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Photovoltaic/Thermal (PV/T)/ground dual source heat pump: Optimum energy and economic sizing based on performance analysis

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ABSTRACT

Dual or multisource heat pumps were conceived to obviate to the defects of a single source, such as outside air, ground, water or solar radiation. Concerning the latter, the use of Photovoltaic/Thermal (PV/T or PVT) modules allows not only to partially recover the otherwise lost heat, but also to cool the PV and increase its electrical efficiency.

Many studies simulated the possible behavior of combination of PVT with other sources, but generally unglazed PVT collectors were used. Only few results based on coupling glazed PVT to ground source heat pumps are available in literature. The use of glazed PVT increases thermal efficiency of the collector, and the coupling of ground allows to keep the electrical efficiency at high values without the risk of cells damage due to overheating.

A refurbished building located in Northern Italy will be equipped by a PVT dual source heat pump, operating with the ground as source/sink, whereas the PVT drives the heat pump compressor and acts as a dual source. When the heat pump does not need heat or operates for summer air conditioning, the ground is the heat sink both for the heat pump and for the PVT cooling.

A dynamic simulation allowed to size the plant and set up a suitable control logic of the main equipment. Very high efficiency and low primary energy consumption are demonstrated for the whole plant, thanks also to the high energy independency from the grid.

1. Introduction

In 2017, the residential sector represented 27% of final energy consumption in EU: natural gas accounted for 36% of the final energy consumption in households, electricity for 24%, renewables for 18% and petroleum products for 11% [1]. The main use of energy was for heating (64% of final energy consumption in the residential sector). When insulating technologies and the new heating and cooling equipment are considered, this datum is surprising. Nowadays, a building can be (better, has to be, following the Directive 2010/31/EU on the energy performance of buildings, now amended by Directive 2018/844/EU) realized with a minimum requirement of energy for heating, cooling, and electricity, largely satisfied by renewable energies [2,3]. These buildings are referred to as Nearly Zero Energy Buildings (NZEB).

In this study, the retrofitting of a school building located near Feltre in Northern Italy allows to use several modern technologies with the purpose of realizing something more of a NZEB, that is a Plus Energy Building. This is a building not only energy self-sufficient on a yearly basis, but able to provide energy to nearby buildings, or to the grid.

Some proposed technologies are well known: ground heat pump, PV modules, thermal solar collectors, thermal bridges reduction, careful thermal insulation, heat recovery on ventilation air [4,5]. The novelty in this study is in the coupling of the different components, with a control logic conceived to exploit all the energy contributions.

A particular novelty is the use of photovoltaic cogeneration (PV/T or PVT). A PV module turns, say, 18% of solar radiation into electricity. As a matter of fact, 82% is then lost (Fig. 1). Moreover, its dissipation is detrimental for the efficiency as it produces an increase in the PV cell temperature. As a consequence, cell efficiency is reduced of the order of 0.2–0.5% for every 1°C increment in the PV module temperature for crystalline silicon cells.

A PVT module exploits the thermal fraction, as the PV cells are connected to a device that exchanges heat with a fluid (usually air or water) [6-8]. PVTs represent really a small niche in the actually huge PV market. The possibility of producing at the same time heat and electricity was conceived in the late 1976 [9]. However,

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Nomenclature

Acronym	Meaning
AHU	Air Handling Unit
DHW	Domestic Hot Water
DPB	Discounted Payback Period
EU	European Union
HP	Heat Pump
HVAC	Heating, Ventilation, Air Conditioning
NG	Natural Gas
NPW	Net Present Worth
NZEB	Nearly Zero Energy Building
PCM	Phase Change Material
PV	PhotoVoltaic
PVT	PhotoVoltaic/Thermal
Symbol	Meaning Unit
EM_1	Thermal energy produced by the PVT field kJ
EM_2	Thermal energy from/to the ground field kJ
EM_3	Thermal energy from/to the Heat Source Tank kJ
EM_4	Thermal energy to the evaporator (from Heat Source Tank or from Cold Tank) kJ
EM_5	Thermal energy from the condenser to the Hot Tank or to the Heat Source Tank kJ
EM_6	Thermal energy to the Cold Tank (cooling load) kJ
EM_7	Thermal energy from the Hot Tank to the DHW Tank kJ
EM_8	Thermal energy from the Hot Tank to thermal load (heating load) kJ
EM_9	Thermal energy from the DHW Tank + PRE-Heating DHW Tank to DHW load kJ
EM_{HP}	Electric energy consumed by the dual source heat pump / chiller kJ
EM_i	Electric energy consumed by pumps (i = pumps of this nomenclature) kJ
EM_{PVT}	Electric energy produced by the PVT field kJ
E_{res}	Electric energy consumed (and so thermal energy produced) by the auxiliary resistor inside the DHW Tank and the Hot Tank kJ
EM_{PV}	Electric energy produced by the PV field kJ
E_{HEX1}	Thermal energy from the condenser to the DHW Tank by means of the Hot Tank kJ, $E_{HEX1} = EM_7$
E_{HEX2}	Thermal energy produced by the PVT field for the direct production of DHW kJ, $E_{HEX2} = EM_1$ when A = N = 1
$E_{PVT\text{-}ground}$	Thermal energy produced by the PVT field to regenerate the ground kJ, $E_{PVT\text{-}ground} = EM_2$ when A = B = N = E = 0 and C = 1 and P5 = ON
$E_{PVT\text{-}source\_tank}$	Thermal energy produced by the PVT field used as heat source at the heat pump evaporator kJ, $E_{PVT\text{-}source\_tank} = EM_1 = EM_3$ when A = B = N = E = C = 0 and P6 = ON and P5 = OFF
$E_{source\_tank\text{-}evap}$	Thermal energy from the Heat Source Tank to the heat pump evaporator, $E_{source\_tank\text{-}evap} = EM_4$ when F = G = Q = R = 0 kJ
$E_{source\_tank\text{-}cond}$	Thermal energy from the heat pump condenser to the Heat Source Tank, $E_{source\_tank\text{-}cond} = EM_5$ when I = H = 0 and Q = R = 1 kJ
$E_{hot\_tank\text{-}cond}$	Thermal energy from the heat pump condenser to the Hot Tank, $E_{hot\_tank\text{-}cond} = EM_5$ when I = H = 1 kJ
$E_{cold\_tank\text{-}evap}$	Thermal energy from the Cold Tank to the heat pump evaporator, $E_{cold\_tank\text{-}evap} = EM_4$ when F = G = 1 kJ
EER	Energy Efficiency Ratio of the heat pump when operating as chiller, $EER = EM_4$ (when F = G = 1) / EM_{HP}
COP	Coefficient Of Performance of the heat pump, $COP = EM_5$ (when I = 1) / EM_{HP}
EER_{syst}	Energy Efficiency Ratio of the system, $EER_{syst} = EM_4$ (when F = G = 1) / (EM_{HP} + EMi)
COP_{syst}	Coefficient Of Performance of the system, $COP_{syst} = EM_5$ (when I = 1) / (EM_{HP} + EMi)
PER_{syst}	Primary Energy Ratio of the system (electrical efficiency η_{el} = 1/1.95 = 51.3% by Italian Decree DM 26/06/2015), PER_{syst} = ((EM_4 (when F = G = 1) + EM_5 (when I = 1)) / (EM_{HP} + EMi))•η_{el}
PER_{plant}	Primary Energy Ratio of the whole plant (electrical efficiency η_{el} = 1/1.95 = 51.3% by Italian Decree DM 26/06/2015), PER_{plant} = ((EM_4 (when F = G = 1) + EM_5 (when I = 1) + E_{HEX2}) / (EM_{HP} + EM_i + E_{res}))•η_{el}
$EP_{gl,nren}$	Non-renewable specific primary energy consumption: ratio between the equivalent non-renewable primary energy of the electricity from the grid to feed the plant consumption and the useful area of the building (A_{build} = 2435 m^2), $EP_{gl,nren} = E_{el,from\_grid}$ / ($\eta_{el} A_{build}$) kWh m^{-2} y^{-1}
η_{th_PVT}	Solar thermal efficiency: ratio between thermal energy produced by PVT and total solar ration on the collectors plane (A_{PVT}: aperture area of the PVT field), $\eta_{th\_PVT} = EM_1/(S•A_{PVT})$
η_{el_PVT}	Solar electric efficiency: ratio between electric energy produced by PVT and total solar ration on the collectors plane (A_{PVT}: aperture area of the PVT field), $\eta_{el\_PVT} = EMPVT$ / ($S•A_{PVT}$)
η_{tot_PVT}	Solar total efficiency: ratio between thermal + electric energy produced by PVT and total solar ration on the collectors plane (A_{PVT}: aperture area of the PVT field), η_{tot_PVT} = (EM_1 + EM_{PVT}) / ($S•A_{PVT}$) = $\eta_{th\_PVT} + \eta_{el\_PVT}$
S	Total solar radiation on the PVT (or PV) collectors plane kJ m^{-2}

in the beginning the main purpose was a PV cooling to increase the electric efficiency. Various designs employing air, liquid, heat pipe, phase change materials (PCM), and thermoelectric modules are possible [10]. Air PVT and ventilated PV façade systems have widely been applied to cool PV cells and to produce low grade thermal energy for space heating in residential applications [11]. No high electrical and thermal efficiency are possible with respect to water cooled PVT collectors due to the low density and small heat capacity of air.

Liquid-based PVT collectors allow some advantages with respect to air cooled due to higher specific heat capacity of coolants employed, leading to improved overall performance and less

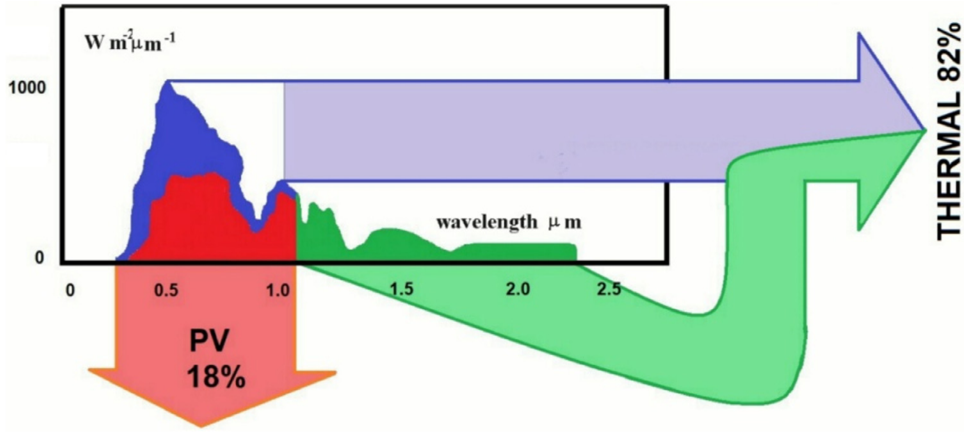

Fig. 1. A possible share of solar irradiation between electricity and thermal effect for a crystalline silicon PV module.

temperature fluctuations. The typical configuration comprises of metallic sheet-and-tube absorber, in which heat extraction is attained via forced fluid circulation through series/parallel-connected pipes adhered to the rear of PV collector [11–13]. Water is largely used as working fluid; however, refrigerants that can undergo phase change at a relatively low temperature can be adopted in systems combining PVT collectors and solar-assisted heat pumps.

