THERMAL ANALYSIS OF TWO COMBINED SOLAR-HEAT PUMP DOMESTIC WATER HEATING SYSTEMS

Juliana Emanuella Gonçalves, juliana@lepten.ufsc.br
Allan Ricardo Starke, allan.starke@lepten.ufsc.br
Sergio Colle, sergio.colle@ufsc.br
Universidade Federal de Santa Catarina (UFSC), Departamento de Engenharia Mecânica, Laboratório de Engenharia de Processos de Conversão e Tecnologia de Energia (LEPTEN) - Campus Universitário – CEP 88040-970 – Florianópolis – SC

Abstract. *In this paper, two configurations of combined solar-heat pump domestic water heating systems are proposed. Focus is given on large daily hot water consumers, such as multi-residential buildings, hospitals or hotels, in order to explore their higher potential for energy and cost savings, and improve the economic justifiability of the proposed systems through the economy of scale. The annual thermal performance of the systems was assessed and compared to the conventional solar system using TRNSYS software. Simulations were performed for a typical meteorological year (TMY) using climate data of Florianópolis, Brazil. Actual manufacturers catalog data were used to simulate both the solar collector and the heat pumps performances. Results show that introducing a heat pump into a conventional solar system leads indeed to enhanced thermal performance. Operating costs reduction between 76-88% can be achieved in combined systems with 100 m² of solar collectors. Further analysis must include economic performance indicators in order to assess the economic feasibility of the proposed systems.*

Keywords*: combined solar-heat pump systems, domestic water heating, solar thermal energy, solar collector.*

1. INTRODUCTION (Times New Roman, 10pt, bold, upper-case)

The most recent Solar Heat Worldwide - Markets and Contribution to the Energy Supply report was release in 2015 by the International Agency of Energy (IAE, 2015). Such report comprises solar thermal market data obtained from 60 countries by the end of 2013, covering an estimated 95% of the worldwide market. According to the report, Brazil has the third (2055 MWth) and fourth (4671 MWth) largest total capacity of unglazed and glazed solar collectors in operation worldwide, respectively. However, regarding the total capacity per 1000 inhabitants, Brazil occupies the 9º positon for unglazed collectors (10.2 kWth/1000 inhab.) and only the 33º position for glazed collectors (23.1 kWth/1000 inhab.). As such, this paper intends to provide technical data to promote the use of thermal solar energy in buildings in Brazil, and therefore bridge the gap between the total capacity and the total capacity per inhabitants of solar collectors in operation.

Previous studies carried out on combined solar-heat pump systems, have demonstrated that substantial energy savings can be obtained through such systems (Chu and Cruickshank, 2014; Kamel et al., 2015). A heap pump can be integrated into a conventional solar system both in a direct connection, wherein the solar collectors are used as the heat pump evaporator; or in an indirect configuration, in which the heat pump is integrated as a closed unit. Depending on how the heat pump is connected to the collectors and storage tanks, different effects can be expected to enhance overall system efficiency, e.g. increasing the collector field efficiency and operating time through pre-cooling the inlet fluid, taking advantage of its COP to provide energy to the users.

Due to the traditional high servicing and maintenance cost charged in Brazilian market, and since this work aims at promoting the use of solar thermal energy in Brazil, focus will be given on noncomplex combined system configurations. Therefore, the indirect connection is here exploited since it relies on more reliable operation and would require minor intervention in case of system failure. The additional cost due to both the heat pump and a possible second storage tank, however, do not justify the energy savings for a single-family dwelling (Banister and Collins, 2015; Sterling and Collins, 2012). If the system is applied to a larger load, such as multi-residential buildings, hospitals or hotels, the economy of scale can thus be explored, which improves economic feasibility.

The aim of this paper is therefore to assess the thermal performance of two combined solar-heat pump water heating systems applied to large domestic hot water consumers. A case study is presented using climate data of Florianopolis-SC, Brazil. Parametric analysis will be carried out in order to evaluate the influence of systems' input parameters such as collector area, storage volume and heat pump capacity. Results will be evaluated according to performance indicators consistent to the literature.

