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Energy Performance of Annual Operation of Heat Pump Coupled with Ground Ice Storage and Photovoltaic/Thermal modules

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Abstract. The use of heat pumps for annual climatization is a suitable mean of achieving the 2030 European decarbonization target (-55% with respect to 1990 CO_2 emissions). The use of seasonal energy storages allows to store one of the two contemporaneous effects (thermal and cooling energy) produced by the heat pump to be used successively when it is necessary. This paper focuses on a dynamic simulation to size the heating and cooling plant and define a suitable control logic for a refurbished building located in northern Italy. The plant is set up by an electric heat pump in annual operation, coupled with a ground ice storage. Ice produced during heating operation is used in summer to face cooling loads (free cooling). PhotoVoltaic/Thermal modules allow to increase the grid independency and to reduce primary energy consumption of the plant as they can be suitably cooled in any season, by recovering heat for domestic hot water or the ground. A dynamic simulation of the system allows for a full description of the behaviour of the ice tank during the charging and discharging processes. In addition, a primary energy performance analysis of the whole plant is reported, also in comparison to a dual source heat pump configuration for the same building.

1. Introduction

Heat pumps are one of the most important solutions considered by the European Commission (EC) to face global climate change by means of the European Green Deal [1] and, more recently, by the RePowerEu strategic plan [2]. EC aims to eliminate the natural gas boilers by 2029 and double the market for heat pumps by getting 10 million individual heat pumps by the next 5 years.

Energy storages can be useful to limit the size and improve the annual operation efficiency of a heat pump due to better matching between the energy supply and demand [3]. Even if several kinds of storage are available, thermal energy storage (TES) is one of the key technologies for efficient energy use, as it is best suited for thermal heating and cooling applications, above all in buildings where heating and cooling energy needs are present contemporaneously [4] [5].

Thermal energy can be stored in three different ways, i.e., sensible, latent, and thermochemical storage. Depending on the temperature range and application, many types of substances are available in each way [6]. In the past, the authors of the present article studied the use of latent storage by the use of Phase Change Materials (PCM) in solar cooling and heating plants [7] [8]. Regarding storage period, TES can be short term (hours, diurnal) or long term (seasonal) [9].

A heat pump produces both heating and cooling effect during its operation; for such a reason in temperate climates, it could be of great interest to store one of the two to be used successively. Therefore, seasonal TES can be of interest from an energy performance point of view. The main issue to face is to provide a storage of sufficient size to store, for example, the optimum quantity of cooling energy produced at the evaporator during a whole heating period, and then to release it during the following cooling season. Xu et al. reviewed three different technologies (sensible, latent, and chemical) for seasonal heat storage in solar systems and related projects [10]. In [11], different seasonal thermal energy storage methods using a heat pump were compared in terms of coefficient of performance (COP) and solar fraction varying collector area and storage volume, and for different types of buildings. Sommer et al. [12] developed a simplified hydrogeological model to determine the thermal performance of the large-scale application of aquifer thermal energy storages considering the influence of well-to-well distances, the role of aquifer thickness, thermal radius, and heat loss. Moreover, the method proposed by the authors allowed to determine the amount of thermal interference that is acceptable from an economical and environmental perspective.

Among seasonal TES, ground ice storages (I-TES) can be coupled with heat pumps, providing some positive effects in unbalanced thermal loads buildings. In fact, the ice formed during winter by the heat released by the stored water to the heat pump evaporator can be used during summer, when the ice can liquefy, allowing free cooling. In an annual energy analysis, the energy spent and related costs to store this potential in the long term should be taken into account. This solution could reduce the nominal cooling power of the chiller and save energy in the cooling season, with advantages from an energy savings point of view. Very few studies are present in such a topic. For example, a heating and cooling system with heat pumps, solar thermal collectors, ice thermal energy storage, and borehole thermal energy storage was presented for a building complex in Oslo, Norway, [13]. Moreover, D'Ingeo presented the case study of a ground ice storage coupled to a gas-fired absorption reversible heat pump for the annual climatization of a commercial building [14]. In [15], one year of operation of a pilot plant allowed the validation of the simulation model of an ice storage buried in the ground. The annual simulations of different control logic, areas and types of solar collectors and volumes of ice storages for the cities of Strasbourg [16] and Zurich [17] allowed to analyze the influence on system performance. In the past, the authors of the present paper presented a study by annual dynamic simulations by TRNSYS 17 [18] of a reference residential building located in Milan (North of Italy) to evaluate various aspects of a ground ice storage (modeled by TRNSYS type 343) coupled with a reversible heat pump system [19]: the correct size of the storage in relation to the building heating and cooling demands, the optimum thickness and position of thermal insulation of the storage, and the best shape of ground ice storage were evaluated.

