Renewable Energy 221 (2024) 119708

Contents lists available at ScienceDirect

Renewable Energy

journal homepage: www.elsevier.com/locate/renene

Energy analysis of multi-source heat pump system: A real case study application

Giuseppe Emmi [a,*], Eleonora Baccega [a], Silvia Cesari [a], Elena Mainardi [b], Michele Bottarelli [a]

[a] Department of Architecture, University of Ferrara, Via Quartieri, 8, 44121 Ferrara, Italy
[b] Department of Engineering, University of Ferrara, Via Saragat, 1, 44121 Ferrara, Italy

ARTICLE INFO

Keywords:
Heat pump
PVT
Multi-source
Geothermal energy
Energy efficiency
Renewable energy

ABSTRACT

In the last years the HVAC sector has seen the growing interest in using the so-called multi-source heat pump systems. This type of plant aims to improve the energy performance of the heat pump. As widely known the heat pump represents the most promise solution in terms of potential for exploitation of renewable energies. The sun and air can be used as alternative sources to the ground during the middle season or when they are available and convenient, while the ground could be used during the peak periods of the seasons keeping good energy performances of the system. The present work summarizes the monitoring results of a large-scale experimental plant used for the air conditioning of a 100 m^2 snack bar in Ferrara, Italy. The energy behaviour of the plant has been monitored for one year and its performance has been compared with the existing plant. The results have proved an energy saving between 20 % and 70 % in cooling mode. In heating the proposed plant is less competitive due to the share of renewable energy considered in the existing heating system. The work shows how these systems are potentially suitable for the electrification and decarbonisation process in the building sector.

1. Introduction

The European building stock includes a large amount of existing and at the same time energy consuming buildings [1]. The building sector accounts for approximately 42 % of the energy used by all the sectors in 2020, making it the major source of greenhouse gas (GHG) emissions and environmental pollution. In Europe the heating and cooling needs in 2020 accounted for about half of its total gross final energy consumption [2]. This condition has persisted for over a decade, despite many actions have been promoted at European and national level to lower these needs and reach the 20 % energy-saving target, the 20 % increase of renewable energy exploitation and 20 % of emission reduction by 2020 [3]. More recently new ambitious environmental targes have been set to 55 % and 100 % GHG emissions reduction by 2030 and 2050 respectively. All the actions in this field must also include activities aimed at improving energy efficiency and increasing renewable energies exploitation, which must respectively achieve an increase of 32.5 % and a share of 32 %.

The heat pump (HP) technology in heating and cooling applications uses electricity to work. This generation system has been widely discussed in the last years as the central technology in the global energy transition and electrification. The HP is considered as the most promising technology for energy security, reducing user costs and for tackling climate change [4]. In fact, the HP can exploit renewable energy from a large variety of heat sources. Furthermore, heat pumps use refrigerants with low or very low GWP (Global Warming Potential) and TEWI (Total Equivalent Warming Impact). For these reasons, the HP system is also considered a promising technology to decarbonise the provision of space heating and cooling in buildings without directly producing GHG and air pollutant emissions. Around 10 % of space heating needs were globally met by HPs in 2021, but the number of installations is growing rapidly with sales at record levels. In the last years an increasing interest has been directed to HP systems capable of exploiting different heat sources in a single plant, the so-called hybrid systems, or multi-source heat pump (MSHP) systems [5,6].

The main idea driving this trend is to make available to the reversible HP a stable high-temperature source or low-temperature sink in heating and cooling respectively. By following this approach many studies, simulations and some experiments have been carried out to demonstrate the reliability and the energy saving potential of these systems [7]. The most common HP system present in the market and in the building sector is the air source heat pump (ASHP) which uses the air as source and sink. The availability of air everywhere and the relatively easy installation make the ASHP system the technology with the greatest

* Corresponding author. Department of Architecture, University of Ferrara, Via Quartieri 8, 44121 Ferrara, Italy.
E-mail addresses: giuseppe.emmi@unife.it (G. Emmi), eleonora.baccega@unife.it (E. Baccega), silvia.cesari@unife.it (S. Cesari), elena.mainardi@unife.it (E. Mainardi), michele.bottarelli@unife.it (M. Bottarelli).

https://doi.org/10.1016/j.renene.2023.119708
Received 15 August 2023; Received in revised form 21 October 2023; Accepted 21 November 2023
Available online 24 November 2023

Nomenclature

AHU	Air Handling Unit
AM	Air Mass
ASHP	Air Source Heat Pump
AW	Air to Water
BHE	Borehole Heat Exchanger
COP	Coefficient Of Performance
DHW	Domestic Hot Water
EER	Energy Efficiency Ratio
GHE	Ground Heat Exchanger
GHG	GreenHouse Gas
GHX	Geothermal Heat Exchanger
GSHP	Ground Source Heat Pump
GWP	Global Warming Potential
HP	Heat Pump
HVAC	Heating Ventilation Air Conditioning
I	Electric Current, [A]
MSHP	Multi-Source Heat Pump
MSHHP	Multi-Source Hybrid Heat Pump
NZEB	Nearly Zero Energy Building
P	Power, [W] or [kW]
PEB	Plus Energy Building
P&ID	Piping and Instrumentation Diagram
PV	PhotoVoltaic
PVT	PhotoVoltaic Thermal
q	Volume flow rate, [L/h]
RF	Radiant Floor
SAGSHP	Solar Assisted Ground Source Heat Pump
SGSHP	Solar-Ground Source Heat Pump
STC	Standard Test Conditions
T	Temperature, [°C]
TEWI	Total Equivalent Warming Impact
V	Voltage, [V]
ZEB	Zero Energy Building

Subscript

A	Air
c	Commercial
cb	Condensing Boiler
e	Experimental
el	Electrical
G	Ground
in	Inlet
m	Measured
mpp	Maximum Power Point
n	Nominal
OC	Open Circuit
out	Outlet
PV	PhotoVoltaic
PVC	PhotoVoltaic Cooling
S	Solar
SC	Short Circuit
th	Thermal
W	Water
WF	Working Fluid

Symbol

Δ	Difference
η	Efficiency

number of installations. On the other hand, its efficiency is significantly affected by ambient air temperature to which is added the defrosting issue at the evaporator side during the heating period. For these reasons, the ASHP systems are not the best solution for the energy efficiency point of view in the cold climates. The MSHP system represents an attractive alternative to the ASHP applications [8].

In literature many studies have investigated this type of HP systems. Si et al. [9] suggested a novel solar-ground source heat pump system (SGSHPS) for an office building. The system was designed for the heating, cooling and tap hot water production. The simulations returned the best results with the ground heat exchangers (GHEs) and the solar thermal collectors installed in series. Deng and Yu [10] investigated a solar and air source heat pump water heater system. They used a mathematical model to evaluate the performance of the proposed HP system, and then compared with that of the conventional one. They concluded that the system has a better performance at a low solar radiation. In the paper [11], Mehrpooya et al. reported the results of a research aimed to the design and optimization of a combined solar collector and ground source heat pump (GSHP) system. The study analysed the system from economical and technical points of view obtained a seasonal coefficient of performance (COP) of 4.14. Razavi et al. [12] propose a Solar Assisted Ground Source Heat Pump (SAGSHP) system for heating and tap hot water production in a residential building in the city of Zahedan, Iran. The system uses the sun and the ground as heat sources for the HP. They investigate the energy and thermal behaviour of the plant by means of simulations. Their system reached COP values of 3.75 and 3.52 as maximum and minimum respectively. Emmi et al. [13] analysed the energy performance of a SAGSHP in cold climates. They use the heat from the solar system to thermally recharge the ground during the cooling period. They found a reduction of about 10 % in seasonal energy performance of the heat pump for each location over the ten years of operating period when the only the ground was used as thermal source for the HP. Han et al. [14] studied a novel multi-source hybrid heat pump (MSHHP) system with seasonal thermal storage. They investigated the differences between the novel MSHHP system and a GSHP system in different climates by using a mathematical model of the heating and air conditioning system. The results of this study show an energy saving of about 30 % of the novel plant against an increase in expenditure of approximately 10 % corresponding to a payback period of 4 years. Chen et al. [15] used a full simulation model of a space heating system assisted by photovoltaic thermal (PVT) collectors and GSHP system. They investigated the energy behaviour considering the energy, environmental, economic, and flexibility performance considering the photovoltaic (PV) contribution, ambient temperature, solar radiation and electricity price. The results of their study showed an energy saving of 32 % with a corresponding cost saving of about 9 % compared to a common GSHP system. Cruz-Peragon et al. [16] have studied the design and the optimization of a hybrid GSHP system. They considered unbalanced thermal load profile of the building the available space for the installation in the design process of the boreholes heat exchangers (BHEs) field of the system. Their plant uses two different HPs with air and ground as heat source respectively. The authors used experimental data for the design of the GHEs field while the whole system was analysed by means of dynamic energy simulations. The optimal layout of the plant has obtained a 33 % energy saving showing 89 % of the total building load provided by the ground source. Besagni et al. [17] presented the results of a solar assisted HP for the air-conditioning and domestic hot water (DHW) production in a real small plant in Milan. The HP used in the plant has two evaporators connected in series for the exploitation of air and hot water as heat sources. The hot water is produced by PVT collectors. They found a significantly increase of the energy performance thanks to the water-source evaporator and to the avoided defrost cycles.

Jie et al. [18] studied the operation of a novel indirect expansion solar-assisted multi-functional HP. The system can provide heating, cooling, and hot water production. They compared the energy behaviour of the system with and without the solar loop. They found the evaporating and condensing heat exchange rate increased by 37.4 % and 32.3 %, respectively by using the solar system. In the same plant of the previous case study Cai et al. [19] investigated the effects of the indirect expansion on their system. They carried out a parametric analysis of the system by using the simulation model. In water heating mode they concluded that the simultaneously operating of the solar loop together with the multifunctional heat pump system can be an energy efficiency method. Furthermore, in space heating mode, the optimal management of the system needs an optimal control of the working fluid mass flow rate according to the temperature level and thermal load.

