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Dynamic modelling of a dual-source heat pump system through a Simulink tool

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Abstract. In this paper, the performance of a reversible Dual-Source Heat Pump (DSHP) system, able to exploit renewable energy from, alternatively, air and ground sources, is evaluated by using Matlab-Simulink. The actual source exploited depends on a simple control strategy on the basis of the current external air temperature. Yearly dynamic simulations have been carried out by coupling the DSHP to a detached residential building located in Bologna, in which heating and cooling loads are strongly unbalanced, and coupled to a Borehole Heat Exchangers (BHEs) field. Different case studies have been analysed in which the length of the borefield has been modified. The obtained results show that an optimal switching temperature can be determined to maximise the Annual Performance Factor (APF) for a fixed BHE field length. Additionally, it has been demonstrated that DSHPs can be very useful in order to reduce the total required length of the borehole heat exchangers and, consequently, the associated costs, and to solve the problems linked to the ground temperature drift, which can be originated by the presence of an undersized borefield and/or by unbalanced building loads. As a consequence, DSHPs can be suggested for the retrofitting of traditional ground-coupled heat pump systems in presence of undersized BHEs.

1. Introduction

During the last years, EU regulations have requested the reduction of the primary energy demand of many human activities, especially of space heating and cooling of buildings. Heat pumps are one of the most suitable heat generators for this purpose, since they permit to reduce the primary energy consumption of buildings HVAC systems by far.

As well-known, Air-Source Heat Pumps (ASHPs) performance is strongly influenced by the external air temperature. In particular, it decreases when the ambient temperature drops in winter and when the external air temperature grows up in summer [1,2]. Furthermore, ASHPs efficiency is deeply reduced by defrost cycles, which occur when the ambient temperature is low and the relative humidity is high. The frost formed on the heat pump external heat exchanger can be removed in different ways, for instance, by resorting to an electric resistance, by stopping the compressor, or by inverting the cycle. The last way, considered in this paper, is generally adopted when the heat pump can operate in reversible mode [3].

On the contrary, Ground-Coupled Heat Pumps (GCHPs) performance is more stable along the year and typically higher than that related to air-source models [4,5], since the ground has a more stable

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temperature, which is usually warmer than that of the air during winter and cooler during summer. However, HVAC systems coupled to the ground are more expensive and their installation costs increase with the total borehole field length [6].

A Dual-Source Heat Pump (DSHP) system, able to exploit, alternatively, air and ground, allows to exploit the positive aspects of both thermal sources, by using the geothermal energy source when the external air temperature is excessively low during winter (avoiding in this way also energy losses due to defrost cycles) and switching to the aerothermal energy source when the ambient temperature overcomes a reference value which depends on the control strategy. In this way, undersized borehole heat exchangers could be installed and, as a consequence, the whole investment cost of the HVAC system could be reduced.

The aim of this work is to compare the DSHP seasonal and annual performance with that obtained by traditional units, in which only ambient air or only the soil is used as external heat source (or sink), in order to evaluate the benefits of this technology.

2. Model

2.1. Building data

Yearly dynamic simulations have been carried out by coupling a heat pump system to a detached residential building, located in Bologna (Northern Italy, 44°29’N, 11°21’E), in which heating and cooling loads are strongly unbalanced. The detached residential building is a single-story house composed by four thermal zones, as shown in figure 1, and by a non-heated attic [7].

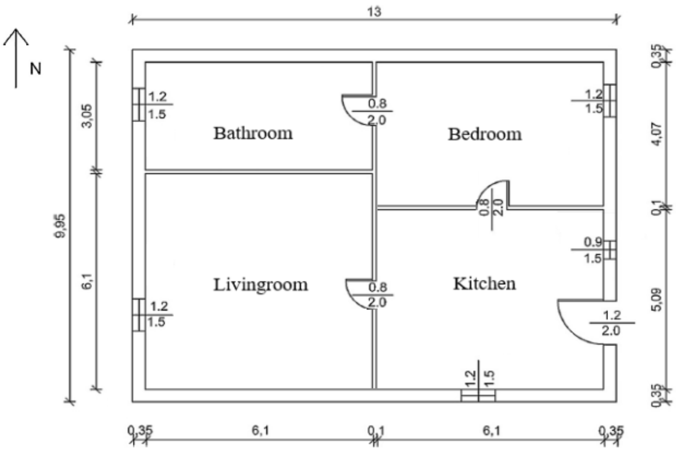


Figure 1. The detached residential building planimetry.