1.1. Motivation for this study

To the best of our knowledge, very few studies are available in literature on using TRNSYS and type 343 to simulate ground ice storage coupled with a heat pump in annual operation, and none features the coupling with PhotoVoltaic Thermal (PVT) modules. A PVT module is a hybrid collector that exploits the thermal fraction, as PV cells are connected to a device that exchanges heat with a fluid (usually air or water) [20]-[22]. Liquid-based PVT collectors are largely more diffused than air-cooled: in the typical configuration, a metallic sheet and tube absorber extracts heat by forced fluid circulation through series / parallel connected pipes adhered to the rear of the PV collector [23]. More recently, nanofluids have been investigated to improve overall performance [24]. Thermal levels can provide domestic hot water (DHW) or ambient heating, but utilization also as heat pump source can be useful, as it is a correct compromise between useful energy and a moderate PV temperature.

Lazzarin and Noro [25] have already presented a dynamic simulation of a PVT dual source reversible heat pump plant. The system was proposed for the retrofitting of the gymnasium of a high school near Belluno (North-East of Italy). In that case, ground and glazed PVT were used as source/sink, with the latter also driving the heat pump compressor. In this paper, the authors propose a variation of that scheme: instead of the ground (vertical probes) + the heat produced by the PVT system, an underground ice thermal energy storage acts as heat source of the electric heat pump. The advantage is the production of ice, which can be used in summer for cooling (free cooling). In the proposed scheme, the thermal energy produced by PVT is used for DHW production. The eventual excess of heat produced by the PVT and the condenser of the chiller (when the latter is in operation if the ice should not be sufficient to face the cooling loads of the building) is released to the ground. Regarding the plant configuration studied in [25], no multisource heat pump is designed, and the ground acts only as a heat sink, so horizontal probes (less expensive than vertical) are provided.

The purposes of the study are the following:

- to choose the configuration (ground ice storage volume and shape, and thickness of the thermal insulation) that optimizes the energy performance of the system (in terms of non-renewable primary energy ratio, non-renewable primary energy specific consumption, electrical independency from the grid);
- to compare the energy performance of the system with respect to the original configuration (that of reference [25]) and a traditional one;
- to extend the comparison also from an economic point of view.

As the main novelty of this study, dynamic simulations by TRNSYS with a time step of 15 min using a specific type for buried vertically stratified thermal energy storage (type 343) are used to set up a suitable scheme of the HVAC plant and its control logic to face the contemporaneous demands of DHW, heating, and cooling. Furthermore, results in terms of ice volume fraction inside the storage for two years of operation are presented and the optimization of the energy performance of the plant is reported. These can give useful advice for the first design of the system.

The rest of the paper is organized into the following sections: Section 2 reports on the modeling hypotheses of the HVAC plant and the working mode of the ground ice storage heat pump system; Section 3 reports the main results of the simulations as a comparison between the different alternatives based on energy indexes, the ice volume fraction, and the economic analysis. The best volume of the I-TES and size of PVT field are researched as well. Finally, in Section 4, some remarks and conclusions are reported.

2. Methods

2.1. The building model

The building is part of an old high school building (completed in 1960) located near Belluno, North-East of Italy (46°1’ N, 11°54’ E). The climate is rather severe in winter (3100 degree-days). A large gym (33 m x 25 m x 8.40 m) on two levels is the main part, bathrooms with showers and toilets, changing rooms, and technical rooms are located at the ground floor. At the first floor there are an office, a small gym, and a bar, whereas at the second floor six laboratories are going to be refurbished with the aim of constructing a nearly zero energy building (NZEB).

In the refurbishment, the outer walls and the roof are going to be carefully insulated, with an average thermal transmittance of approximately 0.15 W m^{-2} K^{-1}. The glazing system and the floor to the ground will have a thermal transmittance of 0.7 W m^{-2} K^{-1} and 0.5 W m^{-2} K^{-1} respectively.

The building (total floor area 2435 m^2, enclosed gross heated volume 11060 m^3) is divided into 20 thermal zones in the TRNSYS model. The demands of space cooling, heating, and DHW are satisfied by the HVAC system, which also includes two air handling units (AHU) for ventilation (Figure 1).

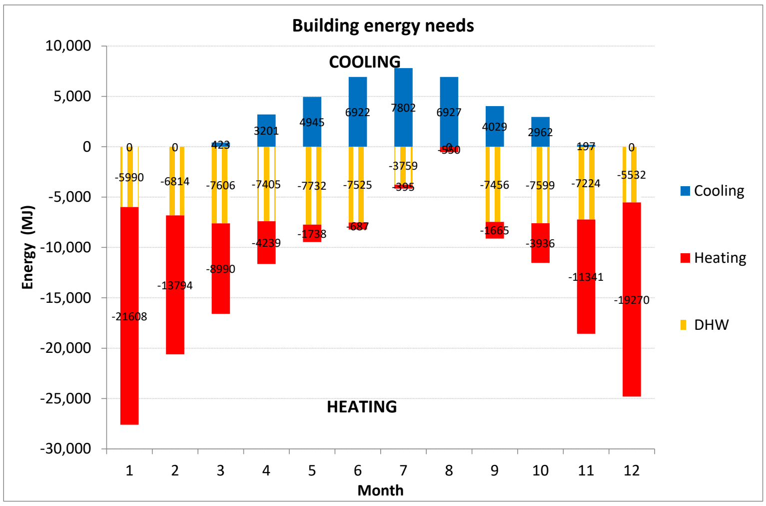


Figure 1. Monthly energy needs in terms of heating, cooling, and DHW.

