An optimization of the switching temperature for an energy system with a double-source heat pump

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Abstract

This paper presents a dynamic analysis of a double-source air-conditioning system, aiming at optimally combining, from the economic point of view, the use of the air and the soil as heat source/sink for a heat pump. The energy model is developed in TRNSYS environment, where a novel component, named Type, for the modelling of the heat pump is adopted. The purpose of the analysis is to investigate the performances of the optimized system, obtained by varying the switching temperature and the sizes of the borehole field and the storage tank, for the climate conditions of two different localities, Helsinki and Athens.

Key Innovations

- A system with a novel double-source heat pump TRNSYS Type is investigated.
- An economic optimization of the switching temperature is obtained with GenOpt.
- A reduction of the borehole field is obtained by using the double-source heat pump.
- An energy analysis is performed on the optimized system, for a cold and a warm locality.

Practical Implications

When speaking about multi-source energy systems, the optimization of the plant is fundamental; the optimum switching temperature in the double-source heat pump leads to the reduction of the geothermal field.

Introduction

Enhancing the efficiency of the energy systems for the air-conditioning in buildings and reducing the related primary energy consumption through the exploitation of renewable energy sources have become of great interest in the late years. In the field of multi-source energy systems, the double-source, or hybrid heat pump represents a valuable solution to increase the efficiency of the system, compared to a more common air-source heat pump and, at the same time, to reduce the size of the borehole field that should be coupled to a ground source heat pump (Grossi et al. 2018). These solutions can, therefore, decrease the initial investment costs, depending on the plant configurations and on the boundary conditions (Catalina et al. 2011), such as the climatic area. Indeed, despites the dual-source heat pump requires two heat exchangers, one for the air and one for the ground, the second one is generally more expensive than the air-side heat exchanger, thus this solution can still lead to an investment costs reduction. Nevertheless, borehole heat exchangers, depending on the energy demand of the building, need a drilling area that is considerable and that might be not available in the nearby of the building. Therefore, when leading to the borehole field size decrease, the addition of an air-side heat exchanger could be a good compromise, although its related noise production and visual impact should be considered.

Furthermore, an optimisation of the energy system should be carried out to avoid unnecessary oversizing of the components and to further increase the efficiency of the heat pump (Hou et al. 2020).

For this reason, more often optimization methods are combined with building simulation analysis (Nguyen et al., 2014). The optimization algorithms can be used to achieve an optimum system design configuration (Bany Mousa et al. 2019) that aims at minimizing, for example, the economic impact of the system or the greenhouse gases emissions. In the field of energy systems combined with renewable energy sources, Bany Mousa et al. (2019) evaluated several combinations of photovoltaic panels and Fresnel collectors for industrial rooftop applications, with single and multi-object optimizations from the economic, environmental and technical perspectives.

Boonbumroong et al. (2011) used GenOpt (Wetter, 2001) to identify the best configuration of a solar assisted heat pump (SAHP) that minimizes the total cost (investment and operating costs) of the plant.

The present study consisted in the dynamic energy modelling of a double-source system where a heat pump exploits both the air and the soil as heat source/sink, for providing space heating and cooling to a historic building. The dynamic energy model of the building and of the plant system is developed in TRNSYS (University of Wisconsin, 2012) environment where a novel TRNSYS "Type" (TRNSYS element, containing the mathematical model of the component) for the heat pump is employed, which can be used for different simulation applications. The optimization process was carried out using GenOpt, with the aim of identifying an optimal switching temperature for the investigated case study, which allows increasing the efficiency of the system, reducing the size of the borehole field and decreasing the thermal drift of the ground temperature, due to unbalanced operating condition. The simulations are carried out for an historic non-residential building, under different climatic

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conditions, to make it possible a comparison of the optimal switching temperature for the different locations. Through this approach, it is possible to investigate and predict in detail the thermal and electrical behaviour of double source heat pump systems, which can be used in the context of both retrofitted and new buildings, under different weather conditions.

Methods

In this section, the energy system, modelled in TRNSYS, is described. A novel TRNSYS Type was used to simulate the operating conditions of the hybrid heat pump. In particular, the Type uses the polynomials of the compressors to derive the electrical energy absorbed by the compressor and the heat exchanged at the load and source sides of the heat pump. The Type is very flexible, as it can be used to simulate a reversible heat pump, which can produce heat at high or low-temperature through a cascade or single-stage configuration. Moreover, the efficiency of the heat exchangers can be modified by the user, who, in this way, can easily consider different types of heat exchangers. Thus, different heat sources fluids, like water, water and antifreeze mixture and air can be employed by modifying the specific heat of the heat carrier fluid. When operating with air as the thermal source, the model can consider the penalization in the heat pump performances due to the defrost cycles (as a function of the climatic data of the location), whose frequency and duration can be set by the user.

