Field tests of a novel solar-assisted dual source multifunctional heat pump

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Abstract

In this paper, we present the results of field study concerning a novel solar-assisted dual-source multifunctional heat pump, installed in a detached house in Milan. The system couples hybrid photovoltaic/thermal (*PVT*) panels with multifunctional and reversible heat pump. The proposed system is equipped with an "*air source*" evaporator and a "*water source"* evaporator, connected in series and operated alternatively, based on the ambient conditions and system parameters. The "*air source*" evaporator is an external unit; conversely, the "*water source"* evaporator is connected with a storage tank, fed by the *PVT* system. The *PVT* system is connected with the heat pump by two storage tanks to be used to produce domestic hot water and to be used in "*water source"* evaporator. Based on the operating conditions, the hot water is sent to one of the storage tank. The proposed system has been tested experimentally; the results show that the system was able to maintain high efficiency in the different seasons and was able to produce domestic hot water. It was found that the use of the "*water source"* evaporator was able to compensate the performance degradation of the "*air source"* source evaporator caused by the low ambient temperature.

Keywords: Solar assisted heat pump, Hybrid photovoltaic/thermal panels, Dual Source, Domestic hot water, PVT, SAHP

1. Introduction

Multifunctional heat pumps are widely used for heating and cooling applications, as well as for the production of domestic hot water (*DHW*), by exploiting different heat sources (i.e., ground source, air source or water source). In particular, the coefficient of performance (*COP*) of air-source heat pumps depends on the ambient temperature, T_{amb} (viz. T_{amb} is related to the evaporating temperature, T_{eva}). As a consequence, in the winter season, when heating is needed and T_{amb} is lower, the system performance decreases, owing to the lower evaporating temperature. Conversely, considering a reversible heat pump in the summer season, when cooling is needed and T_{amb} is higher, the system performance decreases, owing to the higher condensing temperature. In this respect, the coupling between solar technologies and heat pumps—"*solar-assisted heat pumps*" (*SAHPs*)—is a promising technology to overcome the above-mentioned limitations, to reduce the consumption of energy resources and to meet the targets set by the recent regulations (i.e., European Union targets). Generally speaking, heat pumps can be coupled with photovoltaic (*PV*) panels, thermal collectors or hybrid photovoltaic/thermal (*PVT*) panels, as outlined in the literature surveys proposed by Kamel et al. (Kamel, Fung e Dash 2015) and by Mohantaj et al. (Mohanraj, et al. 2017). In practical applications, heat pumps are coupled with thermal or *PVT* panels, in direct or indirect expansion configurations. In direct expansion *SAHP*, the solar panel corresponds to the evaporator of heat pump; conversely, in indirect expansion *SAHP*, an intermediate heat exchanger is used to couple the solar system and the heat pump. In addition, *SAHP* systems can be classified into three sub-categories: (a) parallel systems, (b) series systems and (c) dual-source systems. In parallel systems, the heat pump receives energy from the ambient, and the solar energy is supplied directly for either space heating or for *DHW* production. In series systems, solar energy is supplied to the evaporator of the heat pump, thus raising the evaporating temperature (thus, increasing the *COP*) and cooling the solar collectors (thus, increasing the efficiency of the solar panels). In the dual-source systems, the evaporator can receive energy from either the atmosphere or from the solar energy, depending on the ambient conditions and system operation .

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This paper contributes to the existing discussion concerning multifunctional *SAHP* for heating and cooling applications, as well as for the production of *DHW*. In the last decades, different systems have been proposed and, in the following, a brief literature survey is proposed to better outline the framework of this research. Wang et al. (Wang, et al. 2011) experimentally investigated, in a laboratory-scale experimental setup, an indirect dual-source ("*air-source*" and "*water-source*" evaporators) *SAHP* for space heating and cooling and water heating. A storage tank, connected to thermal panels, was used to supply heat to the "*water-source*" evaporator or to produce *DHW*. They reported *COP* = 4, in the heating mode. Bridgeman and Harrison (Bridgeman e Harrison 2008) experimentally investigated, in a laboratory-scale experimental setup, the performance of an indirect series *SAHP* for water heating. They observed *COP* = 2.8 – 3.3, depending on the evaporator and condenser temperatures. Loose et al. (Loose, et al. 2011) performed field tests of various combined *SAHP* systems with different heat sources, for space heating and water heating. The system employed solar thermal collectors and geothermal heat pump with borehole heat exchangers. The collectors fed the storage tank directly when the sufficient solar radiation was available. Otherwise, low grade energy from the collectors would be used in the heat pump for space heating. Bakirci et al. (Bakirci e Yuksel 2011) investigated the performance of an indirect SAHP system for space heating. The solar collectors directly charged a storage tank which was linked to an evaporator to provide a heat source for the heat pump. They observed *COP* in the range of 3.3 – 3.8. Bai et al. (Bai, et al. 2012) theoretically studied, by using a TRNSYS model, an indirect combined hybrid *PVT SAHP* system for *DHW* production. Year round performance results were simulated under the different climatic conditions (Hong Kong and different locations in France) and an average *COP* = 4.9 has been observed.

From the above-mentioned literature survey, as well as from the literature surveys proposed by different authors (Hepbasli e Kalinci 2009, Ozgener e Hepbasli 2007, Parida, Iniyan e Goic 2011, Tian e Zhao 2013), it is observed that there is a lack of field studies concerning practical demonstration and fields studies concerning multifunctional dual source ("*air-source*" and "*water-source*" evaporators) *SAHP*. To this end, this paper presents the field tests of a novel solar-assisted dual-source multifunctional heat pump, installed in a detached house in Milan. The system couples hybrid *PVT* panels with a multifunctional and reversible heat pump. The paper is organized as follows. First, the experimental setup and methods are presented and described; second, the experimental results are commented and discussed; finally, main conclusions are drawn

2. Experimental setup and methods

In this section, the experimental setup and the experimental methods are presented and described. First, the multifunctional heat pump system and main characteristic of the components are presented and discussed (Section 2.1). Second, the details concerning the operation procedures are outlined (Section 2.2). Finally, the experimental techniques (Section 2.3) and performance parameters (Section 2.4) are described and commented.