Some words to describe the results of Figure 1: in the second half of January there is the maximum heating load (-44.6 kW), whereas the maximum cooling load (21.6 kW) is at the beginning of June (the building is still fully operating, that is, open to students and the professors, and gym is also open to extra-school activities). The (small) heating energy need present during the summer months is dedicated to post-heating coils in the AHUs. The DHW need (2000 L per day at 45 °C) is a considerable quota of the total heat demand, especially in the mid-season months.

2.2. The HVAC plant model

The original feature of the HVAC plant is the ground ice tank that provides the heat source of the water/water heat pump, which satisfies mainly the heating load. Ice formed during winter season is stored to be used in the mid and summer months to face the cooling load. The solar section is made up of glazed PVT (variable area, 60 m^2 or 30 m^2) and plain PV (60 m^2 area); these values were previously found to be the optimum [25]. The PVT thermal energy provides DHW heating or at least pre-heating when possible, whereas the electricity produced by the PVT and PV is mainly used to drive the heat pump and the auxiliaries of the plant (pumps and electric resistors of the hot tanks).

Figure 2 reports a simplified scheme of the plant that is set up by five main loops:

- PVT - Ground loop: it is dedicated to exchange the excess heat from the PVT field by means of a horizontal ground heat exchanger. This situation occurs mainly in summer and/or when the Pre-Heating DHW Tank and DHW Tank are already satisfied (i.e., T_{dhw}>45 °C). The P1 pump is activated if global solar radiation in the plane of the panels is greater than 50 W m^{-2} and, after 2 min, if the temperature of the PVT outlet is greater than 30 °C;
- DHW loop: 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 (i.e., the PVT output temperature is greater than the Pre-Heating DHW Tank output temperature), it is heated by the PVT cooling water. If the set point temperature of the DHW Tank (45 °C) is already reached (eventually supplemented via the HEX1 heat exchanger by the Hot Tank), water from the PVT is directed to the ground to be cooled (say below 30 °C). 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;
- Heat Pump - Chiller loop: Hot Tank satisfies the heating load, receiving heat by the heat pump condenser. The Ice Tank is the heat pump cold source, and it produces ice that can be usefully liquified in summer to face the cooling load, thus producing a free cooling effect. The Cold Tank is dedicated to cooling load. It is cooled by the Ice Tank or, if necessary, by the heat pump

evaporator. Even in summer, useful heat for post-heating of AHU or DHW heating can be provided by the heat pump condenser by means of the Hot Tank; at the same time, the excess heat from the PVT or heat pump condenser is exchanged to the ground, maintaining at an acceptable temperature the glazed PVT;

- Heating – DHW loop: it is the circuit that allows the Hot Tank to contribute to the DHW and the heating load;
- Ice Tank loop: it operates mainly in summer, allowing to cool the Cold Tank (free cooling).