Lazzarin and Noro [20] studied the energy performance of a MSHP system and they highlighted the fundamental role of the monitoring data as a guide for the optimal management of the system, in particular when the system uses different technologies and sources even in the long term. Sakellariou et al. [21] investigated the energy performance of a SAGSHP system. The system used a PVT solar field and a very shallow GHEs field. Their study was focussed in the sensitivity analysis on the energy conversion side by varying some parameters of the system. The obtained an improvement of the solar production up to 20 % by using the optimal combination of the system's parameters. A similar study was published by Sommerfeldt and Madami [22]. They investigated the energy performance of a multi-family house in a heating-dominated climate. Their work is a parametric study focused on technical and economic aspects of SAGSHP system. They found a possible reduction up to 18 % and 50 % of the boreholes' length installation and spacing by obtaining the same energy performance of the system without the solar loop. They concluded that the small ground space need due by PVT use has a large potential to increase the GSHP market and at the same time it should be a new way to promote solar energy diffusion. Nouri et al. [23] simulated different configurations of SAGSHP for the heating, cooling, and hot water production of a house in Iran. The study investigated three different layouts of the system, two of them with indirect expansion mode, in parallel and series configuration and the third in direct expansion mode. They found that the optimum configuration was the one with indirect expansion in parallel. The system obtained a COP of 3.96 with a corresponding payback of 13 years compared to a conventional HP system. Considering the environmental and natural gas exporting costs, the payback time was reduced to 6 years. They concluded that their system can be used in Iranian cold climate.

As seen in the literature very often the results in most of the research deriving from simulation activities rather than experimental data. On the other hand, when real plants are investigated, the size of the systems are quite small, and the use of the buildings is fictive and not real.

The present research describes the thermal and energy behaviour of a large MSHP system installed in the North of Italy. The case study is a real installation in a University of Ferrara building used as snack bar. As reported in the literature, there is a lack of works and studies addressed at the analysis of real large installations. The goal of this work is to increase and improve knowledge in this field by trying to partially fill the lack of real experiences in this area. The description and the results of the monitoring activities, which aim to enrich the knowledge in this field made it possible to highlight advantages and potential issues related to this technology and similar applications to the one described in the present work. The document compares the proposed MSHP system with the conventional plants also including the use of district heating network and summarizes the energy saving potential of this technology.

2. Methodology

The current literature includes the results of many studies and research about the use of MSHP systems for heating, cooling, and DHW production. As widely known, the energy saving potential of these systems is related to the possibility of using different heat sources and sinks in heating and cooling mode respectively. The HP uses the best source between the available during the operation of the plant to improve the energy performance of the system. According to the type of installation the sources and sinks can be used simultaneously or individually by the HP. Among the many works in this field, most of them are simulation-based studies which investigating the optimal design of the systems and components of the plants. Instead, few jobs are the result of monitoring activities in experimental plants and often the monitoring data are aimed to investigate single part of the plant or small installation at laboratory scale. The experimental data are then used for the calibration of single part of the MSHP system not considering the entire plant. The present work investigates a large real MSHP system and summarizes the results of a monitoring campaign of the first year of operation. The experimental plant was designed and built within the European project IDEAS funded by the H2020 programme [24]. The operation of the system and the exploitation of the renewable energy sources are the results of the management rules developed and used within project for the exploitation of the sources and sinks in such type of system.

2.1. The case study

The present work investigates the energy performance and the exploitation of renewable energy sources and sinks of a large scale experimental MSHP system. The results highlight the energy behaviour of the plant and aspects related to the use of the heat sources/sinks during the heating and cooling operation of the plant. The new system has been installed in a snack bar, which is at the service of the Department of Biomedical and Specialties Surgical Sciences of the University of Ferrara. The MSHP system is used for HVAC of the 100 m^2 snack bar developed as a single room and located inside the University building complex. The snack bar is normally open from 7:00 a.m. to 5:00 p.m. during the workdays of the week and it is used by students, professors, employees, and external users of the building complex. For the installation of the IDEAS system, it was necessary to identify a building owned by the University of Ferrara which would make available a neighbouring area of ground and enough space on the roof. The ground has been used for installation of the horizontal GHEs field while the surface on the roof has been used for the installation of the PVT field and of the air to water heat exchanger.

2.2. The existing building

The part of the building where the snack bar is located is not recent and its envelope is that of the era in which it was built and has not undergone any energy refurbishment over the years. The choice of the case study with a poor building envelope represents a good testing field for the IDEAS system and its potential in renewable sources exploitation. The good result of this experimental activity in an existing building surely represents a push towards this promising and interesting technology to be applied to new efficient buildings. In fact, considering the same useful area, the future buildings, but similarly the current new and refurbished ones, are characterized by significantly lower energy needs compared to our case study. Lower heating and cooling demand and lower thermal peak load reduce the sizes of system components and the required space for installation of the system. At the same time, extremely cold climates should not be constraining for the use of these technologies. The current thermal insulation levels used in the building envelope and the begun path toward the nearly zero energy buildings (NZEB), the zero energy buildings (ZEB) and the plus energy buildings (PEB) could help a wide diffusion of this technology in appropriate contexts for the installation.

The external walls of the snack bar area are floor-to-ceiling windows with painted iron frame and single glazed glass. Differently, adjacent walls to other indoor spaces and small parts to the external environment are brick walls not thermally insulated. The flat roof is characterized by two different levels in the indoor side. The part covering the snack bar area is higher than over ancillary spaces, and a false ceiling is installed in the whole space of the snack bar. Before the installation of the new MSHP system, an all-air system was installed in the snack-bar, and it was used for the air conditioning of the room and to ensure the required fresh air change rates. The air distribution of the existing plant was installed within the false ceiling of the room. The installation of the MSHP has not affected the distribution system, which has kept its existing layout. Some changes have been made to the air handling unit (AHU) to be able to use the new system with the existing plant for the air conditioning. A view of the snack bar is reported in Fig. 1. The picture on the left shows the external with of the snack-bar with the glazed walls and the picture on the right shows the internal space.

2.3. The new experimental plant

Before the installation of the new IDEAS MSHP system, the HVAC of the snack bar was guaranteed by the AHU with one finned coil heat exchanger for the heating and cooling in winter and summer respectively. The main hydraulic distribution of the heat carrier fluid in the University building complex is a four pipes type and the change of the season layout at the snack bar level is obtained through manually operated three-way valves. The heating and cooling network of the building complex to which the snack bar is connected is fed by a substation of the district heating network (DHN) of Ferrara and by an air-to-water chiller respectively.

The IDEAS system was built between the end of the year 2021 and the beginning of 2022. The indoor components and devices of the plant are installed in a technical room close to the snack bar. The core of the plant is represented by the water-to-water HP, which operates between buffer tanks with its primary loops. The source-side has a single buffer tank (named BF1) while on the user side two buffer tanks connected in series are fed by the HP (named BF2.1 and BF2.2). The HP is of a standard type with on/off type compressor. Two pumps are mounted on board and the primary loops of the HP are hydraulically connected with the previous buffer tanks on the source and user side respectively. This layout of the system allows to avoid any hydraulic balancing issues during the operation of the plant. Furthermore, the storage tanks have been designed according to the size of the HP to avoid excessive switching on/off of the HP since the compressor is not inverter driven type. The pumps installed inside the HP are oversized for the application but there was not the possibility to change them. As mentioned before in the text, the IDEAS MSHP system needs the use of buffer tanks to avoid hydraulic issues and at the same time the management of the sources and terminal units requires a control of the fluid flow rate in the secondary loops of the plant. These particular needs made it necessary to install two additional pumps on the secondary loop side as described hereafter in the text.

On the source side, the buffer tank BF1 is connected to the secondary loop of the system and the pump P1 fed the main loop. In this loop, three different thermal source/sink sections can operate separately or simultaneously: the geothermal heat exchangers section (named GHX), the air section (named AHX), which includes a drycooler unit, and the solar section (named PVT_c) made by installing a hybrid photovoltaic thermal modules field. This last section includes the connection of two experimental compound parabolic photovoltaic thermal collectors installed for research aim (named PVT_e) and not considered in the present analysis because of their contribution thermal and electric are negligible in the context of the investigated plant. The GHX loop uses flat panel GHEs installed in trenches. More details about this type of GHEs and their applications can be found in Refs. [25–28]. The solar section includes a solar field of commercial PVT panels (PVT_c) obtained from the use of common PV panels in which an aluminium roll bond heat exchanger is applied on the back of the PV panel, for this reason the PVT panel can be considered as an unglazed type. The AHX section is performed by a dry-cooler with inverter-driven axial fan. The Fig. 2 shows some pictures of the heat source sections of the IDEAS system and the properties of the components of the IDEAS system are summarized in Table 1.

The existing plant of the snack bar has the AHU at the user side for the air-conditioning. The existing AHU has been modified integrating a new section connected to the IDEAS plant. The new experimental plant has the possibility of using, individually or simultaneously, the existing AHU as modified with the new IDEAS section and the novel RF system installed in the snack bar during the renovation of the floor. The hydraulic connection of the IDEAS system to the existing AHU has been obtained by the installation of a new finned coil heat exchanger in addition to the original one. The two finned coil heat exchangers (the old one and the new one) are placed in series to the air flow. This choice became necessary to avoid the mixing between the working fluids of the existing plant and of the new experimental IDEAS system. This design approach ensures HVAC operation regardless of the state of the experimental plant, especially in the first start-up period, which has required a lot of time spent for the testing and updating the management and logging data systems. The RF terminal unit, differently from the standard one, uses the PCM technology by the installation of two different PCMs in plastic containers embedded into the screed under the pipes. The PCMs have the following nominal melting points: 27 °C for heating and 17 °C for cooling respectively. The piping and instrumentation diagram (P&ID) of the IDEAS system is reported in Fig. 3. The diagram shows all the hydraulic connection of the system and the layout of the AHU that includes the existing section (AHU UNIFE) and the new section (AHU IDEAS).

As it can be seen from the same figure, the sections of the three heat sources are connected in parallel and the tank BF1 acts as a collector of the total flow rate processed by pump P1 serving all the three loops. The direct loop is connected with the GHXs field, which represents the main heat source/sink of the MSHP system. It should be highlighted that one of the main goals of IDEAS project is to investigate the behaviour of the MSHP with the GHXs field undersized, as in the case of the investigated experimental plant. The sharing of the total heat carrier fluid flow takes place by controlling the valves V1 and V2 showed in the P&ID diagram, for the air and solar loop respectively, while the total flow rate is controlled by changing the speed of the inverter driven pump P1.

a) b)

Fig. 1. View of the snack bar: a) external, b) internal.