2. SYSTEMS DESCRIPTION

In this paper, two configurations of combined systems are proposed:

(A) a dual-tank solar system combined in parallel to an air-source heat pump (ASHP), in which the solar loop and the heat pump operate independently since each one is connected to a different storage tank (Fig. 1-A); and,

(B) a dual-tank solar system combined in series to a water source heat pump (WSHP) (Fig. 1-B). In this case the energy can be supplied to the user either via the heat pump or the heat exchanger. In the particular case where both storage tanks operate with water, the heat exchanger is not necessary. By setting the heat exchanger effectiveness to be 1, a bypass between the storage tanks can be mathematically established, increasing the simulation model versatility.

An electric-based heating system and a single tank traditional solar system are also undertaken in this analysis as reference systems. Due to the intermittent nature of solar energy, all solar systems include an auxiliary heater to ensure demand is met during off-sun periods.

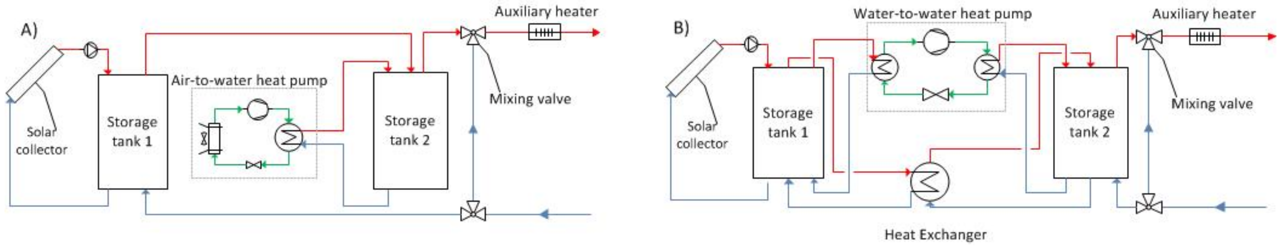

Figure 1. Combined systems proposed in this paper: A) a dual tank solar system combined to an air source heat pump (ASHP); and B) a dual tank solar-assisted water source heat pump (WSHP) system.

3. SIMULATION MODELS

All the systems were modeled in TRNSYS software (Klein et al., 2010), and simulated for a typical meteorological year (TMY) using climate data of Florianópolis-SC, Brazil. All of the systems were simulated with the same hot water profile and total daily load, and delivered domestic hot water at constant temperature, which creates a common basis for comparison, since each system deliver the same amount of energy for the same period.

The simulations were performed for a multi-residential building with a daily hot water consumption of 20 m^3 at 40 °C. The hot water profile used in this work is shown in Fig. 2, and was obtained by Salazar (2004) from in situ measurements of a 90-dwelling complex in Florianópolis. Make-up water stream was considered to be at 20°C. The main system components are the solar collectors, the storage tanks and the heap pumps, which are further described in this section.

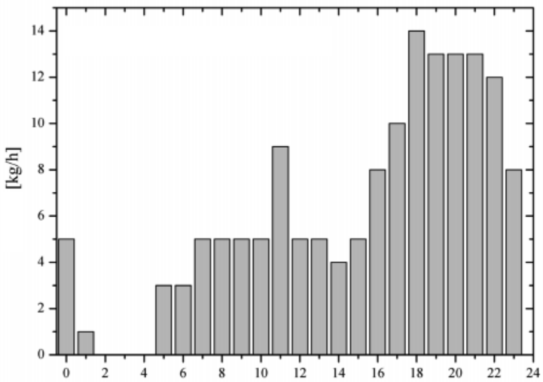

Figure 2. Daily hot water draw profile obtained by Salazar (2004).

The solar collector was modeled as flat-plate collector, using TRNSYS Type 1b. This component comprehends 5 possibilities for considering the effects of off-normal solar incidence; in this work, a second order quadratic function is used to compute solar thermal efficiency. Results based on standard tests of collector efficiency must be provided as input for this component. The solar collector used in this work corresponds to Jelly Fish JFS10 collector, manufactured by TOSI Ltda. Main input data for the TRNSYS Type 1b are detailed in Tab. 1.