2.1 Experimental setup

The multifunctional heat pump system (Fig. 1) has been designed and installed in a detached house (Fig. 2) located in Milan, at RSE Spa headquarter. The detached house has an heating/cooling area equal to 64 m^2, with a nominal load equal to 4 kW_{th} (indoor temperature equal to 20 °C and outdoor temperature equal to -5 °C). Fig. 1 displays the layout of the multifunctional heat pump, which is composed by five parts: (a) a solar system, (b) a *DHW* storage tank and an "*intermediate-temperature*" storage tank, (c) a reversible heat pump, (d) terminal heating and cooling systems (viz. fan coils) and (e) circulating pumps (Tab. 1). Further details concerning the different components of the system are provided in the following.

Solar system. The solar system consists in seven *PVT* panels and a *PV* panel (Fig. 2). The *PV* panel has the same size and cells characteristic of the *PVT* panels, to compare the performance of the two technologies. The seven *PVT* panels (south-oriented, 45° titled angle) have 1.75 kW_{el} nominal power and are of the roll-bond technology (Fig. 3a). They are composed by polycrystalline silicon cells, a steel heat-exchanger and a 0.002 m insulation. In the heat exchanger part of the *PVT*, water/ethylene glycol mixture has been used as working fluid, to prevent icing-related issues, with a nominal flow rate equal to 0.630 m^3/h. It is worth noting that the comparison between *PV* and *PVT* panels is not presented here and is a matter of ongoing research activities.

Storage tanks. The *DHW* storage tank (0.186 m^3 in volume) has been used to produce hot water; conversely, the "*intermediate-temperature*" storage tank (0.300 m^3 in volume, Fig. 3b) has been used as "*water-source*" from the heat pump, instead of the "*air-source*" one, to improve the performance in cold days and avoid defrosts and reduce temperature fluctuations. Inside the "*intermediate-temperature*" storage tank water/ethylene glycol mixture has been used as working fluid, to prevent icing-related issues. The "*intermediate-temperature*" is connected to the heat pump, by a plate heat exchanger (Fig. 3c) and the maximum flow rate "*intermediate-temperature*" towards the "*water-source*" heat exchanger is 1.45 m^3/h.

Heat pump unit. The heat pump is a R410A reversible heat pump, 7 kW_{th} nominal heating capacity and 3.47 nominal COP (water leaving temperature equal to 35 °C and air temperature equal to 7 °C), equipped with an electronic expansion valve and a variable speed compressor.

Circulating pumps. As shown in Fig. 1, the experimental system includes circulating loops: (a) solar collector loops; (b) "*intermediate-temperature*" storage tank to heat pump; (c) indoor terminal fan-coil side loop and (d) *DHW* circulating loops. Details on pumps used are provided in Table 1.

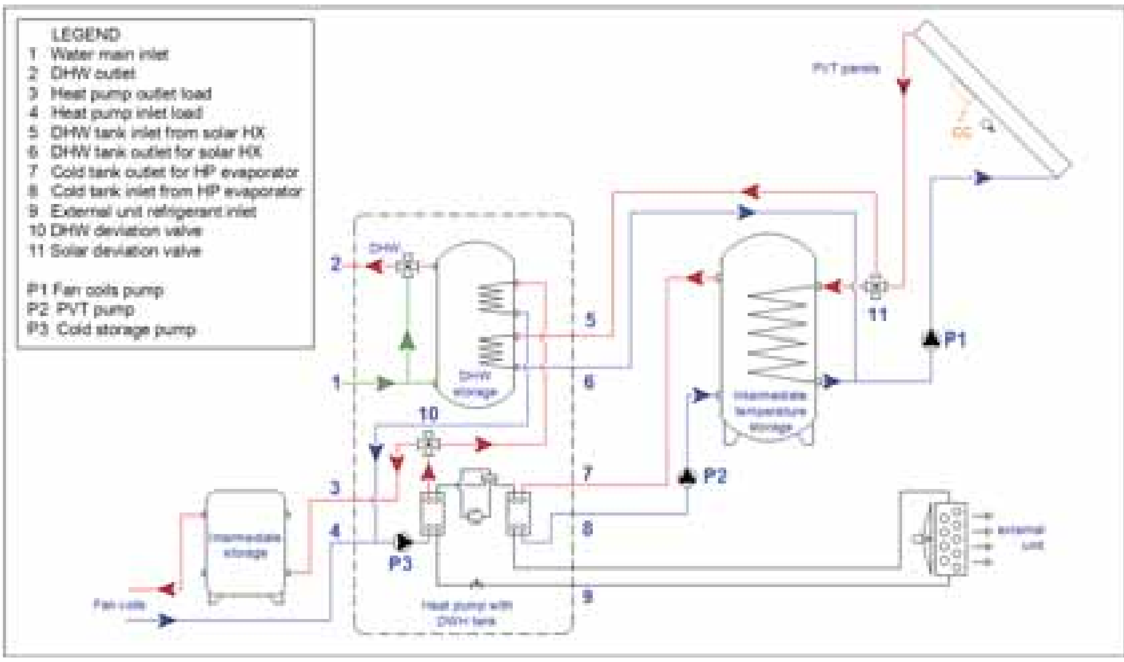


Fig. 1: Experimental layout

Fig. 2: Photo of the detached house at RSE Spa

(a) PVT roll-bond panels

(b) "*intermediate-temperature*" tank

(c) Plate heat exchanger between the heat ump and the "intermediate-temperature" tank