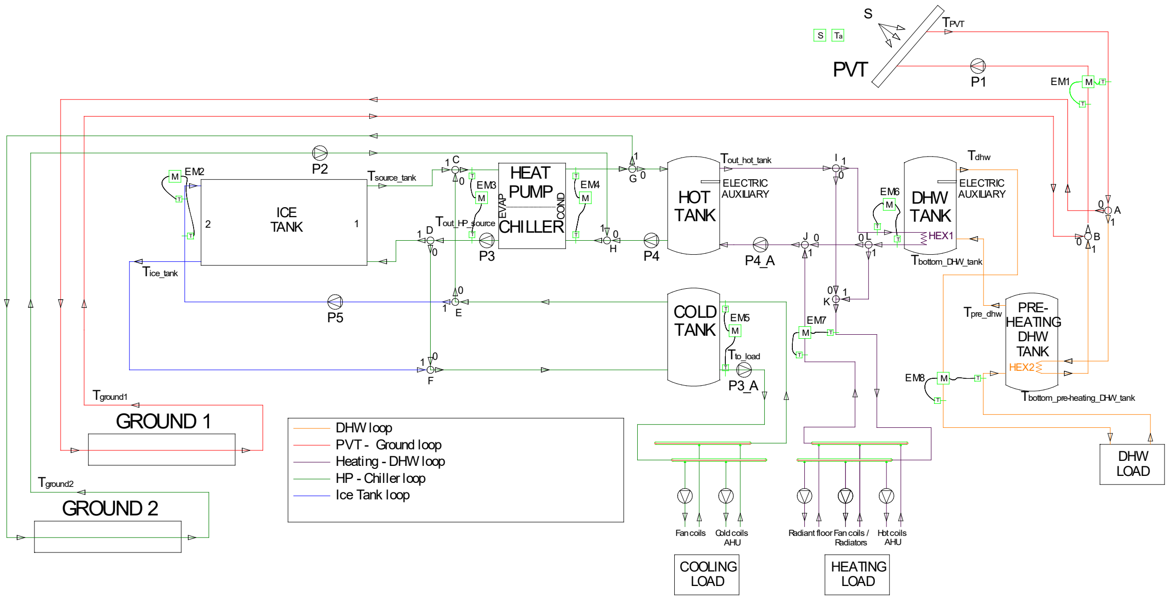


Figure 2. Simplified scheme of the HVAC plant.

2.3. *I-TES Type*

Type 343 models an underground cylindrical (or conical) with circular bases stratified I-TES. In our case, the storage is vertically subdivided into six elements, each one is passed by two different circuits (Figure 2): circuit 1, connected to the HP evaporator, charges the storage (ice) during the heating operation of the system; circuit 2, connected to the cooling load, discharges the storage, allowing for using the ice during the cooling operation. An insulation layer on the top is settable by the type (two very different values are considered in the simulations, 5 and 20 cm). Setting the thermophysical properties of the storage materials and the soil, and the heat transfer and absorption coefficients at the ground surface allows to determine the heat losses/gains of the storage.

Equation (1) allows to determine the volume of the storage by considering the annual amount of energy exchanged by the HP evaporator with the heat source during the heating season; in the base case the volume results to be 621 m^3:

$$V_{I-TES} = \frac{E_{evap,HP}}{\left(\rho_{H_2O} \cdot c_{H_2O} \cdot \Delta T_{H_2O} + \rho_{ice} \cdot r_{ice}\right)} = \frac{208}{(1000 \cdot 4.187 \cdot (7-0) + 917 \cdot 333.5)} = 621 \text{ m}^3 \quad (1)$$

where:

$V_{I\text{-}TES}$ = volume of I-TES (m^3);

$E_{evap,HP}$ = energy exchanged by the HP evaporator during heating operation as a result of the simulations (GJ);

ρ_{H2O} = liquid water density (kg m^{-3});

c_{H2O} = liquid water specific heat (kJ kg^{-1} K^{-1});

ΔT_{H2O} = temperature difference between the minimum for the cooling operation and the freezing (K);
ρ_{ice} = solid water density (kg m^{-3});
r_{ice} = latent heat from the solidification water (kJ kg^{-1}).

In the base case, the cylindrical shape is considered with a base diameter equal to 10.5 m; the height is determined by 6 elements of 1.2 height each. In each element, a 16 mm ID polyethylene tube, 120 m long, was located to exchange the requested amount of heat during the operation. The structure is made of 0.2-m thick concrete walls, and it is located 2 m under the ground surface. Polyethylene is considered as insulation material on the top of storage: as a matter of fact, placing the insultation on the bottom or on the side of the storage does not allow for better energy performance [19].

2.4. Systems to be compared

Table 1 reports the different alternatives considered here. Alternative n. 2 is the base case as previously described, whereas the alternatives n. 1 and n. 3 differ only for the size of the I-TES (respectively a reduced and a greater diameter). The main idea is to test the energy performance of the system by varying the volume of the I-TES. Alternative n. 4 tests an increase in the thickness of the insultation material (from 5 to 20 cm), whereas case n. 5 investigates the energy performance with a different diameter/height ratio but the same volume of the I-TES with respect to the base case. All alternatives n. 1-5 are set with 60 m^2 of glazed PVT to be comparable with the previous configuration of the plant studied in [25]: in case n. 6, 30 m^2 area of PVT is set up to test the performance of the system with a smaller renewable thermal and electric energy production.

Table 1. Description of the alternatives considered in the present study.

Alt. n.	Alternative	Diameter I-TES (m)	Insulation material thickness (cm)	Solar field PVT (m^2)	Height of the six elements (m)
1	8m – 5cm – 60PVT	8	5	60	1.2
2	105m – 5cm – 60PVT	10.5	5	60	1.2
3	125m – 5cm – 60PVT	12.5	5	60	1.2
4	105m – 20cm – 60PVT	10.5	20	60	1.2
5	125m – 5cm – 60PVT - short	12.5	5	60	0.847
6	105m – 5cm – 30PVT	10.5	5	30	1.2

To quantify the non-renewable primary energy savings achievable with the use of the I-TES heat pump coupled with the PVT system, a reference system is considered equipped with a natural gas-fired condensing boiler (100% mean efficiency on lower heating value) and an air-cooled chiller [25]. Table 2 reports the efficiency indices for the energy analysis reported in the next Section3.