Many experimental and theoretical studies have been proposed in the recent past on the performance of PVT water cooled flat plate collectors [14,15]. Performance strongly depends on the channels absorber design, the glazed or unglazed configuration, the flow rate, and the inlet temperature of thermal fluid. An experimental study was carried out by the authors in order to investigate the efficiencies obtainable by some prototypes of PVT water cooled flat plate collectors [16,17].

Water cooled PVT is the most widespread technology. The thermal levels can provide domestic hot water (DHW) or ambient heating, but a better compromise between the useful energy and a moderate PV temperature indicates a better utilization as heat pump source. A first proposal of using a PVT panel to supply a heat pump appeared not economically viable [18]. The attractive feature of PVT was essentially the best use of a limited area for collecting solar energy [19]. The high cost of PV and the absence of commercial PVT limited the researches mainly to computer simulations, with few experiments related above all to DHW heating [20].

Recently, the performance of a roll-bonded PVT heat pump system was analyzed [21]. The results indicate that the proposed system can offer refrigeration for building space cooling demand in summer with considerable performance and long-term stable operation, provided that nighttime and rainy/overcast daytime PVT working as condenser is scheduled. Another study [22] presents the first results of a field study concerning a solar-assisted dual-source multifunctional heat pump, installed in a detached house in Milan, in four operation modes. In this case, the PVT panels are used, by employing two storage tanks, to produce DHW and to provide a heat source to the HP evaporator, showing interesting and promising results.

Bertram et al. reported a possible improvement from 4.2 to 5.4 for the seasonal performance factor for a dual source heat pump system ground-PVT with respect to only ground source [23]. The penalization for the PV electricity production was estimated at 5%. They used unglazed PVT.

In a recent study, a review of energy-efficient measures and renewable energy technologies in NZEBs discusses the application and suitability of PVT coupled to ground source heat pumps [24].

Most of the considered solutions are unglazed for fear of a possible overheating of the PV cells. A recent market survey supplies a short list of manufacturers, most of them with unglazed products [25].

Instead, in this study glazed PVTs are used, as the Authors believe that a glazed PVT can give the best results once the PV cells temperature is controlled by a proper coupling with the ground heat exchanger. The retrofitting of part of an old school building in the North of Italy allows to analyze the energy performance and the economic viability of a dual source glazed PVT/ground heat pump system in a severe winter climate. The contemporaneous DHW, heating and cooling demand during mild and hot months makes necessary to set up a suitable scheme of the HVAC plant and its control logic.

Section 2 illustrates the detailed modeling hypotheses of the building retrofitting, the HVAC plant, the working mode and the principle of the dual source (glazed PVT and ground) heat pump system. Section 3 contains the performance evaluation method, the results and discussions of this study, firstly on a monthly basis for the preferred solution, subsequently on a yearly basis for the comparison of different alternatives. Finally, Section 4 contains some concluding remarks of the study.

2. Methods

2.1. Building retrofitting and thermal loads

The building that is going to be refurbished in 2019 is part of an old (completed in 1960) high school building of Feltre, situated in the province of Belluno, northern Italy. The climate is rather severe in wintertime (3100 degree-days). The gym and the laboratories will be refurbished by the Belluno Province Administration (a public Authority charged with the public education service) with the aim of realizing a NZEB. The main part of the refurbished building is a large gym (33 m × 25 m × 8.40 m) expanding on two levels. Changing rooms, bathrooms with toilet and showers, and technical rooms are located at the ground floor; an office, a small gym and a bar are at the first floor. At the second floor, six laboratories will be refurbished and made newly available.

The total floor area is 2435 m^2, with an outward surface area of 2505 m^2, and an enclosed gross heated volume of 11,060 m^3. The building will be carefully insulated in this refurbishment, with outer walls and roof allowing for an average thermal transmittance of approximately 0.15 W m^{-2} K^{-1}, floor to the ground of 0.5 W m^{-2} K^{-1}, and glazing system of 0.7 W m^{-2} K^{-1}.

Based on Trnsys 17 [26] dynamic simulation, the building is divided into 20 thermal zones. Each zone is defined by means

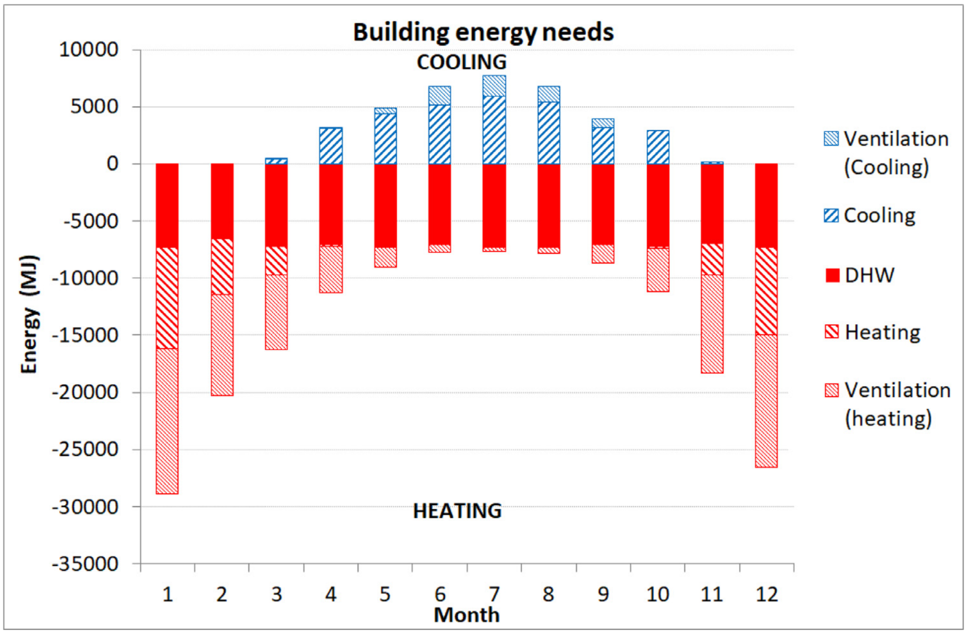

Fig. 2. Monthly energy needs in terms of heating, cooling, ventilation (hot and cold coils of AHUs), and DHW.

of scheduling of presence of people, type of activity, lighting and other internal gains, humidity and air temperature set points. The HVAC system provides ventilation by two air handling units (AHU), space heating and cooling, and DHW production. Fig. 2 reports monthly energy needs, obtained by adding up the heating and cooling loads calculated with a 0.25 h simulation time step.

The maximum cooling load (21.6 kW) occurs at the beginning of June, when the school is still fully operating (open to students and professors, gyms open to extra-school activities as well). The maximum heating load (−44.6 kW) occurs in the second half of January. However, even during summer months some heating is required for post heating ventilation. Ventilation loads in heating season are prevailing with respect to room heating due to the excellent thermal insulation and the careful reduction of thermal bridges in the refurbished building. DHW needs (2000 L per day at 45°C) are a large quota of the total heat request. They are concentrated in short periods mainly during the evening due to training sessions of local sport teams.

2.2. The HVAC plant

The original feature of the HVAC plant is the solar section composed of glazed PVT (and plain PV as later illustrated) that drive a heat pump which satisfies all the loads, safe that the PVT thermal component provides DHW heating or at least preheating, when possible, or acts as heat pump cold source.

A simplified scheme of the plant is represented in Fig. 3, where the various components are connected via suitable storage tanks.

As a matter of fact, the plant is set up by four main loops:

1 PVT - Source Tank - Ground loop: it includes the dual heat source (PVT and vertical tube ground boreholes) for the heat pump. The PVT excess heat (mainly in summer) recharges the ground;
2 DHW loop: on the right of Fig. 3, two storage tanks are committed to the DHW service. Water from the mains arrives at the Pre-Heating DHW Tank where, if suitable temperatures can be obtained, it is heated by the PV cooling water when this is not directed to the Heat Source Tank or to the ground. The heating to the set point temperature is completed in the DHW Tank, supplemented via a heat exchanger by the Hot Tank. The presence of two tanks for DHW production allows to satisfy the large request (2000 L per day) and, at the same time, to usefully cool down the PVT by the thermal exchange with the low temperature of the fresh water from the mains;
3 Heat Pump - Chiller loop: Hot Tank satisfies the heating load, receiving heat by the heat pump condenser. The Heat Source Tank is the heat pump cold source and it is alimented either by the ground or by the PVT.
The Cold Tank is dedicated to cooling loads. It is cooled by the heat pump evaporator. During the heat pump operation as a chiller, useful heat for post-heating or DHW heating can be provided by the heat pump condenser. In summer, the eventual excess heat from the PVT or heat pump condenser is used to recharge the ground, maintaining at an acceptable temperature theglazed PVT;
4 Heating – DHW loop: it is the circuit that allows the Hot Tank to contribute to the DHW and heating loads.
The control logic is based on a survey of solar radiation intensity (S), determining an operation threshold between 50 and 100 W m^{-2}, and on a comparison between the various suitable tanks set-points and the available temperatures in the different circuits as depicted in Fig. 3.
A radiant floor provides space heating in the large gym, which is cooled by an all air system that provides the ventilation service (6600 m^3 h^{-1}) by an air handling unit (AHU). The other rooms (small gym, bar, laboratories and offices) are equipped with fan coils, whereas toilets are heated only (by radiators), and laboratories are served also by an independent AHU (7000 m^3 h^{-1}) for ventilation.