Fig. 3: Details of the experimental setup

Tab. 1: Details on circulating pumps used (see Fig. 1 for further details)

Code (Fig.1)	Location	Technical details	Power [W]
P1	Solar circulating pump	DEB Evolplus Small 60/180 M PWM	5 - 100
P2	"*intermediate temperature*" circulating pump	WILO Stratos Para 25/1-7 PWM	5 - 70
P3	Fan-coil circulating pump	WILO Stratos TEC RS 25/7 PWM	3 - 70
P4	*DHW* circulating pump	WILO ZRS 12/7	86 (fixed speed)

2.2 Operation modes and control

The multifunctional heat pump has been tested starting from 17th January 2017 and the monitoring is still ongoing. The system has been tested in three operation modes in order to study the influence of the different operation parameters; each mode was controlled aiming to (a) minimize the electricity consumption by the heat pump, using solar thermal energy, and (b) maximizing the production of electricity of *PVT* panels keeping their operating temperature as low as possible. The operating principles for each mode are provided in the following.

- **Mode#1. Heating mode without *DHW* production (from 17/01/2017 to 13/03/2017).** The internal set-point temperature was set at $T_{set\text{-}point}$ = 22 °C, with night attenuation of 4 °C (from 22.00 to 7.30). In this operation mode, the temperature of the water produced by the heat pump and sent to the fan-coils was set by using a climatic curve: T_3 = 43 °C with T_{amb} = 0 °C and T_3 = 35 °C with T_{amb} = 15 °C. In this operation mode, valve *V1* was set to deviate all thermal energy produced by the *PVT* panels to the "*intermediate-temperature*" storage tank. The "*intermediate-temperature*" storage tank, is used as the "*water-source*" for the heat pump when T_{amb}, is low and the resulting *COP* would decrease. Changes from "*air-source*" evaporator to "*water-source*" evaporator are obtained by switching on/off the pump *P2* and switching on/off the external "*air-source*" unit.

- **Mode#2. Heating mode with *DHW* production (from 13/03/2017 to 24/05/2017).** The operation mode *mode#1* has been modified as follows: (a) a daily profile of *DHW* production was set to produce 1*50 l* (corresponding to, approximately, 4 KWh); (b) valve *V1* was set to deviate the glycol-water mixture, at the outlet of the *PVT* panels, depending on the temperature of the storage tanks (a) towards the "*intermediate-temperature*" *DHW* storage ($T_{intermediate\text{-}temperature}$ < 38 °C), (b) towards the *DHW* tank ($T_{intermediate\text{-}temperature}$ ≥ 38 °C) or towards the "*intermediate-temperature*" storage tank ($T_{DHW,tank}$ ≥ 58 °C); (c) the *DHW* storage tank set-point temperature was set to $T_{DHW,set\text{-}point}$ = 48°C and its lower temperature has been set $T_{DHW,mantenance}$ = 42°C. In the case, the temperature of the *DHW* storage tank would fall below $T_{DHW,mantenance}$, the heat pump would be used been used to increase the *DHW* tank temperature.
- **Mode#3. Cooling mode with *DHW* production (from 24/05/2017, ongoing).** The internal set-point temperature was set at $T_{set\text{-}point}$ = 24 °C. In addition, the operation mode *mode#2* was modified as follows: (a) a daily profile of *DHW* production has been set to produce 1*50 l* (corresponding to approximately, 3 KWh; the DHW corresponding power is lower compared with the previous operation mode, owing to the higher inlet water temperature, T_1); (b) valve *V1* was set to deviate the glycol-water mixture, at the outlet of the *PVT* panels, depending on the temperature of the storage tanks towards the *DHW* storage tank ($T_{intermediate\text{-}temperature}$ ≥ 36 °C) or towards the "*intermediate-temperature*" storage tank ($T_{DHW,tank}$ ≥ 57 °C);).

2.3 Measurement system and procedure

All the main variables, to describe mass and energy balances, have been measured. The flow rate in each circuit has been measured by an electromagnetic flowmeter meter (E&H Promag P50, ±0.2% read value). All the inlet and outlet temperatures of the main equipments, the supply and return water temperatures of the different locations, were measured by RTD Pt100 4wire 1/5DIN, inserted inside the pipes. The indoor and outdoor (near the *PVT* panels) temperature and humidity were measured by an Pt100 4wire hygrometer (Siap+Micros). The solar radiation intensity has been measured by a thermopile pyranometers (pyranometer Kipp&Zonen CMP11), mounted at a 45° inclined angle near the *PVT* panels. The power consumption of the heat pump and the circulating pumps (solar pumps, intermediate-storage tank, fan-coil pump) were measured by multifunction electric meters (Shark 100, ±0.1%, and FRER MonoNano, ±0.5%). Evaporating and condenser pressures were measured by pressure transducers (Keller series 21Y). All the temperature probes were verified with a calibration procedure by using thermostatic batch, at RSE Spa. All data were recorded automatically at every 6 seconds interval in a data logger (Advantech ADAM 5000 and 4000 data logging devices) to be further post-processed.