Table 2. Energy performance indices (refer also to Figure 2).

Index	Description	Unit
PER_{syst}	Primary Energy Ratio of the whole plant (electrical efficiency η_{el} = 1/1.95 = 51.3% by the Italian Decree DM 26/06/2015) PER_{syst} = $= ((EM_4 \text{ (when G=H=0)} + EM_5 + E_{HEX2} + E_{res}) / (EM_{HP} + EM_i + E_{res}))\cdot \eta_{el}$	-
$EP_{gl,nren}$	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 the thermal energy produced by PVT and the total solar radiation on the collector plane (A_{PVT}: aperture area of the PVT field) $\eta_{th\_PVT} = EM_1 / (S{\cdot}A_{PVT})$	-
η_{el_PVT}	Solar electric efficiency: ratio between electric energy produced by PVT and total solar radiation on the collectors plane (A_{PVT}: aperture area of the PVT field) $\eta_{el\_PVT} = EM_{PVT} / (S{\cdot}A_{PVT})$	-
η_{tot_PVT}	Solar total efficiency: ratio between thermal + electric energy produced by PVT and total solar radiation on the collector plane (A_{PVT}: aperture area of the PVT field) $\eta_{tot\_PVT} = (EM_1 + EM_{PVT}) / (S{\cdot}A_{PVT}) = \eta_{th\_PVT} + \eta_{el\_PVT}$	-

All details about the types used in the TRNSYS model, the thermal and electric characteristics of PVT and PV, the nominal data of the heat pump and the power consumption and mass flow rate of the pumps are reported in [25].

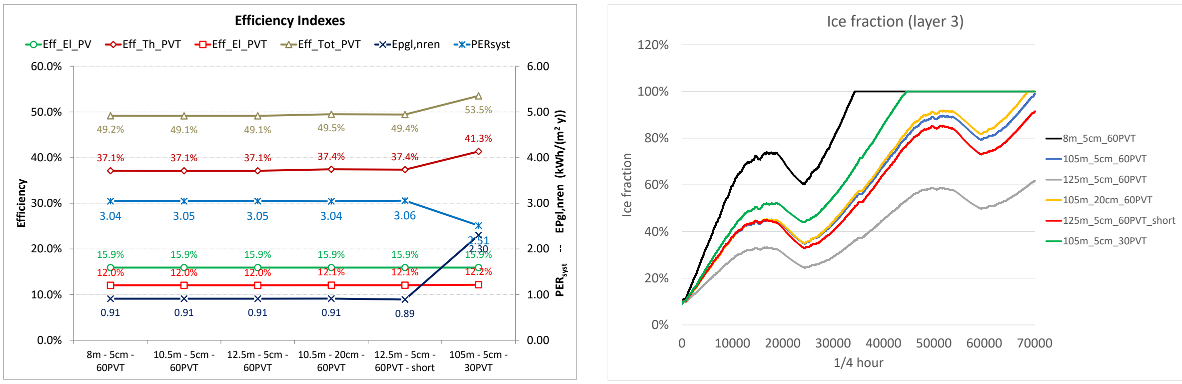
3. Results and Discussion

First, a comparison between the different alternatives in terms of energy performance and ice volume fraction is reported on an annual basis (Section 3.1). Successively, the main results of the monthly analysis are discussed for the most favorable alternative (Section 3.2). Finally, a simplified analysis of the different alternatives from an economic point of view is reported (Section 3.3).

3.1. Annual energy performance

In terms of the global efficiency of the plant, the volume and configuration of the I-TES have a very limited influence: the best configuration is n. 5 with the highest PER_{syst} (3.06) and the lowest $EP_{gl,nren}$ (0.89 kWh m^{-2} y^{-1}) (Figure 3(a)). The reduction of the volume of the I-TES (alt. n. 1 vs. alt. 2) is not beneficial because the storage saturates quicker. This is apparent in Figure 3(b) where the ice fraction of the layer (segment) n. 3 during two years of simulation is reported: by reducing the volume (alt. n. 1, black line) the 100% ice fraction is reached earlier with respect to the base case (alt. n. 2, blue line); on the contrary, increasing the diameter of the ice storage (alt. n. 3) reduces the ice fraction during the simulation (grey line). The effect of a faster saturation of I-TES is a slight decrease of the heat pump COP in the following heating season as a result of the lower evaporation temperature. Increasing the thickness of the insulation material (alt. n. 4 in Figure 3(a), yellow line in Figure 3(b)) does not allow the plant to substantially achieve better energy performance for the same reasons. Instead, increasing the diameter/height ratio of the I-TES is beneficial (alt. n. 5 vs. n. 2). In fact, by varying the insulation and the diameter/height ratio of the I-TES, it is possible to control the amount of energy stored (transformed into ice) and the amount dissipated through the boundaries. The formed ice is then used during the summer to satisfy the requested cooling loads, and the quantity of ice remaining at the end of the cooling season will affect the energy performance of the I-TES heat pump coupled system in the following winter.