The net surface area is equal to 111.6 m^2 (gross floor area 129.4 m^2), the net conditioning volume is 301.5 m^3 and the surface to volume ratio is equal to 1.32 m^{-1}. Table 1 reports the main geometrical characteristics of each room, including external walls surface area, net floor area and net volume. Additionally, the infiltration ratio is set to 0.5 vol h^{-1} in every zone. The thickness and transmittance values of the opaque and clear components are reported in table 2. The level of the thermal insulation is medium and would not respect the current Italian transmittance limits [8]. Indeed, in the external walls and in the slab on grade floor only 6 cm of polystyrene (conductivity 0.038 W m^{-1} K^{-1}) have been inserted, while 8 cm have been selected for the roof. The clear components are low-emissivity (equal to 0.1) double glasses (4/12/4) filled with Argon, with a glass transmittance of 1.5 W m^{-2} K^{-1}; the frame to total surface ratio is 20% and the frame transmittance is 2 W m^{-2} K^{-1}. The external opaque components and the slab on grade transmittance values have been calculated according to the standards UNI EN ISO 6949 [9] and UNI EN ISO 13370 [10], respectively. Moreover, occupancy and equipment heat gains have not been considered. The indoor set-point temperature is 20 °C during winter and 26 °C during summer. The heating season starts on October 15th and ends on April 30th, while the cooling season goes from June 15th to August 31st. The climatic data from METEONORM [11] have been selected. The minimum and maximum external air temperature values in Bologna are -7 and 35 °C in winter and summer, respectively.

The heating and cooling loads of the building have been evaluated by means of the ALMABuild tool [12,13]. The maximum heating load is reached in January and is equal to 6.15 kW, whilst the maximum cooling load is reached in August and is equal to 1.66 kW. As evident, the building loads are strongly unbalanced, with a heating to cooling load ratio equal to 3.7.

Table 1. Main geometrical characteristics of each room.

Thermal zone	External wall surface area [m^2]	Net floor area [m^2]	Net volume [m^3]
Bathroom	29.9	18.6	50.2
Bedroom	32.9	24.8	67.0
Livingroom	39.0	37.2	100.5
Kitchen	36.0	31.0	83.8

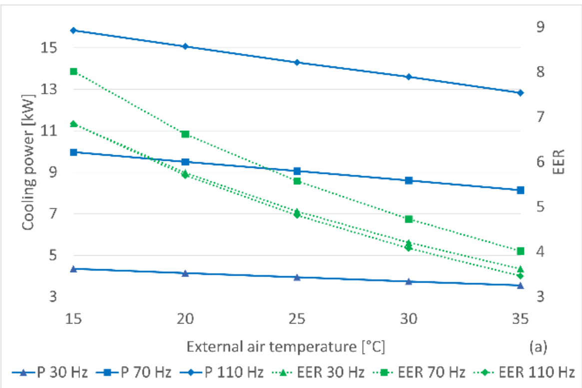
Table 2. Main characteristics of the opaque and clear components.

Component	Thickness [cm]	Transmittance [W m^{-2} K^{-1}]
External wall	34	0.38
Internal wall	10	1.79
Floor	33.5	0.28
Ceiling	39	1.04
Roof	32	0.401
External door	15	2.5
Window	2	1.77

2.2. Heat pump system

Three different inverter-driven heat pumps, coupled one at a time to the same building, have been considered: an ASHP (air-to-water heat pump), a GCHP (brine-to-water heat pump) and a DSHP (air/brine-to-water heat pump). The DSHP, modelled by considering the characteristic curves of the ASHP and GCHP, is able to exchange energy, alternatively, with both air and ground. Moreover, a back-up system is not present.

Figures 2 and 3 report the heat pump performance data from the manufacturer technical datasheets, in air-source and ground-source modes, respectively. The curves refer to a load water temperature at the heat pump inlet equal to 15 °C in summer and 35 °C in winter. In particular, figure 2 shows the heat pump thermal capacity in heating and cooling mode, COP and EER as functions of the external air temperature, whilst figure 3 plots the same data as functions of the BHE fluid temperature at the heat pump inlet.



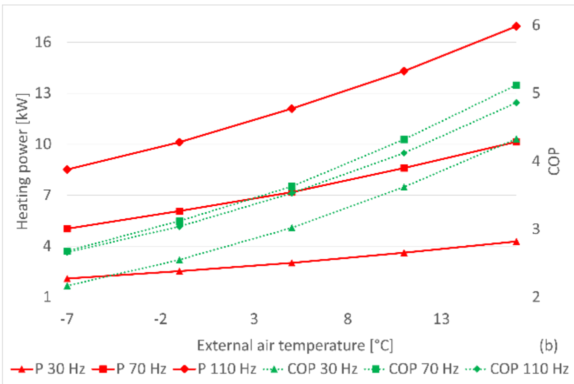


Figure 2. (a) Cooling power and EER and (b) heating power and COP (air-source mode) as functions of the external air temperature, for three values of the inverter frequency (30 Hz, 70 Hz, 110 Hz).