Type534 was used to model the storage tanks, and corresponds to a vertical cylindrical tank with constant volume. The model accounts for storage fluid interactions with the heat exchangers, with the environment through thermal losses, and with multiple inlet/outlet couple flow streams through the storage tank. In this work, fixed outlet nodes and temperature-seeking inlet nodes were used to model each storage tank, which means that the user must specify the outlet locations (nodes) for each of the streams, while the inlet flow streams will be directed to the node closest in temperature to the incoming stream. The stratification phenomena observed in storage tanks is modeled dividing the tank into isothermal temperature node. Each constant-volume node is assumed to be isothermal and interacts thermally with the nodes above and below through conduction and convective heat transfer. The number of nodes, and hence the stratification degree, is specified by the user. For annual simulations, up to 10 nodes are considered enough to represent the stratification phenomena (Duffie and Beckman, 2013).

Table 1. Solar collector parameters/inputs (single solar collector Jelly Fish JFS10).

Parameter	Value	Unit
Fluid specific heat	4.179	kJ/kg K
Tested flow rate	70	kg/h m²
Intercept efficiency	0.77	-
Efficiency slope	15.372	kJ/h m² K
Efficiency curvature	0	kJ/h m² K²
First order IAM	0.1065	-
Second order IAM	0	-
Collector slope	35	°

Storage tank stratification can be explored to improve either single components efficiency or the entire system performance. For instance, connecting the storage tank bottom node to the solar collectors' inlet enhances collector efficiency because it takes advantage of a coolest fluid stream. On the other hand, the stream flow between the storage tank and the user is connected to the upper node in order to provide the user with the highest temperature as possible, and hence to reduce auxiliary heater requirement.

TRNSYS types used to model both the ASHP and the WSHP require an external file to determine the heat pump performance. The input file provides the power consumption and the energy transfer as a function of the load and source inlet temperatures and mass flow rates. With knowledge of the energy transfer rate, mass flow rate and input temperatures, the load and source outlet temperatures are calculated and used in the simulation. Since normalized performance data is provided, from a single set of data, different scale factors can be simulated in order to reduce/increase the heap pump output.

In this work, the heat pump files correspond to catalog performance data of commercial heat pumps, namely the air-to-water Aeromax Plus high-temperature heat pump manufactured by Kingspan; and, the water-to-water Aquazone 50PSW025-420 heat pump manufactured by CARRIER. The Coefficient of performance (COP) and heat capacity at nominal conditions for both heat pumps are shown in Tab 2.

4.1 Control strategy

The control strategies adopted for all systems aim at minimizing the auxiliary energy consumption, while ensuring water supply to the user at the desired temperature.

When coupling solar collectors to heat pumps, higher temperatures are achieved at the source side, improving heat pump performance. However, heat pumps test operating conditions in general do not cover such ranges. An issue to be considered concerns therefore the lack of heat pump operating data for higher source side temperatures. For both combined systems, the component used to model the heat pumps is not able to extrapolate data. If values outside the range are provided as input, the performance will be returned based on the closest point.

In this work, both the ASHP and the WSHP are not allowed to work outside of the operating temperature range specified by the manufacturer. By means of temperature controllers, the heat pump turns off whether the source or load inlet temperatures are outside the operating range. Table 2 gives the abovementioned temperature ranges for both heat pumps. The WSHP source inlet limit provided in the manufacturer catalog is -6.7 °C; however, in the system proposed here, since the heat pump source fluid is water, the lower limit was considered to be 2 °C due to water freezing occurrence. The entire range could be explored by replacing the source fluid with a mixture of ethylene glycol and water, for instance; however, the system initial cost would be higher.

Table 2. Heat pumps operating temperature ranges.

	Nominal COP	Nominal heating capacity [kW]	Inlet temperature range [°C]	
ASHP	3.7	43.6	Source (air)	-9 to 34
			Load (water)	26 to 54
WSHP	4.4	9.08	Source (water)	2 to 25
			Load (water)	21 to 42

The first operating mode for all the systems corresponds to the solar collector circulation loop, which depends only on the solar availability, and thus operates independently to the rest of the system. The subsequent modes rely on energy stored from operating Mode 1, except the Auxiliary heater mode, which is only enabled if no other mode is possible.

The conventional system follows the simplest strategy: Mode 1 is enabled if solar gain is available; if solar energy collected is not enough to satisfy the energy demand, the auxiliary heater is activated.