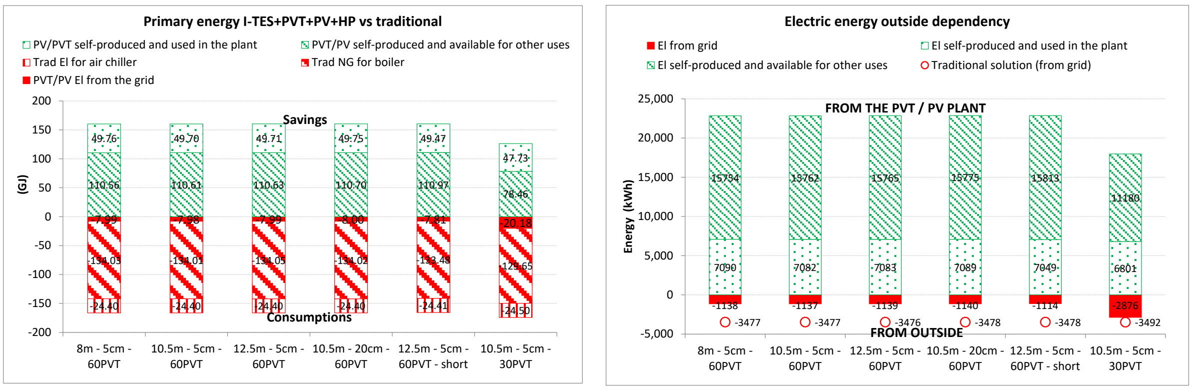
Reducing the PVT area (case n. 6) is not beneficial because the same I-TES volume (case n. 2) is saturated much faster (green line in Figure 3(b)) because the heat pump has to operate for longer during the year to satisfy the quota of DWH load that the PVT does not cover.



(a) (b)

Figure 3. (a) Annual values of the efficiency indexes (Eff_El_PV=PV electrical efficiency; Eff_Th_PVT=PVT thermal efficiency; Eff_El_PVT=PVT electrical efficiency; Eff_Tot_PVT=PVT total efficiency; EPgl,nren=annual specific consumption of non-renewable primary energy; PERsyst=primary energy ratio of the whole system); (b) ice fraction of layer 3 during two years of simulation.

Figure 4(a) reports the non-renewable primary energy balance of the different alternatives compared to the reference solution, considering that the latter uses the same pumps as the proposed one (except for the solar pump, P1, and the P5 in Figure 2). The bars reveal a relevant negative item due to the natural gas demand for the conventional plant (134 GJ), with a comparatively small consumption of the air-cooled chiller (24.4 GJ). The proposed plant takes some electricity from the grid only in the cold months, around 8 GJ that becomes 20.2 GJ in alt. n. 6 (respectively, 1140 and 2880 kWh_{el} in Figure 4(b)), offering from March to October an energy surplus around 111 GJ (around 15800 kWh_{el}) available for other electric uses of the building. On an annual basis, the traditional plant primary energy consumption is 158 GJ, whereas the proposed plant has a surplus of 111 GJ (around 78 GJ in case n. 6). This is even greater than the primary energy surplus of 49 GJ reported by the best configuration of the multi-source heat pump system in the previous study [25]. The better performance of the scheme proposed in this study compared to the one previously studied [25] is also confirmed by the lower specific annual non-renewable primary energy consumption of the plant (0.89 vs 3.9 kWh m^{-2} y^{-1}).



(a) (b)

Figure 4. Comparison of the different alternatives with the traditional one on the basis of (a) non-renewable primary energy consumption/saving; (b) electricity from the grid and from the PVT and PV plants.

3.2. Monthly energy performance of the best alternative

As stated in the previous section, the best configuration is n. 5 with the highest PER_{syst} (3.06) and the lowest $EP_{gl,nren}$ (0.89 kWh m^{-2} y^{-1}). Obviously, the best performances occur during the summer months (Figure 5), when free cooling is possible. Anyway, the primary energy ratio of the system is always greater than 2 during the year.

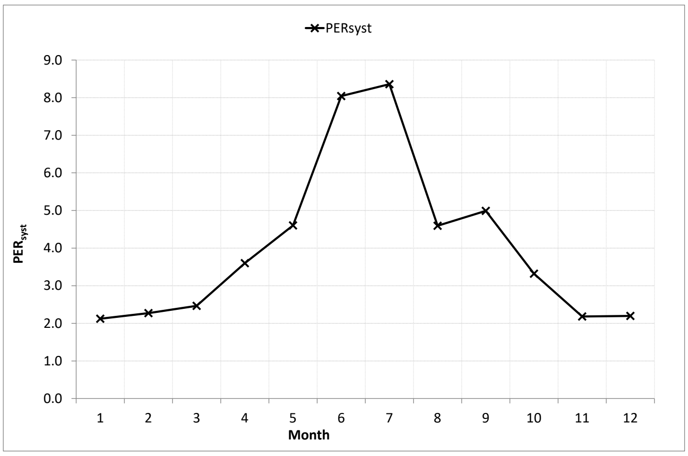


Figure 5. Monthly non-renewable primary energy ratio of the whole system for the best alternative.

