Dual source heat pump, a high efficiency and cost-effective alternative for heating, cooling and DHW production

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Abstract

This article presents the characteristics and performance of an innovative dual source heat pump (DSHP) for heating, cooling and domestic hot water (DHW) production. The research work was carried out in the framework of the H2020 European project: Geot€ch 'GEOthermal Technology for economic Cooling and Heating'. The DSHP is able to choose the most favourable source/sink in such a way that it can work as an air-to-water heat pump using the air as a source/sink, or as a brine-to-water heat pump coupled to the ground. The DSHP is manufactured as an outdoor 'plug & play' unit, working with R32 refrigerant and including a variable speed compressor, which gives full capabilities for an efficient modulating operation. The DSHP was fully characterized in steady state conditions at the IUIIE laboratory. In order to assess its dynamic performance and to identify key control strategies to optimize its annual operation, a complete integrated model of the DSHP system in TRNSYS including the DSHP and all the other system components was developed. A first energy assessment, carried out for an office building located in the Netherlands, proves that the DSHP system would be able to reach a similar efficiency than a pure ground source heat pump (GSHP) system with half the ground source heat exchanger area needed. Therefore, the DSHP system could become a cost-effective alternative solution for heating, cooling and DHW production in buildings, as the initial investment would be significantly reduced compared to GSHPs, with similar or even higher energy efficiency.

Keywords: dual source heat pump; geothermal energy; energy efficiency

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1 INTRODUCTION

According to the International Energy Agency, buildings account for almost one third of the final global energy consumption, and they are an important source of CO_2 emissions. In particular, heating, ventilation and air-conditioning systems (HVAC) account for roughly half of global energy consumption in buildings. The sector is expanding, so it is bound to increase its energy consumption. Therefore, reduction of energy consumption and the use of energy from renewable sources in the building sector constitute important vectors to reduce the greenhouse gas emissions. When it comes to space heating and cooling using shallow geothermal energy as a renewable energy source, ground source heat pump (GSHP) systems become one of the most efficient heating and cooling renewable technologies currently available. These systems use the ground as a heat source or heat sink, depending on the season, in order to provide buildings with heating and cooling, respectively. However, they imply the use of refrigerants in the heat pump refrigeration cycle that might have an impact in the ozone layer depletion and global warming.

Fortunately, the current trend is to switch to new refrigerants with no impact in the ozone layer and a low global warming potential. Nowadays, the GSHPs that are in the market are working with these type of refrigerants, such as HFCs or HFOs (e.g. R32). Regarding the direct and indirect emissions, the current GSHPs are usually factory shield equipment, so the direct emissions of refrigerant are negligible and practically the totality of the refrigerant is recovered at the end of the heat pump life.

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Furthermore, as the power consumption of these systems is lower than conventional ones, the indirect emissions are also reduced.

GSHP systems have proved to be more efficient than conventional air-to-water heat pumps, as demonstrated by Urchueguía *et al.* [1], who concluded that GSHP systems can lead up to a 40% savings in annual electricity consumption, in comparison to air to water conventional heat pumps. Nevertheless, one of the main disadvantages of GSHPs is their high investment cost. Therefore, a reduction in both construction and operation costs is required for these systems to become successful, especially for Southern European countries where the market of GSHP systems has not taken off yet.

A possible approach to save energy in GSHP installations is to combine it with another thermal source in the form of hybrid systems. In the case of heating dominated areas, they are combined with solar thermal energy as reported in Ref. [2]. A good review of this sources combination can be found in Ref. [3]. In the case of heating and cooling systems, a common practice is to combine GSHPs with a cooling tower or a dry cooler, to use the ambient as an extra heat source/sink. It is also possible to combine GSHPs with thermal energy storage, mainly by means of phase change materials, as described in Ref. [4]. In fact, both a hybrid system and a thermal storage device can be combined as described in Ref. [5].

Hybrid systems, combining ground and air as heat sources, have two basic advantages. On one hand, given the cost of an air heat exchanger (cooling tower or dry cooler) the ground heat exchanger can be significantly reduced, reducing the total cost of the system. On the other hand, if the operation of the system is optimized, the possibility of employing the most adequate heat source could lead to a considerably higher seasonal performance and consequently to the reduction of the energy consumption.

In order to further reduce the size of the installation, its cost, and simplify the operation, some researchers tried to develop a dual source heat pump (DSHP), implementing the dual source: water and air, directly into the heat pump design. In Ref. [6], the authors developed a DSHP utilizing groundwater and air sources. They found an improvement in the Performance Factor ranging from 2 to 7% compared with the system only employing the groundwater source, and from 4 to 18% when compared with the system only employing the air source. They pointed out that higher system performance could be achievable with the use of variable speed for the compressor and the circulation pump. A recent review on hybrid ground heat pumps can be found in Refs. [7] and [8].

The possibility of having the dual source available right at the heat pump has several advantages: first the cost will be certainly lower than the one required for the external air source heat exchanger alternative (dry cooler or cooling tower), the unit will be much more compact since it is integrated in the heat pump and it takes the advantage of a superior heat transfer on the refrigerant side, and the elimination of the temperature difference produced by the intermediate BPHE, and for this same reason the COP of the unit when operating with the air source, will be superior. All this at the marginal cost of adding an extra heat exchanger, and the necessary valves and control to switch from one source to another into the heat pump design. Furthermore, as other hybrid options, the DSHP will significantly help to reduce the size and corresponding cost of the ground heat exchanger and at the same time will offer an interesting potential for energy reduction if an adequate operation strategy is developed for making use of the most convenient heat source at each moment.

The Geot€ch project 'Geothermal Technology for economic Cooling and Heating' [9] is a 4 years' duration project (2015–18) which intends to demonstrate the next generation of GSHP systems with a high energy efficiency but also with lower system costs with respect to those already existing in the market. One of the aims of the project is to develop system solutions that make the best use of hybrid heat pump and control technologies so that efficient replicable 'plug & play' whole systems can be offered to the housing and small building market sectors. To this end, an efficient and comparative low cost 'plug & play' system for providing the heating, cooling and DHW needs has been designed and it is going to be installed in three demosites located in Italy, the United Kingdom and the Netherlands, respectively.

An innovative DSHP has been developed which is capable of making optimal use of ground or air environmental heat sources according to operating and climate conditions. The heat pump is able to select the most favourable source/sink (ground or air) depending on their temperatures or other sensed parameters.

This article first describes the design and characteristics of the DSHP designed in the project. Then, a TRNSYS model of the complete system, including all the integrated system components (DSHP, ground source heat exchanger, air conditioning and DHW hydraulic loops), which has been developed in order to assist both in the optimal design and energy optimization of the operation of the system, is presented for the demo site located in the Netherlands. Finally, the paper presents an initial analysis of the system operation and performance based on the model results for a whole year of operation.