In this work, the heat pump is employed for the provision of space heating and cooling to a historical building. The heat pump is connected to a thermal energy storage from which heat is withdrawn for the air-conditioning of the analysed building, whose terminal units are fan-coils.

For the 5 years simulations, carried out with a timestep of 6 minutes, the Test Reference Year (TRY) data for the cities of Helsinki and Athens are obtained from EnergyPlus database (Energy Plus, 2020).

The energy system

The thermal load profile of a two-story historic non-residential building, with a heated area of 364 m^2 and a volume of 973 m^3, is investigated. The building envelope is mainly composed of solid brick walls and the structures are not thermally insulated. The windows are single glazing and the external upper covering of the building is a wooden roof. The building was simulated in TRNSYS environment (Type 56). Table 1 summarizes the main thermal properties of the building envelope.

Table 1: Thermal properties of the building envelope for the case study.

	Thickness [cm]	**U-value** [W/(m^2 K)]
External Wall	50	1.49
Adjacent Wall	50	1.32
Ground Floor	66	1.27
Intermediate Floor	36	1.61
External Roof	10	2.50
Windows	-	4.7

Table 2 shows the monthly values for the thermal energy demand and the annual heating and cooling demands, for the two case studies. The thermal load for Helsinki is heating-dominant, with a heating/cooling ratio around 35, Athens is cooling-dominant, with a heating/cooling ratio equal to 0.6.

The heating and cooling distribution system is linked to the heat pump load-side through a thermal storage. The set point temperature for the storage is set to 42°C for heating operation and to 8°C for cooling (with a dead band of 5°C).

The borehole heat exchangers (BHEs) are simulated using TRNSYS Type 557a and are double U-tube. The main properties for the borehole field are presented in Table 3. On the other hand, at the source/sink-side of the heat pump, a switch temperature ($T_{switch,H}$ or $T_{switch,C}$) defines the thermal source or sink to be used in heating and cooling respectively. Indeed, the investigated machine is a double-source or hybrid heat pump, which employs either the air or the ground, depending on the temperature of the external air, selecting the most favourable temperature to increase the energy efficiency. In particular, in heating operation, when the outdoor air is higher than the $T_{switch,H}$, the heat pump uses the air as the heat source, otherwise it uses the ground. On the contrary, in cooling operation, when the outdoor air temperature is lower than the $T_{switch,C}$, the air is used as the thermal sink, instead of the ground.

The calculation of the thermal load of the building allows the determination of the heat pump size. For this reason, in heating the peak load of 120kW for Helsinki and 40kW for Athens have been considered, while in cooling the peak load is of 22 kW for Helsinki and of 44 kW for Athens. After establishing the heat pump size to match all the building load, an optimization was carried out in order to find the best size for the thermal storage tank, the number of borehole heat exchangers and the switching temperature in heating and cooling.

The optimized system is compared to the reference system, which employs only the ground as thermal source/sink and whose size is designed using the ASHRAE method (ASHRAE, 2011).

Table 2: Thermal Energy Demand for the building, for the Helsinki and Athens.

	Helsinki	**Athens**
	[kWh]	[kWh]
Jan	-16966	-4557
Feb	-15853	-4525
Mar	-13239	-3018
Apr	-7856	-749
May	-2056	0
Jun	0	6252
Jul	1022	9728
Aug	1113	9591
Sep	0	5416
Oct	-8769	0
Nov	-14268	-1429
Dec	-16395	-4278
HEATING	-95402	-18556
COOLING	2735	30987

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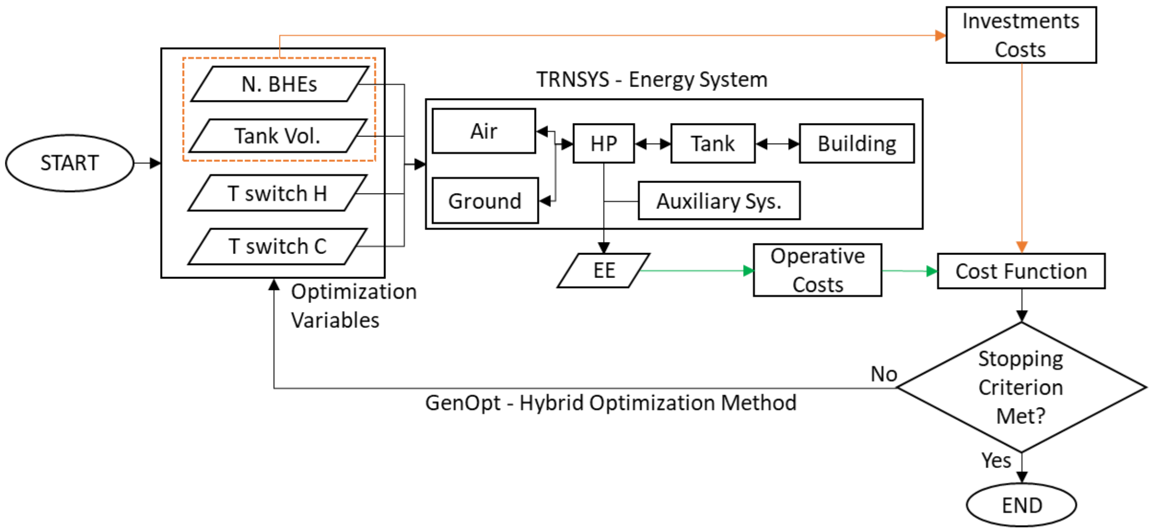