2.4 Performance evaluation

The performance of the system has been evaluated based on the mass and energy balances, based on the recorded data (flow rates and temperatures). In particular, the energy fluxes across every component has been computed as follows:

$$Q = \dot{m} c_p (T_{inlet} - T_{outlet}) \quad \text{(Eq. 1)}$$

In Eq. (1), T_{inlet} and T_{outlet} refer to the inlet and outlet temperatures, m is the mass flow rate, c_p is the specific heat of water. Based on the heat fluxes and the electric power measured, the *COP* (during heating mode; *mode#1* and *mode#2*, Eq. (2)) and the *EER* (during the cooling mode; *mode#3*, Eq. (3)) have been computed as follows:

$$COP = \frac{Q_{HP \rightarrow fan-coil} + Q_{HP \rightarrow DHW-tank}}{P_{el}} \quad \text{(Eq. 2)}$$

$$EER = \frac{Q_{HP \leftarrow fan-coil} + Q_{HP \rightarrow DHW-tank}}{P_{el}} \quad \text{(Eq. 3)}$$

In Eqs. (2-3), P_{el} is the electric power provided to the systems; $Q_{HP\rightarrow fan\text{-}coil}$ and $Q_{HP\leftarrow fan\text{-}coil}$ is computed based on T_3 and T_4 and $m_3 = m_4$ (please refer to Fig. 1 for the location of the subscripts) and refer to the heat transfer from the heat pump towards the fan-coils and vice-versa; conversely, $Q_{HP\rightarrow DHW\text{-}tank}$ refers to the het flux provided from the heat pump to the *DHW* tank and is computed as follows:

$$Q_{HP\rightarrow DHW-tank} = \sum_{t=0}^{n=N} \rho V_{DHW,tank} c_p (T_{t=n} - T_{t=n-1}) \qquad \text{(Eq. 4)}$$

In Eq. (4), $V_{DHW,tank}$ is the volume of the *DHW* tank, ρ is the density of water, and *t* is the time variable. Please note that Eq. (4) is computed under the following constrains: (a) the temperature of the *DHW* storage is below $T_{DHW,maintenance}$, (b) the temperature of the *DHW* storage tank is increasing with time; (c) heat pump status is ON.

To study the influence of the "*water source*" evaporator and the "*air-source*" evaporator, Eq. (2) has been modified as follows, based on the status of the storage tank circulating pump (pump *P2*, Fig. 1):

$$COP_{\text{"water-source"}} = \frac{Q_{HP\rightarrow fan-coil} + Q_{HP\rightarrow DHW-tank}}{P_{el}} \quad if\ pump\ P2 = ON \qquad \text{(Eq. 5)}$$

$$COP_{\text{"air-source"}} = \frac{Q_{HP\rightarrow fan-coil} + Q_{HP\rightarrow DHW-tank}}{P_{el}} \quad if\ pump\ P2 = OFF \qquad \text{(Eq. 6)}$$

3. Experimental results

In this section, the experimental results are presented and discussed. First, the ambient conditions in the monitored period are presented. Second, the performance of the heat pump are presented and commented with reference to the three different operating modes. Finally, the influence of the "*water-source*" and the "*air-source*" evaporators on the performance are commented.

3.1 Ambient conditions

In order to provide an overview of the heat pump working conditions, Fig. 4 displays the value of ambient conditions in the monitored period. In particular, Fig. 4 displays the daily averaged values of the ambient temperature and relative humidity. It is worth noting that the heat pump was operated in a quite broad range of operating conditions (i.e., daily averaged T_{amb} ranged between 0 and 33 °C; conversely, the instantaneous values of T_{amb} ranged between -5°C and 40 °C).

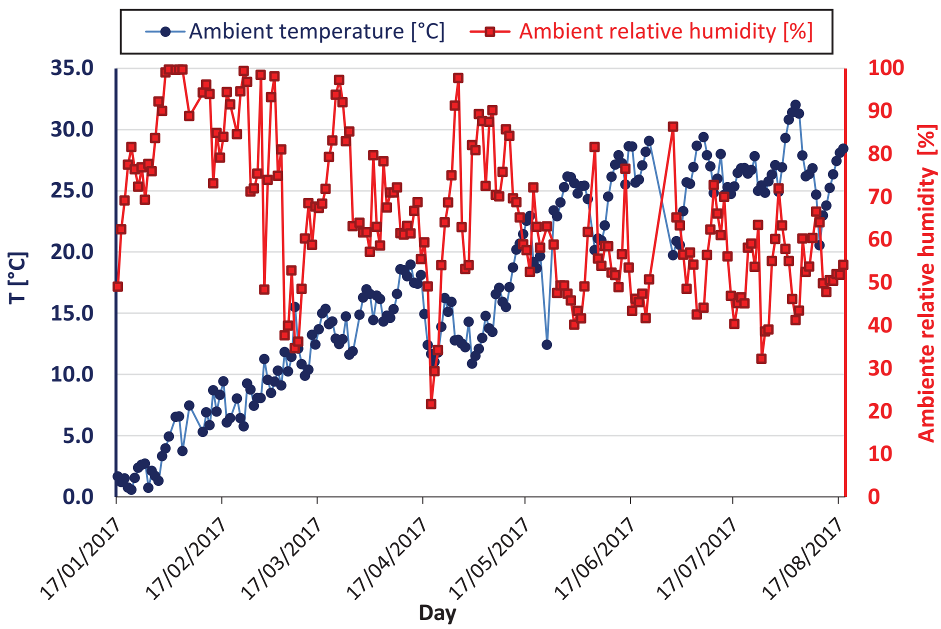


Fig. 4: Daily-averaged ambient conditions: ambient temperature and relative humidity

3.2 Seasonal performance

Fig. 5 displays the daily-averaged performance of the multifunctional heat pump in the three operating modes described in Section 2.2. In addition, in Fig. 5, T_{amb} is displayed for the sake of clarity. The performance of the multifunctional heat pump are computed by using Eq. (2)—the *COP*—and Eq. (3)—the *EER*. In the operation *mode#1*, the useful effect of the heat pump is $Q_{HP\rightarrow fan\text{-}coil}$ (there is no *DHW* production; $Q_{HP\rightarrow DWH\text{-}tank} = 0$); conversely, in the other operation modes (viz. *mode#2* and *mode#3*) the useful effect of the heat pump consists in both $Q_{HP\rightarrow fan\text{-}coil}$ and $Q_{HP\rightarrow DWH\text{-}tank}$. Please note that the electric consumption, P_{el}, considered in the evaluation of *COP* and *EER* does not account for the contribution of the auxiliaries (i.e., circulation pumps, stand-by consumption, etc…): (a) the solar circulating pump (its consumption was approximately 4.1 % of the total energy consumption); (b) the "*intermediate-temperature*" circulating pump (its consumption was approximately 1.2 % of the total energy consumption); (c) the Fan-coil circulating pump (its consumption was approximately 3.7 % of the total energy consumption), (d) the DHW circulating pump and all other auxiliaries (their consumptions were approximately 14.2 % of the total energy consumption).

