to the range 33.75-48.75 Hz in summer period, whereas in winter there are some cases in which the value tends to the maximum (fc=80 Hz). The latter fact always happens in BC but never in CC, while it sometimes happens for DWCC and EHXC.

The fan frequency tends to the minimum value because a higher fan frequency means an increase in the fan energy consumption, which is much higher than that of the condenser capacity. There are some cases, in which the optimal value is not exactly the minimum, mainly because the condenser capacity increase compensates the higher fan energy consumption, but it never exceeds 25 Hz. The temperature difference in the condenser influences not only the energy consumption of the circulation pump, but also the compressor consumption, being its impact greater in the case of the compressor consumption. This, similarly to the latter case, explains why this variable tends to the minimum value, because in case of a higher temperature difference (lower secondary fluid flow rate) the increase of compressor consumption is higher than the decrease of the circulation pump. Regarding the number of plates in the BPHE of the EHXC, it always tends to the maximum number of plates, which is 22, independently of the case, location and season. Evidently, the higher is the number of plates in the BPHE the more efficient is the heat exchange on it, allowing thus the system to have a better efficiency and always compensating the extra consumption introduced by the circulation pumps due to a higher pressure drop. The increasing mass flow rate of the secondary circuit allows the BPHE increase also the heat transfer efficiency in the indirect connection and thus the global efficiency of the system. However, the higher is the mass flow rate, the higher is the consumption of the circulation pumps, affecting directly the SPF4 of the system. This is the reason of not having the largest value of mass flow rate in the secondary circuit of the BPHE as the optimal. Regarding the compressor, the optimal values depend on the period analyzed, summer or winter, as well as on the climate and type of indirect connection considered. The differences between the optimal operation point for the different locations and season respond to a dependence of the compressor consumption with the ambient temperature and the difference concerning the indirect connections analyzed respond to a dependence of the compressor consumption with the inlet temperature to the condenser.

Table 2. Optimal variable values for SPF4 in winter and summer for each of the climates analysed.

Location	Case	ff (Hz)		fc (Hz)		dTc (K)		m1/m2 (-)		Np (-)		SPF4	
		Winter	Summer	Winter	Summer	Winter	Summer	Winter	Summer	Winter	Summer	Winter	Summer
HEL	BC	20.63	20.63	80.00	48.13	3.00	3.38	-	-	-	-	3.93	5.46
	CC	25.63	21.25	41.25	33.75	3.00	3.00	-	-	-	-	2.87	4.05
	EHXC	20.00	20.00	80.00	37.50	3.00	3.00	0.78	0.78	22.00	22.00	3.06	4.33
	DWCC	**20.63**	**20.00**	**80.00**	**45.00**	**3.00**	**3.00**	**-**	**-**	**-**	**-**	**3.80**	**5.30**
STR	BC	20.63	20.00	80.00	48.75	3.19	3.31	-	-	-	-	4.36	5.29
	CC	25.00	20.00	41.25	48.75	3.00	3.31	-	-	-	-	3.16	3.79
	EHXC	20.00	21.25	80.00	38.13	3.13	3.13	0.74	0.81	22.00	22.00	3.33	4.23
	DWCC	**20.00**	**20.00**	**80.00**	**45.31**	**3.25**	**3.25**	**-**	**-**	**-**	**-**	**4.19**	**5.14**
ATH	BC	20.00	20.00	80.00	48.75	3.44	3.31	-	-	-	-	5.02	6.50
	CC	21.25	20.00	37.50	32.19	3.00	3.38	-	-	-	-	3.66	4.71
	EHXC	22.50	20.00	41.25	38.13	3.13	3.44	0.91	0.80	22.00	22.00	3.93	5.05
	DWCC	**20.63**	**20.00**	**49.38**	**43.75**	**3.00**	**3.81**	**-**	**-**	**-**	**-**	**4.80**	**6.28**

In this way, *Table 2* becomes a guide to select the best operational values and to choose the optimal indirect connection for any multi-variable air source heat pump system for sanitary hot water production in Europe, since the three representative climates are included in the analysis.

4. Conclusions

This research work assesses the design and operation of a multi-variable ASHP system for SHW production, where up to five variables concerning the operation of the system were analyzed: fan frequency (ff), compressor frecuency (fc) and secondary fluid temperature difference at the condenser (dTc). The assessment was performed for the three representative climates in Europe: Athens (warmer climate conditions), Strasbourg (average climate conditions) and Helsinki (colder climate conditions).

Regarding the design, the optimal indirect connection between the HP and the water storage tank (TES) was determined. Results show the double wall condenser case (DWCC) as the best option in terms of system global efficiency with a slightly lower efficiency 3.66 % than the direct connection case (BC). The coil heat exchanger inside the storage tank (CC) is the worst option in terms of energy efficiency with a 27.1 % lower energy efficiency than the BC. Finally, the external brazed plate heat exchanger case (EHXC) shows a slight advantage of 5.41 % regarding the CC.

Regarding the optimal operating conditions, it was concluded that the fan frequency always tends to the minimum value of the optimization range considered (ff=20-25 Hz). Similarly, the secondary fluid temperature difference in the condenser always tends to the minimum value within the range considered for the optimization (dTc=3-3.5 K). Regarding the variables associated only to the EHXC, the optimal value for the number of plates of the heat exchanger always tends to the maximum value considered in the range, 22 plates, and the mass flow rate in the secondary circuit of the external heat exchanger to a range of 0.74-0.91 times the mass flow rate in the primary circuit. Finally, regarding the compressor frequency (fc), different trends were observed depending on the type of indirect connection and climate conditions considered. This is mainly due to the influence of the ambient temperature and the inlet water temperature to the condenser for each case considered: (fc=80 Hz) in colder ambient temperature conditions for all types of indirect connection except for the CC, or to a range of (fc=33.75-48.75Hz) in warmer ambient temperature conditions and always for the CC.

Results obtained could become a design and operation guide for any multi-variable air source heat pump system for sanitary hot water production in Europe.

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