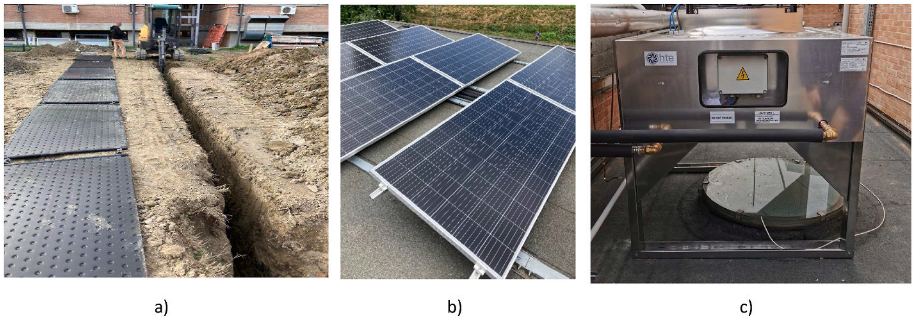

Fig. 2. Pictures of the IDEAS systems: a) GHXs installation b) PVT_c field c) AHX (drycooler).

Table 1
Properties of the IDEAS system.

Description	Value	m.u.
Horizontal Geothermal Field – Flat panel GHEs		
Number of flat-panel GHEs	41	–
Dimensions	2 x 1	m x m
Total length of the trench	82	m
Depth of installation (bottom)	2.3	m
Spacing	3	m
Solar Field – PVT panels (STC: 1000 W/m^2 - AM 1.5–25°C)		
Number of PVT panels	14	–
Number of cells	72	–
Dimensions	1.002 x 1.979 x 0.040	m x m x m
P_{max}	400	W
V_{mpp}	41.03	V
I_{mpp}	9.69	A
V_{oc}	50.39	V
I_{sc}	10.26	A
η_{PV}	20.17	%
P_{max} coefficient	−0.37	%/K
$\eta_{0\ th}$	0.516	–
$b_{1\ th}$	11.044	W/(m^2 K)
Air heat exchanger – finned coil heat exchanger (air T_{in} 35°C - working fluid T_{in}/T_{out} 50/40°C)		
$P_{nom\ th}$	20.7	kW
Heat Pump		
$P_{heating}$ (User 40/45 °C – Source 0/-3 °C)	25.3	kW
COP_H	3.38	–
$P_{cooling}$ (User 12/7 °C – Source 30/35 °C)	25.0	kW
COP_C	3.65	–
Buffer Tank		
BF1	800	L
BF2.1 + BF2.2	500 + 500	L

A similar approach is used on the user side of the plant. In this case, the valve V4 together with the inverter driven pump P2 operate to control the thermal load at the finned coil heat exchanger. The RF is fed by a standard pump group designed for RF systems; its internal mixing valve operates to control the supply temperature of the working fluid to the RF loop.

2.4. The monitoring system

The thermal and electrical plant sections have been monitored through the installation of dedicated measurement devices and logging system. Furthermore, a thermal energy meter has been installed in the existing finned coil heat exchanger loop to measure and record the thermal energy consumption of the existing system in case of use. Not all the data from the monitoring system have been used in the present work, the following details are reported for information purposes to give an overview of the system installed. The monitoring chain includes:

- Thermal energy meters for the monitoring of the following loops: $HP_{source\text{-}primary}$, $HP_{user\text{-}primary}$, GHX, AHX, PVT_c, PVT_e, AHU_{UNIFE}, AHU_{IDEAS} and RF.
- Electric energy meters for the HP and for the auxiliary devices of the plant: pump P1 and P2 on the source side and user side at the secondary loop respectively and AHX.
- T-type thermocouples, used to monitor the RF.
- PT100 temperature sensors used to monitor temperatures in the buffer tanks (BF1, BF2.1 and BF2.2).
- 4-wires PT1000 sensors with Modbus connection used to monitor: the reference temperature in the GHXs field, the rear surface

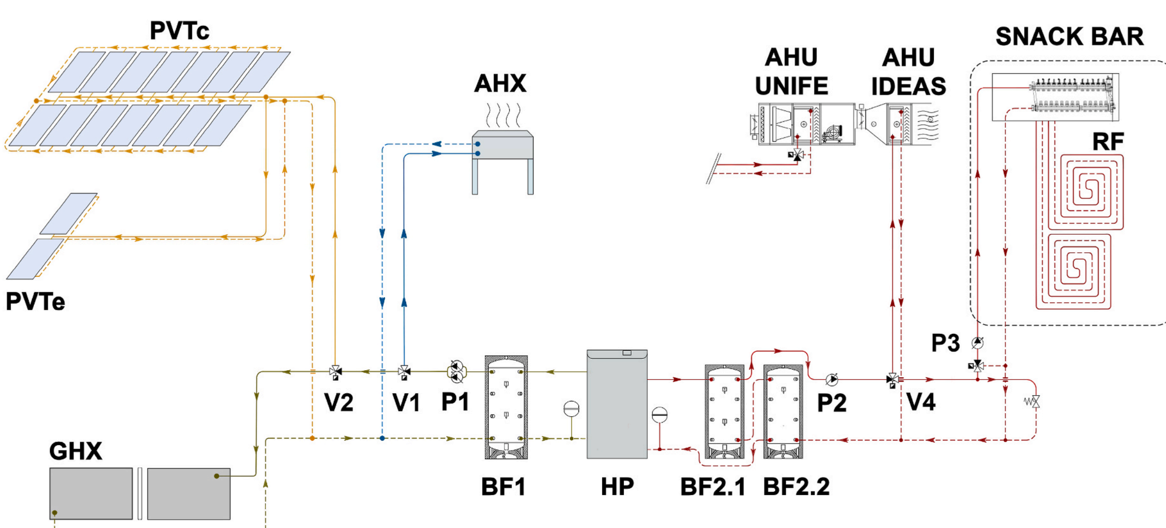

Fig. 3. Piping and instrumentation diagram of the IDEAS experimental plant.

temperature of the commercial PVT panels and the air supply temperature at the AHU.
- a room humidity and temperature sensor, with Modbus connection installed in the snack bar.
- a cabled weather station (DAVIS Vantage Pro 2 type). The station allows the monitoring of outdoor air temperature and relative humidity, solar radiation, rainfall, wind speed and orientation.

The positions and the main properties of the measuring devices are reported in Fig. 4 and in Table 2 respectively.

3. The management of the experimental MSHP system

3.1. The management algorithm of the thermal sources

The exploitation of the renewable heat sources in MSHP systems is more complicated compared to a single heat source installation, like a ground source HP system from which the multi-source approach has been started. The main goal in managing the plant is to keep the buffer tank on the sources side thermally charged (i.e. as warm as possible in winter, and as cold as possible in summer) and ready for the HP to obtain a better COP, operating over smaller operating temperature differences between source and user side. The best source to be exploited, or the best mix of sources has to be chosen at any time by the rules of the source management algorithm used by the system. The approach used to pursue this objective can be based on the choice of the most favourable temperature between the available sources or on the one with the estimated higher thermal power rate. The temperature of heat carrier fluid at the air loop and solar loop essentially depends only on the weather conditions. Indeed, these available sources are not affected by the previous operation of the system in short and long period and also by the thermal load of the building, as it happens in the case of the shallow ground heat exchangers field. As such, these sources can be exploited whenever convenient. At shallow depths, soil temperature naturally varies along the seasons but not in phase with them and the average air temperature. This makes the ground a very interesting renewable thermal energy source as it will remain relatively cold in summer and warm in winter,

Table 2
Main properties of the measuring devices.

Parameter	Accuracy/Reference standard	m. u.
Thermal energy meters [29]		
Volume flow rate	$\pm(2 + 0.02 {*} q_m/q_n) \rightarrow$ or ±5 % if greater	%
Difference of temperature (Measured)	$\pm(0.5 + 0.15/\Delta T)$	%
Difference of temperature (Calculator)	$\pm(0.5 + 0.05/\Delta T)$	%
Electrical energy meters [30]		
Electrical Energy	IEC 62053-21 - Class 1	–
T-type thermocouples [31]		
Temperature	±0.5	°C
4-wires PT100 sensors – Class A [32]		
Temperature	±0.2 (@25 °C); 0.31 (@80 °C)	°C
4-wires PT1000 sensors – class B [33]		
Temperature	±0.43 (@25 °C); 0.70 (@80 °C)	°C
T and RH sensor [34]		
Temperature	±0.2	°C
Relative humidity	±3	%
Weather station [35]		
Temperature	±0.3	°C
Relative humidity	±2	%
Solar radiation	±5	%

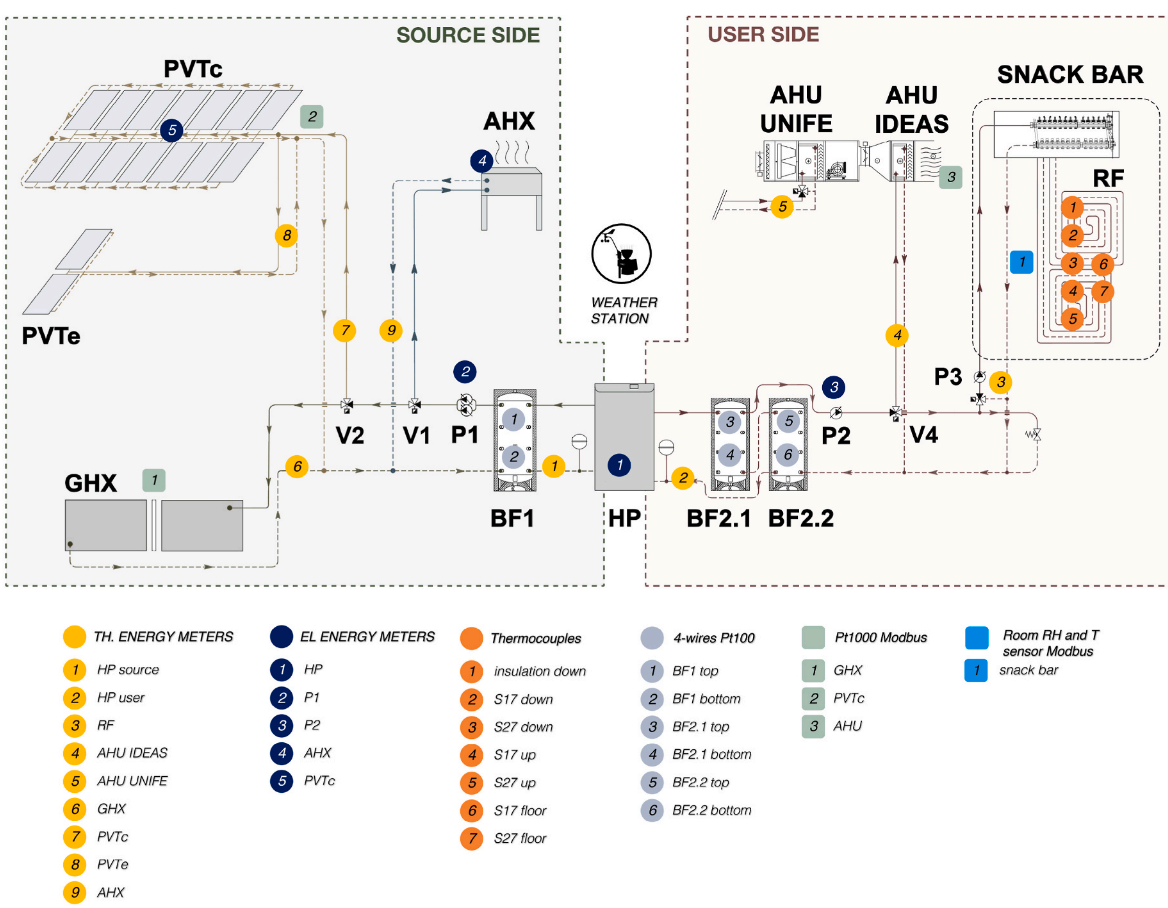

Fig. 4. Measuring devices installed in the IDEAS experimental plant.

especially at the beginning of the seasons. In normal conditions, the undisturbed ground temperature over 5–10 m deep is almost constant close to the average air temperature with a thermal gradient of 0.03 °C/m. In zones with anomalous gradient of temperature this value can increase up to 0.7–0.8 °C/m [36] while at low deep the ground temperature is heavily affected by the weather.