Figure 6(a) reports the solar energy balance on a monthly basis. Even if thermal losses generally exceed the other items, a good contribution to DHW heating is offered throughout the whole year, but mainly in mid-season months: e.g., in October and November such contribution is 39% and 35% of the global solar ration, respectively. In terms of electricity production, the best performances of the PVT field are during hot months (from May to August) in absolute terms, but during winter months in relative terms (e.g., in July there is a production of 1131 kWh$_{el}$, that is, 10.9% of the solar radiation, whereas in January the electricity production is 882 kWh$_{el}$ that is 14.4%).

Figure 6(b) confirms the significant contribution of PVT to the heating of DHW. PVT satisfies the whole DHW demand from March to October, and the Hot Tank provides the complementary part (auxiliary electric resistance contribution is very limited).

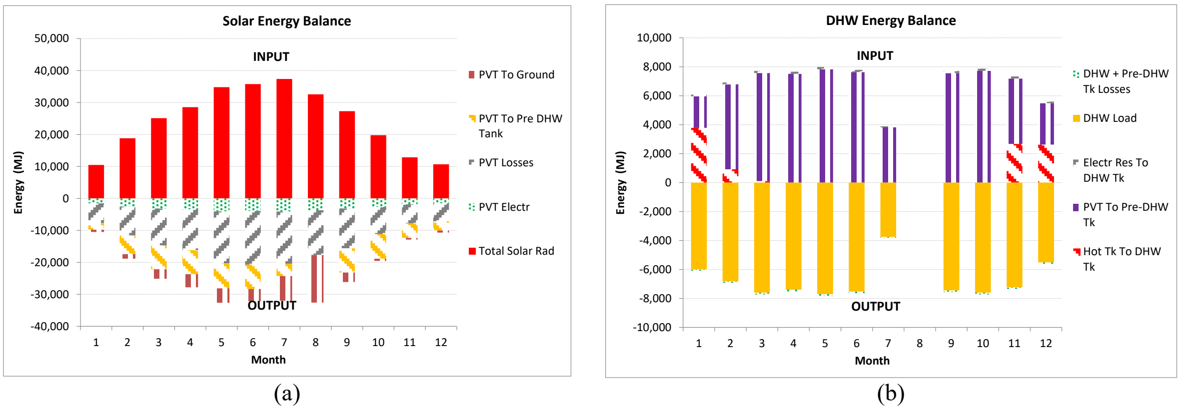
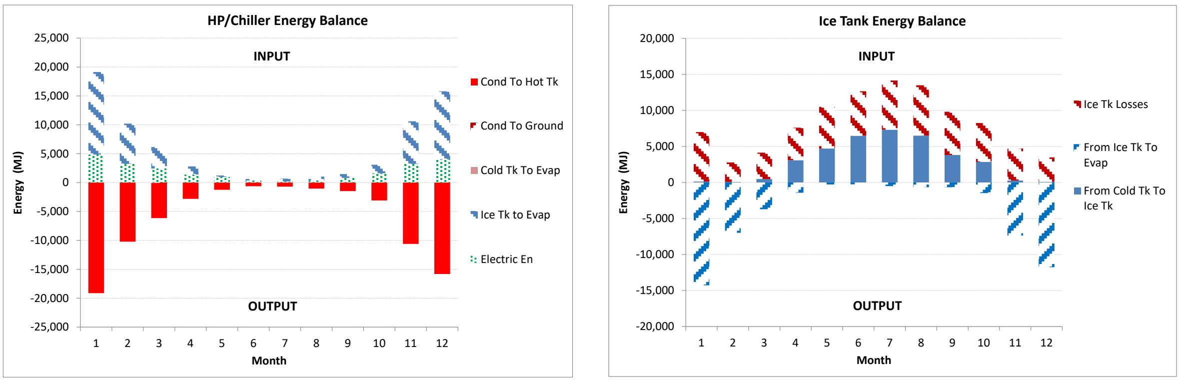


Figure 6. (a) Monthly PVT energy balance; (b) monthly DHW Tank energy balance.

Figure 7(a) reports the monthly energy balance of the heat pump. Its operation is always as a heat pump (mainly from October till April), with the I-TES as cold source and the Hot Tank as heat sink (useful effect). In summer, the heat pump operates in a very limited time supplying the condensation heat to the Hot Tank (a useful heat recovery for DHW and for contributing to the limited heat loads of the hot coils of AHUs), whereas no energy is directed from the condenser to the ground. As a matter of fact, the I-TES is suitable designed to provide full free cooling, and no electric chiller is required during the cooling season (Figure 7(b)).