2.3. Modeling of the main equipment

Trnsys type 50c (based on type 1 modeling the solar thermal collector based on Hottel-Whillier equation) is used in order to simulate the glazed PVT collector. In this type, thermal performance of the collector depends on the solar radiation incidence angle on the plane of the collector. Thermal and electrical performance of the PVT depend on the main geometrical, constructive, electrical and thermal specifications reported on Table 1 (data

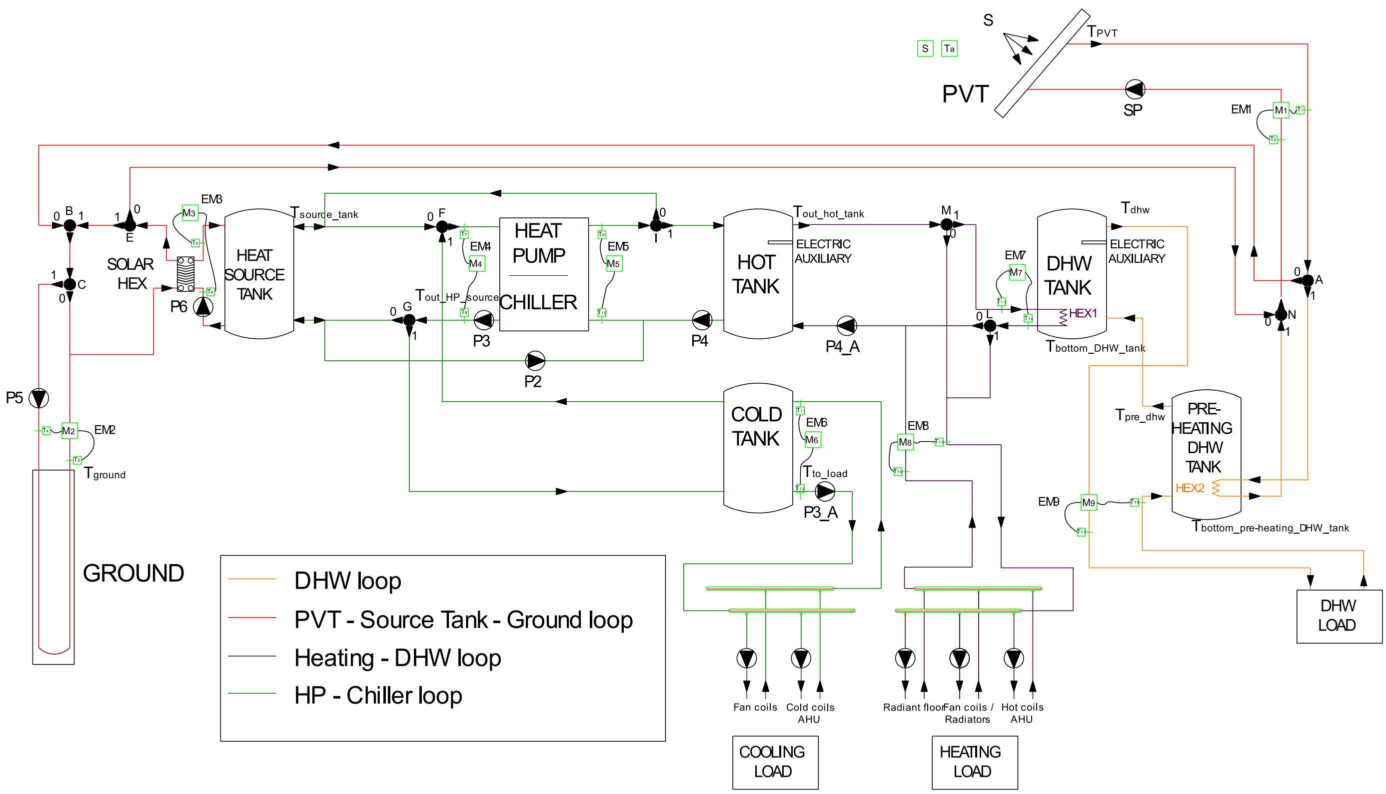

Fig. 3. Simplified functional diagram of the HVAC plant.

Table 1
Main parameters of the commercial glazed PVT collector.

Parameter	Unit	Value
Cell size	mm	156 × 156
Cell number		72
Cell type		Mono-crystalline
Absorptance coefficient		0.85–0.9
$\tau\alpha$		0.9
Gross Area	m^2	2.1
Opening area	m^2	1.95
Packing Factor		0.9
Front glass	mm	3.2
Glass type		Tempered glass
Extinction coefficient thickness product		0.01
Recommended flow rate	$Lh^{-1}\ m^{-2}$	50
Electrical Specifications (Values tested under STC)		
Nominal Power (P_{max})	Wp	300
Nominal Voltage (V_{mp})	V	36.6
Nominal current (I_{mp})	A	8.45
Power Temperature Coefficient	%/ °C	−0.43
Module Efficiency		16.0%
Thermal Specifications		
Zero loss Efficiency (η_0)		59%
a_1 (First Order Heat Loss, FRUC)	$Wm^{-2}\ K^{-1}$	3.30
a_2 (Second Order Heat Loss)	$Wm^{-2}\ K^{-2}$	0.02
F_R		0.66
F'		0.66
U_C	$Wm^{-2}\ K^{-1}$	5.03
U_{fl}	$Wm^{-2}\ K^{-1}$	0.25

refer to a real glazed PVT module available on the market). Mass flow rate entering the PVT affects not only its performance, but also the heat exchange with the ground due to the configuration of the plant: as a matter of fact, the mass flow rate value depends on the series/parallel configuration of the solar field. Considering that this study is parametric by varying the solar field between 20 m^2 and 60 m^2 (Table 2), that is between 12 and 36 PVT modules, the selected configuration is two equal groups where the collectors are in parallel with a group in series with the other. Such a choice allows to have a solar field total mass flow rate that is suitable for the ground field loop, a feature which is required as the flow may operate in a single circuit that connects the solar section with the ground as previously described.

The ground field is modeled by type 557a with a ground thermal conductivity of 2.87 W m^{-1} K^{-1}, and a storage heat capacity of 2016 kJ m^{-3} K^{-1}. The ground field is composed by nx100 m in

Table 2
Size of the solar and ground fields, and of the storage tanks for the alternatives considered.

Alternative	Solar field PVT (m^2)	N. boreholes (100 m depth each)	Pre-DHW Tank (L)	DHW Tank (L)	Hot Tank (L)	Cold Tank (L)	Heat Source Tank (L)
20 m^2 - 5	20	5	1500	500	1500	1000	2000
40 m^2 - 3	40	3	1750	750	1500	1000	2000
40 m^2 - 4	40	4	1750	750	1500	1000	2000
60 m^2 - 3	60	3	2000	1000	2000	1000	2500
60 m^2 - 5	60	5	2000	1000	2000	1000	2500

Table 3
Nominal data of a commercial electric driven water/water heat pump/chiller (data supplied at 12/7°C; 40/45°C).

Variable	Unit	Value
Pcooling	kW	27.2
Pthermal	kW	35
Pelectric	kW	7.75
Total Energy Ratio		8.03
Water flow rate heat exchanger use/cold	L s^{-1}	1.30
Water flow rate heat exchanger recovery/hot	L s^{-1}	1.67

a row vertical tube U heat exchangers, with an outer diameter of 32 mm and a thickness of 2.9 mm, distance 6 m, n = 3-4-5 as a function of the solar field area as described in Table 2. The tanks are modeled by type 4a (Hot Tank, Cold Tank, Heat Source Tank) and type 60d (Pre-DHW Tank, DHW Tank). Both the Hot Tank and the DHW Tank have a 3 kW auxiliary electric resistance to supplement the heating and DHW load respectively, if necessary. This means that when the water temperature in the upper part of the tank is below the set point (50°C and 45°C respectively, with a dead band of 3°C) and the main generators (PVT and heat pump) are not available, the auxiliary electric resistances operate. Table 2 reports also the suitable size of the tanks for each alternative considered in this study.

Finally, type 927 is used to model the water-water heat pump based on nominal data from a manufacturer concerning thermal power, cooling power and electrical power consumption at various heat source and sink temperatures (Table 3). The capacity of the heat pump has been selected on the basis of the peak thermal load calculation (heating, DHW, and cooling) as reported in Section 2.1. The performance of the heat pump has been calculated by the Trnsys type 927 at the different operating temperatures of the heat source and sink. These have been considered varying in useful ranges according to source and sink temperatures (respectively T source tank $\leq$ 25°C and Hot Tank or Heat Source Tank max 52°C). In order to simulate the real behavior of the heat pump, a controlled flow diverter (R) and a controlled flow mixer (Q) are added. Table 4 reports the outlet of the valves and the status of the pumps for the three main operation modes of the heat pump.

3. Results and discussion

A full system energy performance evaluation is the starting point for a rational sizing of the various plant components: PVT area, length of the geothermal probes, and tanks capacities. The length of the ground probes can be reduced by increasing the PVT field, as the contribution of the solar energy is greater both in electric and thermal energy. Moreover, more solar energy can be directed to recharge the ground during summer months. Therefore, an energy and economic analysis has been carried out for each alternative listed in Table 2.

As additional roof surface was available even after the installation of the largest PVT area, a further 60 m^2 area plain PV is considered to increase the electrical production.

The main results of the monthly analysis are here illustrated for the most favorable alternative (see next Sections 3.1 and 3.4). The alternative considers three ground probes, 100 m each, and 60 m^2 PVT (plus the above considered 60 m^2 of plain PV). The results are reported as energy balance for the main plant sections (Fig. 4). The meaning of the symbols in Figs. 3 and 4 and of the energy performance indexes used in next sections is listed in the Nomenclature.

Subsequently, a comparison between the different alternatives in terms of energy and economic results on an annual basis is reported.

3.1. Monthly energy balances of the preferred alternative

A solar energy balance is given in Fig. 5 for the PVT section. The balance brakes down the positive input, that is the solar radiation, into five fractions:

- Thermal losses, that is the largest fraction;
- Electric energy;
- Contribution as heat pump heat source;
- Thermal energy to the DHW pre-heating;
- Ground recharging.

Even if thermal losses generally exceed the other items, a good contribution is offered in winter by PVT as heat pump cold source (about a half of the required energy). In the most unfavorable month, January, 4950 MJ (1374 kWh) are available for the heat pump at a temperature higher than allowed by the ground source. The produced electricity is estimated at 1582 MJ (439 kWh) and thermal losses at 4337 MJ (1204 kWh).

In July, the produced electricity is of 4602 MJ (1278 kWh), with 6664 MJ (1851 kWh) of thermal energy to DHW pre-heating and 5757 MJ (1599 kWh) to ground recharging. In July, the overall useful energy supplied by PVT is almost comparable to the heat losses, with a 48% overall monthly efficiency (electrical efficiency

Table 4
Status of pumps and valves control variables for the HP-Chiller loop (refer to Fig. 3).