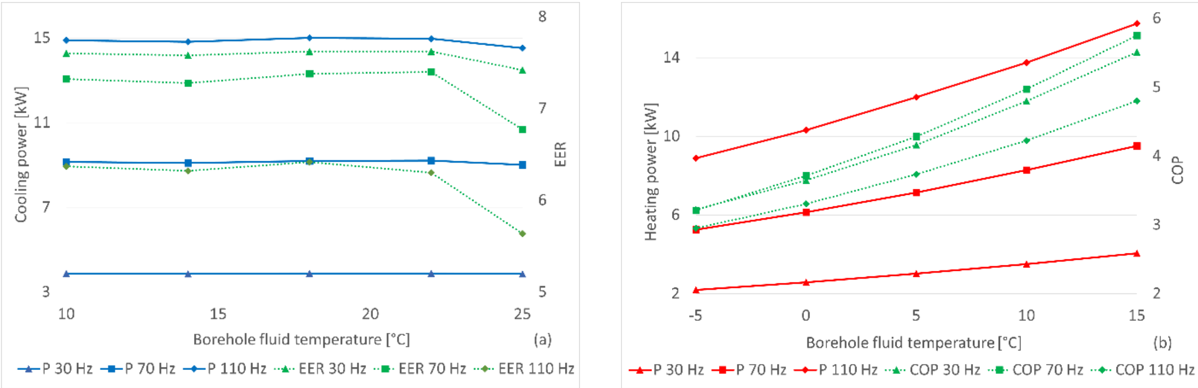


Figure 3. (a) Cooling power and EER and (b) heating power and COP (ground-source mode) as functions of the borehole fluid temperature at the heat pump inlet, for three values of the inverter frequency (30 Hz, 70 Hz, 110 Hz).

From figures 2 and 3, it is interesting to note that the EER values of the brine-to-water heat pump are more stable with respect to those of the air-to-water heat pump, in which the EER values plunge with the increase of the external air temperature. In both air and ground-source mode, the heat generators are able to cover the maximum heating load of the building without using a back-up system.

If a heat pump operating in ground-source mode is selected (GCHP or DSHP), a hydronic loop with a single-speed circulation pump is considered to connect the heat generator to a BHE field. In this paper, different borefields with vertical single U-tube BHEs have been considered. The BHEs are made of high-density polyethylene and the gap between BHEs and ground is filled by Termoplast PLUS [14], a commercial sealant mortar. In each BHE, a mixture of water and 60% Freezium [15] flows. Freezium is a commercial antifreezes borehole fluid, based on organic salts (potassium formate), with low viscosity (even at low temperatures), high thermal conductivity, no toxicity and biodegradability. The main characteristic of the BHEs (borehole diameter D_b, internal pipe diameter $D_{p,i}$, external pipe diameter $D_{p,e}$, pipe conductivity k_p, grout conductivity k_{gt}, shank spacing s) and of the borehole fluid (specific heat capacity $c_{p,f}$, density ρ_f, freezing temperature T_{ice}) have been reported in tables 3 and 4, respectively. The thermal conductivity and diffusivity values of the ground are equal to 1.97 W m^{-1} K^{-1} and 8.8E-7 m^2 s^{-1}, respectively. The annual mean temperature of the soil is set to 13.2 °C.

Table 3. BHE characteristics.

D_b [m]	$D_{p,i}$ [m]	$D_{p,e}$ [m]	k_{gt} [W m^{-1} K^{-1}]	k_p [W m^{-1} K^{-1}]	s [m]
0.13	0.0262	0.032	1.6	0.355	0.085

Table 4. Thermal properties of the borehole fluid.

$c_{p,f}$[1] [J kg^{-1} K^{-1}]	ρ_f[1] [kg m^{-3}]	T_{ice} [°C]
3070	1217	-25

[1]Property at 5 °C.

Three different case studies have been analysed in this work: heat pump system based on a GCHP (case A), heat pump system based on an ASHP (case B), heat pump system based on a DSHP (case C). In each case study, the heat pump has been coupled to the emitters (fan-coils) by mean of a hydronic loop in which an inertial tank of 500 l and a single-speed circulation pump are present. The thermal storage is located in the heat pump delivery pipe. The HVAC system is activated 24/24 h both during the heating and the cooling season. The water tank has the aim to stabilize the water temperature in the hydronic loop in presence of defrost cycles or partial loads, in order to reduce the heat pump on-off cycles. The cycle inversion due to the defrosting occurs when, at the same time and for at least 10 minutes, the ambient temperature is lower than 6 °C and the relative humidity is higher than 50%. In this case, the heat pump revers its cycle following a three-periods scheme [3].