ASHP system operation modes are detailed in Tab. 3. In this configuration, the heat pump works as a back-up device, similar to the auxiliary heater. However, since the heat pump takes advantage of its COP, its operation is preferred over the auxiliary heater activation. Since the storage tanks 1 and 2 are connected is series, whenever the user requires hot water, water flows from upper node of the storage tank 1 to the node closest in temperature of the storage tank 2 (Mode 2). If the temperature reaching the user is below 40°C and the heat pump is able to operating within its temperature ranges, Mode 3 will be enabled. If the demand is not satisfied yet, the Auxiliary heater mode is enabled. ASHP system strategy control is therefore pretty similar to the conventional solar system operation.

Table 3. ASHP system operation modes.

	Description	**Condition**	**State**
Mode 1	Solar loop charges storage tank 1	$(T_{in} - T_{out})_{col} > 5°C$	Solar circulation pump ON
Mode 2	Solar energy stored in storage tank 1 is transferred to storage tank 2	User requires hot water	
Mode 3	Heat pump is enabled as a back-up heater, taking advantage of the COP	$T_{inf,source} < T_{in,source} < T_{sup,source}$ $T_{inf,load} < T_{in,load} < T_{sup,load}$ $T_{user} < T_{desired}$	Heat pump ON
Mode auxiliary heater	The auxiliary heater is activated to heat the water up to the desired temperature	$T_{user} < T_{desired}$	Auxiliary heater ON

WSHP system operation modes are described in Tab. 4, and the flow chart of the system control strategy is shown in Fig. 3. Since the heat pump and the bypass connection operate in parallel within the WSHP system, its control strategy is more complex than the others. The first operating mode operates independently to the rest of the system, the same as before. Once the storage tank 1 is charged by the solar loop, Mode 2 as well as Mode 3 is possible. The second operating mode prioritizes the bypass between storage tanks. If upper node temperature of the storage tank 1 is higher than upper node temperature of the storage tank 2, the bypass is enabled (using the heat exchanger). Otherwise, Mode 3 is activated and the heat pump is used to transfer heat from storage tank 1 to storage tank 2, taking advantage of its COP. However, Mode 3 is only enabled if the heat pump source and load inlet temperatures are within the operating range.

Table 4. WSHP system operation modes.

	Description	**Condition**	**State**
Mode 1	Solar loop charges storage tank 1	$(T_{in} - T_{out})_{col} > 5°C$	Solar circulation pump ON
Mode 2	Storage tank 1 provides energy to storage tank 2 through the bypass connection	$T_{tank1} > T_{tank2}$ $T_{user} < T_{desired}$	By pass ON
Mode 3	Heat pump transfer energy from storage tank 1 to storage tank 2, taking advantage of its COP	$T_{inf,source} < T_{in,source} < T_{sup,source}$ $T_{inf,load} < T_{in,load} < T_{sup,load}$ $T_{user} < T_{desired}$	Heat pump ON
Auxiliary heater mode	The auxiliary heater is activated to heat the water up to the desired temperature	$T_{user} < T_{desired}$	Auxiliary heater ON

The heat pump operation decreases the storage tank 1 temperature, leading to two different effects. On one hand, lower temperatures prevent the bypass activation and, on the other hand, increase solar collector efficiency, which in turn increases the amount of energy collected. During high radiation days, the energy collected is enough to enable the bypass connection. During low radiation days, the heat pump carries out the heat transfer between tanks for most of the day. As such, energy is transferred through the bypass when solar energy is available, and through the heat pump during off-sun periods. Finally, as for the previous systems, if the desired consumption temperature is not achieved yet, then the auxiliary heater will be enabled.