A suitable control algorithm at the source side must firstly include a list of all the possible system operating states in terms of set of sources exploited, then, at each time it must attribute a score to each state to evaluate the advantage that could be obtained by their exploitation. Finally, it must decide which state to activate and operate the valves and devices accordingly. The set of possible source system states, on the base of the plant scheme in Fig. 3, is summarized in Table 3, where a distinction is made between operating states aiming at recharging the buffer tank on the sources side (primary modes) when the heat pump is switched on, and those aiming at performing side operations such as recharging the ground source or cooling the PVT panels from overheating during hot summer days (secondary modes). For primary states, the evaluation is made on a thermal power basis, by estimating the thermal power that could be delivered to the buffer tank by each of them in a similar way to what done in the ε-NTU method for heat exchangers. All the secondary loops on the source side work in parallel mode mixing the flow rates of each circuit before supply the working fluid to the buffer tank. The algorithm estimates the maximum thermal power of the available heat sources defining which is the state of the system that has to be used by the plant between the ones listed in Table 3. Consequently, the management algorithm controls the position of the valves (V1 and V2) and the speed of the pump P1. In case of the exploitation of two heat sources the share of the total volume flow rate is decided by the user setting the parameters (position of valves and pump speed) for each layout of the plant. The algorithm is extremely flexible because of the user can modify the parameters of the code in function of the plant and its properties. More details about approach, method and parameters used in the management algorithm are reported in Ref. [37].

3.2. The management of the terminal units at the user side

The IDEAS system can work in different modes at the user side by using the RF and the AHU. The fan of the AHU is always on and not controlled by the IDEAS system, independently from the operation mode of the terminal units, to guarantee the air change rate in the snack bar. The following layouts can be used for the HVAC of the indoor space:

- RF layout: In this mode the RF is used for heating and cooling and the fresh air is supplied at external ambient conditions. This operating mode is suitable during mid-seasons. In summer, the algorithm calculates the supply heat carrier fluid temperature at the RF in function of indoor temperature and relative humidity, avoiding the surface condensation.
- AHU layout: In this mode the RF is always switched off and the HVAC is managed by the AHU and the finned coil heat exchanger hydraulically connected with the IDEAS system. The system works as an all-air system.
- RF + AHU layout: the system works as an air-water plant. If the indoor ambient condition is close to the setpoint temperature, the RF operates normally as in the first case, while the AHU supplies air to the room with a temperature close to the setpoint value. In cooling mode, the dehumidification is guaranteed suppling low temperature working fluid to the AHU heat exchanger. If the indoor temperature is far from the setpoint temperature the AHU is managed to reach the desired temperature.

Table 3
Possible states of source/sink exploitation.

State	Type	Name/ Description	Flow rate to ground	Flow rate to air (valve V1)	Flow rate to solar (Valve V2)
1	p	Only Ground	100	0	0
2	p	Only Air	0	100	0
3	p	Sun	0	0	100
4	p	Ground + Air	X_{G*}	100 - X_{G*}	0
5	p	Ground + Solar	X_{G**}	0	100 - X_{G**}
6	s	Ground + Air	100 – X_A	X_A	0
7	s	Ground + Solar	100 – X_S	0	X_S
8	s	Solar + Air	0	1 – X_{PVC}	X_{PVC}

Legend of the table: p - primary mode (HP on), s - secondary mode (HP off), X value represents the fraction of the total mass flow rate fed by the pump P1, the share of the volume flow rate is decided by the user changing the position of the valves during the hydraulic balance of the plant. Each layout of the system has a different position of the valves.

In all the configurations the management rules of the terminal units operate for keeping the indoor temperature close to the setpoint decided by the user in heating and cooling. The set point temperatures used during the operation of the HVAC system were 20 °C and 26 °C in heating and cooling period respectively. The algorithm acts on the valve V4 and the speed of the pump P2 changing the thermal load on the IDEAS finned coil heat exchanger, while the supply temperature at the RF terminal unit is controlled by a standard pumping unit with 3-way valve and controller on board. The controller receives the supply temperature value from the IDEAS management algorithm and works itself for the control of the RF system.

4. Results of the operating period

The results of the monitoring campaign include the data for the first season in heating and cooling respectively. The IDEAS system started working for its first time at end of May 2022 and the collection data reported in the present paper are updated until the end of January 2023. In this first period, few issues have emerged over time and their resolution have required adequate technical times with the consequent stop of the experimental plant. During the stop of the IDEAS system, the existing plant was used for the HVAC of the snack bar. For this reason, the energy analyses have been performed for time windows in which the complete or enough datasets of monitoring data were available for the evaluation of the energy performances of the system. The details of the energy analysis are described in the following sections.

4.1. Analysis of the thermal source/sink sections of the IDEAS MSHP system

During the period of the two seasons analysed, a large amount of working fluid (about 12,000 m^3) flowed through the GHXs loop. From the end of May 2022 to January 2023 the total amount of heat extracted from the ground was equal to 6880 kWh_t (this thermal load includes the HP operation in winter and the de-superheating of the ground in summer), while the heat injected was equal to 11,570 kWh_t. The trends of the cumulate heat exchanged with the ground are shown in the chart in Fig. 5. The lines in red and blue represent the heat injected and extracted respectively, while the lines in light blue and green show the trend of the external air temperature and the reference temperature of the ground measured by a PT1000 sensor at about 1.5 m deep in the middle between the GHXs lines.

The present energy analysis evaluates the heat loads from July 2022 to September 2022 and from the end of November 2022 to the end of January 2023 for the cooling and heating period respectively. These time windows have been considered because of the availability of enough data about the operating condition of the system and within these periods some time windows have been selected for the energy analysis. As it can be seen from the chart, in the cooling period the heat extracted from the ground is not equal to zero. The meaning of this

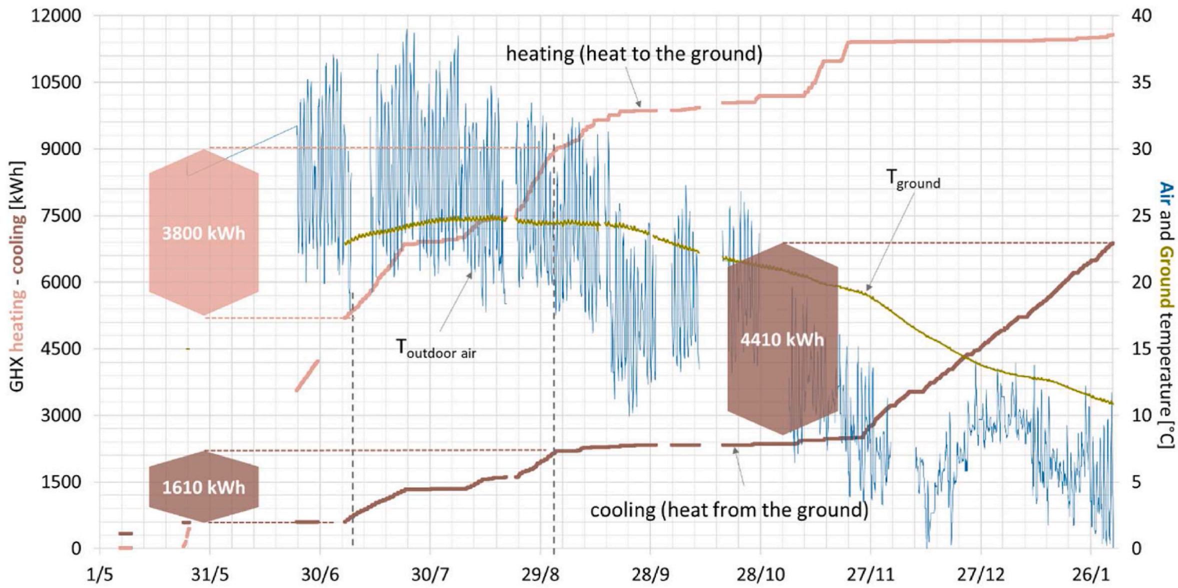

Fig. 5. Heat exchanged with ground, external air temperature and ground temperature.

unexpected increase of the heat loads is due to the use of the AHX and PVT field over the night to thermally discharge the ground when the HVAC system was switched off. One of the main objectives of the IDEAS project was the possibility of using an undersized GHEs field thanks to the use and the exploitation of air and sun as alternative sources. In summer period the system can operate with the air and the PVT field for the de-superheating of the ground. In fact, during the night whenever the air temperature makes possible this type of operation by the system, the heat is extracted from the ground and the heat exchanges with the air by means of the AHX loop. Simultaneously the solar loop is used with the air loop to exchange heat with the sky by the PVT field. In the heating period, the presence of favourable environmental conditions has made possible to use the solar loop to thermally recharge the ground or the BF1 tank only at the end of January 2023. In the first half of November 2022 the heat injected in the ground was due to a not correct operation of the existing plant solved at the end of the month. Globally, the thermal load profile on the ground side is not balanced if the total extraction and injection of heat are considered. The ratio between the two heat loads is equal to 1.58. The unbalancing of the thermal loads, despite the use of the de-superheating in the summer, is however quite limited. The data in the chart provide an idea of the magnitude of the thermal loads on the ground side in a context of building and weather similar to the one being investigated, i.e.an application in not thermally insulated building in a typical climate of the regions located in the northeast of Italy. What is particularly interesting is not the result related to the building as such, but the result obtained in an application where the load on the user side is real and not simulated as reported in most of the works present in the literature.