For case A (GCHP), a borefield consisting in 2 BHEs, each 60 m long, has been considered. The borehole heat flux per unit of length, exchanged between BHEs and ground, typically ranges between 30 and 50 W m^{-1} [16,17]. A value of 34 W m^{-1} has been chosen for sizing the borehole field. For case B (ASHP), obviously no BHEs are considered. For case C (DSHP), 4 different configurations of the borehole field have been studied: 2 boreholes each 60 m long (case C1); 1 borehole 90 m long (case C2); 1 borehole 70 m long (case C3); 1 borehole 60 m long (case C4). In cases C2-C4 the total BHEs length is undersized with respect to the building energy demand.

2.3. *Software*

The considered building and heat pump system have been modelled in ALMABuild, a Matlab toolbox, operating in the Simulink environment, for the dynamic simulation of coupled building-HVAC plants [12,13]. ALMABuild is composed by different libraries for the modelling of the main components of buildings and HVAC systems. The HVAC plant described in this paper has been modelled in Simulink by means of components from both ALMABuild and CARNOT, another Simulink library for simulation of HVAC components. In figure 4 a simplified operational scheme has been presented.

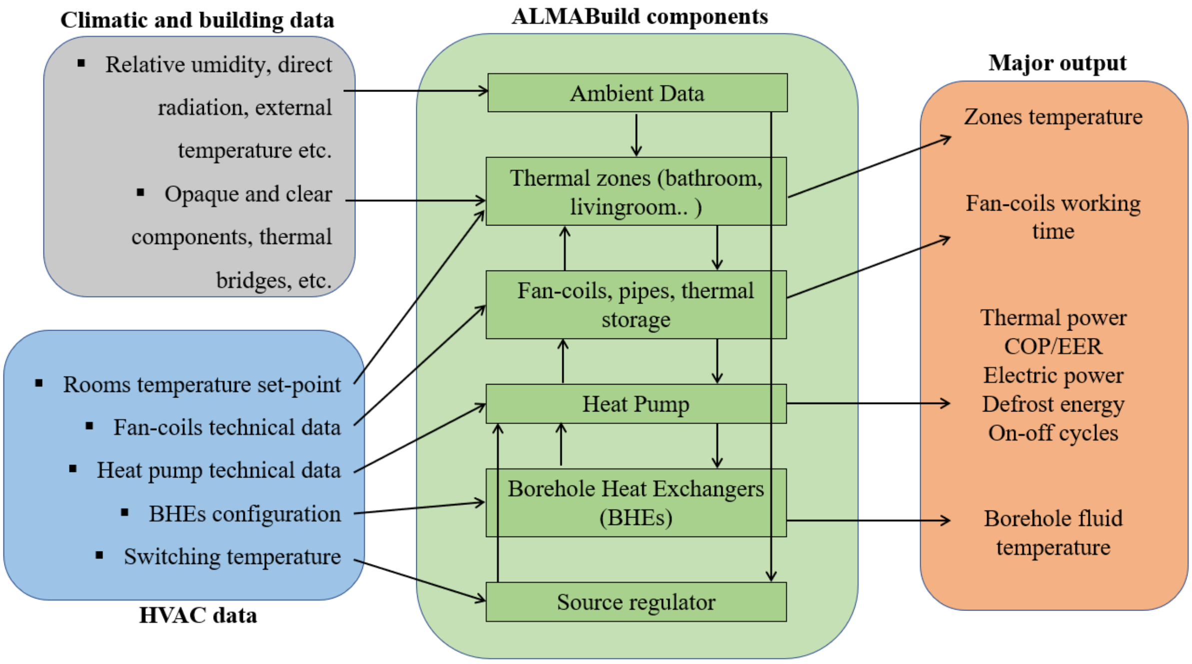


Figure 4. Software operational scheme.

Climatic, building and HVAC data (input of the software) are implemented in the ALMABuild components (thermal zones, fan-coils, heat pump etc.). Several outputs are provided by the dynamic simulations, among which the thermal zones temperature values and the heat pump working conditions. The Simulink layout of the whole system is shown in figure 5, where the main components (fan-coils FC, Thermal Storage, Heat Pump, BHE, PI controller, thermal zones, Circulation Pump) have been labelled.

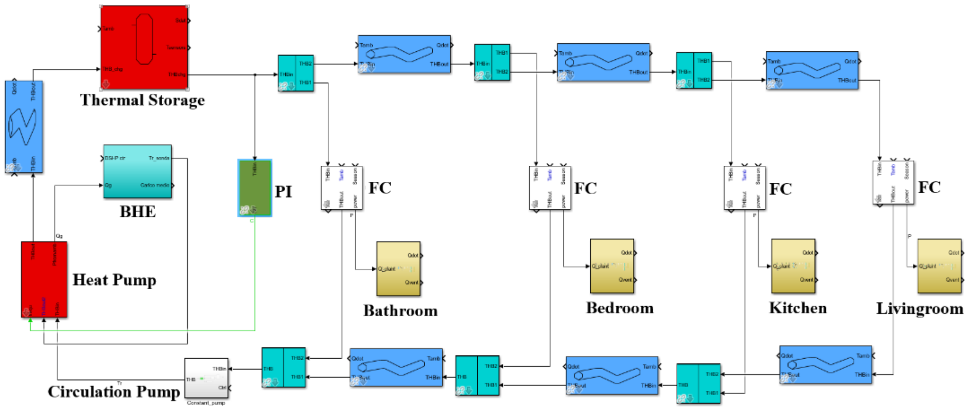


Figure 5. Simulink layout of the coupled building-HVAC system.