Pumps/Valves	1. Operation as heat pump with contemporaneous cooling demand	2. Operation as heat pump without contemporaneous cooling demand	3. Operation as chiller (with cooling demand) with saturation of the Hot Tank
I,H	1	1	0
G,F	1	0	1
P2	OFF	OFF	ON
P3	ON	ON	ON
P4	ON	ON	OFF

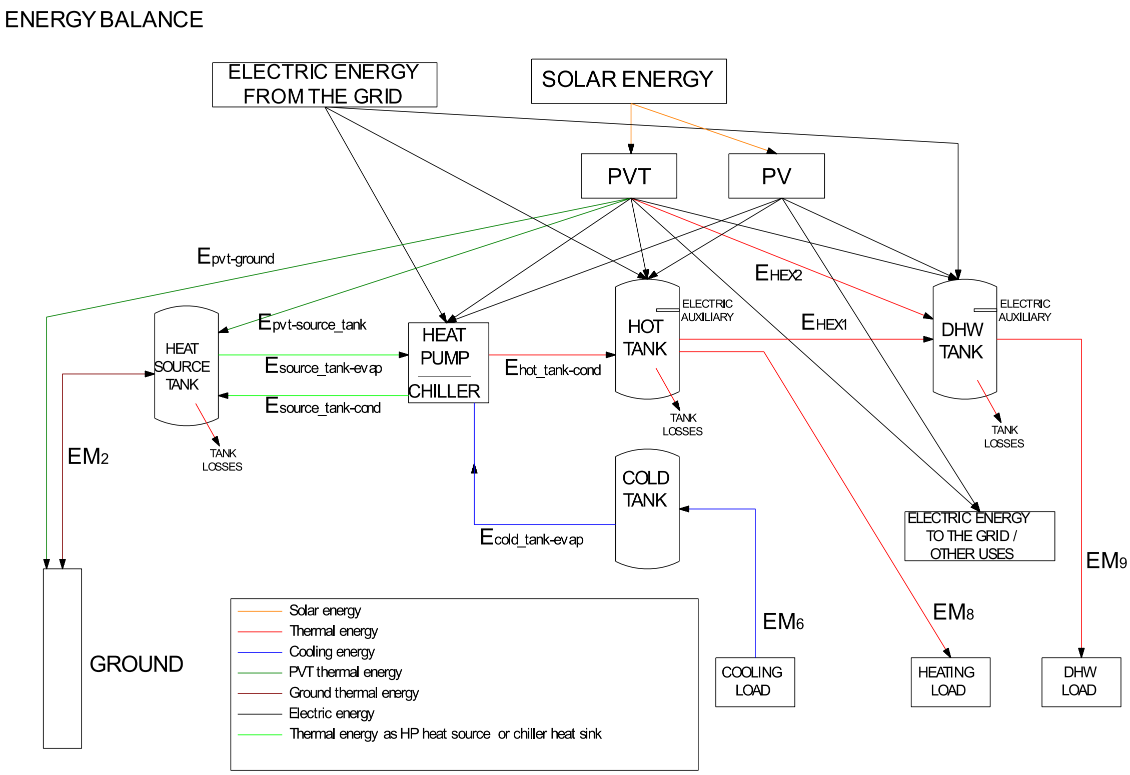

Fig. 4. Energy balance of the main equipment.

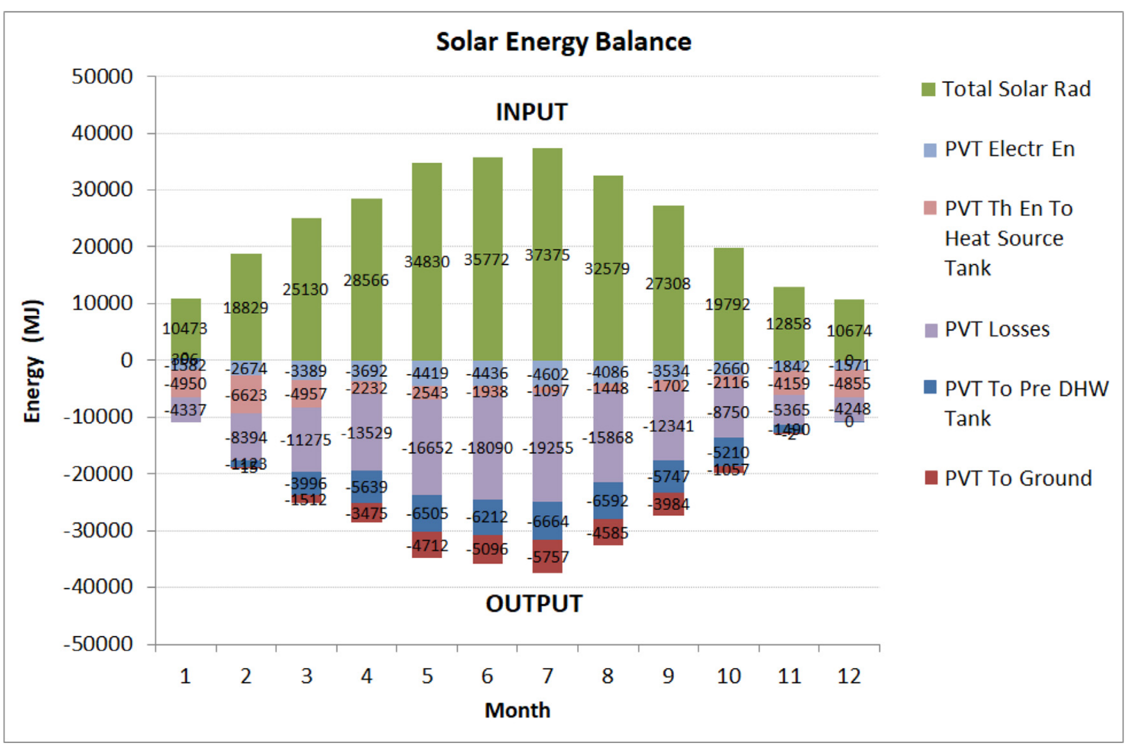

Fig. 5. Monthly PVT solar energy balance.

12%, thermal efficiency 36%). As a matter of fact, around 20% of the incident solar energy is used for DHW pre-heating in summer (it increases till 26% in October).

The significant contribution of PVT to DHW heating is well represented in Fig. 6. Here, the negative component of the balance is the load plus a small heat amount due to the tank thermal losses. PVT satisfies from 70% to more than 90% of DHW demand for the whole period from April to October, whereas the complementary part is provided by the Hot Tank (auxiliary electric resistance contribution is very limited).

As far as the heat pump behavior is concerned, Fig. 7 reports the energy balance. When operating as heat pump (during colder

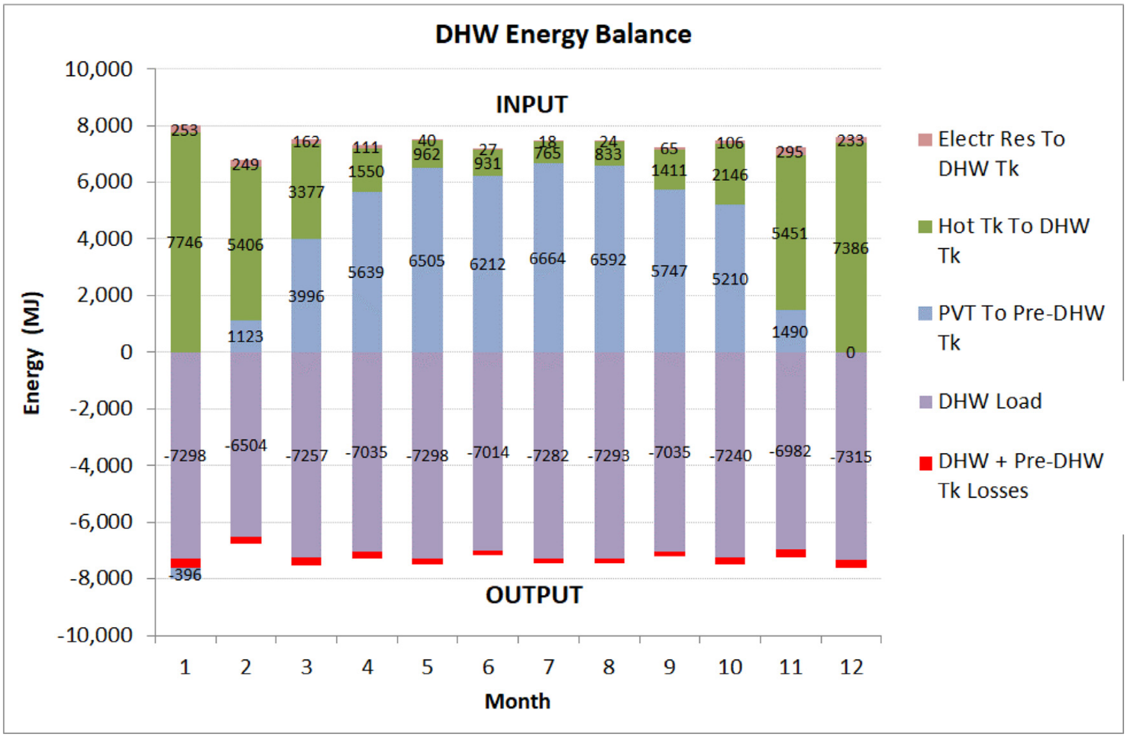

Fig. 6. Monthly DHW energy balance.

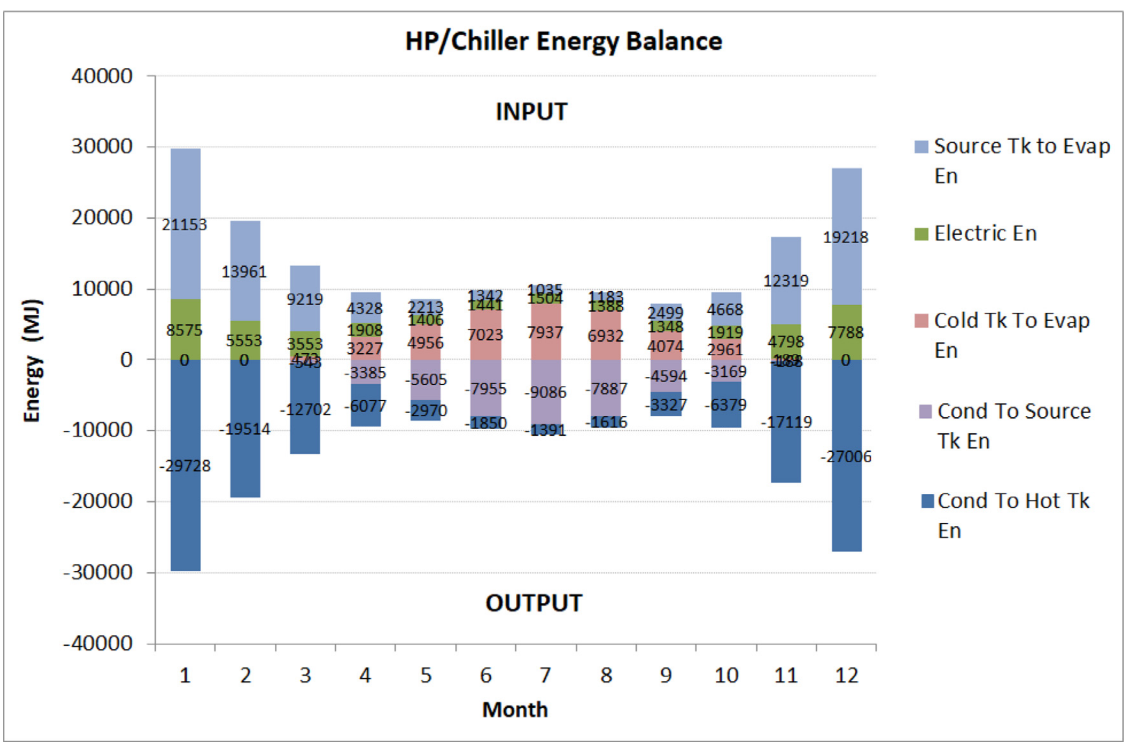

Fig. 7. Monthly heat pump/chiller energy balance.

months) the cold source is the Heat Source Tank, and the heat sink (useful effect) is the Hot Tank. In summer, the heat pump operates mainly as a chiller producing the useful effect at the Cold Tank, and supplying the condensation heat either to the Hot Tank (useful heat recovery for DHW and for contributing to the limited heat loads of the hot coils of AHUs) or to the Heat Source Tank.