The secondary loop of the plant at the source side is equipped with the inverter-driven pump P1 used for the exploitation of the three thermal sources (ground, air and sun). The different sources can work individually or in parallel on the same buffer tank (BF1) as it can be seen from the P&ID in Fig. 3. The buffer tank BF1 receives all the working fluid flow rates processed by the individual loop. The speed of the pump is decided by the management algorithm of the system as function of the rules implemented in the management software. In the whole monitored period, the total amount of electrical energy consumption of pump P1 at the secondary loops was equal to 640 kWhe. The pumping costs of this device are relatively low if compared to the total amount of heat exchanged with the ground, which is about 29 times the electric consumption. The total amount of P1 electrical energy consumption has been assigned to the GHXs loop nevertheless all the secondary loops at the source side always use the pump P1. This assumption does not change the above consideration; indeed, it strengthens it even more since the numerator quantity in the report only considers the heat exchanged to the ground. Considering all the three loops the ratio would be even higher.

The exploitation of the air source through the AHX loop is shown in Fig. 6. As it can be seen in the chart, the air source is used exclusively during summer, as could be expected. During the heating period the temperature of the ground is on average higher than that of the air and the system management rules obviously evaluate the ground as the best heat source to use. On the other hand, the experimental plant was not designed to solve the frosting issue at the finned coil heat exchanger of the AHX unit and only in the middle season, at the beginning and at the end of the heating season, the air source could be used instead of the ground source. The AHX loop, i.e. the drycooler, needs electrical energy in order to work with the air, to this need a part of the consumption of the P1 could also be attributed to this device but, for the sake of simplicity, reference will be made to fan consumption only. Considering the total amount of the heat exchanged with the air, equal to 4550 kWh_t, which corresponds an electrical energy consumption of about 770 kWh_e it gets a ratio of about 6 between the two quantities, a decidedly lower result if compared with the consumption of pump P1 attributed entirely to the heat exchange with the ground as describe above in the text. This simple consideration highlights how this section of the system, and its device can affect the global energy performance obtainable from the whole system. Surely, a further optimization of the working conditions in terms of difference of temperature between the working fluid and the air should improve the efficiency of this loop, but the result will not be comparable with the ground loop and its electrical energy demand. However, it is obvious that a high electrical energy consumption like the one in the AHX section could be justified by the unavailability of space for the installation of the GHEs with a consequent reduction in the initial installation costs.

The chart reported in Fig. 7 depicts the heat exchanged at the solar loop of the system. Differently from the previous charts in Figs. 5 and 6, in addition to the heat exchanged, the trends of the solar radiation on the horizontal surface and the temperature of the PVT_c panels are included. This section of the plant has been used for two different aims. In the summer period, the section was cooled during the day to increase the electrical photovoltaic conversion and during the night it is used to de-superheat the ground section, this last already mentioned before in the text. For the highlighted period (July and August), the cooling activity

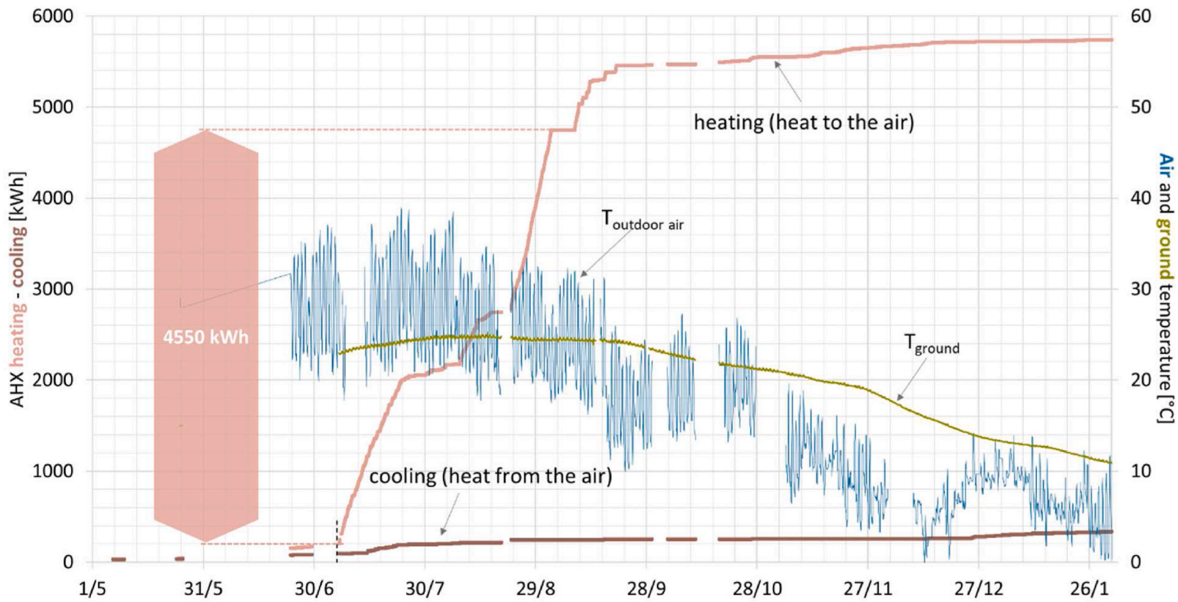

Fig. 6. Heat fluxes at the AHX loop.

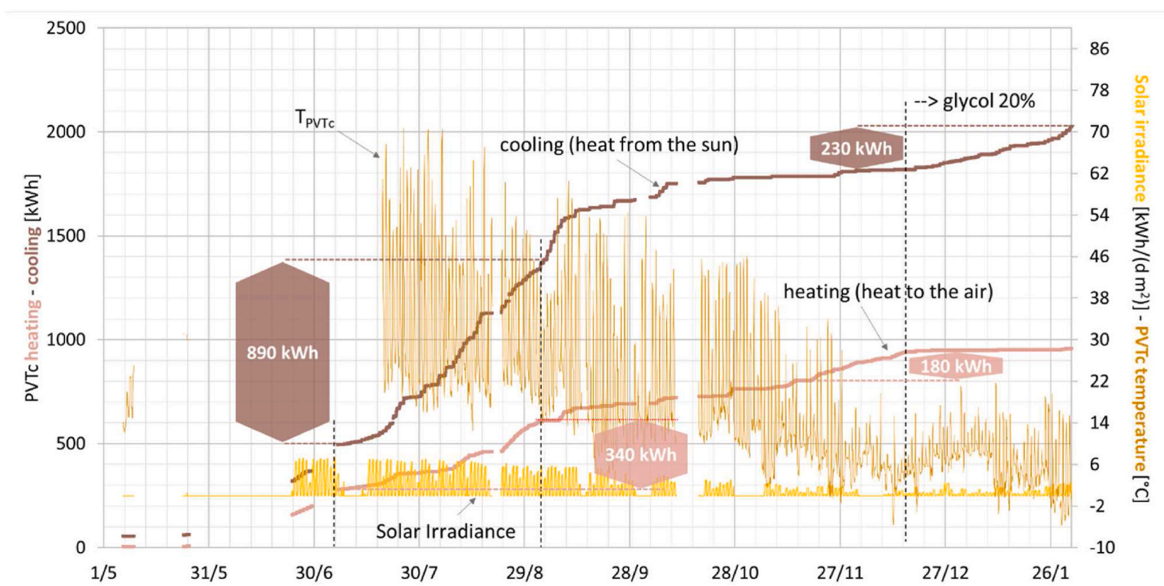

Fig. 7. Heat fluxes at the PVTc loop.

accounted for 890 kWh_t, whilst the de-superheating for 340 kWh_t. The temperature on the rear side of the PVT panels just occasionally achieved values over 60 °C (the limit at which the system activates to cool the panels), even with high solar irradiance (>800 W/m^2). During the winter period, until end-January, foggy days occurred several times so that the solar contribution to the system was quite limited. Just from end-December it was possible to gather the solar contribution (230 kWh_t).

Regarding the first aim of the solar loop operation, i.e. the cooling of the PVT module in summer, the increase of the photovoltaic production was quite limited. This action of the plant needs the operation of the air loop (the AHX) and as describe before in the text, this last represents a high electricity consumption device. In the specific case, the additional electricity production due to cooling is not able to cover the consumption of the air loop. A comparison between the electricity production during two similar days is reported in Fig. 8. The chart shows the power production of the commercial PVT panels for two different days with the same solar irradiance. During the 4th of August the solar section was cooled by the working fluid of the plant, whilst during the 5th of August it was not. The minimum and maximum air temperatures were almost comparable between the two days: 21.3 °C and 36.6 °C for the 4th, 22.5 °C and 37.5 °C for the 5th.

The peak power production was 3327 W_e in the cooling mode and 3091 W_e in standard mode when the maximum solar irradiance was 852 W/m^2 on a horizontal surface. Therefore, the cooling mode allowed a net increasing of 7.6 % in peak power conversion. In terms of efficiency, the cooled panel achieved 14.3 %, whilst the not cooled one 13.3 %. Considering the cooling performed until 8p.m., the energy balance is reported in Table 4. The extra-production occurring due to the cooling is 1.4 kWh_e, made possible due to the work of P1 and AHX (4.0 kWh_e). Therefore, the cooling of PVT panels is only functional to increase the peak power (+7.6 %) and not to increase the energy production.