The long-term effects on the ground of the seasonal building loads have been taken into account by means of the *g-functions* method [18], in which dimensionless thermal response functions simulate the ground temperature trend produced by a constant dimensionless thermal load. By means of *g-functions*, the BHE fluid outlet temperature is evaluated at every simulation time step [5], whilst the thermal resistance of the single U-tube BHEs is calculated with an analytical expression [19].

In order to evaluate the heat pump performance, the GCHP, ASHP and DSHP data about thermal power, COP and EER have been implemented in the Simulink model for different values of sink and source temperature. The control strategy of the heat pump is operated by a PI controller, whose monitored variable is the outlet water temperature of the thermal storage (user side). The PI control strategy is based on a hysteresis cycle (reported in figure 6), 5 K wide and centred on the reference value, which has been set to 40 °C and 10 °C in winter and summer, respectively. According to the on-off logic, the heat pump is turned off when the water temperature, supplied to the fan-coils, exceeds 42.5 °C in winter or becomes lower than 7.5 °C in summer, whereas the heat pump is turned on when the water temperature goes lower than 37.5 °C in winter or exceeds 12.5 °C in summer.

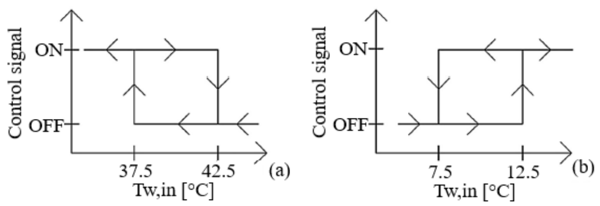


Figure 6. Hysteresis cycle for the heat pump regulation, (a) heating mode and (b) cooling mode.

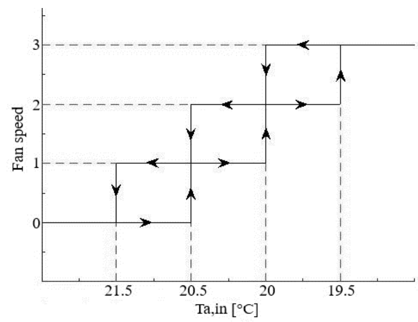


Figure 7. Hysteresis cycles for the fan-coils regulation (heating mode).

The terminal units, one for each room, are three-speed fan-coils. The reference air temperature values in the thermal zones during winter are 20.5, 20 and 19.5 °C for low, medium and high fan-coil velocities, respectively. The fan-coil hysteresis cycles are shown in figure 7. The corresponding reference temperature values during summer are 25.5, 26 and 26.5 °C. The characteristics of the selected fan-coils are specified in table 5, where $P_{h,r}$ is the thermal zone heating load, $P_{h,s}$ is the fan-coil thermal power, $m_{a,h}$ is the air mass flow rate in heating mode, $P_{c,r}$ is the thermal zone cooling load, $P_{c,tot,s}$ is the fan-coil cooling power, $m_{a,c}$ is the air mass flow rate in cooling mode, m_w is the water mass flow rate, ΔT_{fc} is the water temperature difference between fan-coil inlet and outlet, $T_{w,in}$ is the water temperature at fan-coil inlet, $T_{a,in}$ the ambient temperature and RH is the relative humidity.

The energy losses linked to the on-off and defrost (air-source mode) cycles have been considered by introducing corrective coefficients for the heat pump performance [7].

Table 5. Fan-coils characteristics.

Thermal zone	$P_{h,r}$ [W]	$P_{h,s}$[1] [W]	$m_{a,h}$[1] [m^3 h^{-1}]	$P_{c,r}$ [W]	$P_{c,tot,s}$[2] [W]	$m_{a,c}$[2] [m^3 h^{-1}]	m_w [l h^{-1}]
Kitchen	1740	1890	442	480	1540	341	253
Living room	1900	1890	442	500	1540	341	253
Bathroom	1140	1170	319	350	900	233	158
Bedroom	1400	1410	319	330	1330	233	317

[1]Heating reference conditions: ΔT_{fc} = 5 °C, $T_{w,in}$ = 40 °C, $T_{a,in}$ = 20 °C, fan high speed.