A preliminary energy assessment of the system was presented in the 16th International Conference on Sustainable Energy Technologies—SET 2017 with title 'Modelling and energy analysis of a DSHP system in an office building' [10]. This article describes the dual heat pump design and experimental performance, as well as an updated version of the model including the details of the building thermal loads (heating, cooling and DHW), as well as an initial energy assessment for the system.

2 INNOVATIVE DSHP

The innovative concept of the dual source heat pump presented here is the possibility of having both air and ground source/sink all integrated in the same refrigerant circuit. This compact solution will allow using either the air or the ground source/sink, whichever is more convenient from an efficiency point of view, therefore, leading to a superior seasonal performance compared with current technology, and to a significant reduction of the size of the ground heat exchanger with the corresponding reduction on the system cost.

2.1 Design requirements and conditions

The heat pump has to satisfy all the heating, cooling and DHW demand of a small multifamily house or office. It must be reversible in order to produce heating during winter and cooling during summer. In addition, it has to satisfy the domestic hot water (DHW) needs. The unit has been conceived as a 'plug & play' unit for domestic use, therefore, it must be as much compact and simple to operate as possible, as well as extremely automatic and reliable.

Nominal heating capacity should be around 8–10 kW. Anyhow, in order to have a unit that can be employed in a large number of different applications, also being able to adapt to a broad range of thermal demands, it was decided to employ a variable speed, inverter driven, scroll compressor.

The targeted efficiency for yearly production of heating, cooling and DHW is an annual performance factor >3.5. DHW production is not a negligible thermal load, especially because of its relatively high temperature. In order to increase the efficiency of the unit, it was decided that the DHW will be generated by the HP at 55°C. Legionella treatment of the DHW tank will be done by an electrical resistance in the tank following regulations and recommendations for this kind of small open loop systems.

The decision about which should be the employed refrigerant is very difficult, given the current status of the European F-gas regulation and the volatile evolution of the regulations on refrigerants internationally. On one hand, it seems clear that the use of HFCs will be severely and progressively restricted. On the other hand, the alternatives to the current employed refrigerants are not clear at all, and for many of the options there are not commercially available components yet, e.g. compressors.

The possible choices of refrigerant at the moment of designing the unit were:

- R410A: This is the standard refrigerant for this kind of application. It has good characteristics but a high value of GWP, therefore, it does not seem to be in line with the trend indicated in the new EU F-gas regulation (No. 517/2014), thus not being usable in a medium term horizon.
- R32: There is still a very limited compressor availability, but it seems that a large part of the AC OEMs are migrating to this old refrigerant because, despite being an HFC, it has considerable lower GWP than the standard refrigerant of the sector (R410A). HP cycles with R32 tend to produce an excessively high discharge temperature at the compressor which could be limiting the compressor life. This problem can be solved by sequentially injecting some refrigerant liquid at the suction, in order to cool down the compressor.
- R1234ze: This HFO refrigerant is also an alternative with almost negligible GWP. However, it is an expensive refrigerant, and very new in HP applications, therefore, there is a very little availability of components. Moreover, there is still some controversy about its long-term behaviour, so it could be facing also restrictions in future regulations.

After the consideration of all the possible alternatives and the consultation with several compressor manufacturers, finally R32 was chosen as the best solution for short and medium term. The use of R32 pushes the option of an outdoor installed unit. This has led to the use of an air to water heat pump with an extra BPHE for the ground loop, as the frame for the evolution to the DSHP. Furthermore, in order to be able to produce DHW apart from heating water, a dedicated BPHE for DHW production was incorporated in the design. Figure 1 shows the basic structure and components.

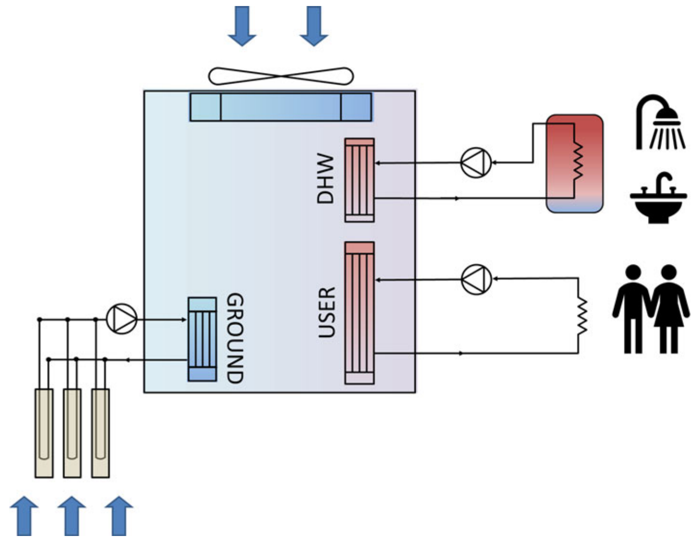

Figure 1. *Basic structure and components of the dual source heat pump.*

2.2 System layout

The system layout is designed to satisfy all the operating modes of the heat pump, i.e. heating, cooling and DHW production both during winter and during summer. The targeted market is houses or small offices, therefore, heating and cooling are completely separated by the intermediate seasons. Only DHW demand can exist simultaneously with either heating or cooling demands.

The DSHP must work in nine different operating modes, which are summarized in Table 1. They are classified in winter and summer mode: when the system operates in summer mode, it will work as a chiller while when it operates in winter mode, it will work as a heat pump. Apart from the working modes corresponding to the heat pump, Table 1 also shows three extra working modes for the system (MS, M10 and M11) which correspond to free-cooling operating conditions. The heat pump will be switched off in mode MS and M10, but it will be switched on in M11 for the production of DHW using the air as a source. Working mode MS—Midseason indicated in Table 1 corresponds to those moments of the year in which the external ambient temperature gets very mild values in the range, for instance, of 21 ± 1.5°C in cooling mode, and 19 ± 1.5°C in heating mode. In these conditions, no active heating or cooling is needed and therefore the heat pump will be switched off.

A great number of refrigerant circuit layouts can be considered in order to keep the right circulation of refrigerant through the desired components. There are clearly six different topological operation modes, allowing to produce heating, cooling or DHW, in winter and summer, from the two available source/sinks air and

Table 1. *Heat pump and system operating modes.*

	Condenser	Evaporator	Operating mode
Summer	Air	User	M1—Summer Air
	Ground	User	M2—Summer Ground
	–	–	M10—Free Cooling
	–	–	MS—Midseason
	DHW	User	M3—DHW User
		Air	M6—DHW Air
		Ground	M8—DHW Ground
		Air	M11—Free-Cooling + DHW Air
Winter	User	Air	M4—Winter Air
	–	–	MS—Midseason
	User	Ground	M5—Winter Ground
	DHW	Air	M7—DHW Air
		Ground	M9—DHW Ground

ground. There is an additional mode when it is possible to produce DHW in summer at the same time that the unit is producing chilled water (this mode is called full recovery).