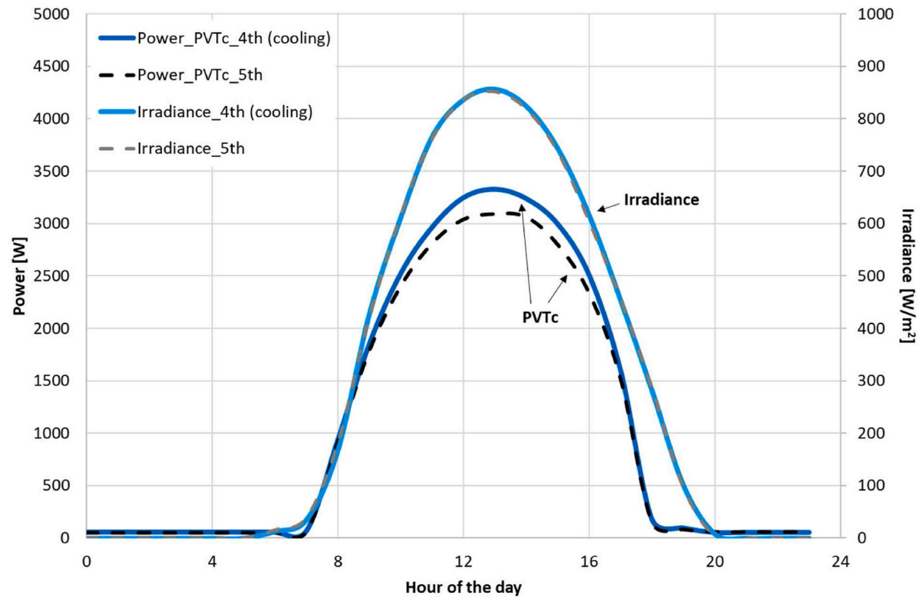

Fig. 8. Electricity production with and without cooling at the PVT_c loop.

Table 4
Electrical and thermal energy balances for the PVT_c loop with and without the cooling of the panels.

Day	4th August Cooling mode		5th August standard mode
Section/ Device	electrical energy [kWh_e]	thermal energy [kWh_t]	electrical energy [kWh_e]
P1	1.1	–	–
AHX	2.9	34.3	–
PVT_c	25.8	41.4	24.4
PVT_c – P1 - AHX	21.8	–	24.4 (+11.9 %)

4.2. Energy analysis of the IDEAS system

The energy analysis of the IDEAS system has been carried out highlighting the differences in terms of energy performance with the existing plant serving the snack bar. As already described above, in the summer period, the cooling demand is satisfied by the use of an air-to-water chiller serving the whole building complex and not only the snack bar. In winter the DHN supplies heat to the building complex and therefore to the heating loop of the snack bar. The energy analysis has been carried out considering the heat fluxes at the primary loops of the plant, i.e. evaluating the heat loads at the condenser and evaporator side of the HP in heating and cooling period respectively. This approach allows to evaluate all the subsystems' behaviours at the user side after the HP, thus making the assessment is performed with the same operation conditions, not having changed or influenced in any way the management method of the terminal units. On the user side the system has worked always with the AHU + RF layout as described in section 3.2. The comparison of the two systems, i.e. the existing and the new MSHP system, needs the energy costs of the existing plant at the thermal power plant level to guarantee the same heating and cooling services as the IDEAS system. To achieve this goal, some assumptions and literature data have been used for the evaluation of the existing generation systems, since direct measures of the energy consumption were not available for the existing thermal power plant of the building complex. In particular, for the cooling operation a simulation model of the air-to-water chiller has been developed for this aim. The polynomials of a screw compressor working with R134a refrigerant fluid [38] (the same compressors family of the existing plant) and the typical pinch points were used in the definition of the energy model (5 °C at the evaporator and 12 °C at the condenser). The simulation model has been developed according to the model described in Ref. [39], for the calculation of the energy performances of the chiller. For the heating mode, the data of the DHN of Ferrara have been considered. The available data of the DHN state that the 55 % of the heat is obtained from the geothermal source and waste material in waste-to-energy plants. Considering the particularity and exceptionality of that DHN the share of heat from renewable energy sources supplied by the DHN has been reduced to 40 % in the energy analysis. This value has been used to suggest a threshold value to obtain a similar result in terms of primary energy consumption between the existing system and the proposed IDEAS system. This threshold can be used as reference in preliminary energy analysis when DHN is available as an option for the heating purpose.

In cooling mode, two different time windows in the whole period have been analysed, having available for these periods enough data for the calculation of energy performance of the equivalent existing system as described above in the text. If available, the energy fluxes at the user side and source side of the heat pump were considered together with the electric energy consumption logged by the monitoring system. The days between July 15th and July 23rd and the period between August 23rd and September the 1st have been used for the analysis in cooling mode, while the days between November 27th and January 31st were used for the heating period. The charts in Fig. 9 and Fig. 10 show the heat exchanged between the HP and the buffer tanks at the source side and user side respectively. In the same charts the trends of air temperature and ground temperature are reported. The results in terms of heat exchanged with the buffer tank and the electric energy consumption are summarized in Table 5 and Table 6 for cooling and heating respectively. The electric energy includes the overall consumption of all the devices on the source side and not only the consumption of the HP.

In cooling mode, the energy carrier is the same, i.e. the electric energy, therefore the results in terms of energy demand and primary energy saving can be easily compared. From Table 5 it can be seen that for the two period the energy saving of the investigated MSHP system is of about 20 % comparing the energy demand of the existing plant. The charts in Figs. 11 and 12 show the comparison of the EER values for the AW chiller and IDEAS system in the two periods. As expected, the EER values of the IDEAS system are quite higher than the ones obtained for the AW chiller. As it can be seen in the charts, the undersize of the GHEs loop, which represent one of the investigated topics of the IDEAS

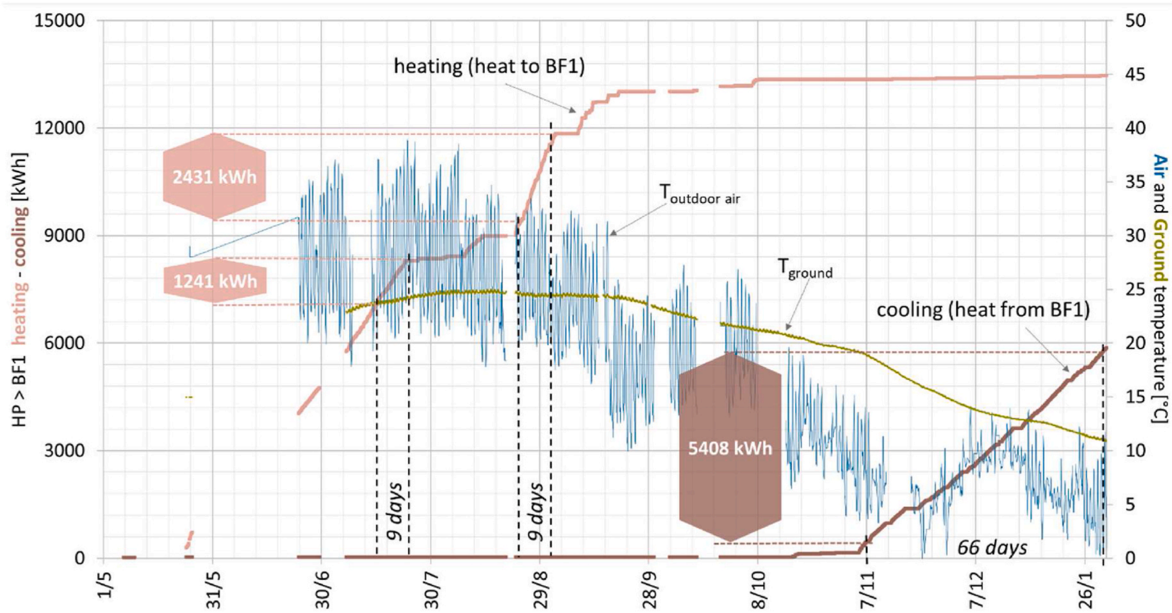

Fig. 9. Heat load – Primary loop Source side.

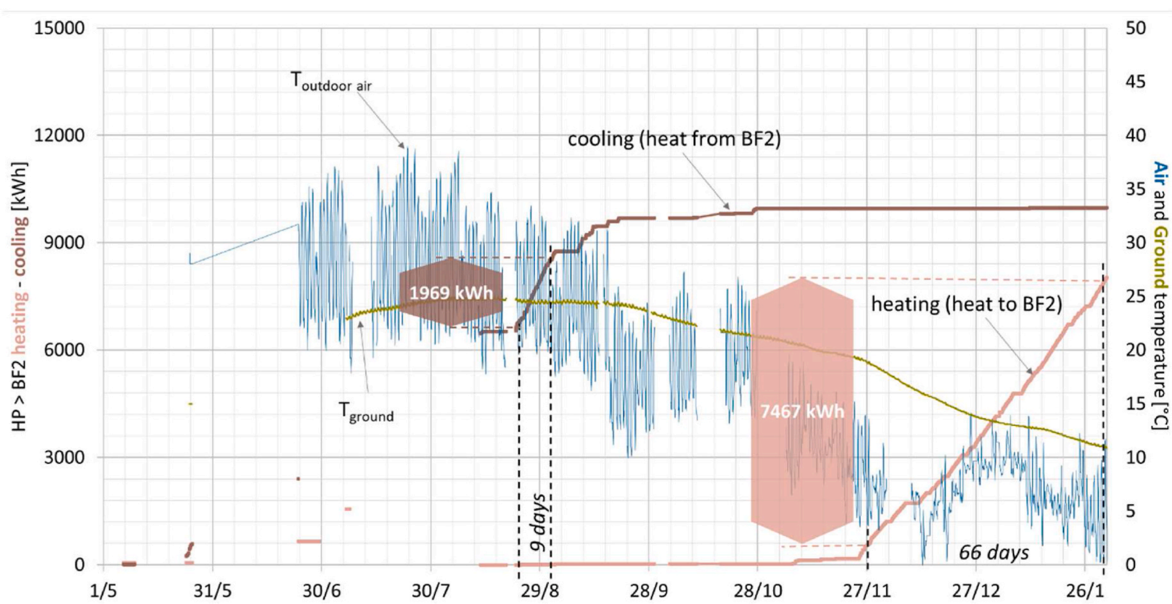

Fig. 10. Heat load – Primary loop User side.

project, significantly affects the energy performance of the HP. In fact, the trend of the EER values follows the shape of the air-to-water chiller' EER but always with higher values. This behaviour is due to the thermal load of the building that is quite high during the end of the morning and in the afternoon according to the climate and use of the snack bar. During this period the IDEAS system increases the number of activations of the HP and therefore the heat injection rate to the ground significantly increases. Consequently, the ground goes in thermal suffering at the end of the afternoon with possible issues for the following days. Instead, thanks to the management of the IDEAS system, the ground can be de-superheated during the night and thermally regenerated for the following day. In fact, has it can be seen in the chart the energy performances over the two periods are quite constant, this means that the ground seems to be not affected to a thermal drift.

In heating mode, the comparison is quite different because two different energies carriers are used in the plants. This difference needs the use of conversion coefficients to obtain a quantitatively comparable quantity, i.e. the primary energy. From the results in the table the whole energy performance of the IDEAS system is similar to the existing system, which uses a DHN with a large share of renewable energy. A comparison with a more common generation system with a condensing boiler (η_{cb} = 98 %) increases the primary energy saving up to 40 %, making the IDEAS system a valid solution compared to a traditional system.