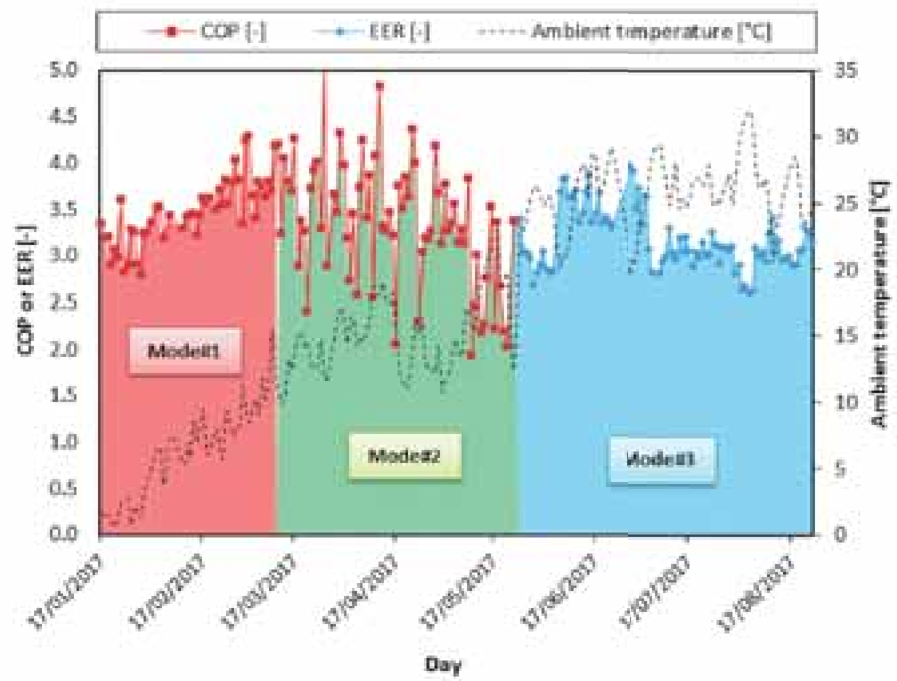


Fig. 5: Daily-averaged performance of the multifunctional heat pump depending on the operation mode and ambient temperature

In the operation mode *mode#1*, the daily-averaged *COP* is in the range of 3 and 4.5; as expected, *COP* increases with time, owing to the increasing daily-averaged T_{amb}. The relationship between T_{amb} and the heat pump performance can be further understood, by considering the layout of the system. Indeed, the performance of an heat pump may be related (for fixed component design), to the evaporating and condensing temperatures/pressures. In the present case: (a) the condensing pressure is related to the internal conditions, that, for a fixed set-point are periodical with time; (b) evaporating temperature T_{eva} is related to T_{amb}, owing to the variable speed compressor and the electronic expansion valve. Therefore, a variation in the ambient temperature, mainly affects T_{eva} and, thus, affect the performance of the system. To better discuss this concept, Fig. 6 displays the relationship between the variables of the heat pump (i.e., evaporator pressure and condensing pressure), T_{eva} and *COP*. The reader may refer to the studies proposed by Kuang and Wang (Kuang e Wang 2006) and Ma and Zhao (Ma e Zhao 2008) for a more detailed discussion on the role of variable speed compressors in heat pumps. In addition, in Fig. 6 is also displayed the influence of the "*water-source*" evaporator. This point is further discussed in Section 3.3. In the operation mode *mode#2*, the performance of the system, compared with *mode#1*, shows a larger variability and are slightly lower. This behavior can be explained based on the system operations as well as on the ambient conditions. First, in this period, beside DHW production, $Q_{HP\rightarrow fan\text{-}coil}$ is very low and, in some days, $Q_{HP\rightarrow fan\text{-}coil} \approx 0$, owing to the high T_{amb} (the internal set-point temperature can be achieved also with very low heat pump load). Therefore, *COP* is mostly related to $Q_{DHW\text{-}tank}$. Second, it should be noted that

$Q_{HP \rightarrow DHW\text{-}tank}$, is produced at higher temperature compared with $Q_{HP \rightarrow fan\text{-}coil}$; therefore the expected performance of the heat pump is reduced. In this respect it is well known that Eq. (2) does not take into account the grade of heat produced, as it is related to energy balances and neglect the entropy/exergy concept. In the operation mode *mode#3*, in the cooling mode with DHW production, the daily-averaged *EER* is in the range of 3 and 4.5. The discussion concerning the relationship between the ambient conditions and the system performance is similar to the above-discussion for the operation mode *mode#1*. It is worth noting that, owing to the high ambient temperature, in the summer season, the *PVT* panels are able to contribute to the maintenance temperature of the *DHW* storage tank, thus reducing $Q_{HP \rightarrow DHW\text{-}tank}$ and, thus, the energy consumption of the heat pump.