The components which must be interconnected in the right order, depending on the operation mode, are the inverter driven compressor, three BPHEs (USER, providing heating or cooling to the building; GROUND, connecting the unit with the source/sink ground loop; DHW), and finally one air coil (round tube plate fin heat exchanger—RTPFHX), which can actuate as a condenser or evaporator depending on the mode. Additionally, the unit requires an adequate liquid receiver to store all liquid refrigerant that will be in excess for some of the modes and operating conditions, and at least one expansion valve to control superheat and, in some occasions, limiting the maximum discharge temperature by injecting some liquid refrigerant at the compressor suction.

A significant number of solutions for the refrigerant circuit have been analysed and evaluated from the point of view of cost, reliability and efficiency, in that same order. The best solution found is based on the utilization of 10 solenoid valves, which always work on the adequate direction by combining them if necessary with a check valve, a unique liquid line, with liquid receiver and sight glass, and one direction expansion valve for the control of the superheat at the inlet of the compressor.

2.3 Design and selection of components

This unit is going to incorporate a variable speed compressor in order to be able to adapt the performance to the wide range of operating conditions and to minimize part load losses. Scroll technology has been chosen because its superior efficiency at these operating conditions.

The optimum compressor size would be the smallest possible one that could provide the demand in the worst conditions running at maximum speed. In this way, it would be able to cope with the peak demand but still have a good system performance for most of the time in part load and a wide modulation capacity. However, the commercial availability of compressors for the chosen refrigerant (R32) was very small at the moment of designing the heat pump. Among the available compressors, it was chosen the smallest variable speed compressor available for R32. The selected compressor is model XHV-25 (R32) by Copeland.

As explained above, the adopted DSHP concept is based on an air to water heat pump. The frame dimensions of the heat pump prototype were selected in order to have space enough to fit all the necessary components. The air coil consisted of a Round Tube and Plate Fin (RTPF) heat exchanger occupying a whole side of the frame. On the other side of the coil the fans were situated. The RTPF consisted of two rows of tubes of 8 mm diameter, with five refrigerant circuits. Two ECM fans (A3G450-AC28-51 by EBM) were selected with continuous variable speed control.

Both BPHEs (USER and GROUND) are employed either as evaporator and condenser depending on the operation mode, therefore, they have been designed both as evaporators and both include the corresponding distributor. The plate models finally selected were the F80AS (asymmetric plate) and the F85 by SWEP.

The BPHE for the DHW production works always as a condenser, so a condenser model has been selected for this service: plate model B26 (asymmetric) by SWEP.

Asymmetric plate models by SWEP have been selected in order to minimize the refrigerant charge and reduce the pressure drop on the water loops.

Electronic expansion valve (EEV) Model E2V14 by CAREL was employed to keep control of the superheat. As the unit has a large number of different operation modes and the evaporator changes from one heat exchanger to another, the control of the superheat is metered at the compressor suction, which is common for all operation modes. For the same reason there is no way to implement pressure compensation. Anyhow, the pressure drop through the different evaporators has been estimated to be low.

A CAREL control system (pCO5+) is employed to control the EEV, the switching of the solenoid valves, the speed of the compressor, fans, and the water and brine circulation pumps, and all the required safety switches and alarms. Additionally, the control system measures the discharge temperature and if this temperature increases over a certain threshold it controls the EEV in a way that some pulses of liquid refrigerant are sent to the compressor suction in order to cool down the compressor.

The selection of the adequate diameters for the pipes of the different lines of the unit has been based first on two main targets: first, avoiding excessive pressure losses; and second, insuring a minimum velocity in order to get the return of the oil at low compressor velocities. This is especially critical for the suction line. For those pieces of pipe that change their role from one mode to another, for instance, from part of the liquid line to part of an expansion line, the most critical criteria has been selected in order to satisfy the design conditions.

The three required circulation pumps are also included in the frame of the heat pump: ground pump, user pump and DHW pump.

A modelling study was performed by means of IMST-ART [11] in order assist the design of the heat pump, for each of the different operating modes. IMST-ART has been very useful to size the BPHEs, to design the coil circuitry and all the pipes of the refrigerant circuitry. Besides, it has provided an initial estimation of the heat pump performance.

Figure 2 shows the basic layout and composition of the heat pump in its frame. As it can be seen, the air coil is situated in the front, and the fans in the back panel. The three BPHEs, the compressor, the liquid receiver and the three circulation pumps are visible in the picture. It should be pointed out the clean layout of the heat pump and the compactness of the unit.

2.4 Testing campaign and unit performance

A first set of 29 steady test points were performed, including the most important operation modes and conditions, which was employed to analyse the unit performance and prepare the general testing campaign. These tests allowed to check the adequate refrigerant charge and check the effective oil return and have a first estimation of the unit performance.

Additionally, with these results, the compressor efficiencies were evaluated and it was found a slight decrease of efficiency from the estimated values obtained from the R410A results supplied by the compressor manufacturer. The compressor efficiency correlations were conveniently readjusted to the experimental data and they were introduced in the IMST-ART software. Then a comparison between predicted and measured results was performed for all the available test points. A very good agreement was found between the predicted and the measured results, so it was decided to employ the software to explore the variability of the unit performance when the input variables were changed, covering the whole range of possible variation for them. For instance, for winter-ground mode, the input variables are inlet brine temperature to the evaporator, brine flow rate (or brine temperature variation across the evaporator), inlet water temperature to the user condenser, user loop mass flow rate (or water temperature variation across the condenser) and compressor frequency. For air source modes, the brine temperature and the brine flow rate, are not employed and instead, the input variables are the outdoor air temperature and the air flow rate (or the fan frequency).

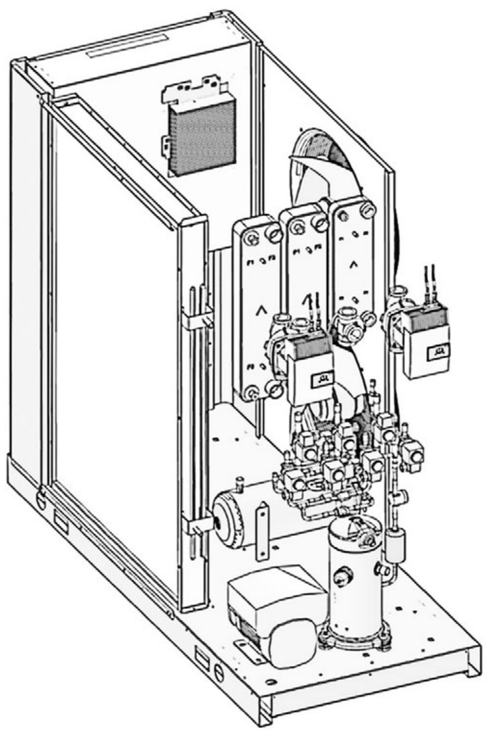

Figure 2. *Dual source heat pump plug & play unit.*

IMST-ART was employed to evaluate the unit performance in the whole range of variation for Winter-ground heating mode, totalling 640 test runs. The performance results, mainly condenser capacity, evaporator capacity and compressor consumption for all 640 points, were employed in order to find a convenient polynomial, containing linear and quadratic terms, and also some crossed variables terms. The study was performed employing the fitting by linear regression. The study was carried out manually by inserting new terms and retaining those that have obtained good estimation indicators until no further significant improvement was reached in the adjusted R^2 and the maximum relative error. The finally selected polynomials were able to predict all the 640 performance points with a maximum error lower than 4%, and a much lower average error.