[2]Cooling reference conditions: ΔT_{fc} = -5 °C, $T_{w,in}$ = 10 °C, $T_{a,in}$ = 26 °C, RH = 60%, fan medium speed.

3. Results

In this section the results of the dynamic simulations are presented. Performance indicators are used to compare the different heat pump systems in order to find out the best configuration (equation (1)). The Seasonal Coefficient of Performance (SCOP) is the winter performance indicator, that is the ratio between the total thermal energy provided by the heat pump during winter and the corresponding electric energy used by the heat pump compressor. The Seasonal Energy Efficiency Ratio (SEER) is the summer performance indicator, namely the ratio between the total cooling energy provided by the heat pump during summer and the corresponding electric energy used by the heat pump compressor. The Annual Performance Factor (APF) is the annual performance indicator, ratio between the total energy supplied by the heat pump during the year and the corresponding total electric energy used.

$$SCOP = \frac{\sum_{j=0}^{\tau_h} E_{h,j}}{\sum_{j=0}^{\tau_h} E_{el,j}} \qquad SEER = \frac{\sum_{j=0}^{\tau_c} E_{c,j}}{\sum_{j=0}^{\tau_c} E_{el,j}} \qquad APF = \frac{\sum_{j=0}^{\tau_h+\tau_c} E_{h,j} + E_{c,j}}{\sum_{j=0}^{\tau_h+\tau_c} E_{el,j}} \tag{1}$$

3.1. GCHP, case A

In case A, the building is coupled to a HVAC system based on a GCHP. The performance of a brine-to-water heat pump is weakly influenced by the external air temperature and thus, when the BHEs are properly sized, it is quite stable along the season. In case A, the borehole field is composed by 2 vertical single U-tube boreholes, each 60 m long. In table 6 the GCHP energy performance parameters, obtained in correspondence of the first year, have been reported.

Table 6. Seasonal and annual performance factors, GCHP.

SCOP	SEER	APF
3.95	6.24	4.07

The obtained value of APF is close to that of SCOP, since most of the building energy demand is linked to the winter season. The value of SEER is about 60% greater than that of the SCOP, due to a cool source temperature and an oversized borefield with respect to the summer loads.

Furthermore, on-off cycles are frequent in the mid-seasonal months, in which the heat pump works at partial loads. The annual number of the heat pump on-off cycles reaches 837, where 507 cycles are related to the winter period and 330 to the summer period. As a consequence, the average number of on-off cycles per day is 2.6 during winter and 4.3 (+65%) during summer. This large difference is due to the oversizing of the heat pump with respect to the summer energy demand.

In figure 8 the annual trend of the BHE fluid temperature at the heat pump inlet is illustrated. The fluid reaches a minimum temperature of 0 °C during the heating season and a maximum one of 20 °C during the cooling season, and this guarantees good seasonal performance.

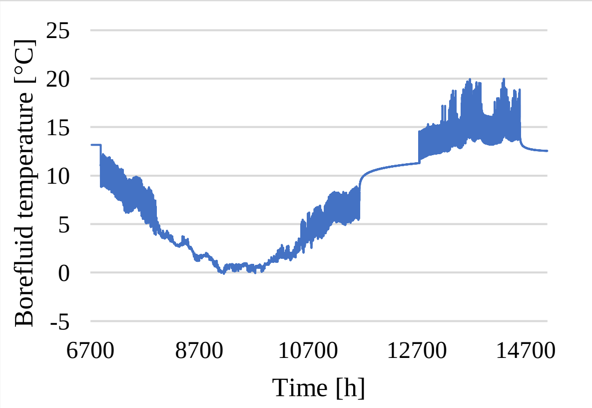


Figure 8. Annual trend of the BHE fluid temperature at the heat pump inlet, GCHP.

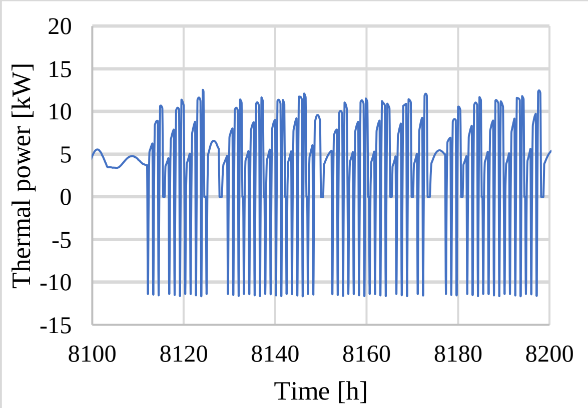


Figure 9. Heat pump power supplied from December 3rd to 7th, ASHP.