Figure 1: Scheme of the interaction between the energy system and the optimization models.

Table 3: Characteristics of the BHE field.

	Helsinki	Athens
Ground		
Specific heat	1000 J/(kg K)	
Density	2500 kg/m^3	
Undisturbed temperature	5.2 °C	17.9 °C
Thermal conductivity	2.2 W/(m K)	
Thermal gradient	0.03 °C/m	
Specific volume heat capacity	2500 kJ/(m^3 K)	
Pipe		
Length of each borehole	100 m	
Thermal conductivity	0.35 W/(m K)	
Outer diameter of pipe	32 mm	
Inner diameter of pipe	26 mm	
Center-to-center distance	78 mm	
Distance between BHEs	7 m	
Fluid		
Composition	Water-Glycol (20%)	Pure Water
Specific heat	4.02 kJ/(kg K)	4.186 kJ/(kg K)
Density	1022 kg/m^3	1000 kg/m^3

The optimization

For the optimization, a hybrid model, combination of the Particle Swarm Optimization (PSO) algorithm and the Hooke-Jeeves algorithm is used, through the TRNOpt interface of TRNSYS, which calls GenOpt program. More in detail, the optimization process starts with the PSO method: an established number of simulations is carried out in parallel, with different combinations of the optimization variables. In particular, a set of potential solutions (particles) to the optimization problem evolve during the simulation process by a given number of iterations. This evolution is driven by some update equations, which model the position of the particles in the next iteration (or generation). The algorithm stops when all the generation of particles, which are given as the inputs to the TrnOpt software, have been tested or when convergence conditions are satisfied (Baños et al. 2011).

After that, the overall minimum point obtained with the first algorithm becomes the starting point for the Hooke-Jeeves algorithm.

The flow chart in Figure 1 schematizes the interaction between the optimization model and the energy system dynamic model. The number of borehole heat exchangers and the volume of the tank determine the investment costs and together with the switching temperatures variables are given as parameters to the energy system model. The same model allows the computation of the total electrical energy absorbed, from which the operating costs are calculated. The total cost can therefore be obtained and compared with the results of the other simulations in order to modify the optimization variables.

Indeed, the results of the optimizations provide the optimal values for the parameters, which minimize the cost function expressed in Equation (1):

$$C_{TOT} = C_0 + \sum_{n=1}^{lifetime} \frac{(1+t_i)^n}{(1+t_d)^n} \cdot c_{kWh} \cdot E_{grid} \quad (1)$$

where, C_{TOT} is the sum of the investment costs (C_0), and the operating costs, calculated over a lifetime of the plant of 20 years, assuming an inflation rate t_i of 1% and a discount rate t_d of 3%. In Equation (2), c_{kWh} are the costs of the electrical energy, set to 0.2 €/kWh (evaluated for the year 0), and E_{grid} is the electrical energy, expressed in kWh. The simulation time of 5 years is chosen for monitoring the effect of the decrease in the efficiency due to the thermal drift of the ground, while allowing an acceptable computation time. In particular, the optimization is carried out by minimizing the cost of total electrical energy that is absorbed during the 5 years. Therefore, by considering the lifetime of 20 years, it is assumed that the total energy absorbed during the 5 years is equally distributed every year and then kept constant for the 20 years of operation. This is a simplification because, in real applications, for each year there might be an increase in the electrical energy absorbed, due to the

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decrease in efficiency of the heat pump, which is related to the thermal drift effect of the ground.

The investment costs of the storage tank C_{Tank} are calculated as in Equation (2) and depend on the tank volume V_{Tank}, expressed in m³.

$$C_{Tank} = 890.77 \cdot V_{Tank} + 636.58 \;\; [€] \qquad (2)$$

Instead, for the BHEs field, the investment cost was set to 50 €/m per unit length of the installed borehole heat exchanger. The operating costs are related to the HP, the fan and the circulators. Table 4 reports the most important parameters used in GenOpt for the optimization.