Then, taking these performance polynomials as the true Response Surface (RS) a Design Of Experiments (DOE) was carried out. Several DOE methodologies were tried and tested, in order to find which one was able to give the best compromise between the number of test points and accuracy in the determination of the corresponding RS. The best compromise was found with the Central Compact Design methodology and it turned out that selecting only one of the two orthogonal blocks and central star it was sufficient to get a very good estimation of the RS all over the entire domain of the five independent variables, resulting in only 30 test points.

Once the test matrices were elaborated for each operating mode, the test campaign was followed until all the necessary points were tested for all the seven operating modes. The test matrices were adequately corrected when the variation of some parameter could go beyond the testing capabilities or the test point was in an area of no interest for the targeted application, for instance when the unit will not be in operation because it is much more profitable to employ the free cooling system.

All test results were systematically analysed in order to detect possible operation problems or mistakes, and be able to repeat them if necessary. After all the test results became available, the compressor correlation was checked again and readjusted, and the performance for each test point was also evaluated with the software. This allowed a basis to analyse the results and to check the unit performance and the possible existence of testing mistakes. A comparison between experimental and predicted performance was carried out for each operating mode. This comparison allowed to deeply analyse the results and check all the individual test points. A very good agreement between estimated and measured results was found all across the different test matrices for all the seven operating modes.

Figure 3 shows the heating capacity, the compressor power input, and the COP (calculated only taking into account the compressor consumption) for Winter Ground operating mode. The results are shown as a function of the compressor frequency, for a hot water supply temperature of 45°C, 0°C of return temperature at the brine loop, a 3 K brine temperature difference through the evaporator and 5 K through the condenser. Figure 4 shows the same results for a hot water supply temperature of 35°C.

As it can be observed, both the heating capacity and the compressor consumption increase with the compressor speed in a quite linear way. COP values are considerably high regardless the low temperature considered at the inlet of the evaporator, 0°C (brine return temperature from the ground heat exchanger). The maximum COP is obtained around 40 Hz for both applications, with a significant performance decay.

Figures 5 and 6 show the same performance, in this case for Winter Air operating mode with the air at a dry bulb temperature of 7°C (wet bulb temperature of 6°C), keeping the rest of the parameters (Tco and dTc) with the same values as in Figures 3 and 4.

As it can be observed, the heat pump COP is higher in Winter Air than in Winter Ground. However, this is only due to the fact that the air temperature is much higher than the return brine temperature, so the performance between both working modes cannot be fairly compared. In any case, the high values of air operation at moderate temperatures 7(6) supports the concept of taking the heat from the air when the air temperature is high or mild, reserving the ground for the lowest ambient temperatures. Again, the maximum COP is obtained around 40 Hz of compressor speed.

Once the results for each operating mode became available, they were employed to fit the developed performance polynomials, which will be afterwards employed for the system models and for the optimization of the system control.

2.5 DSHP model

The heat pump will be considered as a black box in the TRNSYS integrated model. For that purpose, the performance of the unit will be calculated by means of polynomial correlations which depend on the working conditions (different source and distribution temperatures and water/brine flow rates) of the heat pump

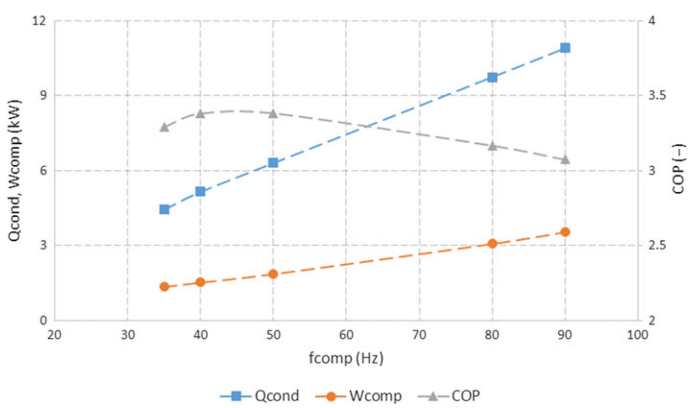

Figure 3. *Condenser capacity, compressor power input and compressor COP in Winter Ground operating mode (Tco = 45°C, dTc = 5 K, Tei = 0°C, dTe = 3 K).*

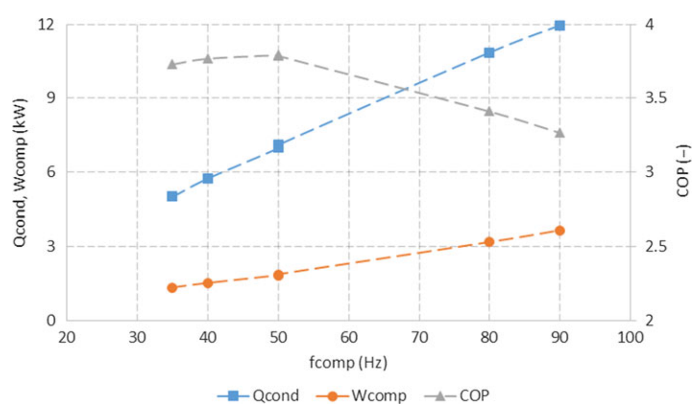

Figure 5. *Condenser capacity, compressor power input and compressor COP in Winter Air operating mode (Tco = 45°C, dTc = 5 K,* T_{air} *= 7(6),* f_{fan} *= 50%).*

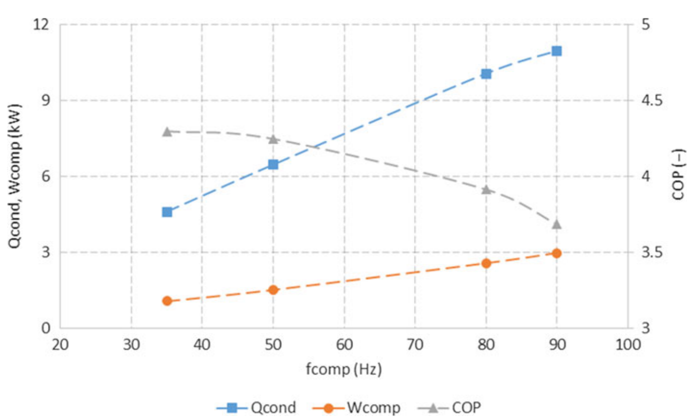

Figure 4. *Condenser capacity, compressor power input and compressor COP in Winter Ground operating mode (Tco = 35°C, dTc = 5 K, Tei = 0, dTe = 3 K).*