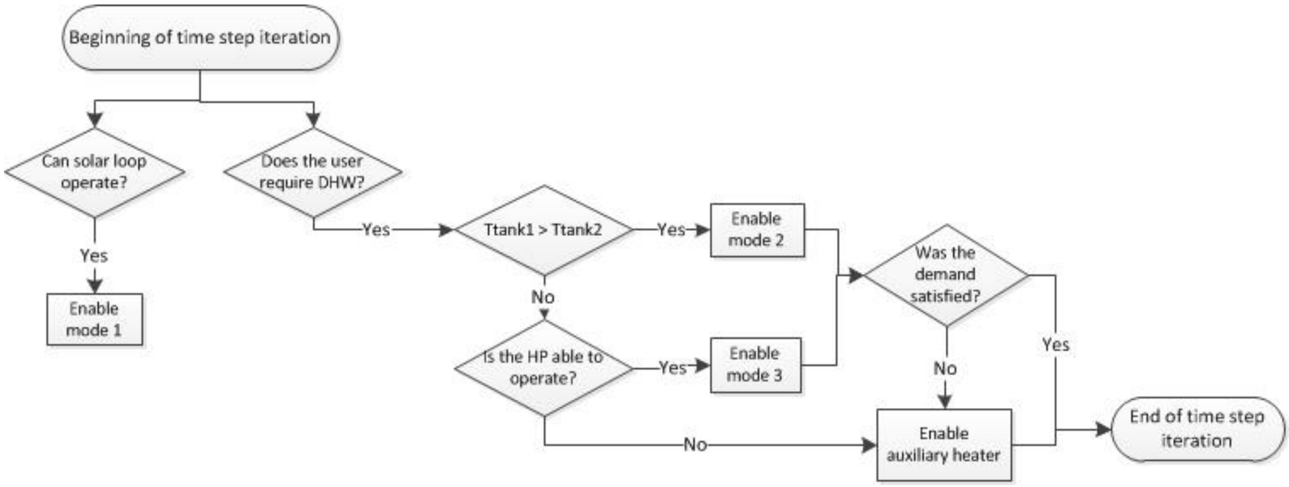

Figure 3. Flow chart of WSHP system control strategy.

4.2 Performance figures

Chu and Cruickshank (2014) conducted a review on solar-heat pump combined systems applied to the residential sector. The authors concluded that a large variety of configurations and parameters exist, which makes difficult to compare differing systems and establish an optimal design. Moreover, different authors use different performance criterion and this inconsistency also added to the difficulty of comparing the studies of different systems. Therefore, the performance indicators are used in this work as defined in IEA SHC Task44 (Malenković, 2013). The most important ones are the seasonal performance factor of the system, the seasonal performance factor of the HP, and the solar fraction, defined as follow.

The energy quantities involved in the following calculations are the heat power domestic hot water delivered to the user (Q_{DHW}); the solar heat power, (Q_{solar}); the heat power HP evaporator ($Q_{evap,HP}$); the heat power HP condenser ($Q_{TOT,el,HP}$); the electricity consumption of entire HP ($P_{TOT,el,HP}$) which accounts for the compressor and fan consumptions; the electricity consumption of all components ($P_{el,TOT}$), which in this work accounts for the heat pump and the auxiliary heater consumption (P_{aux}).

The seasonal performance factor of the system (SPF_{system}) gives its final energy efficiency and is calculated as the overall useful energy output to the overall driving final energy input for the adopted boundary, as in Eq. (1).

$$SPF_{system} = \frac{\int \dot{Q}_{DHW}.dt}{\int P_{el,TOT}.dt} \tag{1}$$

The seasonal performance factor can also be used to express the efficiency of a single component, e.g. heat pump, or a subsystem. As such, the seasonal performance factor of the heat pump (SPF_{HP}) is the ratio between its heating capacity and the overall electricity consumption over a certain period of time, according to Eq. (2).

$$SPF_{HP} = \frac{\int \dot{Q}_{cond,HP}.dt}{\int P_{TOT,el,HP}.dt} \tag{2}$$

The solar fraction (SF) definition adopted in this work is based on the green boundaries show in Fig. 4, and includes the thermal losses of the storage tank, which are considered as internal losses to the system. Higher storage losses increase the need for additional heat leading to lower solar fractions. The fraction is hence limited to one, reached when no additional heat is delivered. It can also become negative when the losses exceed the solar contributions for small-scale systems. The SF is defined as follows:

$$SF = 1 - \frac{\int \dot{Q}_{cond,HP} + P_{aux}.dt}{\int \dot{Q}_{DHW}.dt} \tag{3}$$