Table 4: Range and type of variables used into the TrnOpt interface.

	Units	Type	Step	Extremes
Helsinki				
N. of BHEs	[-]	Discrete	1	From 8 to 17
Storage Tank	[m³]	Discrete	0.5	From 2.5 to 3.5
$T_{switch,H}$	[°C]	Continuous	1	From 5 to 15
$T_{switch,C}$	[°C]	Continuous	1	From 8 to 25
Athens				
N. of BHEs	[-]	Discrete	1	From 4 to 11
Storage Tank	[m³]	Discrete	0.5	From 1.5 to 3
$T_{switch,H}$	[°C]	Continuous	1	From 6 to 20
$T_{switch,C}$	[°C]	Continuous	1	From 10 to 28

Results

In this section, the results of the optimization and the energy analysis for the two climates are presented. The optimal values that have been found from the optimization process are shown in Figure 2 and are reported in Table 5, where also the costs of the plant, considering the established lifetime, are presented for the two case studies.

Figure 2, in particular, reports the results of the optimization (the total cost of the plant), obtained by varying the sizes of the BHEs field and the storage tank and the switching temperatures, for the case of Helsinki.

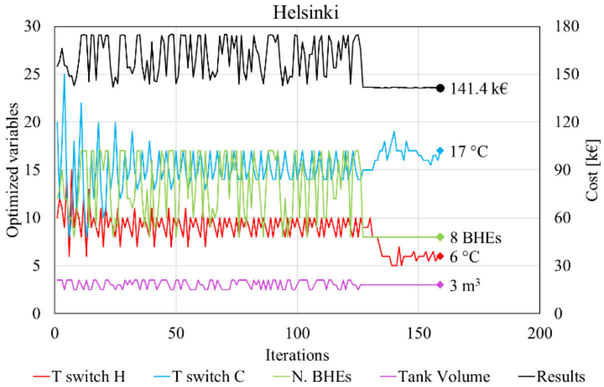

Figure 2: Optimization variables and results - Helsinki.

The minimized cost, with the corresponding optimized variables are then highlighted.

As for Helsinki, the same procedure for the case study of Athens is carried out, and the optimization variables and results can be seen in Figure 3.

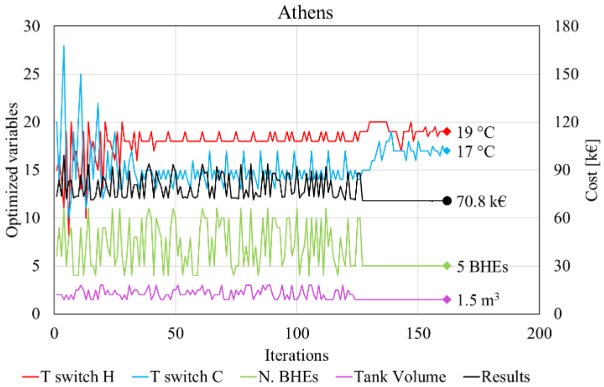

Figure 3: Optimization variables and results - Athens.

In Table 5, also the values for the reference system, which employs only the ground as the thermal source/sink, are reported. It can be seen that for the two case studies the BHEs field is minimized, thus obtaining a reduction of the investment costs. Indeed, for the analysed case studies, where a considerable number of boreholes is involved, the variation of the borehole heat exchangers field is largely the most affecting variable on the total cost of the plant, and its size reduction leads to a relevant decrease in the investment costs. On the other hand, the variation of the switching temperature has a lower influence on the cost, also due to the extreme climatic conditions considered and to the related not favourable external air temperatures for the heat pump operation. With the optimization, a reduction of 18% in the total cost is obtained for the case of Helsinki and of 19% for Athens. If the cost of the additional air-side heat exchanger is considered, this cost saving decreases, but it is still favourable. Indeed, assuming for the air-side heat exchanger a cost of €11000 for the case study of Helsinki and €5000 for the case of Athens, the reduction in the total cost decreases to 11% and 13.4%, respectively, compared to the reference cases.

Table 5: Optimization variables and results for the cases, for the optimized system (Opt.) and the reference system (Ref.).