4.3. Electrical contribution of the PVT field

As described in section 2.3 the IDEAS system includes a solar loop with a PVT field of 14 panels 400 W_p each installed on the flat roof of the snack bar with a slope of 12°. The nominal size of the photovoltaic system is 5.6 kW_p. Even in hypothetical operation at nominal power, the electric load available would not be able to cover all the electric load

Table 5
Energy performance - Cooling mode.

	Air to water chiller (Simulated)	ideas system
From 15/07/22 to 23/07/22 (9 days)		
Cooling load [kWh_t]	993[a]	
Electric energy consumption [kWh_e]	451	357 (−21 %)
EER [−]	2.2	2.8
From 23/08/22 to 01/09/22 (9 days)		
Cooling load [kWh_t]	1969	
Electric energy consumption [kWh_e]	780	684 (−17 %)
EER [−]	2.5	3.0

[a] The cooling load for this period has been calculated by the difference between the heat exchanged at the condenser and the electric consumption of the compressor.

Table 6
Energy performance - heating mode.

	DHN	IDEAS System
From 27/11/22 to 31/01/23 (66 days)		
Heating Load [kWht]	7467	
Electric Energy Consumption [kWhe	194[a]	2594
Share of Fossil Energy (60 %) [kWhf]	4480	–
Primary Energy	4834[b]	4716[b] (−2.4 %)

[a] Pumping cost related to the internal pipe network of UNIFE area ($H = 8$ m_{H2O}, $Q_{WF} = 4\ m^3/h$, $\eta_{pump} = 0.7$).
[b] Electrical energy conversion coefficient, $\eta_{el} = 0.55$.

required by the HP that needs an electric power of about 8 kWe. Therefore, in such a situation the electricity production of the PVT field can be considered totally used by the system. On the other hand, this assumption could easily be supported by the installation of a small electricity storage battery because of this storage could only be recharged when the compressor of the HP is switched off. In fact, when the HP Is switched on the electric energy is always supplied by the grid. The results of the energy analysis including the electricity production in the same period investigated in the previous section are reported in Table 7 and Table 8 for the cooling and heating period respectively. As expected, the positive contribution of photovoltaic production significantly improves overall energy performance of the system. In cooling, the EER increases from 2.8 to 7.3 in the first period corresponding to a reduction in the energy demand of 70 % and consequently the same reduction is achieved for primary energy demand and GHG emission. In the second period, the reduction in energy demand increases up to 38 % and the EER increases up to 4.1. In the heating period the effect of the photovoltaic production is less evident. The installation close to the horizontal position of the PVT modules does not allow to fully exploit the potential production of the system as the inclination is not optimal for the winter period and the electricity production is quite limited. Globally, the primary energy saving increases up to 13.9 % compared to the existing DHN system.

5. Conclusions

The HP system represents nowadays the most promising technology to pursue the EU objectives of electrification and decarbonisation in the building sector. In the last years there has been a growing interest in MSHP systems thanks to the potential energy performance improvement that the system can achieve by using different heat sources and sinks. The results of this study show the energy saving potential of a MSHP systems and it represents one of the few studies in the literature that investigates a real large-scale application. The study shows the use of the different renewable sources, and their energy needs in order to be used in the MSHP system. This preliminary analysis has highlighted that the air loop represents the most energy intensive section of the plant and it must be carefully designed to optimize the operation of the system.

The following conclusions and observations have been obtained from the present study and should be evaluated for a good design approach of this type of systems:

- The use of MSHP system increases the energy saving potential of HP technology.
- Despite the undersizing of the flat-panel GHEs field, the system achieved good energy performance results, higher than the existing plant, especially in cooling mode.
- In cooling mode, a reduction of 20 % in electrical energy demand has been obtained for the case study analysed. This reduction increases up to 70 % if the photovoltaic production of the PTV system is considered in the energy balance of the IDEAS plant.

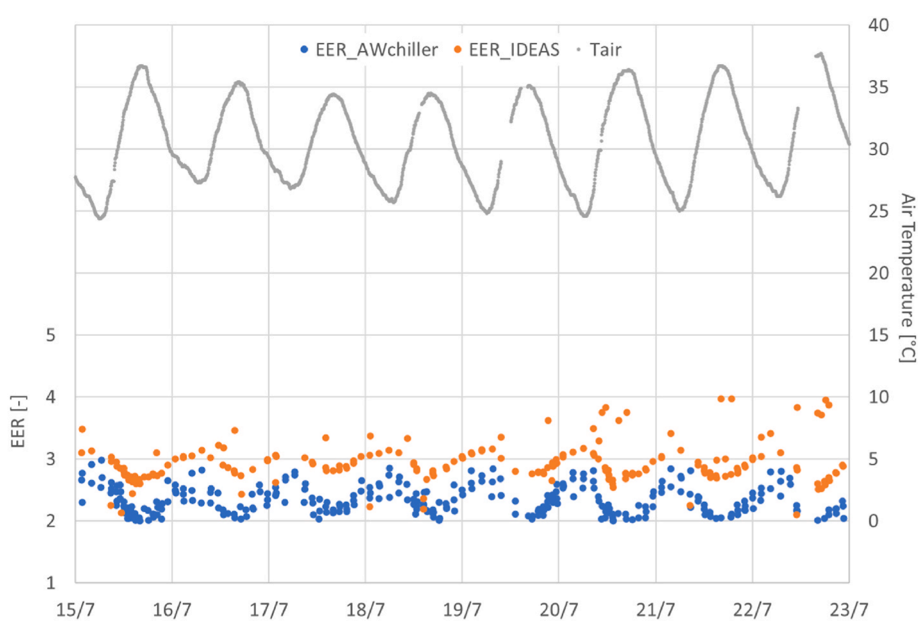

Fig. 11. Comparison between EER values of AW chiller and IDEAS system - From 15/07/22 to 23/07/22.

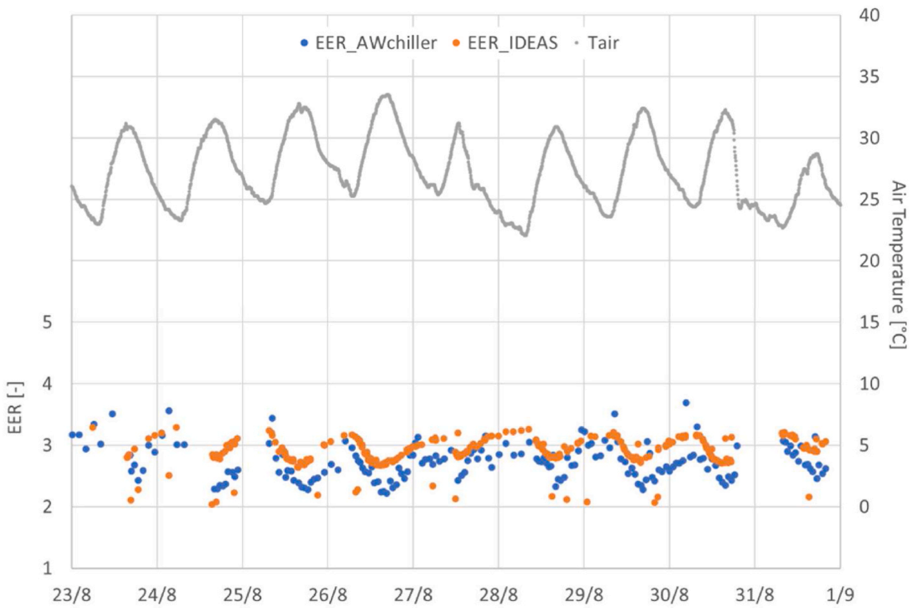

Fig. 12. Comparison between EER values of AW chiller and IDEAS system - From 23/08/22 to 01/09/22.

Table 7
Energy performance with photovoltaic production - Cooling mode.

	Air to Water Chiller (Simulated)	IDEAS System
From 15/07/22 to 23/07/22 (9 days)		
Cooling load [kWh_t]	993[a]	
Electrical energy consumption [kWh_e]	451	357 (−21 %)
Photovoltaic production [kWh_e]	–	222
Net electrical energy consumption [kWh_e]	451	135 (−70 %)
EER [−]	2.2	7.3
From 23/08/22 to 01/09/22 (9 days)		
Cooling load [kWh_t]	1969	
Electrical energy consumption [kWh_e]	780	684 (−17 %)
Photovoltaic production [kWh_e]	–	199
Net electrical energy consumption [kWh_e]	780	485 (−38 %)
EER [−]	2.5	4.1

[a] The cooling load for this period has been calculated by the difference between the heat exchanged at the condenser and the electrical energy consumption of the HP compressor.

Table 8
Energy performance with photovoltaic production - Heating mode.

	DHN	IDEAS System
From 27/11/22 to 31/01/23 (66 days)		
Heating load [kWh_t]	7467	
Electric energy consumption [kWh_e	194[a]	2594
Photovoltaic production [kWh_e]	–	305
Net electric energy consumption [kWh_e]	194	2289
Share of fossil energy (60 %) [kWh_f]	4480	–
Primary energy [kWh_p]	4834[b]	4161[b] (−13.9 %)

[a] Pumping cost related to the internal pipe network of UNIFE area (H = 8 m_{H2O}, $Q_{WF} = 4\ m^3/h$, $\eta_{pump} = 0.7$).
[b] Electrical energy conversion coefficient, $\eta_{el} = 0.55$.

- In heating mode, the MSHP system has similar energy performance if compared to DHN. In this case the contribution of photovoltaic production is limited in the energy balance of the system.
- The primary energy demand of the case study is reduced by 14 % and 40 % in heating and cooling respectively, by using the MSHP system.
- The air source must be managed carefully to limit its energy costs; it represents the section of the system with high energy consumption.

In addition, the real application of the existing building with poor building envelope contributes to support the validity and the energy saving potential of the technological solution used for the air conditioning. The IDEAS system, and the MSHP systems in general, represent an alternative solution to the common heating and cooling systems thanks to their capability to improve the exploitation of renewable resources and at the same time reduce the primary energy needs and the GHG emissions. Consequently, new or refurbished buildings can only facilitate the use of the investigated technology. In fact, they require less energy and lower thermal power compared to existing buildings with consequent advantages with consequent advantages linked to overcoming economic and implementation barriers. It is therefore obvious to think how the future buildings (e.g. the so-called ZEB, ZEB and PEB buildings) will allow a wide diffusion and replicability of the investigated system.