3.2. ASHP, case B

In case B, the HVAC system is based on an ASHP. An air-source heat pump performance is deeply influenced by the external air temperature. In particular, efficiency worsens with the decrease of the external air temperature in winter and with the increase of the external air temperature in summer. In table 7 the ASHP energy performance parameters in correspondence of the first year have been reported.

Table 7. Seasonal and annual performance factors, ASHP.

SCOP	SEER	APF
2.80	4.06	2.87

As evident from table 7, SCOP is 31% lower than SEER and the main reason is that a large number of defrost cycles (1762) is needed during the heating season. As an example, figure 9 shows the thermal power supplied by the heat pump in heating mode from December 3rd to 7th. It can be noted that negative power values (corresponding to defrost cycles) are present several times, even in a short winter period, to melt the ice formed on the evaporator. The annual number of heat pump start-ups is 2873, where 2547 start-ups occur during the winter period and 326 during the summer one. Consequently, the average start-up number per day is 12.9 during winter and 4.2 (about 3 times less) during summer. It is evident that the huge difference mainly depends on the winter defrost cycles. Also in this case the APF value is more similar to the SCOP one due to the predominance of the heating loads.

Moreover, the minimum and maximum external air temperature is -7 °C in winter and 35 °C in summer, respectively. This, together with the need of defrosting cycles, strongly reduces the heat pump performance with respect to that of the GCHP (case A).

3.3. DSHP, case C

In case C, a DSHP is considered, modelled by using the characteristic curves of both the ASHP and the GCHP. The actual source exploited in winter during each simulation time step depends on the comparison between the external air temperature value and a reference value called switching temperature (T_{sw}): when the external air temperature overcomes the switching temperature, energy is extracted from air (air-source mode), otherwise, energy is taken from ground (ground-source mode).

During the cooling season, the heat pump works only in ground-source mode in order to partially recharge the ground, due to the unbalanced building loads, and to maximize the performance.

In case C, the seasonal and annual DSHP performance, with the 4 different borefield configurations described in section 2.2, have been analyzed. Figure 10 shows the APF obtained in each case as a function of the switching temperature in correspondence of the first year. It is worth noting that a switching temperature equal to 26 °C (points at the far right in figure 10) corresponds to simulate a traditional GCHP.

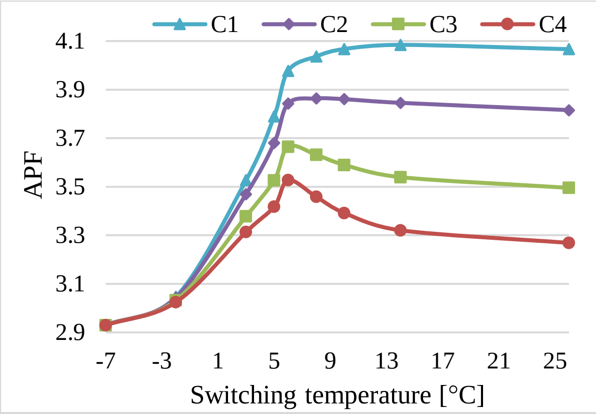


Figure 10. APF as a function of the switching temperature, with different borefield lengths.

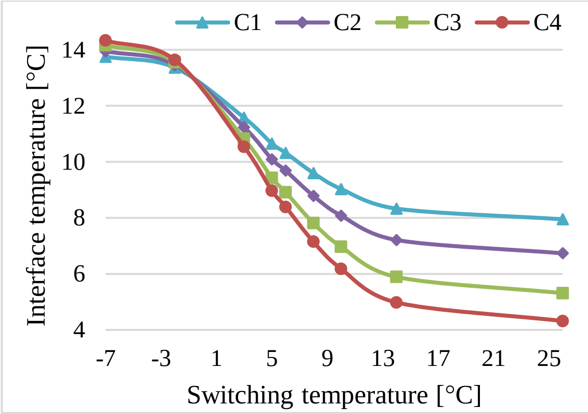


Figure 11. Mean temperature at the BHE-ground interface as a function of the switching temperature, with different borefield lengths.

It is clear from figure 10 that, as the switching temperature decreases, the distance between the curves obtained with the different total boreholes lengths is reduced, and the APF converges to the same value (2.93, point at the far left in figure 10) in correspondence of T_{sw} = –7 °C, where the results are not influenced by the borefield size since the air-source mode is predominant.

Moreover, it is possible to appreciate that, ranging from case C1 (total BHEs length: 120 m) to cases C2 (total BHEs length: 90 m), C3 (total BHEs length: 70 m) and C4 (total BHEs length: 60 m), the DSHP performance decreases for a fixed switching temperature. Indeed, the shorter the size of the borefield, the lower the fluid temperature at the BHE outlet, with consequent worse heat pump performance. As an example, the minimum annual borehole fluid outlet temperature, for a fixed switching temperature of 14 °C, varies from -2.2 °C (case C1) to -14 °C (case C4).