	Helsinki		Athens	
	Opt.	**Ref.**	**Opt.**	**Ref.**
N. of BHEs	8	16	5	10
Storage Tank [m³]	3	3	1.5	2
$T_{switch,H}$ [°C]	6	-	19	-
$T_{switch,C}$ [°C]	17	-	17	-
Results [k€]	140.5	170.3	70.8	87.5

However, the efficiency of the system is lower, as expected, with the reduced borehole field. Therefore, the operation costs for the reference case are 12% lower for the case of Helsinki, with a SCOP (Seasonal Coefficient of Performance) of 3.6 and a SEER (Seasonal Energy Efficiency Ratio) of 8.1 for the 5th year of operation. The COP is the ratio of the energy flux exchanged at the

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condenser in heating operation (load-side of the heat pump) divided for the electrical power absorbed by the compressor. The EER is the ratio of the energy flux exchanged at the evaporator in cooling operation (load-side of the heat pump) divided for the electrical power absorbed by the compressor. Correspondingly, the SCOP is computed as the energy provided to the building during the heating season, over the electrical energy absorbed by the heat pump; the SEER is the thermal energy extracted by the building during the cooling season, divided by the electrical energy absorbed by the heat pump. Similarly, for Athens, the operation costs for the reference case are 20% lower compared to the optimized double-source HP, with a SCOP of 4.3 and a SEER of 4.2 for the 5th year of operation.

Figure 4 shows the switching temperature control, for Helsinki, when the heat pump operates in heating mode. When the air temperature (*T Air* in the figure) is higher than $T_{switch,H}$, set to 6°C, the HP uses the air as the thermal source. A relatively high $T_{switch,H}$, allows avoiding the necessity of defrosting cycles from the HP and to achieve better efficiencies. In Figures 4 and 5, *T Switch H* and *T Switch C* are the set $T_{switch,H}$ and $T_{switch,C}$, *T Ground* represents the temperature of the fluid exiting the BHEs, *COP Air* and *COP Ground* represent, for the corresponding timestep of operation of the HP, the performances of the heat pump in heating operation, when the system is working with air or with the ground as thermal source/sink respectively.

In the same way, Figure 5 shows the switching temperature control for Athens and the relative heat pump performances (EER), when operating in cooling mode. The obtained $T_{switch,C}$ allows to decrease the unbalanced thermal loads in cooling and heating, diminishing the negative effect of the thermal drift.

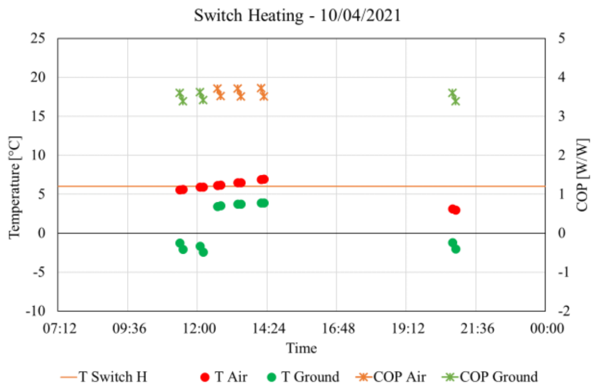

Figure 4: Switching temperature control and performances in heating - Helsinki.

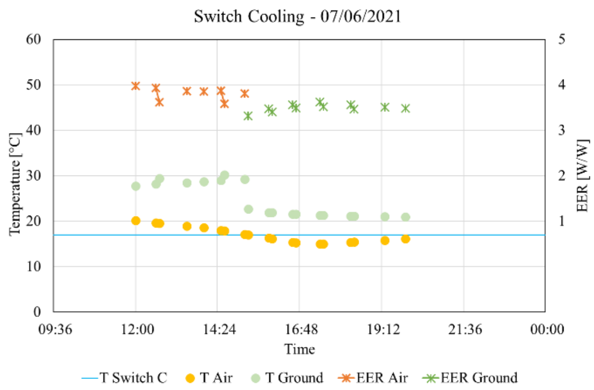

Figure 5: Switching temperature control and performances in cooling - Athens.

Speaking about the phenomenon of the thermal drift at the ground, Figure 6 (a) and (b) reports the average ground temperature for the 20 years of operation, obtained using Type 557a of TRNSYS for the case of the reference energy plant, where only the BHEs field is employed, and the double-source HP. A better response to the unbalanced thermal load can be seen for the reference system: for Helsinki a smoother decrease in the ground temperature is registered; on the contrary, in Athens a slower increase in the soil temperature is obtained from the simulations. The temperature drift of the ground with the double-source heat pump is equal to -4.7°C for the case of Helsinki and +3.3°C for the case of Athens.