CRediT authorship contribution statement

Giuseppe Emmi: Conceptualization, Methodology, Investigation, Writing – original draft, Writing – review & editing. **Eleonora Baccega:** Writing – review & editing. **Silvia Cesari:** Writing – review & editing. **Elena Mainardi:** Conceptualization, Writing – review & editing. **Michele Bottarelli:** Conceptualization, Methodology, Investigation, Writing – review & editing, Supervision.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

This study was funded by the project "IDEAS" financed by the European Union's Horizon 2020 research and innovation program, G.A. No. 815271. The authors declare that there is no conflict of interests.

References

[1] European Commission, Directorate-General for Energy, F. Gerard, T. Smit, K. Rademaekers, et al., Policy Support for Heating and Cooling Decarbonisation – Roadmap, Publications Office of the European Union, 2022. https://data.europa.eu/doi/10.2833/977806. (Accessed 1 April 2023).

[2] EUROSTAT "Shares summary results 2020". https://ec.europa.eu/eurostat/documents/38154/4956088/SUMMARY-results-SHARES-2020.xlsx/e1197d70-9ef0-ba1f-193b-cab0dcf9b13f?t=1642578001746. (Accessed 1 April 2023).

[3] Climate & energy package. https://climate.ec.europa.eu/eu-action/climate-strategies-targets/2020-climate-energy-package_en, 2020. (Accessed 1 April 2023).

[4] The future of heat pump. https://www.iea.org/reports/the-future-of-heat-pumps. (Accessed 1 April 2023).

[5] F. Pelella, G. Zsembinszki, L. Viscito, A.W. Mauro, L.F. Cabeza, Thermo-economic optimization of a multi-source (air/sun/ground) residential heat pump with a water/PCM thermal storage, Appl. Energy 331 (2023), 120398, https://doi.org/10.1016/j.apenergy.2022.120398.

[6] R. Lazzarin, M. Noro, Photovoltaic/Thermal (PV/T)/ground dual source heat pump: optimum energy and economic sizing based on performance analysis, Energy Build. 211 (2020), 109800, https://doi.org/10.1016/j.enbuild.2020.109800.

[7] F. Maranghi, L. Gosselin, J. Raymond, M. Bourbonnais, Modeling of solar-assisted ground-coupled heat pumps with or without batteries in remote high north communities, Renew. Energy 207 (2023) 484–498, https://doi.org/10.1016/j.renene.2023.03.040.

[8] F. Busato, R. Lazzarin, M. Noro, The control of renewable energies to improve the performance of multisource heat pump systems: a two-case study, Appl. Sci. 11 (14) (2021) 6653, https://doi.org/10.3390/app11146653.

[9] Q. Si, M. Okumiya, X. Zhang, Performance evaluation and optimization of a novel solar-ground source heat pump system, Energy Build. 70 (2014) 237–245, https://doi.org/10.1016/j.enbuild.2013.11.065.

[10] W. Deng, J. Yu, Simulation analysis on dynamic performance of a combined solar/air dual source heat pump water heater, Energy Convers. Manag. 120 (2016) 378–387, https://doi.org/10.1016/j.enconman.2016.04.102.

[11] M. Mehrpooya, H. Hemmatabady, M.H. Ahmadi, Optimization of performance of Combined Solar Collector-Geothermal Heat Pump Systems to supply thermal load needed for heating greenhouses, Energy Convers. Manag. 97 (2015) 382–392, https://doi.org/10.1016/j.enconman.2015.03.073.

[12] S.H. Razavi, R. Ahmadi, A. Zahedi, Modeling, simulation and dynamic control of solar assisted ground source heat pump to provide heating load and DHW, Appl. Therm. Eng. 129 (2018) 127–144, https://doi.org/10.1016/j.applthermaleng.2017.10.003.

[13] G. Emmi, A. Zarrella, M. De Carli, A. Galgaro, An analysis of solar assisted ground source heat pumps in cold climates, Energy Convers. Manag. 106 (2015) 660–675, https://doi.org/10.1016/j.enconman.2015.10.016.

[14] Z. Han, L. Qu, X. Ma, X. Song, C. Ma, Simulation of a multi-source hybrid heat pump system with seasonal thermal storage in cold regions, Appl. Therm. Eng. 116 (2017) 292–302, https://doi.org/10.1016/j.applthermaleng.2017.01.057.

[15] Y. Chen, H. Hua, J. Wang, P.D. Lund, Integrated performance analysis of a space heating system assisted by photovoltaic/thermal collectors and ground source heat pump for hotel and office building types, Renew. Energy 169 (2021) 925e934, https://doi.org/10.1016/j.renene.2020.12.117.

[16] F. Cruz-Peragon, F.J. Gómez-de la Cruz, J.M. Palomar-Carnicero, R. Lopez-García, Optimal design of a hybrid ground source heat pump for an official building with thermal load imbalance and limited space for the ground heat exchanger, Renew. Energy 195 (2022) 381–394, https://doi.org/10.1016/j.renene.2022.06.052.

[17] G. Besagni, L. Croci, R. Nesa, L. Molinaroli, Field study of a novel solar-assisted dual-source multifunctional heat pump, Renew. Energy 132 (2019) 1185–1215, https://doi.org/10.1016/j.renene.2018.08.076.

[18] J. Ji, J. Cai, W. Huang, Y. Feng, Experimental study on the performance of solar-assisted multi-functional heat pump based on enthalpy difference lab with solar simulator, Renew. Energy 75 (2015) 381–388, https://doi.org/10.1016/j.renene.2014.09.054.

[19] J. Cai, J. Ji, Y. Wang, W. Huang, Numerical simulation and experimental validation of indirect expansion solar-assisted multi-functional heat pump, Renew. Energy 93 (2016) 280–290, https://doi.org/10.1016/j.renene.2016.02.082.

[20] R. Lazzarin, M. Noro, Lessons learned from long term monitoring of a multisource heat pump system, Energy Build. 174 (2018) 335–346, https://doi.org/10.1016/j.enbuild.2018.06.051.

[21] E.I. Sakellarioua, A.J. Wrighta, P. Axaopoulosb, M.A. Oyinlola, PVT based solar assisted ground source heat pump system: modelling approach and sensitivity analyses, Sol. Energy 193 (2019) 37–50, https://doi.org/10.1016/j.solener.2019.09.044.

[22] N. Sommerfeldt, H. Madani, In-depth techno-economic analysis of PV/Thermal plus ground source heat pump systems for multi-family houses in a heating dominated climate, Sol. Energy 190 (2019) 44–62, https://doi.org/10.1016/j.solener.2019.07.080.

[23] G. Nouria, Y. Noorollahia, H. Yousefi, Designing and optimization of solar assisted ground source heat pump system to supply heating, cooling and hot water demands, Geothermics 82 (2019) 212–231, https://doi.org/10.1016/j.geothermics.2019.06.011.

[24] IDEAS e Novel building Integration Designs for increased Efficiencies in Advanced Climatically Tunable Renewable Energy Systems. https://www.horizon2020ideas.eu. (Accessed 1 April 2023).

[25] M. Bottarelli, M. Bortoloni, Y. Su, On the sizing of a novel Flat-Panel ground heat exchanger in coupling with a dual-source heat pump, Renew. Energy 142 (2019) 552–560, https://doi.org/10.1016/j.renene.2019.04.088.

[26] V. Ciriello, M. Bottarelli, V. Di Federico, Uncertainty-based analysis of variations in subsurface thermal field due to horizontal flat-panel heat exchangers, Procedia Environmental Sciences 25 (2015) 50–57, https://doi.org/10.1016/j.proenv.2015.04.008.

[27] G. Emmi, M. Bottarelli, Enhancement of shallow ground heat exchanger with phase change material, Renew. Energy 206 (2023) 828–837, https://doi.org/10.1016/j.renene.2023.02.079.

[28] M. Bottarelli, E. Baccega, S. Cesari, G. Emmi, Role of phase change materials in backfilling of flat-panels ground heat exchanger, Renew. Energy 189 (2022) 1324–1336, https://doi.org/10.1016/j.renene.2022.03.061.

[29] Maddalena, Ultrasonic compact thermal energy meter Microclima U. https://www.maddalena.it/en/products/thermal-energy/microclima-u/95. (Accessed 1 April 2023).

[30] Eastron SDM 630 series. https://www.eastroneurope.com/products/view/sdm630modbus. (Accessed 1 April 2023).

[31] Delta Strumenti, thermocouple wire (in Italian), https://www.deltastrumenti.it/images/pdf/termocoppie/in_cavetto/Termocoppia_cavetto.pdf. (Accessed 1 April 2023).

[32] 4-wires PT100 RTD. https://www.tc.co.uk/downloads/Rigid-Stem-Resistance-Thermometers-Type-16.pdf. (Accessed 1 April 2023).

[33] S+S Regeltechnik, Sleeve sensor with temperature measuring transducer THERMASGARD® HFTM - Modbus - T3. https://www.spluss.de/en/sleeve-sensor-with-temperature-measuring-transducer-thermasgard-hftm-modbus-t3-modbus-d101-62a6-0000-000-p4138/?search=1. (Accessed 1 April 2023).

[34] S+S Regeltechnik, Room humidity and temperature sensor HYGRASGARD RFTF – Modbus. https://www.spluss.de/en/room-humidity-and-temperature-sensor-hygrasgard-rftf-modbus-modbus-1201-42b6-6000-000-p3398/. (Accessed 1 April 2023).

[35] Davis instruments, weather station vantage Pro2. https://www.davisinstruments.com/vantage-pro2/. (Accessed 1 April 2023).

[36] A. Galgaro, G. Emmi, A. Zarrella, M. De Carli, Possible applications of ground coupled heat pumps in high geothermal gradient zones, Energy Build. 79 (2014) 12–22, https://doi.org/10.1016/j.enbuild.2014.04.040.

[37] G. Emmi, M. Cavazzuti, M. Bottarelli, A management strategy for multi-source heat pump systems, Int. J. Heat Technol. 40 (4) (2022) 879–887, https://doi.org/10.18280/ijht.400403.

[38] https://www.bitzer.de/. (Accessed 1 April 2023).

[39] J. Vivian, G. Emmi, A. Zarrella, X. Jobard, D. Pietruschka, M. De Carli M, Evaluating the cost of heat for end users in ultra low temperature district heating networks with booster heat pumps, Energy 153 (2018) 788–800, https://doi.org/10.1016/j.energy.2018.04.081.