By comparing the results of cases C3 and C4 to each other, it is clear that, adopting an optimized switching temperature, it is possible to achieve with a shorter borefield the same APF value of a traditional GCHP coupled to a longer (and more expensive) field: the APF value of case C4 in correspondence of T_{sw} = 6 °C is even slightly larger than that achieved in case C3 (that has a longer borefield) in correspondence of T_{sw} =26 °C (ground-source mode only). Indeed, the results reported in figure 10 point out that an optimal switching temperature can be determined, to maximise the APF value for a fixed borefield length. As an example, in case C4 the APF in correspondence of T_{sw} = 6 °C (optimal value) is 8% higher than that obtained for a switching temperature of 26 °C and 20% higher than that with T_{sw} = –7 °C. Moreover, adopting a switching temperature of 14 °C even in the well-sized case C1 allows to obtain an APF slightly greater than that of the traditional GCHP (T_{sw} = 26 °C). Indeed, with T_{sw} = 14 °C, energy from the air source is extracted when the external air temperature is higher than the ground temperature and this promotes better heat pump performance (compare the heat pump performance data reported in figures 2 and 3).

In addition, from figure 10 it is possible to observe that, as the size of the BHEs rises, the optimal switching temperature, which maximizes the APF, increases. This trend is coherent since the ground utilization increases with the switching temperature, and longer BHEs guarantee higher borehole fluid temperatures, assuring better performance indicators.

Since the main reason for the differences in the APF values is the source temperature, in figure 11 the mean temperature at the borehole-ground interface is plotted as a function of the switching temperature. By referring to case C4 (red curve in figure 11), if a switching temperature of 6 °C is adopted, the interface temperature (8.4 °C) is by far higher than that reached exploiting only the ground (4.3 °C, with T_{sw}= 26 °C). Due to this large difference (about 4 °C), the system annual performance obtained exploiting both sources (T_{sw}= 6 °C) results substantially better, as confirmed by figure 10. In fact, when both aerothermal and geothermal sources are used with a DSHP, the ground loads are more balanced and the soil has the chance to recover and stay at a warmer temperature. The adoption of a correct switching temperature in presence of unbalanced seasonal building loads and short BHEs (as in case C4) can limit the ground temperature drift. For this reason, a DSHP is particularly indicated to replace traditional ground-coupled heat pumps in presence of BHEs that are undersized (for instance due to modifications of the building thermal needs).

3.4. Comparison

The comparison between the annual performance of the different heat pump systems (with the optimal switching temperature for DSHP cases) is clarified in table 8, where case B (ASHP) is used as a reference.

Table 8. APF values for each case study.

	B	C4	C3	C2	C1	A
APF (% increase)	2.87 (+0)	3.53 (+23)	3.67 (+28)	3.86 (+35)	4.08 (+42)	4.07 (+42)

From table 8 it is evident that, for cases C1-C4 (DSHP), the higher the borefield size, the higher the APF achieved: APF values increase from 23% to 42% with respect to case B by going from case C4 to case C1. Cases C1 and A have the same total BHEs length, but the first configuration refers to a DSHP that extracts energy from air when the external air temperature is higher than the ground temperature, resulting in slightly better winter performance. However, one has to bear in mind that ranging from case C1 to case C4, the investment cost related to the BHEs realization decreases significantly (total BHEs length from 120 m of case C1 to 60 m of case C4), making the adoption of DSHP promising.

4. Conclusions

In this paper, the performance of different heat pump systems, coupled one at a time to the same detached residential building with unbalanced seasonal loads, has been analysed using ALMABuild. Three heat pump configurations have been considered: a Ground-Coupled Heat Pump (GCHP), an Air-Source Heat Pump (ASHP) and a Dual-Source Heat Pump (DSHP), modelled by employing the characteristic curves of the ASHP and GCHP. The DSHP works only in ground-source mode during summer in order to partially recharge the ground, whereas the actual source exploited in winter depends on a simple control strategy: when the ambient air temperature overcomes a reference value called switching temperature, the air source is used (air-source mode), otherwise, the ground source is employed (ground-source mode).

The numerical results demonstrate that, when the borefield is well-sized, the DSHP performance is higher than that related to the ASHP and lower than that of the GCHP. In addition, the smaller the BHEs size, the lower the switching temperature that maximizes the DSHP APF. An annual performance similar to or even larger than that of a GCHP can be obtained by a DSHP with an optimized switching temperature, with a consistent reduction of the investment cost. In presence of undersized BHEs, the adoption of a DSHP with an optimal switching temperature permits to reduce the problems linked to the ground temperature drift, so DSHPs can be selected for the retrofitting of traditional ground-coupled heat pumps in presence of undersized BHEs.

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