As far as the heat pump behavior is concerned, during the cold months the best observation point is the balance of the Hot Tank (Fig. 8). Here, the positive component is the contribution of the heat pump condenser to the tank. This is split into the satisfaction of the heating load and the integration of DHW from the pre-heating temperature to the set point (45°C). An almost negligible fraction must be attributed to Hot Tank losses. A further insight

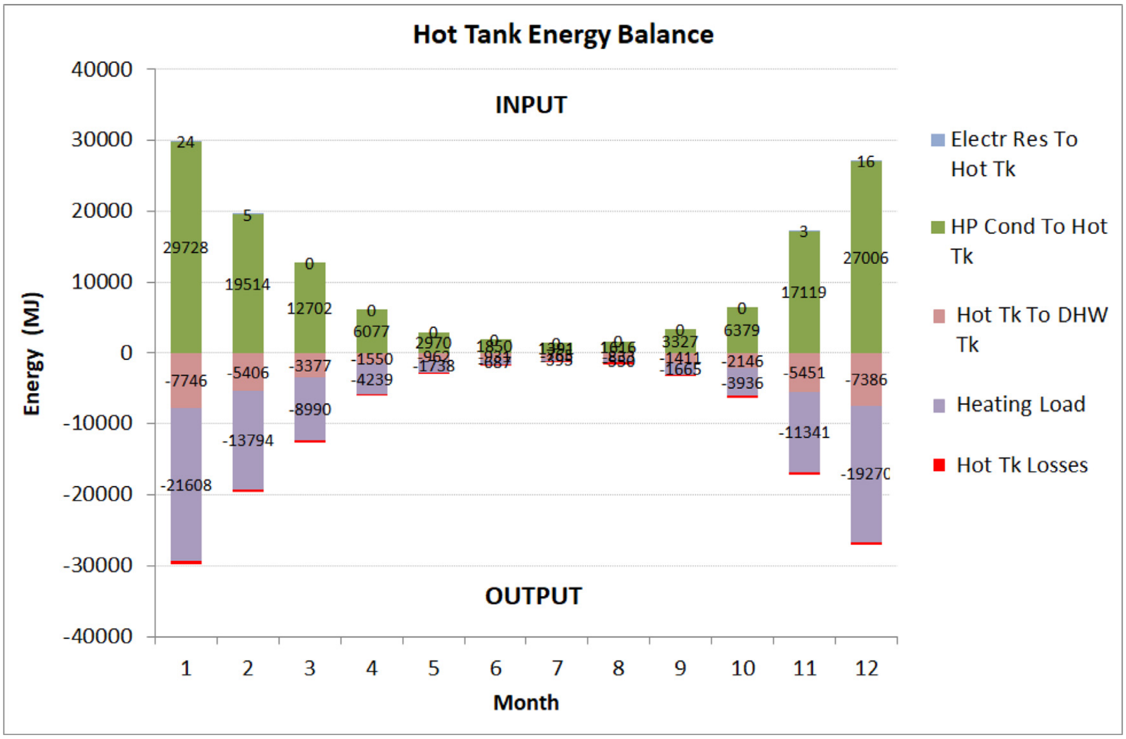

Fig. 8. Monthly Hot Tank energy balance.

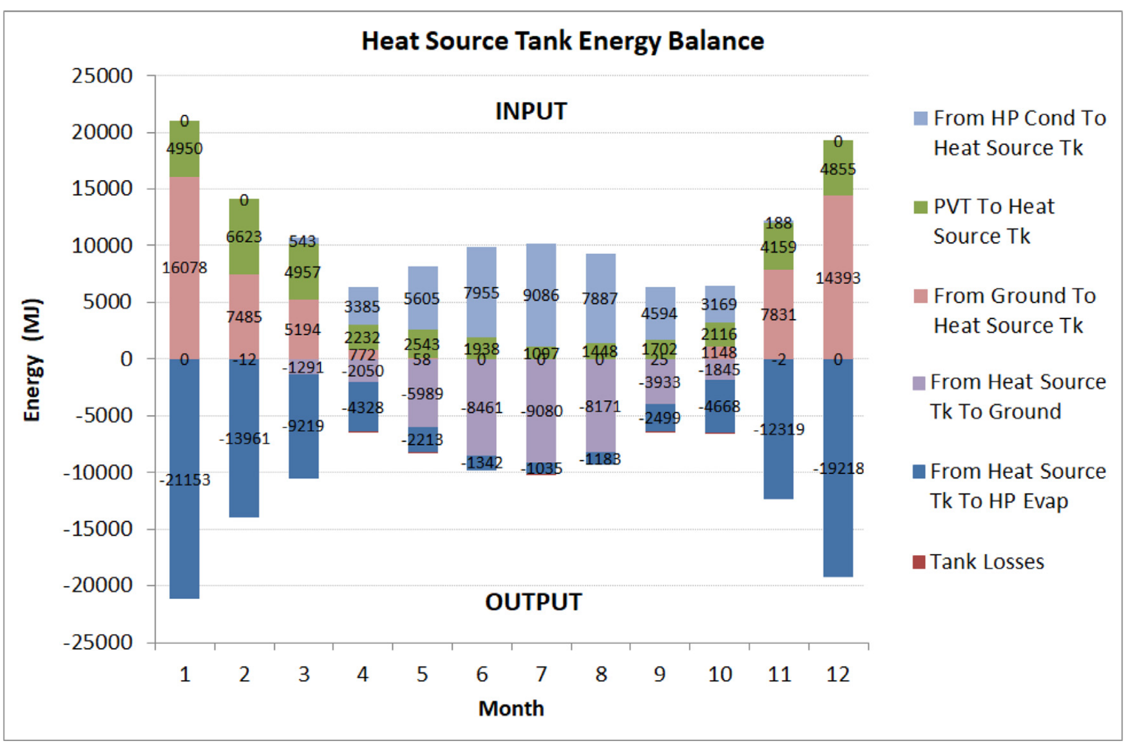

Fig. 9. Monthly Heat Source Tank energy balance.

is allowed by Fig. 9 that reports the balance of the Heat Source Tank. During the cold months, this tank receives the two positive contributions from the ground and from the PVT: this energy is found in practice entirely to feed the heat pump evaporator. However, during the other months the balance is complicated as the heat pump condenser releases to this tank the energy not requested by the DHW heating (mainly satisfied by PVT), or for the AHU post-heating. The output of the tank is then mainly towards the ground, even if a small fraction happens to be supplied to the heat pump evaporator for short periods of the month.

Fig. 10 reports the ground energy balance. During the coldest period (November-February) the ground operates as heat source only (outlet temperature varying in the −3–15°C range), whereas from March till October it is used mainly as heat sink for both the

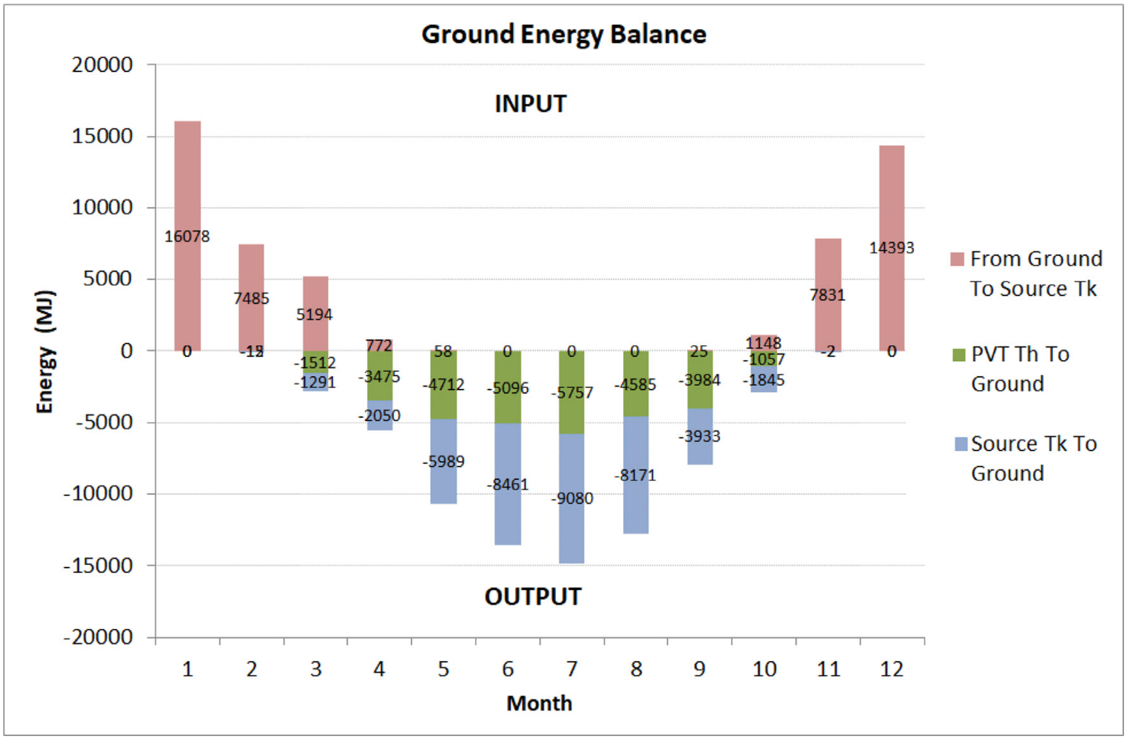

Fig. 10. Monthly ground energy balance.

PVT and the Heat Source Tank (outlet temperature varying in the 8–25°C range). On a yearly basis, the withdrawal is of 52,900 MJ (14,695 kWh), whereas the injection is of 71,000 MJ (19,722 kWh, 8383 supplied by PVT and 11,339 by the heat pump condenser). Then the average ground temperature should slightly increase in the long run.

3.2. Monthly energy efficiency indexes of the preferred alternative

PVT electric efficiency is between 12.3% (July) and 15.1% (January) (Fig. 11). The configuration of the plant allows a suitable PV cells cooling that is beneficial, as during the hottest months electrical efficiency is not far from the nominal value (16% in peak condition) even if the PVT is glazed, and cooling water temperatures sometimes exceed 60°C during summer. Figs. 12 and 13 report the temperature values of the water stream cooling the PVT panels and the PV cells mean temperature in a typical summer and winter day, respectively. The solar pump (SP) starting and turning off depend on the solar radiation and on the comparison between the outlet PVT temperature and the outlet Heat Source Tank temperature: that is why starting and turning off happen in different time of the day along the year. Assuming that the T_out_PVT temperature approximates the mean PVT cells temperature, the comparison between the PV and the PVT cells temperature in Figs. 12 and 13 reveals that the shapes of the curve of PV temperature and solar radiation are similar, unlike the curve of T_out_PVT which is influenced by cooling water as well.

Fig. 14 illustrates the efficiency parameters of the heat pump and of the whole system. They are evaluated as COP for the heat pump operation, and as EER for the chiller operation. The correspondent values for the system, that is taking into account the parasitic power of pumps, are considered as well. This produces a noticeable reduction for the EER when the longer circuits between PVT and boreholes are more frequently active. Both COP and EER maintain high values during the whole year. The primary energy ratio (PER) is particularly high (around 5) for the combined operation of the heat pump producing simultaneously useful heating and cooling for a significant period during the year. In particular, during the mild months (March, April and October) useful heat production is predominant, whereas during the hottest period (form May to September) useful cold production is larger (Figs. 7 and 14). The high efficiency of the whole plant (PER_plant) must be partly attributed to the PVT thermal contribution to DHW heating.