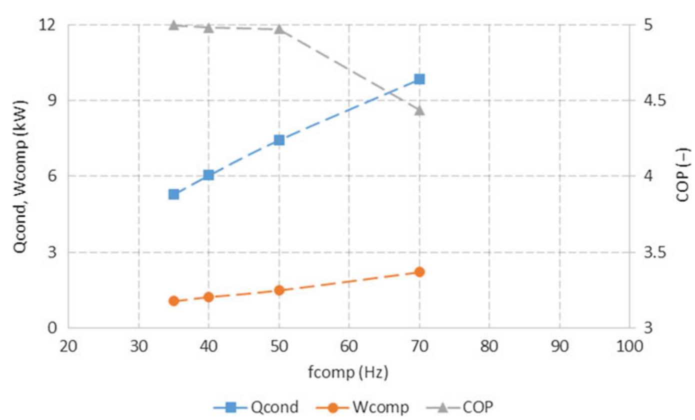

Figure 6. *Condenser capacity, compressor power input and compressor COP in Winter Air operating mode (Tco = 35°C, dTc = 5 K,* T_{air} *= 7(6),* f_{fan} *= 50%).*

for each operating mode. These polynomial correlations were obtained from the experimental results for the condenser capacity, evaporator capacity and compressor power input, and allow a full characterization of the heat pump performance among all the operation modes and a large range of operating conditions. An example of the employed polynomial correlations is presented in the following.

These polynomial correlations were implemented as a new TRNSYS type and integrated in the system model. The main reason for this instead of using the already available heat pump TRNSYS types was the need of accurately reproducing the heat pump performance under different working conditions, which depends not only on the source and load inlet temperature but also on many other variables, as previously stated, for each 1 of the 11 operation modes.

On the other hand, a disadvantage of the already available heat pump types in TRNSYS, is that they calculate the performance of the heat pump based on a fixed number of working points and interpolate the performance when operating under other different conditions. Furthermore, they calculate the heat rejected or absorbed in the source side (ground or air) as the sum (if it is in cooling mode) or the subtraction (if it is in heating mode) of the heat pump capacity and the power consumption, not considering the thermal losses in the heat pump cycle. Contrary to this, the heat pump model based on the correlations calculates the condenser and evaporator capacity, the power consumption and parasitic losses separately, based on the experimental results.

Figure 7 shows a schematic diagram of the heat pump as a black box model for Winter Ground operating mode. The heat pump model receives the following inputs: the secondary fluid inlet temperature to the evaporator, Te_i (K) (brine in this case); the secondary fluid temperature difference across the evaporator, dTe (K); the secondary fluid inlet temperature to the condenser, Tc_i (K) (user loop in this case); the compressor frequency, f_{comp} (Hz) and the secondary fluid temperature difference across the condenser, dTc (K).

Equations (1)–(3) provide the heat pump performance for the Winter Ground operating mode (M5).

$$\dot{Q}_{cond} = f_{comp} \cdot (C_0 + C_1 \cdot Te_i + C_2 \cdot Tc_i + C_3 \cdot Te_i \cdot Tc_i + C_4 \cdot Te_i^2 + C_5 \cdot dTe + C_6 \cdot dTc + C_7 \cdot Te_i \cdot dTe + C_8 \cdot Tc_i \cdot dTc) \quad (1)$$

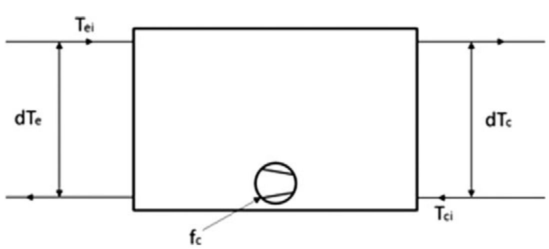

Figure 7. *Black box model for the heat pump in Winter Ground mode.*

$$\dot{Q}_{evap} = f_{comp} \cdot (C_0 + C_1 \cdot Te_i + C_2 \cdot Tc_i + C_3 \cdot Te_i \cdot Tc_i + C_4 \cdot Te_i^2 + C_5 \cdot dTe + C_6 \cdot dTc + C_7 \cdot Te_i \cdot dTe + C_8 \cdot Tc_i \cdot dTc) \quad (2)$$

$$\dot{W}_{comp} = f_{comp} \cdot (C_0 + C_1 \cdot Te_i^2 + C_2 \cdot Tc_i^2 + C_3 \cdot dTe + C_4 \cdot dTc + C_5 \cdot Te_i \cdot dTe + C_6 \cdot f_{comp} \cdot Te_i + C_7 \cdot f_{comp} \cdot Tc_i + +C_8 \cdot f_{comp}^2 \cdot Te_i \cdot Tc_i + C_9 \cdot f_{comp} \cdot dTe + C_{10} \cdot \frac{1}{f_{comp}}) \quad (3)$$

Where, $\dot{Q}_{cond}$ is the condenser capacity (W), $\dot{Q}_{evap}$ is the evaporator capacity (W), and $\dot{W}_{comp}$ is the compressor power input (W). As it can be seen, the polynomials for the heat pump capacities are based on nine coefficients, while the compressor power input requires two extra coefficients, i.e. total 11 coefficients. The compressor power input includes the inverter losses.

When the heat pump is working with the air, for instance Winter Air mode, the same polynomials were found to be adequate with the substitution of dTe, which is related to the brine flow rate in the ground mode, by the fan frequency f_{fan} (%).

Figure 8 shows the comparison between experimental measurements and the correlations results for the compressor consumption and the condenser capacity in Winter Ground operating mode (M5). It can be observed that the adjustment is very accurate, being the maximum deviation lower than 3.2 and 2.6%, respectively. The maximum deviation between experimental and calculated performance values for all operation modes is lower than 5%.

Figures 9 and 10 show the COP maps obtained from the developed polynomials for the Winter Ground and Winter Air operating modes. Top graphs correspond to COP evaluated only with the compressor consumption whereas bottom graphs correspond to COP evaluated with all electric consumptions, i.e. compressor, user circulation pump, and either brine circulation pump or fan. Left graphs correspond to a supply water temperature of 45°C whereas right hand graphs correspond to 35°C. The return brine temperature is 0°C for the Winter Ground case, whereas the air temperature is 7 (6)°C. The compressor frequency is 50 Hz. It should be noticed that the Y axis in Figure 9 shows an opposite trend to the Y axis in Figure 10, since increasing the brine temperature difference across the evaporator in Figure 9 means decreasing the brine flow rate, whereas Y axis in Figure 10 shows the fan frequency, which is proportionally linked to the air flow rate.