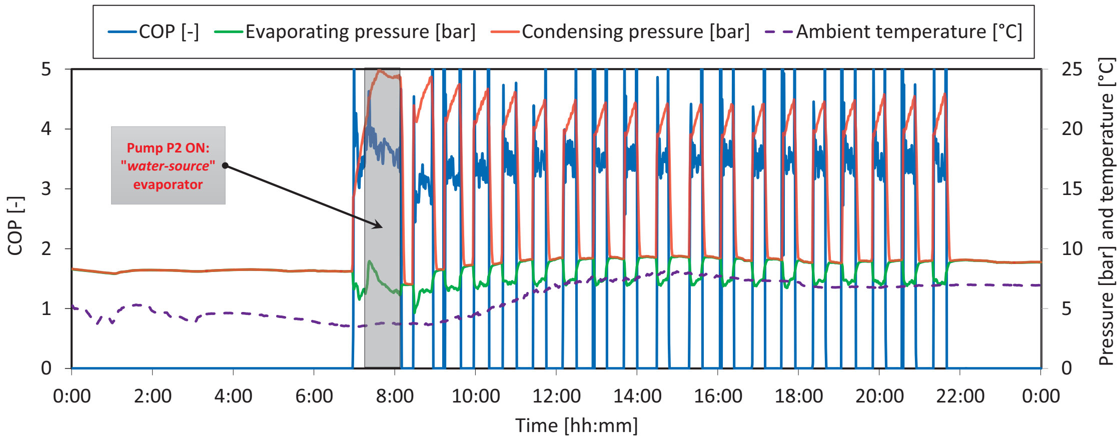


Fig. 6: Daily performance of the multifunctional heat pump: performance parameters (data obtained at 13/02/2017, *mode#1*)

3.3 Comparison between water source and air source

In this section, an insight in the performance of the system is proposed, by analyzing the contribution of the "*water-source*" evaporator and the "*air-source*" evaporator to achieve the averaged performance described in Section 3.2. To this end, by applying Eqs. (5-6) in the in the operation mode *mode#1*, the corresponding "*water-source*" the "*air-source*" performances have been obtained; the results of this analysis have been summarized in Fig. 7, in terms of daily-averaged values. Please note that in some days (viz. the data not displayed in Fig. 7), the "*water-source*" evaporator was not used. Applying the whole dataset, (in the operation mode#1) and applying Eqs. (5-6), an average *COP* increase approximately 35.5% from the "*air-source*" mode to the "*water-source*" mode has been observed. In order to better understand the relationship between "*water-source*"/"*air-source*" evaporators and the COP, Fig. 8 proposes three daily profiles of the heat pump operation.

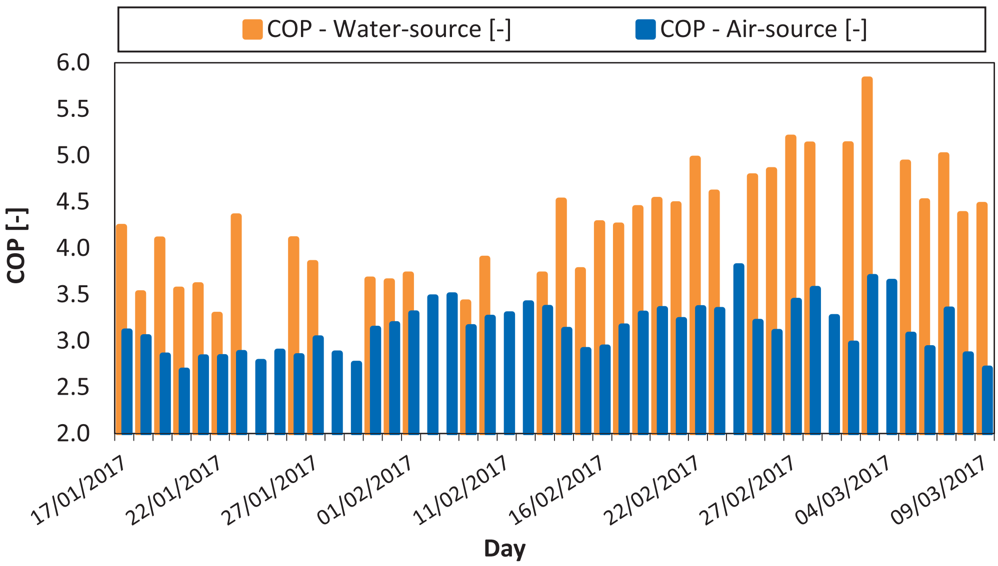
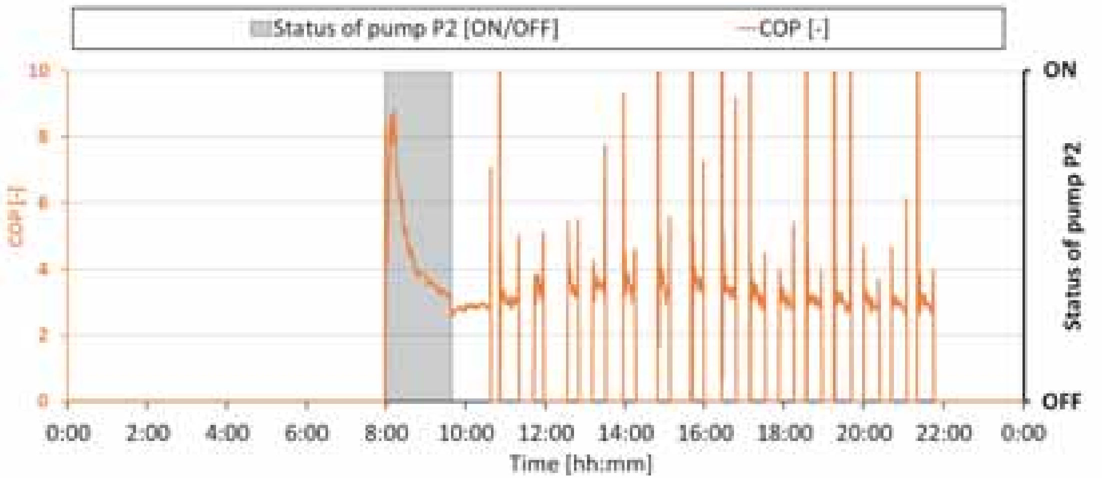
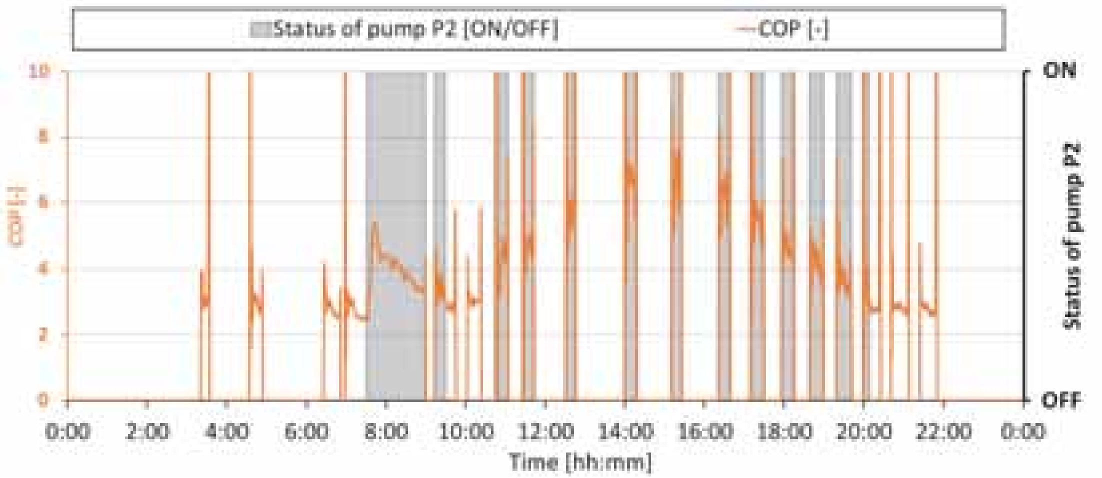