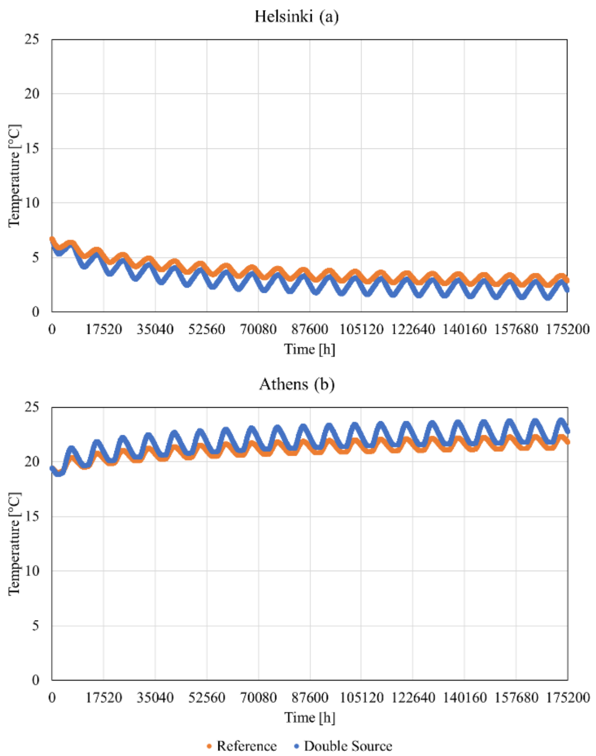

Figure 6: Average soil temperatures for Helsinki (a) and Athens (b), for the reference and double source energy plants.

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As already discussed, from the economic point of view, the reduction of the BHEs field leads to a cost benefit that is greater compared to the effect of the reduced energy efficiency of the double-source HP.

Analysing more in detail the optimized energy systems, Figure 7 provides, for each month of the 2nd and 5th year of simulation, the thermal energy exchanged at the load side of the heat pump, divided by the employed source/sink (the air or the ground) and by use (cooling and heating). For the case study of Helsinki (a), the thermal energy supplied to the building in heating, using the ground as the heat source, is 94% of the total needed. The ground is used as the thermal sink for 94.5% of the cooling demand. In the same way, for Athens (b), these values correspond to 99% and 98% respectively.

The pie charts in Figure 8 highlight the operating hours of the HP, in heating and cooling, using the ground or the air as the thermal source and sink respectively, for the two case studies. Overall, for the case of Helsinki, the HP works for 93% of the time in heating, using the ground as the heat source, while for 5% of the time it works with air. For 1.7% of the time, the HP operates in cooling, using the ground as the thermal sink. On the other hand, for the case of Athens, the HP works for 100% of the time in heating, using the ground as the heat source, while 99% of the time operates in cooling using the ground as the thermal sink.

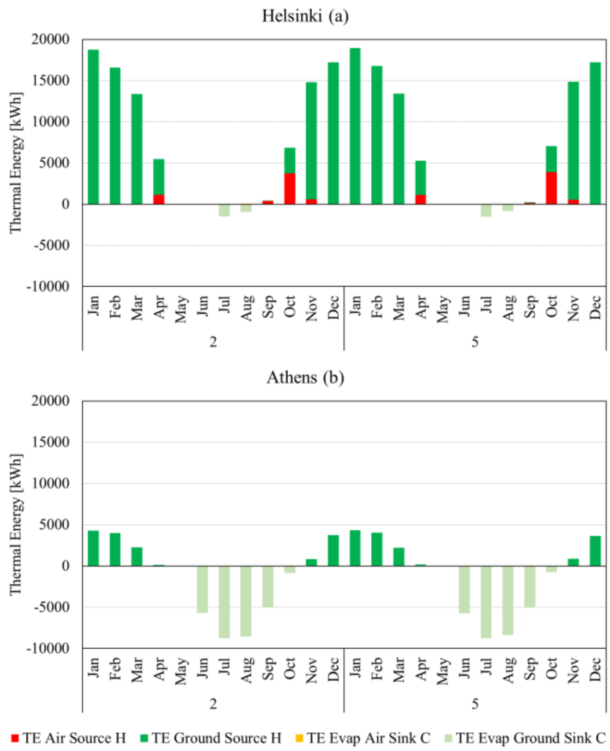

Figure 7: Thermal energy at the load side of the HP, divided by source/sink and by use (heating H, cooling C).

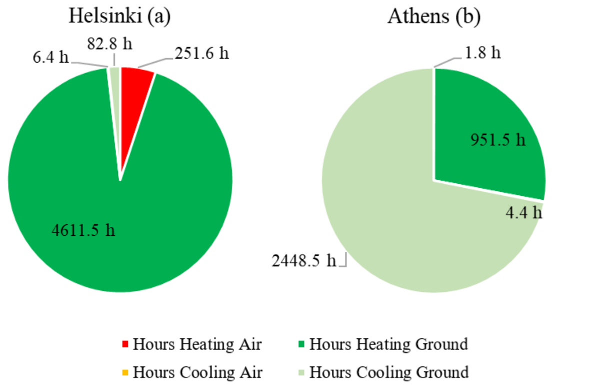

Figure 8: Hours of operation in heating and cooling, divided by source/sink.