As it can be observed in Figure 9, the lower the temperature difference across the evaporator, the higher the compressor COP. In other words, the higher the brine flow rate, the higher the compressor COP. This trend is corrected when it is taken into account the brine circulation losses in the brine loop, since the pumping losses depend on the brine flow rate. The result is that the optimum is situated around 3 K of temperature difference. Interestingly, this is the standard value employed by most of the manufacturers for this parameter, moreover, it is the one

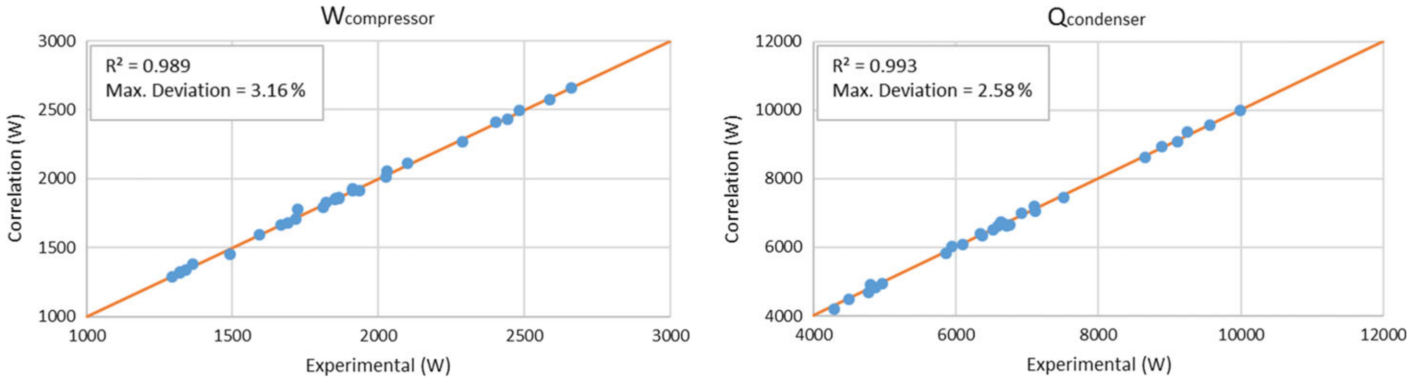

Figure 8. *Experimental measurements versus polynomial correlations for Winter-ground mode: (**a**) Compressor consumption. (**b**) Condenser capacity.*

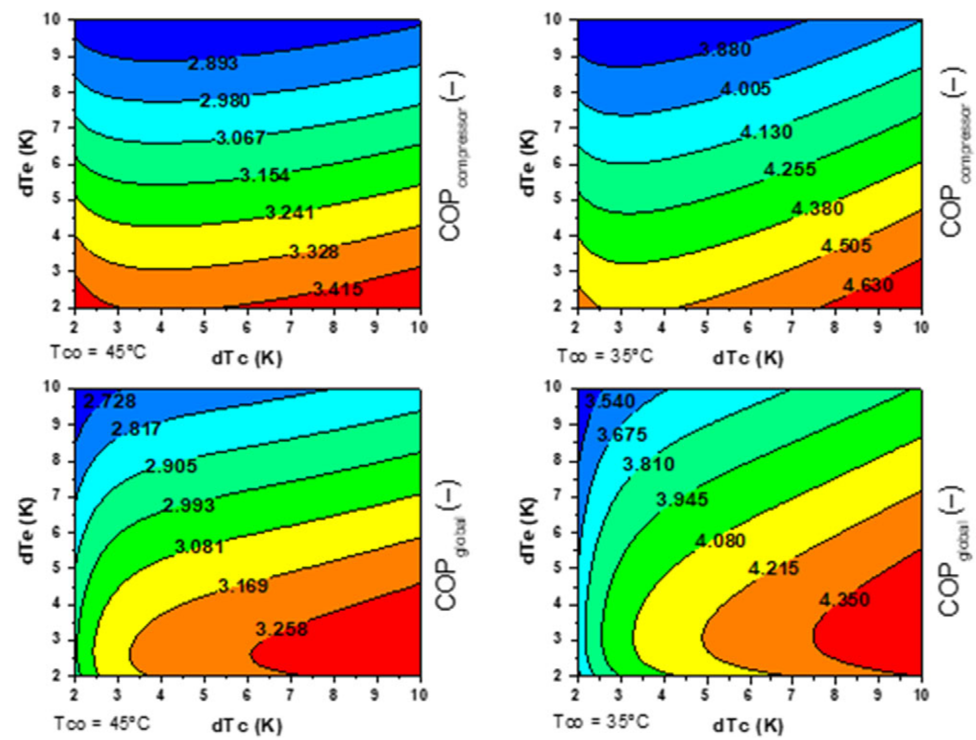

Figure 9. *COP maps as a function of the water temperature difference across the condenser (dTc) and of the water temperature difference across the evaporator (dTe) at different supply water temperatures (Tco = 45°C on the left side, Tco = 35°C on the right side) in Winter Ground operating mode, at 50 Hz of compressor frequency and 0°C brine return temperature.*

employed in the certification testing standards. In contrast, the maps show that the higher the water temperature difference through the condenser, the higher the COP, although the influence is lower than at the evaporator. This means that the lower the user flow rate, the higher is the COP. Low flow rates imply a higher thermal resistance at the condenser so it should contribute to increase the condensing pressure and so to a COP decrease, which is not observed. This is because the mapping has been built with experimental results in which we keep constant the supply water temperature, which is the usual way of control of these kind of heat pumps. This makes that, in fact, when the condenser flow rate decreases, the temperature difference increases so that the user return temperature decreases, allowing a decrease in the condensing temperature that makes the COP increase.

Similar trends can be found in Figure 10: the higher the water temperature difference through the condenser (dTc), the higher the COP, although the influence is again lower than the one of the fan frequency at the evaporator. Again, the lower the user flow rate, the higher is the COP. In this case, it is much apparent the influence of the fan consumption on the global COP. If the pumps and fan consumptions are not taken into account, the higher the fan frequency, the higher the COP (compressor) because the thermal resistance at the evaporator decreases and therefore the evaporation temperature increases. However, if the fan consumption is