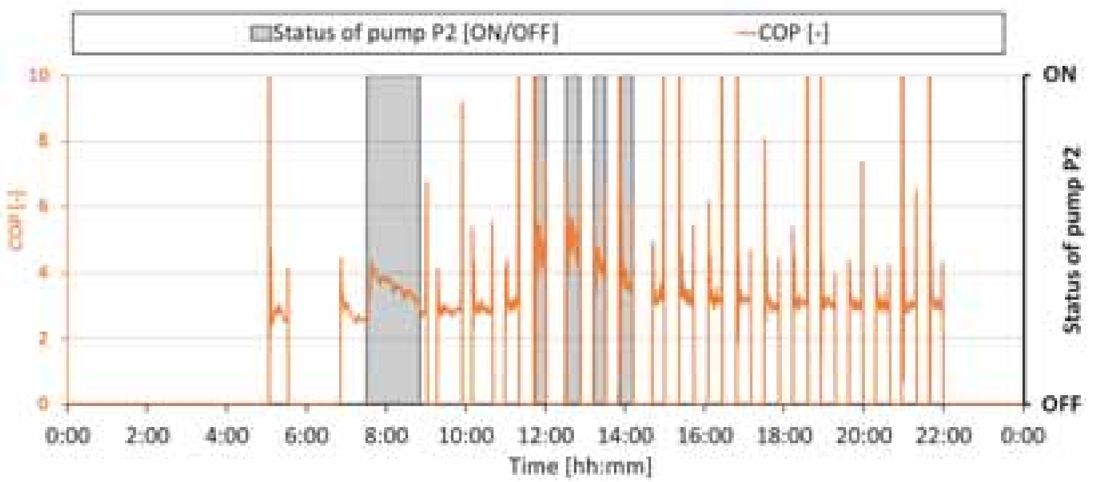
Fig. 7: Daily-averaged performance of the multifunctional heat pump: influence of "*water-source*" and "*air-source*" evaporators (data obtained during operation mode *mode#1*)



(a) data obtained at 17/01/2017, operation mode *mode#1*



(b) data obtained at 23/01/2017, operation mode *mode#1*



(c) data obtained at 27/01/2017, operation mode *mode#1*

Fig. 8: Daily performance of the multifunctional heat pump: influence of "*water-source*" and "*air-source*" evaporators

Fig. 8 clearly displays the significant increase of *COP*, owing to the use of the "*water-source*" evaporator. For example, it can be observed that, in Fig. 8a, by activating the pump *P2* at 8.00 AM the *COP* increases from 2.7 to 8 (instantaneous values). Similar considerations can be drawn for the other days displayed in Fig. 8b and Fig. 8c. It is important to emphasize that the "*water-source*" evaporator avoided the defrost cycles of the aerothermal heat pumps and the subsequent decrease in the performances. In this respect, the reader may refer to the defrost-related issues described by Zhang et al. (Zhang, et al. 2017). It should be noted that the *COP* obtained using water as source has been obtained in the most unfavorable conditions: the morning, when maximum building load and with the highest delivery water temperature. Unfortunately, the use of the water source is limited in time and depends on the thermal energy available in the "*intermediate-temperature*" storage tank. Future studies should be focused on alternative technical solutions for storage tanks (i.e., phase change material, *PCM*, storage tanks), to improve the availability of the "*water-source*".

4. Conclusions, outcomes and outlooks

This paper presented the very first results of field study concerning a novel solar-assisted dual-source multifunctional heat, installed in a detached house in Milan. The system couples *PVT* panels with a dual-source ("*air source*" and "*water source*" evaporators) multifunctional and reversible heat pump. The multifunctional heat pump has been tested starting from 17th January 2017 (the monitoring is still ongoing) in three operation modes: (a) heating mode without *DHW* production; (b) heating mode with *DHW* (150 l) production; (c) cooling mode with *DHW* (150 l) production.