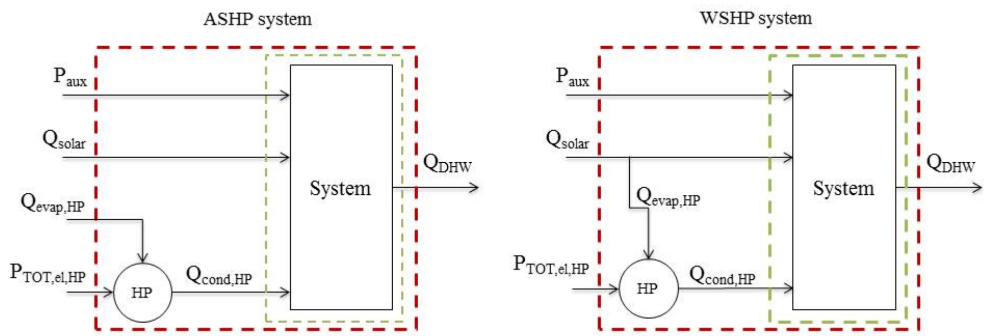

Figure 4. Boundaries used to define the performance figures.

In addition, the parameter free energy fraction (FEF), proposed by Freeman et al. (1979), was also considered in as a performance indicator. The free energy fraction is defined based on the red boundaries show in Fig. 4, and accounts for non-purchased energy ("free" energy) used to meet the demand, i.e. the solar heat power, Q_{solar}, and the heat power HP evaporator, $Q_{evap,HP}$. For the ASHP system, the free energy fraction includes both the solar energy collected and the energy extract from the environment as heat source for the ASHP. For the WSHP system, on the other hand, this fraction accounts only for the solar energy, since the collectors provide energy for both the bypass and the WSHP. However, in both cases the FEF is described by the following equation (Eq. (4)):

$$FEF = \frac{\int (Q_{solar} + Q_{evap,HP}).dt}{\int Q_{DHW}.dt} = 1 - \frac{\int (P_{TOT,el,HP} + P_{aux}).dt}{\int Q_{DHW}.dt} \quad (4)$$

The performance figures are calculated on annual basis. Moreover, since all solar systems are provided with an auxiliary heater, its efficiency was not accounted for. In another words, for the sake of simplicity, it was considered to be 1.

5. RESULTS

The different systems have been evaluated concerning the influence of solar collector area, storage volume and heat pump rate on the performance figures defined previously. Solar collector area was evaluated within the range between 30 and 300 m² for all systems. Two different approaches were adopted to integrate the storage volume into the conventional system. First, the ratio of storage volume to the solar collector area was fixed to be 75 L/m², as suggested in the literature for an optimal design (Duffie and Beckman, 2013). As such, storage volume increases as solar collector area increases. This system will be referred henceforward as Optimal Conventional system. The second approach considers a fixed 10 m^3 or 15 m^3-storage volume. For the combined systems, on the other hand, storage tanks 1 and 2 were set as 10 m^3 each, and further as 15 m^3 each, while solar collector area was evaluated within the same range. Moreover, the heat pump rate considers the range between 13-21 kW for the ASHP, and 9-27 kW for the WSHP. The values of 10 and 15 m^3 were adopted considering the total daily consumption of 20 m^3. All the systems performed better with larger storage volumes. Therefore, only the results for systems with 15 m^3 storage tanks will be presented.

Results show that combining heat pumps to solar systems indeed enhances its performances. The free energy fraction results are presented in Fig. 5, where the entire solar collector area range is shown at the left side, and the portion between 200-300 m² at the right side. The seasonal performance factors for the system and the heat pump are shown respectively in Fig. 6 and 7.

Conventional system with fixed 15 m^3 storage volume performs slightly better than the Optimal Conventional system up to 200 m² (Fig. 5, left), which does not means that the Optimal system is not optimal, but rather that the Optimal Conventional system is too small for the hot water load considered. Above 200 m² (Fig. 5, right), the performance behavior is reversed, and the Optimal Conventional system achieves FEF values very close to the unit, corroborating the above mentioned.

It can also be observed that above approximately 200 m², both combined systems are able to meet the energy demand with nearly no additional energy (either the heat pump or the auxiliary heater). In another words, both combined systems operate as different configurations of a conventional solar system, when provided with large solar collector areas. In this particular case, the ASHP system operates as a solar system with two storage tanks connected in series, while the WSHP system works as a solar system with storage tanks in parallel. The parallel connection allows the surplus energy to be transferred from storage tank 1 to storage tank 2, storing energy within both tanks. The serial connection, on the other hand, only transfers energy between tanks when energy is required by the user, limiting the energy storage to the tank connected to the solar loop. Therefore, above 200 m², the parallel system performs better than the serial one, as can be seen in Fig. 5 (left) and Fig. 6 (right).