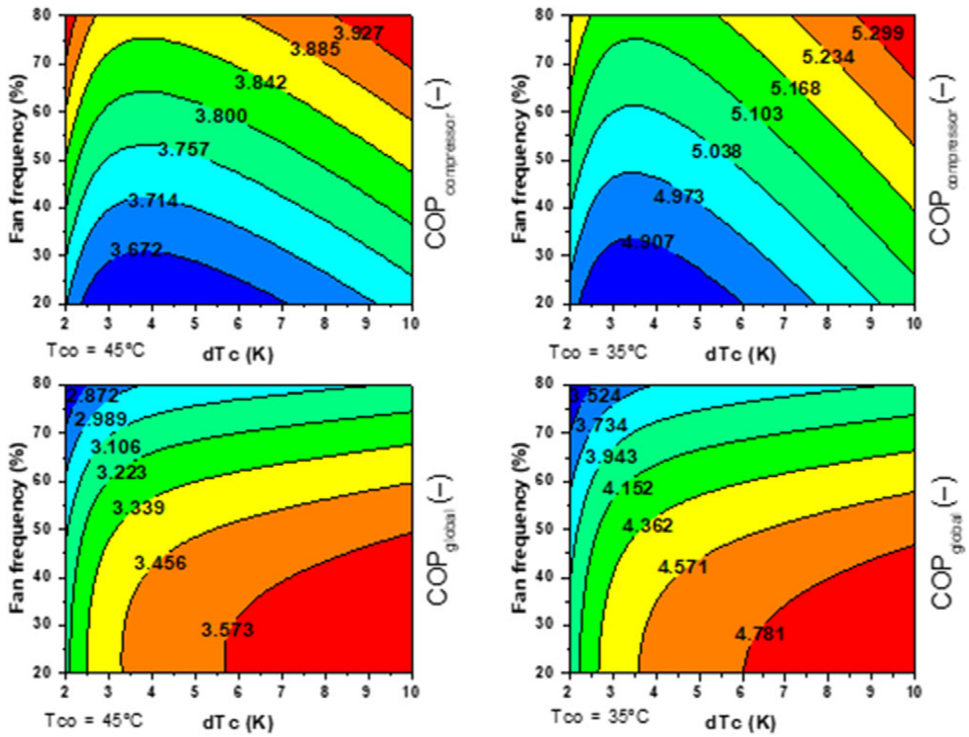

Figure 10. *COP maps as a function of fan frequency and water temperature difference across the condenser (dTc) at different supply water temperatures (Tco = 45°C on the left side, Tco = 35°C on the right side) in Winter Air operating mode, at 50 Hz of compressor frequency and 7(6)°C air temperature.*

taken into account in the COP (global), then, as the fan consumption increases with the fan speed, it reverses the influence and makes that the lower the fan frequency, the higher the global COP.

After analysing Figures 9 and 10, it can be concluded that the influence of the operating parameters on the system performance is very important. Therefore, the minimization of the energy consumption of these complex systems requires the optimization of its operation. The model described in the following is a tool to assist the optimization of the sizing and operation of these type of systems.

3 INTEGRATED SYSTEM MODEL IN TRNSYS

In order to assist both in the optimal design and energy optimization of the operation of the system, a model of the 'plug&play' system in TRNSYS including all the integrated system components (DSHP, innovative ground source heat exchanger, air conditioning and DHW hydraulic loops) is presented for the demo site located in the Netherlands. The layout of the system model is shown in Figure 11.

The heat pump is able to provide heating and cooling to the user, but also DHW. In order to combine the different working modes, it is set that the heat pump will be able to provide DHW during the night (from 00:00 to 06:00). On the other hand, it will only provide heating or cooling during the opening schedule of the building (from 08:00 to 18:00). The volume of the DHW tank is 200 l, but the demand of DHW in the office is 22.5 l/day, corresponding to the demand of three people. This means that the DHW tank has a high inertia, and only a small part of the hot water will be extracted during the day, so the heat pump will work in DHW mode only during short periods in the night. On the user side, the maximum heating load during the year is around 12 kW and the maximum cooling load is around 6 kW. Regarding the circulation pumps, the user loop circulation pump is continuously working during the office schedule (from 08:00 to 18:00), while the ground loop and DHW loop circulation pumps cycle with the compressor operation (they only work when the compressor is working).

3.1 Innovative coaxial BHE

The BHE used in the system model is an innovative coaxial-spiral BHE that was developed by Geothex BV (http://geothex.nl; 20 February 2018, date last accessed). The main innovations consist of an insulated inner pipe that reduces the heat transfer between the inner and outer pipe, together with a ribbed outer channel, which makes the fluid follow a spiral path along the outer pipe. According to preliminary investigations, it is possible to obtain a significant increase on the efficiency when compared to conventional BHEs, especially at low Reynolds numbers [12]. This BHE is under development and optimization inside the framework of the GEOT€CH project.

A model of this new coaxial-spiral BHE was developed based on the thermal network approach, combined with a vertical discretization. This model has been adapted to the new configuration from the B2G dynamic model previously developed for a U-tube BHE configuration and presented in Refs. [13] and [14]. The model reproduces the short-term behaviour of the BHE with a high accuracy, taking into account only the portion of surrounding ground directly affected by the heat injected/extracted during the period considered. A detailed explanation of the model can be found in Ref. [15].

3.2 Selection of the source/sink

It was previously stated that the heat pump is able to work using the ground or the air as source/sink. In this context, a control strategy is needed in order to select the source/sink that, in each moment, will be most favourable in order to obtain the highest efficiency.

As a preliminary control strategy, it was selected a simple strategy, in which the source with the most favourable temperature (highest temperature in heating mode and lowest temperature in winter mode) is selected. In order to prevent the heat pump from changing from one source to another in a short period (as the air temperature changes with a high frequency), a differential controller is used, providing some hysteresis to the control. This means that, the actual ground temperature is used as the reference temperature, and the heat pump changes the source to air when the air temperature is more favourable (considering the dead band of the differential controller). Figure 12 shows the operation of this control for (a) heating mode and (b) cooling mode.

In order to programme this control easily in a control board, it is planned to use the fluid return temperature from the ground loop instead of the ground temperature, as it is easier to measure the temperature of the fluid inside the pipe rather than measuring the surrounding ground temperature.

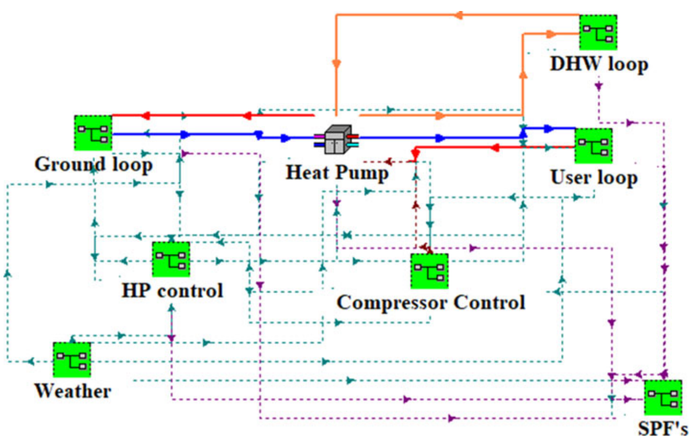

Figure 11. *Integrated system model in TRNSYS layout.*

3.3 Temperature compensation

With the aim of increasing the global efficiency of the system, an optimization strategy based on the supply temperature compensation was implemented in the model. This strategy mainly consists of adapting the supply temperature to the evolution of the outdoor ambient temperature. For example, if the heat pump is providing heating and the outdoor temperature increases, the supply temperature will be decreased, since it will mean a lower heating demand and therefore there will be no need of supplying the water so hot. Analogously, if the outdoor temperature decreases, the supply temperature provided by the system will be increased, since it will mean a higher heating demand. This allows taking advantage of milder outdoor ambient temperatures to improve the coefficient of performance (COP) of the heat pump whenever it is possible. The methodology used in this model is detailed in [16]. According to this methodology, the supply temperature is calculated each simulation time step in such a way that the terminal units are able to meet the user comfort even in the most extreme ambient conditions (maximum and minimum ambient temperatures for the location: 29.3 and −8.1°C, respectively for Amsterdam TRNSYS-Meteonorm weather file). Therefore, the supply temperature will be set depending on the following parameters: current ambient temperature, the maximum and minimum annual ambient temperatures, the desired comfort temperature inside the building (21°C for heating mode and 22°C for cooling mode) and the maximum and minimum limits in the supply temperature. These limits are fixed in order to avoid operating problems in the heat pump (freezing risk, very low pressure ratios in the compressor or very low COPs). In heating mode, the supply temperature limits that have been set are 35 and 45°C; in cooling mode, the supply temperature will have a lower limit of 7°C and a higher limit of 15°C.