4.1 Conclusions and outcomes

The main results of the filed study are as follows:

- the averaged performance (*COP* and *EER*) of the multifunctional heat pump is approximately 3, in the three different operation modes;
- the thermal energy produced by the *PVT* panels have been successfully used to support the "water-source" evaporator in the winter/spring seasons;
- the thermal energy produced by the *PVT* panels have been successfully used to support the production of *DHW* in the summer period;
- the use of "*water-source*" evaporator allows to significantly increase the performance of the plant;
- the use of "*water-source*" evaporator allows to avoid defrost cycles;

4.2 Outlooks

Ongoing research as well as future studies may concern both experimental and numerical researches. On the experimental part, outlooks are as follows:

- extent the field study to the autumn ambient conditions, to provide a complete seasonal overview of the system operation;
- relate the daily and the seasonal operations, to provide a multi-scale evaluation of the heat pump performance;
- propose a comprehensive thermodynamic evaluation (energy and exergy evaluation) of the whole system;
- compare the performance of PV and PVT systems and provide insights in the relationships between operating conditions (i.e., temperatures, flow rates, ambient conditions) and the panels performances (i.e., thermal and electrical energy production);
- study alternative technical solutions for storage tanks (i.e., phase change material, *PCM*, storage tanks), to improve the availability of the "*water-source*".

On the numerical part, future studies would concern the testing and validation of a TRNSYS approach, to extend the results of the present experimental study to different climatic conditions and, finally, to assess the economic feasibility of the proposed system.

5. Nomenclature and abbreviation list

5.1 Acronyms

COP	Coefficient of performance
DHW	Domestic hot water
EER	Energy Efficiency Ratio
HP	Heat Pump
PV	Photovoltaic
PCM	Phase Change Material
PVT	Hybrid thermal-photovoltaic
SAHP	Solar-assisted-heat pump

5.2 Symbols

c_p	Specific heat of water	[kJ/kg K]
m	Mass flow rate	[kg/s]
N	Time discretization in Eq. (4)	[-]
$Q_{HP\rightarrow DWH\text{-}tank}$	Heat transfer from the heat pump to the *DHW* tank	[kW]
$Q_{HP\leftarrow fan\text{-}coil}$	Heat transfer from the heat pump to the fan-coils	[kW]
T	Temperature	[°C]
t	Time in Eq. (4)	[min]
$V_{DHW,tank}$	Volume of the *DHW* storage tank	[m^3]
$V_{DHW,tank}$	Volume of the *"intermediate-temperature"* storage tank	[m^3]
ρ	Density of water	[m^3/kg]

5.3 Subscripts

amb	Ambient conditions
el	Electrical energy
eva	Evaporator
inlet	Inlet condition
outlet	Outlet condition
th	Thermal energy

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7. References

Bai, Y., T. T. Chow, C. Ménézo, and P. Dupeyrat. "Analysis of a Hybrid PV/Thermal Solar-Assisted Heat Pump System for Sports Center Water Heating Application." *International Journal of Photoenergy* 2012 (2012).

Bakirci, Kadir, and Bedri Yuksel. "Experimental thermal performance of a solar source heat-pump system for residential heating in cold climate region." *Applied Thermal Engineering* 31 (2011): 1508-1518.

Bridgeman, Andrew, and S. Harrison. "Preliminary experimental evaluations of indirect solar assisted heat pump systems." *Proceedings of the 3rd Canadian solar building conference. Fredericton.* 2008.

Hepbasli, Arif, and Yildiz Kalinci. "A review of heat pump water heating systems." *Renewable and Sustainable Energy Reviews* 13 (2009): 1211-1229.

Kamel, Raghad S., Alan S. Fung, and Peter R. H. Dash. "Solar systems and their integration with heat pumps: A review." 87 (2015): 395-412.

Kuang, Y. H., and R. Z. Wang. "Performance of a multi-functional direct-expansion solar assisted heat pump system." *Solar Energy* 80 (2006): 795-803.

Loose, Anja, Harald Drück, Nadine Hanke, and Frank Thole. "Field Test and Performance Monitoring of Combined Solar Thermal and Heat Pump Systems." *ISES Solar World Congress, At Kassel, Germany.* 2011.

Ma, Guo-Yuan, and Hui-Xia Zhao. "Experimental study of a heat pump system with flash-tank coupled with scroll compressor." *Energy and Buildings* 40 (2008): 697-701.

Mohanraj, M., Ye. Belyayev, S. Jayaraj, and A. Kaltayev. "Research and developments on solar assisted compression heat pump systems – A comprehensive review (Part A: Modeling and modifications) ." *Renewable and Sustainable Energy Reviews* , 2017: In Press

Ozgener, Onder, and Arif Hepbasli. "A review on the energy and exergy analysis of solar assisted heat pump systems." *Renewable and Sustainable Energy Reviews* 11 (2007): 482-496.

Parida, Bhubaneswari, S. Iniyan, and Ranko Goic. "A review of solar photovoltaic technologies." *Renewable and Sustainable Energy Reviews* 15 (2011): 1625-1636.

Tian, Y., and C. Y. Zhao. "A review of solar collectors and thermal energy storage in solar thermal applications." *Applied Energy* 104 (2013): 538-553.

Wang, Qin, Yu-qian Liu, Guo-feng Liang, Jia-rong Li, Shu-fei Sun, and Guang-ming Chen. "Development and experimental validation of a novel indirect-expansion solar-assisted multifunctional heat pump." 43 (2011): 300-304.

Zhang, Yaning, Qin Ma, Bingxi Li, Xinmeng Fan, and Zhongbin Fu. "Application of an air source heat pump (ASHP) for heating in Harbin, the coldest provincial capital of China." *Energy and Buildings* 138 (2017): 96-103.