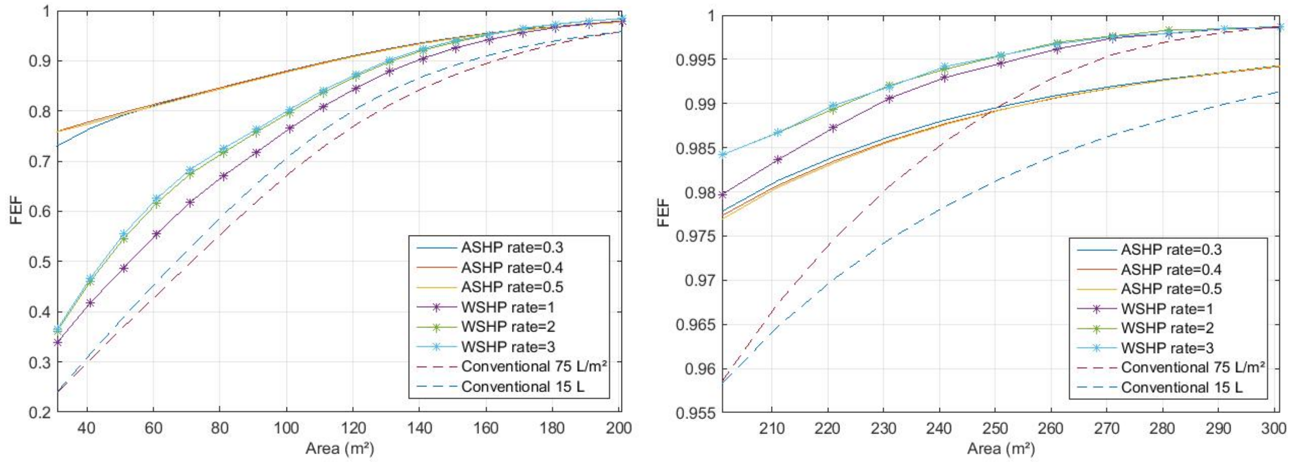

Figure 5. Free energy fraction results for the entire solar collector area range (left), and for portion between 200-300 m² (right).

ASHP requires a minimum solar collector area to operate due to the temperature gap between the main water stream (20°C) and the minimum inlet temperature at ASHP load side (26°C). With 30 m², solar energy collected is enough to raise water temperature from 20 to 26°C, allowing the ASHP to operate. For that reason, ASHP performs similarly to the both conventional systems up to 30 m². Thereafter heat pump is allowed to operate and the system performance changes abruptly, as shown in Fig. 5. ASHP rate has little influence on system's performance. For 50 m² of solar collectors, the smallest capacity ASHP provides as good results as the others do.

Despite higher heat pump seasonal performance factors (Fig. 6, left), WSHP system overall performance is lower than the ASHP system for most of the range where systems still operates as combined systems, i.e. below approximately 180 m² (Fig. 6, right). ASHP operating temperature ranges are larger than WSHP ones for both source and load sides. Moreover, the WSHP source side operating range is further reduced from -6 to 2°C due to water freezing occurrence, cutting the entire range in about 25%. One of the main goals of introducing the heat pump into solar system is to take advantage of its COP. Since ASHP is allowed to operate within larger ranges, taking advantage of the heat pump COP, thus the system requires less auxiliary heat, improving its performance. When the WSHP is not allowed to operate due to limiting ranges, the auxiliary heater is enabled instead, harming the system overall performance. As mention before, the range can be fully explored if a fluid with lower freezing point is used. However, initial system cost would be higher and a more complex control strategy would be required.