3.3. Monthly solar ratio and grid dependency of the preferred alternative

Thermal energy produced by the PVT field can fully cover the DHW demand from February till November, while the whole heat demand (heating + DHW) can be satisfied from April till October.

Electrical solar ratio of the plain PV field and the PVT is very high for a long period during the year as electricity produced by the two fields is firstly used in the components of the plant (heat pump, pumps, and other ancillary equipment). The produced electricity that exceeds local requirements is exported towards the grid. The HVAC plant appears completely energy self-sufficient on a yearly basis, making available electricity for the other uses of the building, or for the grid. The electricity external dependency from the grid can be observed only in January and December, whereas in February and November the self-produced electricity is clearly prevailing. From March to October electricity can be exported as the whole building demand is satisfied (Fig. 15).

3.4. Energy comparison with a conventional system

A comparison has been carried out in energy terms between the proposed plant and a conventional one equipped with a natural gas fired condensing boiler (100% efficiency on lower heating value) and an air cooled chiller (monthly mean EER as reported in Table 5). The comparison is illustrated in Fig. 16 in terms

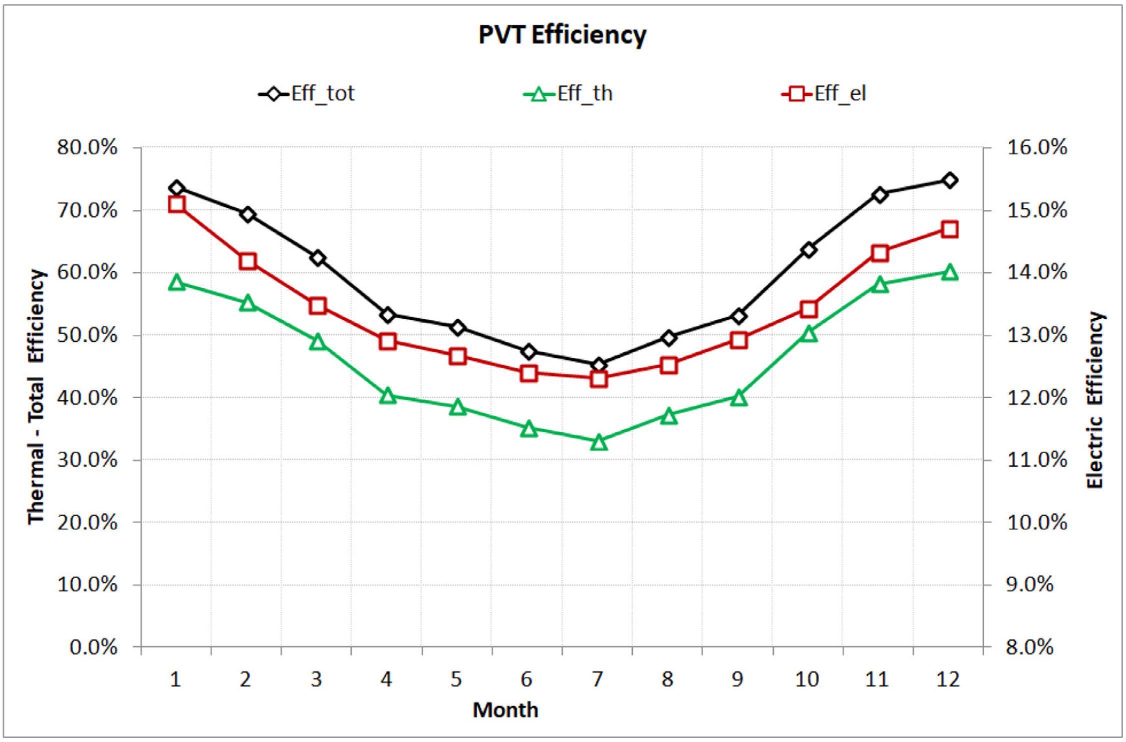

Fig. 11. Thermal, electric and total efficiency of the PVT plant.

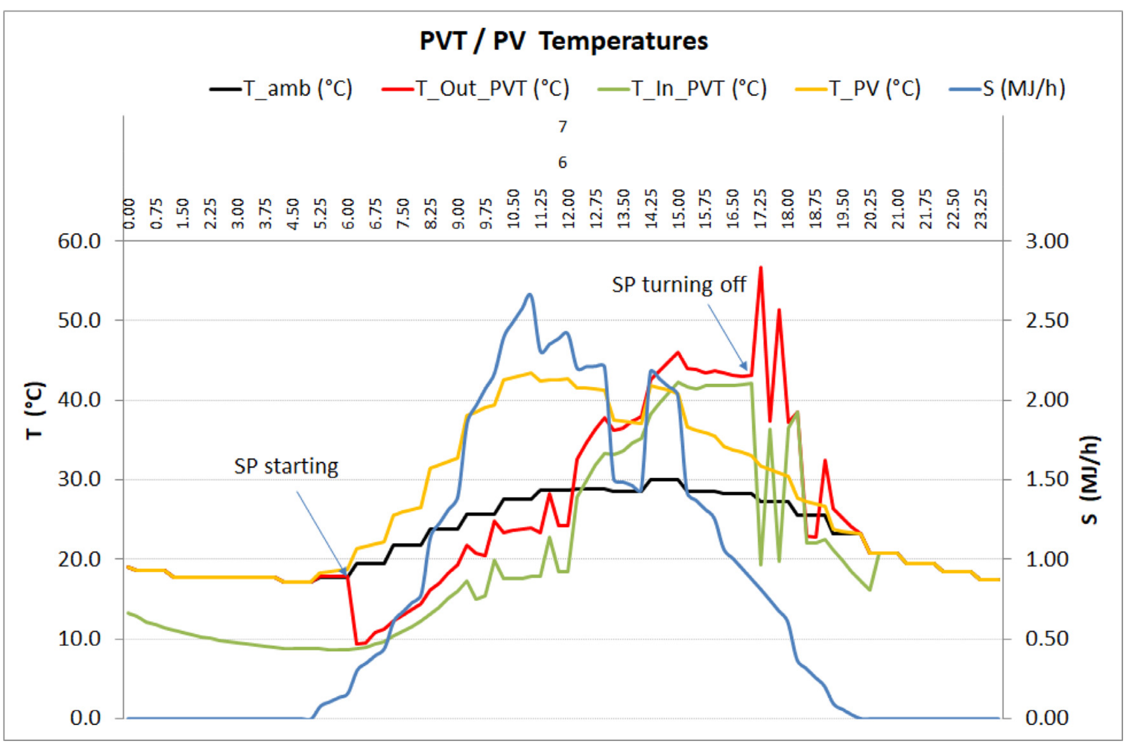

Fig. 12. Outdoor ambient conditions (temperature (T_amb) and global solar radiation on the PVT plane (S)), inlet and outlet PVT temperature (T_In_PVT, T_Out_PVT), and temperature of the PV cells (T_PV) for a typical summer day (6th July).

of primary energy (non-renewable), with a 1.95 factor for the electricity from the grid (average electricity production efficiency 51.3%), and considering that the traditional plant uses the same pumps of the proposed one (except for the solar pump, SP, and the geothermal, P5 in Fig. 3). The monthly bars reveal a relevant negative item due to natural gas demand of the conventional plant, with a comparatively small consumption of the air-cooled chiller. The proposed plant takes some electricity from the grid only in the cold months, offering from March to October an energy surplus to be exported. On an annual basis, the traditional plant primary energy consumption is 198.5 GJ, whereas the proposed plant has a surplus of 116.7 GJ.

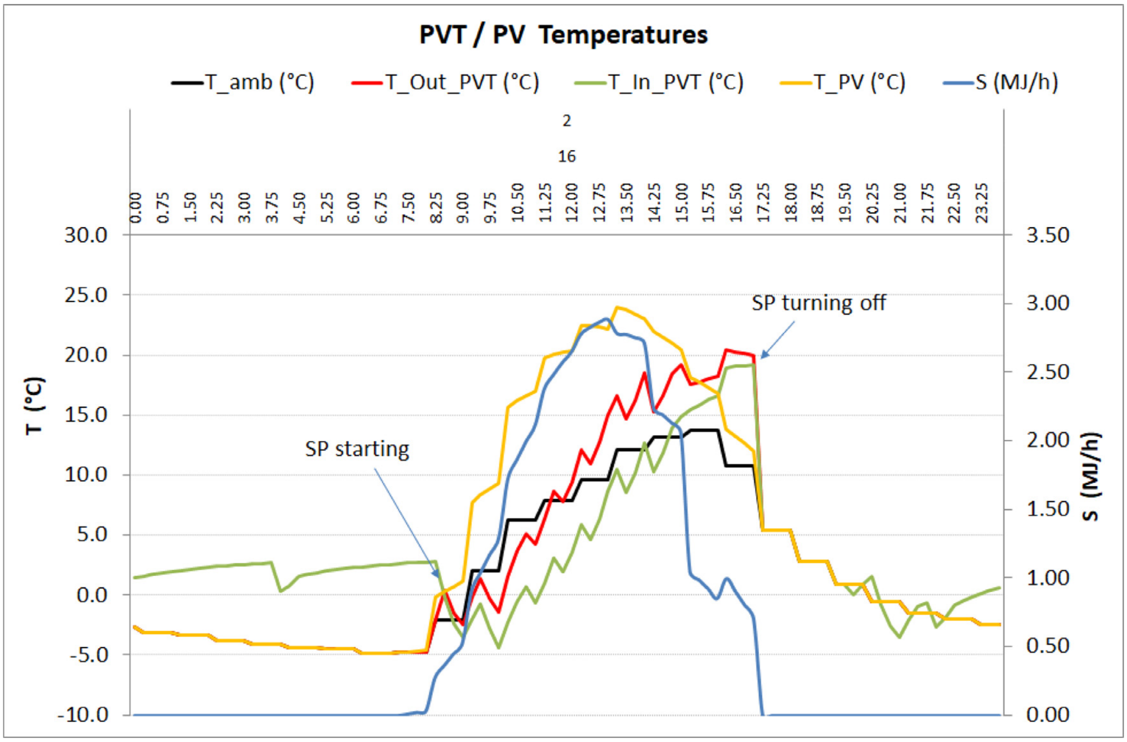

Fig. 13. Outdoor ambient conditions (temperature (T_amb) and global solar radiation on the PVT plane (S)), inlet and outlet PVT temperature (T_In_PVT, T_Out_PVT), and temperature of the PV cells (T_PV) for a typical winter day (16th February).