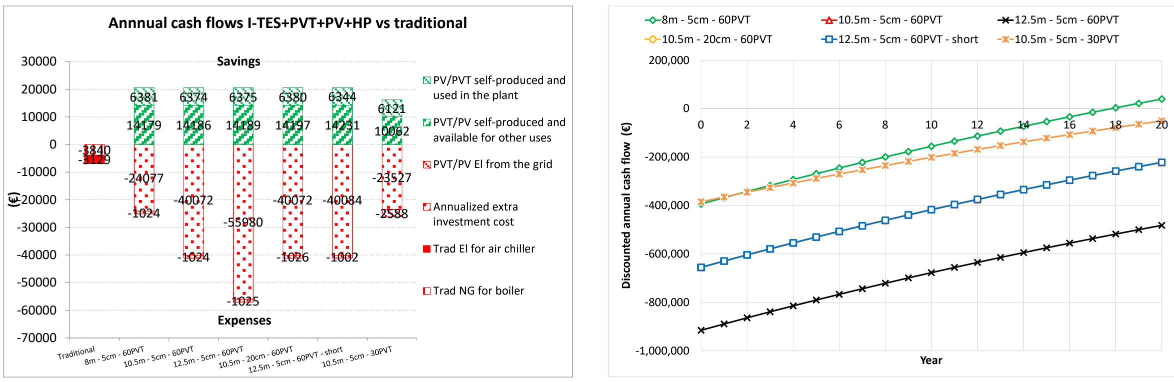
(a) (b)

Figure 7. (a) Monthly heat pump energy balance; (b) monthly I-TES energy balance.

3.3. Economic comparison

A simplified economic analysis is provided taking into account the investment and operative costs of the different alternatives compared to the traditional solution. With regards to investment costs, a reasonable estimate of the specific cost of PVT and PV modules is 300 € m^{-2} and 230 € m^{-2} respectively (full system). Regarding the I-TES, an investment cost of 1000 € m^{-3} is considered. The costs of other equipment (heat pump, chiller, boiler, pumps, tubes, valves, etc.) are not included in the analysis as they are supposed to be the same for both the I-TES+PVT+PV heat pump alternative and the traditional solution. An annual interest rate of 2%, a period of the economic analysis of 20 years, and a unitary cost of electricity and NG of 0.9 € kWh$_{el}^{-1}$ and 1 € Sm^{-3} respectively are fixed.

The comparison between the proposed solutions and the traditional in terms of annual cash flows is reported in Figure 8(a). The I-TES+PVT+PV allows a net annual electricity savings of more than 19500 € (20500 € of saving + 1000 € of expense for the electricity from the grid), but there is an annualized extra-investment cost with respect to the traditional plant that is always higher than the savings (in the best alternative, n. 5, it is 40000 €). The traditional solution would imply an expense of 3130 € for electricity (air/water chiller and pumps) and of 3840 € for NG consumption. The higher PER_{syst} of the proposed plant with respect to the traditional solution does not assure an economic viability. This is due to the very high investment cost of underground ice storage. This is also apparent in Figure 8(b) that reports the discounted annual cash flows of the different alternatives over time: only alternative n. 1 (the one with the smallest I-TES volume) has a payback time lower than 20 years. Instead, with respect to the multi-source heat pump configuration analyzed in [25] serving the same building, an economic advantage is present due to the cost savings in vertical ground boreholes.



(a) (b)

Figure 8. (a) Annual cash flows of the different alternatives compared to the traditional one; (b) discounted annual cash flows of the different alternatives over time.

4. Conclusions

The paper analyzes the energy performance of an I-TES integrated with a heat pump coupled with a PVT plant for annual climatization and DHW production of a refurbished building located in a rather severe climate in winter. The cold energy stored in the I-TES during winter is usefully utilized to satisfy the entire cooling load during summer. The simulations allow designing the correct volume and diameter/height ratio of the cylindrical storage as the best annual performance depends on the equilibrium between ice produced during heating season and consumed during cooling season, taking into account the heat gains/losses with the ground. The simulations allow for also the coupling with the correct area of PVT modules in order to obtain the maximum primary energy ratio and the minimum non-renewable primary energy consumption. For the building and climate considered, the best performances are obtained with an I-TES of 621 m^3 with 12.5 m diameter and 5.1 m height with a 5 cm thick insulation material placed on the top wall of the storage, with a PVT field of 60 m^2.

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. An economic analysis reveals that the great investment cost of the ground ice tank does not allow this solution to be advantageous in this case. As a further development, different climates could be investigated, and a sensitivity analysis of the main energy and economic results could be proposed to better understand the best configuration of the plant.

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