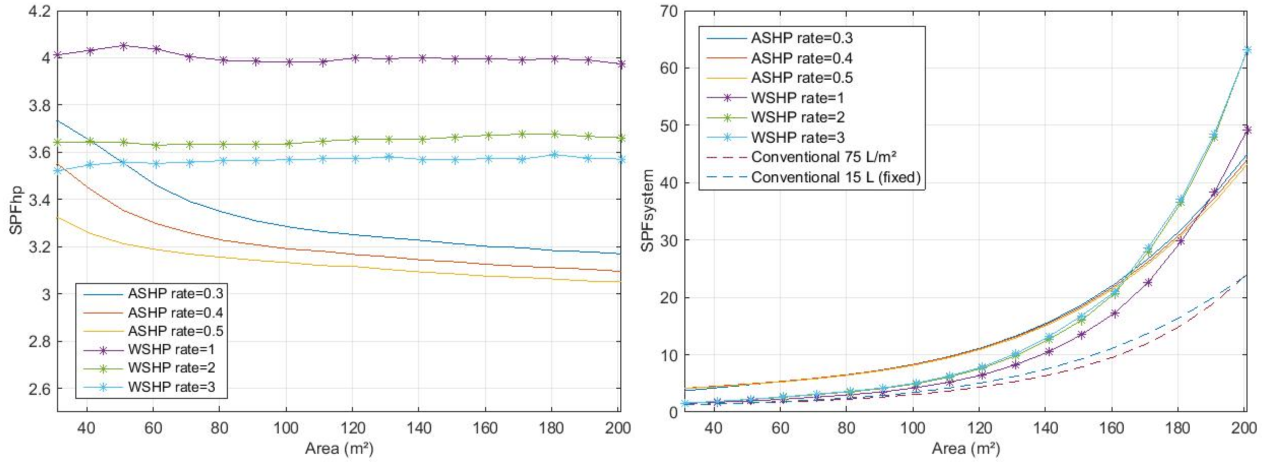

Figure 6. Seasonal performance factor of the heat pump (left) and the system (right).

Table 5 details the results for a fixed 100 m² of solar collector area. Both combined systems present better overall performance indicators in comparison to the conventional systems. Finally, two combined systems with approximately equivalent "sizes" are compared, i.e. the ASPH system with a 17.44 kW resulting capacity heat pump of and the WSHP system with a 18.16 kW resulting capacity heat pump, both with two storage tanks of 15 m³ and 100 m² of solar collectors. Results show the ASHP system is capable of proving 10% higher FEF, which means this system has 10% lower operating costs in comparison with the WSHP system with the same “size”. Moreover, the seasonal performance value of the ASHP system is 70% higher than the equivalent WSHP system.

Table 5. Simulation results for a fixed collector area of 100 m^2.

System	Storage tank 1 volume [m³]	Storage tank 2 volume [m³]	Heat pump rate	Resulting heat pump nominal heat capacity [kW]	SPF_{system}	SPF_{hp}	SF	FEF
Optimal Conventional	7.5	-	-	-	3.09	-	0.68	0.676
Conventional	15	-	-	-	3.48	-	0.71	0.711
ASHP system	15	15	0.3	13.08	8.42	3.28	0.625	0.881
			0.4	17.44	8.37	3.19	0.623	0.880
			0.5	21.80	8.25	3.13	0.622	0.879
WSHP system	15	15	1	9.08	4.26	3.98	0.640	0.765
			2	18.16	4.93	3.64	0.632	0.797
			3	27.24	5.08	3.57	0.633	0.803

6. CONCLUSIONS

Results show that the introduction of a heat pump into the traditional solar system design indeed allows the systems to improve its performance. Given 100 m² of fixed collector area, in comparison with the Conventional system with fixed 15 m³ storage volume, the WSHP system is able to further reduce the operating costs (based on the FEF values) in 13%, and the ASHP system in 24%. Moreover, in comparison with an electric heating system, ASHP system is able to further reduce the operating costs by a minimum of about 70%, when operating with virtually no solar input, case wherein the ASHP provides all the energy required by the user.

It was found that the temperature control harms the overall performance of the WSHP system. The reason is mainly twofold. First, operating temperature ranges of the WSHP are smaller than the ASHP ones for both source and load sides. Second, the WSHP source side operating range is further reduced due to water loop freezing occurrence within the collector loop and the storage tank 1. Since the ASHP is allowed to operate at wider temperature ranges, it can operate for longer periods, which could justify the ASHP system better performance for a given collector area and storage volumes. As an alternative for the system WSHP, a collector fluid with lower freezing temperature such as a glycol-water mixture could be used, enabling the heat pump to operate on the entirely source temperature range. In this case, since the initial costs are expected to be higher, the potential further energy savings must compensate for it. This situation will be evaluated in further simulations.

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