Table 6 reports the electrical energy demanded by the heat pump, divided by contributions, for the 5th year of operation, for Helsinki case study. The increase in the HP electrical energy demand of the 5th year of operation, related to the 1st year of operation, is of 7% for the case of Helsinki. Figure 9 illustrates the mean monthly performances of the HP, during the five years of operation, divided by source/sink and by use.

Table 6: Electrical energy demanded by the HP in heating (H) and cooling (C), by source - Helsinki.

	Air - H [kWh]	Geo - H [kWh]	Air - C [kWh]	Geo - C [kWh]
Jan	0	5974	0	0
Feb	0	5304	0	0
Mar	0	4203	0	0
Apr	302	1264	0	0
May	12	0	0	0
Jun	0	0	0	0
Jul	0	0	-12	-211
Aug	0	0	-12	-111
Sep	36	17	0	0
Oct	1046	945	0	0
Nov	149	4400	0	0
Dec	0	5376	0	0

The trend shows, an increase in the EER values in the 5 years and a decrease of the COP values, due to the unbalanced heating load which negatively affects the heat exchange with the ground.

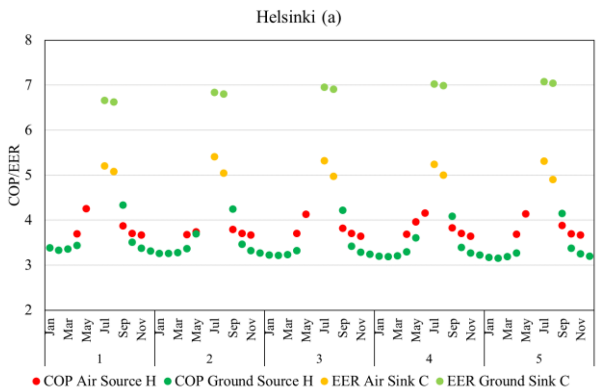

Figure 9: Performances of the HP, divided by source/sink and by use (heating H, cooling C) - Helsinki.

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For the 5th year of operation, the SCOP has a value of 3.2, while the SEER is equal to 6.9, considering both the air and the ground as thermal source/sink. From this graph, it is interesting to notice that, with the selected switching temperature, the air is mostly chosen as the thermal source in the mid-seasons, obtaining an increase in the efficiency of the system during those periods.

Following the same procedure as for the case of Helsinki, Table 7 reports the electrical energy demanded by the heat pump for Athens.

Table 7: Electrical energy demanded by the HP in heating (H) and cooling (C), by source - Athens.

	Air - H [kWh]	Geo - H [kWh]	Air - C [kWh]	Geo - C [kWh]
Jan	0	1091	0	0
Feb	0	1020	0	0
Mar	0	560	0	0
Apr	0	49	0	0
May	0	0	0	0
Jun	0	0	-19	-1688
Jul	0	0	0	-2849
Aug	0	0	0	-2797
Sep	0	0	0	-1616
Oct	0	0	0	-220
Nov	2	219	0	0
Dec	0	904	0	0

In this case, the increase in the HP electrical energy demand of the 5th year of operation, related to the first, is of 3%. Moreover, Figure 10 illustrates the mean monthly performance of the HP. An opposite trend can be observed for the case of Athens, where the performance of the HP when using the ground as a source or sink, during the 5 years, decrease in cooling operation and increase in heating operation, as a consequence of the ground temperature growth. Also in this case, the air is mainly chosen as the thermal source/sink during the first and final months of the heating and cooling seasons. For Athens, for the 5th year of operation, the SCOP has a value of 3.2, while the SEER is equal to 4, considering both the air and the ground as thermal source/sink.

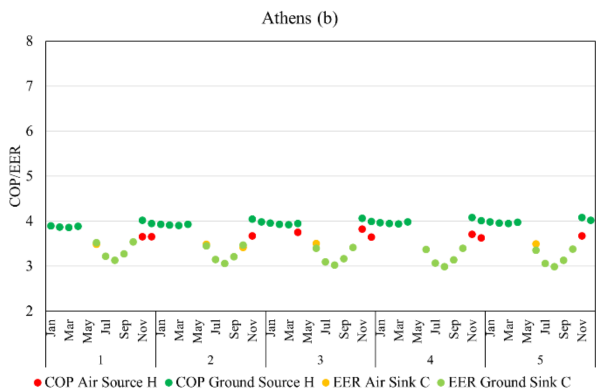

Figure 10: Performances of the HP, divided by source/sink and by use (heating H, cooling C) - Athens.