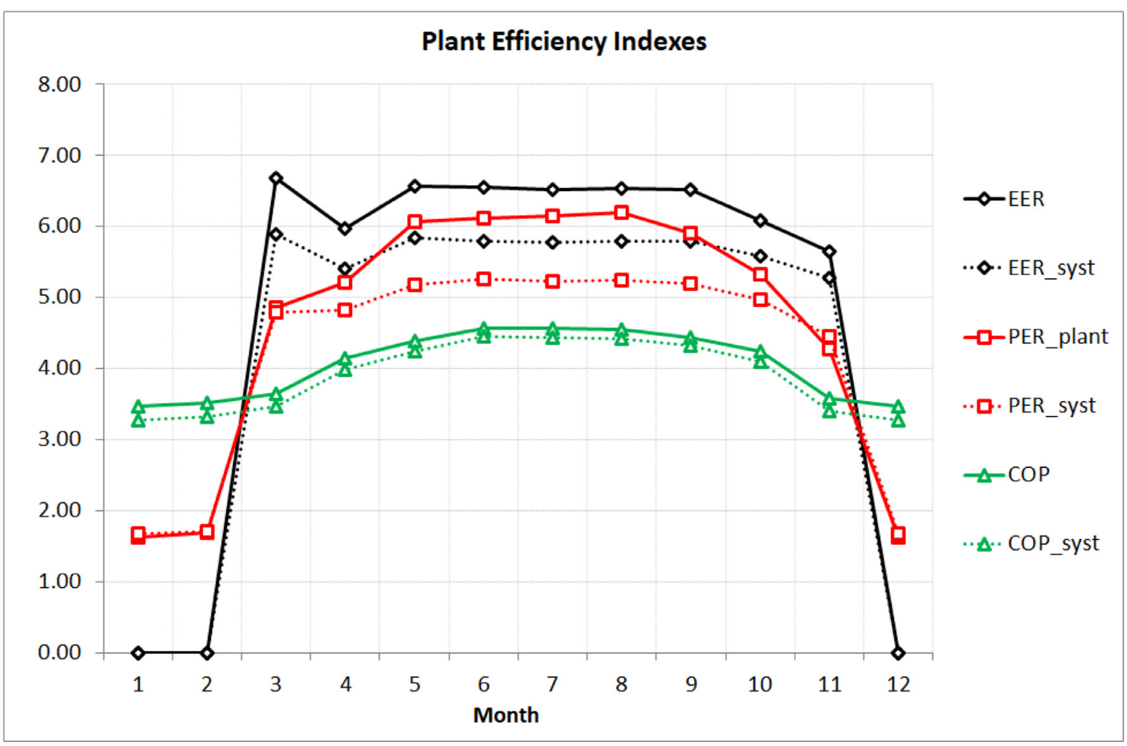

Fig. 14. Heat pump efficiency indexes with pumps parasitic energy (subscript _syst) or without. PER of the whole plant is represented as well (subscript _plant).

3.5. Energy comparison of different alternatives

As previously mentioned, the presented results are referred to a "preferred" alternative. In fact, five different alternatives were studied with different combinations of PVT area and ground probes length as reported in Table 2. Each alternative provides also 60 m^2 of plain PV. The five alternatives are compared in Fig. 17 between them on a yearly basis in terms of primary energy savings, each with the counterpart of a traditional solution (already described in the previous section). The increment of savings is less than proportional to the PVT area as already at 40 m^2 the thermal contribution is in excess of the demand in various summer days, so much the

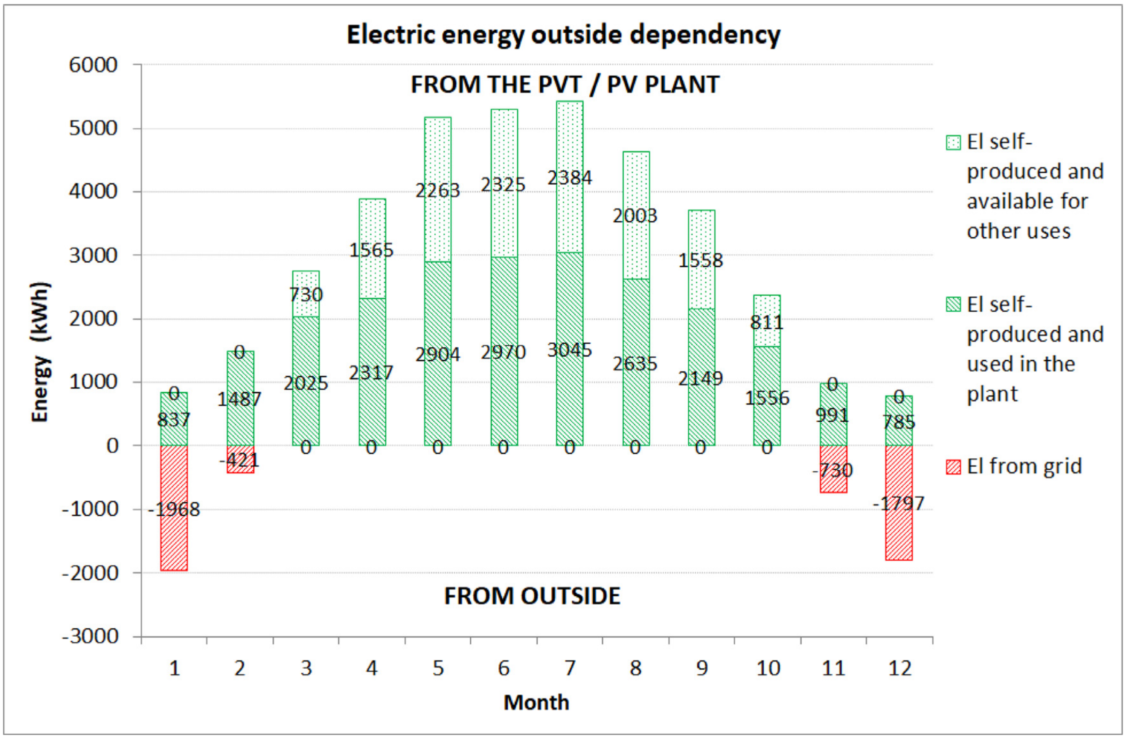

Fig. 15. Monthly electricity balance for the building.

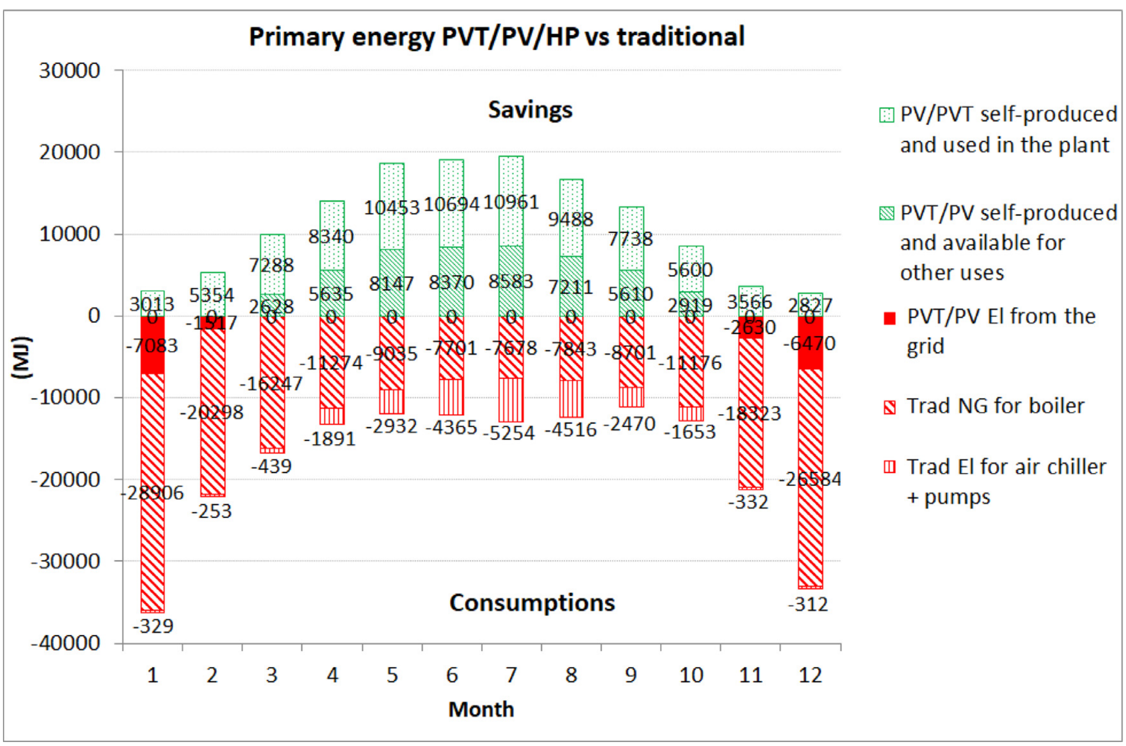

Fig. 16. Monthly comparison between the proposed plant and a conventional plant in terms of non-renewable primary energy.

Table 5
Monthly average values of the EER for the chiller of the "traditional solution".

Month	EERchiller	Month	EERchiller
4	3.7	7	3
5	3.5	8	3.1
6	3.2	9	3.4

more for 60 m^2. This is shown also in Fig. 18, that reports the main performance indexes and the specific annual non-renewable primary energy consumption of the plant: by increasing the PVT field area, its thermal efficiency decreases. No advantage is offered by an increase in borehole length from 300 to 500 m (that instead would increase the capital cost of the plant). The fact is that for a shorter length the ground recharging is more effective, and this balances

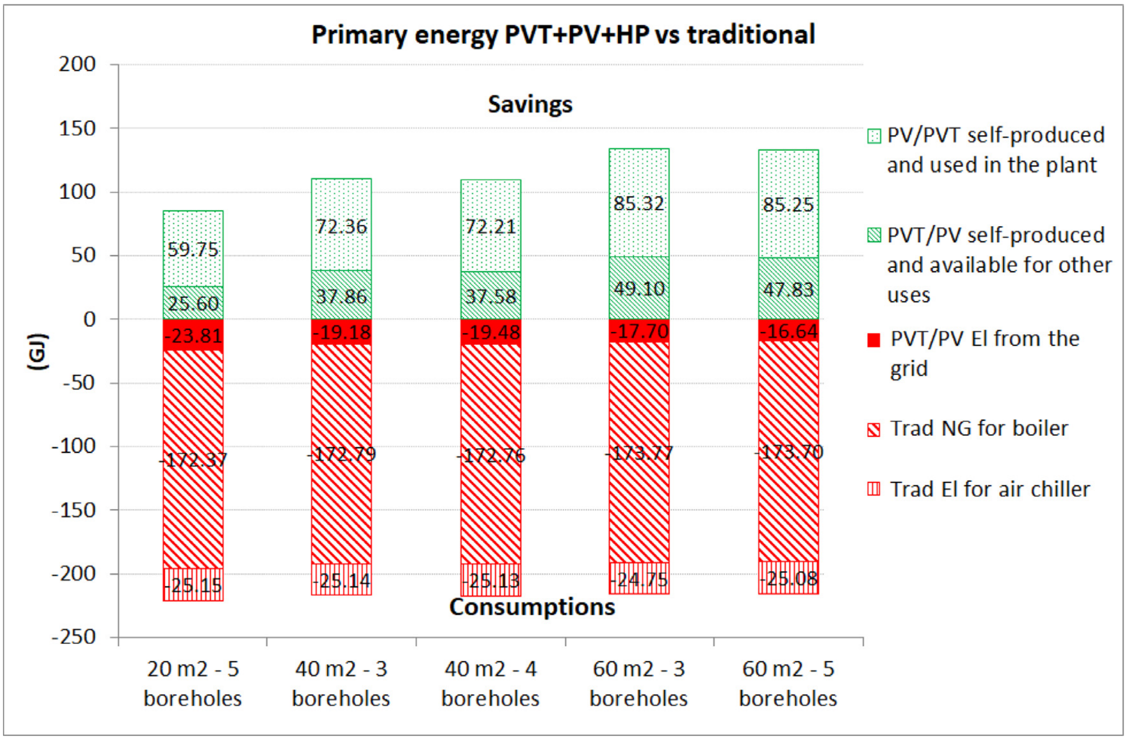

Fig. 17. Non-renewable primary energy comparison between the five considered different alternatives, each compared with the conventional solution.