Table 8 shows the electrical energy demand of the auxiliary devices, for the 5 years of operation and for the case of Helsinki. These values correspond to nearly 5% of the electrical energy demanded from the HP. Moreover, the same table reports the cost of the energy plant, comprehensive of the investment cost of the BHE field, the storage tank and the operating costs (considering the lifetime of 20 years) due to electrical energy absorbed by the HP compressor, the fan of the air-source HP, the circulator of ground source loop, the circulator at the load side of the heat pump and the circulator of the air conditioning distribution system.

Table 8: EE Auxiliary consumption and total costs of the energy plant - Helsinki.

Year	**EE Auxiliary** [kWh]	**Total Costs** [k€]	**Investment Costs** [k€]
1	1370	62.3	43.3
2	1404	19.6	
3	1409	19.8	
4	1383	19.6	
5	1422	20.2	

Similarly, Table 9 contains the electrical energy required by the auxiliary devices (4% of the energy required by the HP), and the plant costs for the 5 years of operation and for the case study of Athens. It can be noticed that the cost of the plant for the case of Helsinki is about 21% higher than the case of Athens.

Table 9: EE Auxiliary consumption and total costs of the energy plant - Athens.

Year	**EE Auxiliary** [kWh]	**Total Costs** [k€]	**Investment Costs** [k€]
1	537	35.6	27.0
2	538	8.8	
3	537	8.9	
4	522	8.7	
5	540	8.9	

Conclusions

The paper presents the dynamic model of a double-source air-conditioning system for the provision of heating and cooling to a historic building, for the climate conditions of Helsinki and Athens. In detail, the heat pump uses either the air or the ground as the thermal source/sink and, at the load side, it is linked to a storage tank. An economic optimization is carried out with the aim of minimizing the cost of the energy plant, varying the sizes of the storage tank and the borehole heat exchanger field and by identifying the best switching temperatures for heating and cooling. These temperatures are compared with the temperature of the external air determining the switch between the air and the ground. The energy model is developed in TRNSYS environment, using a novel double-source heat pump TRNSYS Type and the connection, through the TRNOpt interface, to GenOpt optimization tool.

For the case study of Helsinki, the optimal switching temperatures are equal to 6°C for heating operation and 17°C for cooling operation, resulting in an overall cost of 140.5 k€. This cost includes the investment costs of the borehole field, of the tank and the operating costs of the circulators, the fan and the heat pump, related to the demanded electrical energy and considering a lifetime of 20 years. This result is obtained employing a number of borehole heat exchangers (8 boreholes, 100 m deep) that

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is reduced compared to the design size (16 boreholes, 100 m deep) and with the design size of the storage tank. The optimization, carried out for 5 years of simulation, allows saving nearly 18% of the cost, in comparison to the ground-source heat pump.

For the case study of Athens, the optimal size was obtained with a borehole field composed by 5 boreholes, instead of the design value of 10 boreholes and with a reduced volume of the storage tank. The switching temperature is of 19°C for the heating operation and of 17°C for cooling operation. The reduction of the borehole field leads to a decrease in the overall cost of 19% (with 70.8 k€), compared to the reference case of the ground source heat pump. However, this study highlighted that, in particular for the case of Athens, there is no convenience in using the double-source heat pump rather than the ground-source heat pump: during the cooling period, the air temperature is too high compared to that of the ground. For example, the performances of the heat pump in cooling with a switching temperature of 25°C decrease by more than 11% when using the air instead of the ground as the heat sink.

As a future research activity, it would be interesting to investigate different case studies, such as the application to a well-insulated system or to less extreme climate conditions, where the convenience of employing the double-source heat pump could be more evident.

Nomenclature

Symbols

C	Cost [€]
c_{kWh}	Costs of the electrical energy [€/kWh]
COP	Coefficient Of Performance in heating [W/W]
EER	Energy Efficiency Ratio in cooling [W/W]
E_{grid}	Electrical Energy [kWh]
SCOP	Seasonal Coefficient of Performance [kWh/kWh]
SEER	Seasonal Energy Efficiency Ratio [kWh/kWh]
T	Temperature [°C]
t_i	Inflation Rate [%]
t_d	Discount Rate [%]
U	Transmittance [W/(m^2 K)]
V_{Tank}	Tank Volume [m^3]

Subscripts and apex

C	Cooling
H	Heating
n	Year number n
switch	Switch Temperature
TOT	Total
Tank	Storage Tank

Abbreviations

BHE	Borehole Heat Exchangers
EE	Electrical Energy
GPS	Generalized Pattern Search
HP	Heat Pump
PSO	Particle Swarm Optimization
TE	Thermal Energy
TRY	Test Reference Year

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