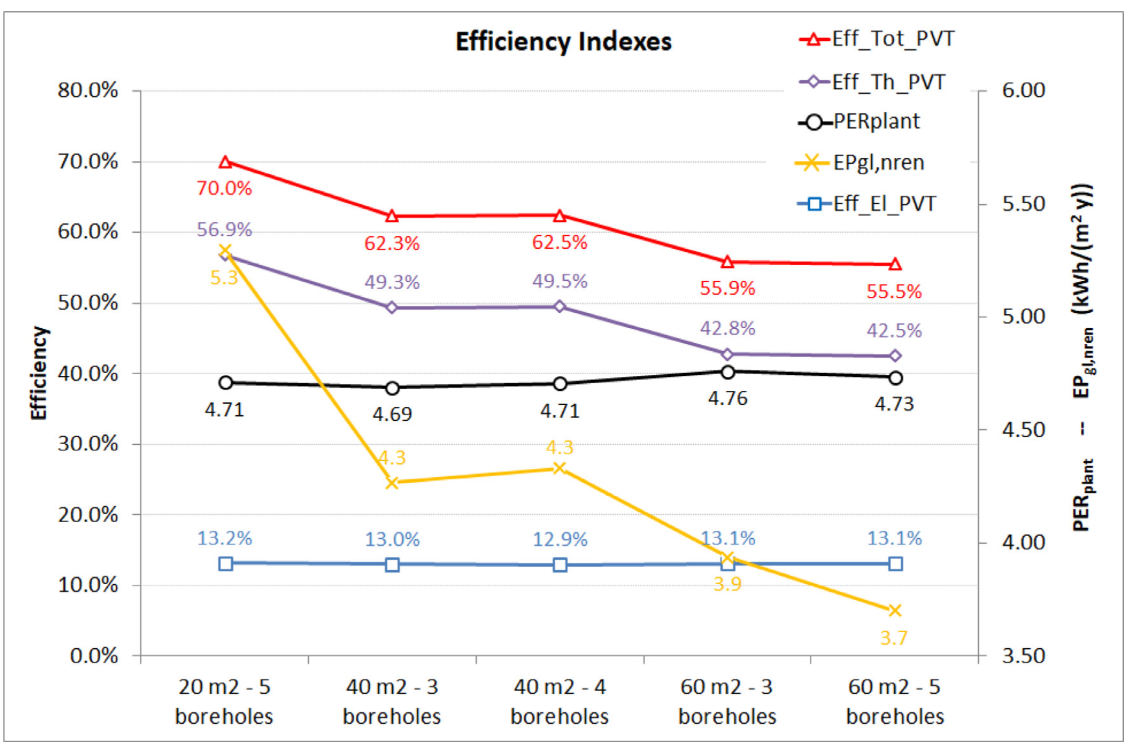

Fig. 18. Annual values of the thermal, electric and total efficiency of the PVT plant, PER of the whole plant, and the specific non-renewable primary energy consumption.

the slightly higher or lower temperature that longer probes may offer to the evaporator or condenser heat pump respectively.

3.6. Economic comparison

As high initial costs are common to renewable energy installations, a more comprehensive analysis should take into account full costs (investment and operative costs in the lifetime of the plant). A simplified economic analysis is here conducted on the base of investment and operative costs for the 60 m^2 PVT – 300 m boreholes – 60 m^2 plain PV solution (that allows the best annual balance between electrical energy self-produced and withdrawn from the grid, and so the highest independency from the grid) compared with the traditional solution.

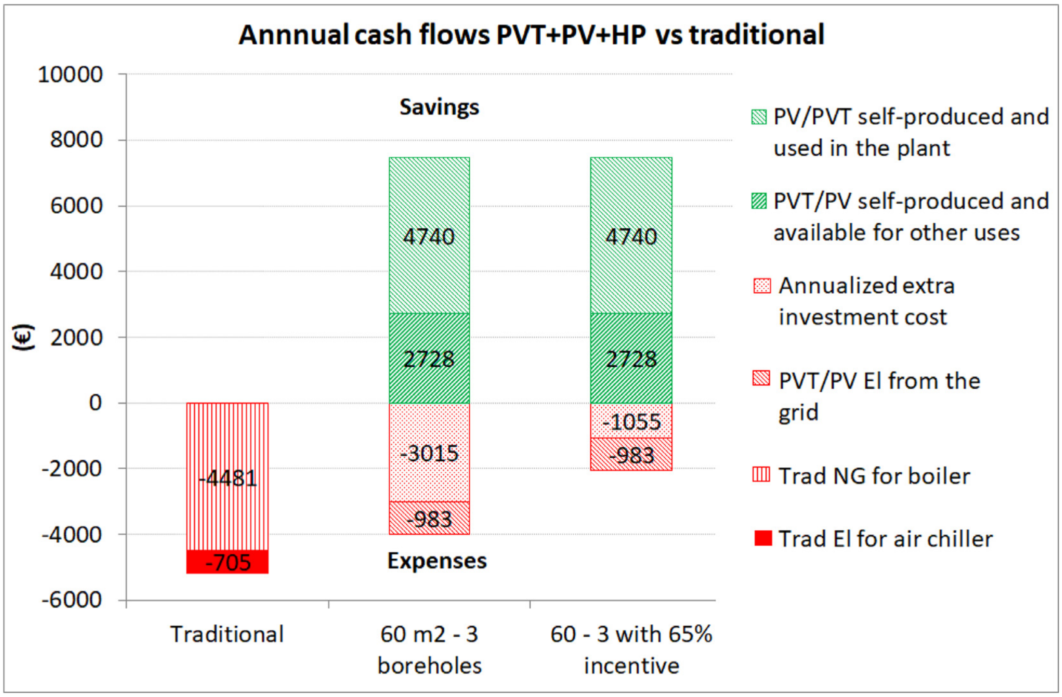

Fig. 19. Annual economic savings obtained by the electricity produced (split into "used in the plant" and "available for other uses") and expenses for the electricity from the grid and for the annualized extra-investment cost with respect to the traditional solution for the preferred alternative, in comparison with the expenses for the electricity from the grid (for the air chiller) and NG (for the boiler) of the "traditional solution". The case with the 65% economic incentive is reported as well.

With regards to investment costs, referring to literature average values [27–30] a reasonable estimate of PV and PVT panels specific cost is 230 € m^{-2} and 400 € m^{-2} respectively. Such a cost is composed by the cost of the module, the cost of the inverter, and other costs (design, installation, frame of the plant, electrical system). Cost of the ground field is calculated considering 10,000 € of fixed cost plus 25 € per meter of ground heat exchanger.

A reasonable hypothesis concerns the cost of other equipment (heat pump, chiller, boiler, pumps, tubes, valves, etc.) that is supposed to be the same for both the dual source heat pump alternative and the traditional solution. Then they are not included in the analysis. Further hypotheses concern the annual interest rate (2%) and the period of the economic analysis (20 years) useful to calculate the actualization factor for the investment costs before mentioned.

The comparison between the proposed solution (60 m^2 – 3 boreholes) and the traditional in terms of annual cash flows is reported in Fig. 19. The former allows a net annual saving of 3470 €, 7468 € of saving with an annual expense of 983 € for the electricity from the grid + 3015 € for the extra-investment cost with respect to the traditional plant. The latter would imply an expense of 705 € for electricity (air/water chiller and pumps) and of 4481 € for NG consumption. The unitary cost of electricity and NG is supposed at 0.2 € kWh^{-1} and 0.9 € Sm^{-3} respectively. The high PER of the proposed plant makes it viable also from the economic point of view, above all considering the economic incentive present in Italy ("Conto Energia Termico 2.0'') (Fig. 19). This incentive provides 65% of the plant total investment cost when the refurbishment of a building leads to a NZEB [31]. For public buildings like schools, such an incentive can be allocated already at the beginning of the intervention.

In terms of discounted payback period (DPB), all the alternatives allow to get an acceptable value, between 5 and 6 years, whereas the most efficient solution features the shortest payback period (5 years). However, consider that the shortest DPB criterion is short sighted for a HVAC plant. In terms of differential (with respect to traditional solution) discounted net present worth (NPW), the most profitable solution is still the multi-source HP plant with 60 m^2 of PVT and 300 m boreholes (+60 m^2 plain PV), that allows to get around 136 k€ in 20 years.

4. Conclusions

The use of glazed PVT offers a technical solution really suitable in view of NZEB design. This is demonstrated by the good energy performance of the plant designed in this study for a refurbished building, coupled to the impressive PV (and then PVT) cost reduction in the last years and the possibility of preventing the overheating in a dual source heat pump combination. From this point of view, the case study here considered is typical: the refurbished building is located in a rather severe climate in wintertime, and the use of the dual source (ground + glazed PVT) heat pump allows very high energy performance and economic viability. The here proposed configuration provides ground regeneration by cooling the PVT mainly during summer, thus keeping the electric efficiency of the cells near to peak value. Moreover, a suitable disposition of storage tanks allows to decouple the heat sources from the heat pump. This is useful in the presence of contemporaneous heating and cooling building loads as in the case here considered, when the heat pump operates as chiller with useful heat recovery.

The design of the plant by means of dynamic simulation considers five alternatives by increasing the solar field (20-40-60 m^2) and decreasing the ground field (500-400-300 m), compared to a traditional solution (NG boiler + air/water chiller). The dynamic simulations by Trnsys revealed that the most efficient solution (highest PERplant and solar fractions, lowest EPgl,nren) is the alternative 60 m^2 PVT + 300 m boreholes. Increasing the solar field to a certain extent permits to reduce the ground field extension with better performances and lower costs, as confirmed also by previous Authors' study [32].

The designed plant proves to be self-sufficient for the electricity on a yearly basis, even exporting electricity to other uses of the building (laboratory equipment, computers, lighting and so on) or to the grid.

CRediT authorship contribution statement

Renato Lazzarin: Conceptualization, Methodology, Software, Data curation, Writing - original draft, Supervision, Writing - review & editing. **Marco Noro:** Conceptualization, Methodology, Software, Data curation, Writing - original draft, Supervision, Writing - review & editing.

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