$$T_{SB} = (1+\beta)\cdot T_{room} - \beta\cdot T_{amb} \tag{4}$$

$$\beta(cooling) = \frac{T_{room} - T_{SB)min}}{T_{amb)max} - T_{room}};\quad \beta(heating) = \frac{T_{SB)max} - T_{room}}{T_{room} - T_{amb)min}} \tag{5}$$

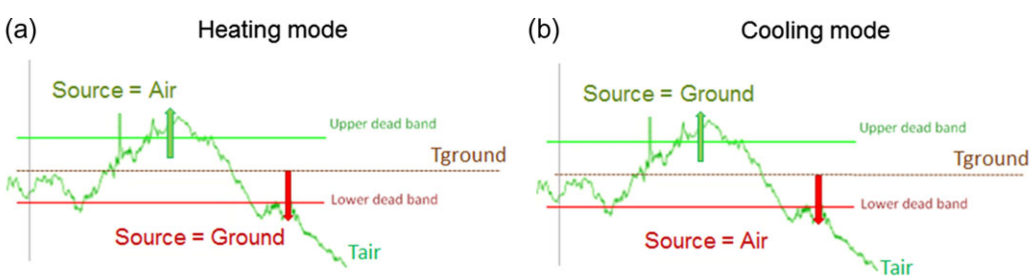

Figure 12. *Selection of the source depending on the air and ground temperature. (**a**) Heating mode. (**b**) Cooling mode.*

Where, T_{SB} is the supply temperature to the building, T_{room} is the comfort temperature desired in each of the air-conditioned spaces, T_{amb} is the ambient temperature, and $T_{amb)max}$ and $T_{amb)min}$ are the maximum and minimum outdoor annual ambient temperatures, and $T_{SB)min}$ and $T_{SB)max}$ are the minimum and maximum supply temperatures for cooling and heating mode, respectively.

4 CASE STUDY AND ANALYSIS OF THE SYSTEM ENERGY PERFORMANCE

The model developed in TRNSYS was used to make an assessment of the system energy performance. In order to study the techno-economical feasibility of the DSHP system, two scenarios were compared. In the first scenario, a GSHP (only ground) system was simulated with a borehole field formed by three BHEs, 70 m deep each one. In the second scenario, a DSHP (air/ground) system was studied. In this case, the length of BHEs is half the one of the GSHP scenario (three BHEs with a depth of 35 m each one). The aim of this comparison is to analyse whether it is possible to obtain a similar efficiency of the system but considering half the length of BHEs, which would mean a significantly reduced investment cost for the DSHP installation compared to that of a GSHP system. The main design and operation parameters of the system considered in the model are shown in Table 2.

4.1 Small office building in The Netherlands

The building that has been used in this analysis is a small office building located in the city of Amsterdam, The Netherlands. This building has two zones with necessities of heating and cooling: an office room and a meeting room. The thermal demand loads have been calculated using a TRNSYS model of this building and then, introduced in the global model of the system as an input. It is assumed that in the building there are three people working in average.

The model considers the different walls of the building and windows, as well as the materials, infiltrations and ventilation required in the building. So, the sensible and latent gains that are needed to meet the comfort inside the different rooms are calculated based on the external conditions (outdoor temperature, radiation on the different facades and humidity), the internal conditions (indoor temperature, humidity) and the overall heat transfer coefficient calculated from the building envelope characteristics. In order to simplify the load profile and the operation of the heat pump, two seasons have been considered: one heating season in which the heat pump provides DHW and heating (thermal load >0 in Figure 13) and one cooling season, in which the heat pump provides DHW and cooling (load <0 in Figure 13). The maximum peak loads are around 11.7 kW in heating and around 5.4 kW in cooling.

Regarding the DHW demand, a profile for DHW in an office building extracted from Ref. [17] and an occupancy of three people is considered. In this profile, the demand (litres per hour) is given for each hour of the day. Figure 14 represents this DHW demand profile during one day.

Table 2. *Parameters of the system.*

Set point heating	35–45°C
Set point cooling	7–15°C
Set point DHW	55°C
Set point free cooling	10°C
Heat pump minimum frequency	30 Hz
Heat pump maximum frequency	70 Hz
Natural heating (Midseason)	On: $T_{amb} > 20.5$; off: $T_{amb} < 17.5$
Natural cooling (Midseason)	Off: $T_{amb} > 22.5$; on: $T_{amb} < 19.5$
Air/ground source, heating mode: ($T_{air} - T_{ground}$)	Ground: <−2; AIR > 2
Air/ground source, cooling mode: ($T_{ground} - T_{air}$)	Ground: <−2; AIR > 2
Office Schedule (air conditioning schedule)	8:00–18:00 on weekdays
DHW Schedule	0:00–8:00 and 20:00–24:00 on weekdays
User buffer tank volume	55 l
DHW tank volume	200 l
DHW profile	22.5 l/day (profile for an office of three people)
BHEs field Dual source system	Three BHEs, 35 m deep
BHEs field Ground source system	Three BHEs, 70 m deep
Thermal demand load profile	Amsterdam
Simulation time step	60 s

4.2 Seasonal performance factor of the system

In order to assess the energy performance of the systems, the seasonal performance factor (SPF) of the systems for a whole year of operation is quantified. The expressions used for the SPFs of the systems (equations from (6) to (9)) were defined according to the SEPEMO-build 'SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector (SEPEMO-Build)' project definition [18]. This project aims at overcoming market barriers to a wider application of heat pumps by developing a universal methodology for field measurement of heat pump systems SPF including a monitoring programme for 46 heat pump installations in six European countries. These expressions are presented in the following equations.

$$\mathrm{SPF1} = \frac{\int_0^t (\dot{Q}_{\mathrm{USER}} + \dot{Q}_{\mathrm{DHW}}) \cdot dt}{\int_0^t (\dot{W}_{\mathrm{HP}}) \cdot dt} \tag{6}$$

$$\mathrm{SPF2} = \frac{\int_0^t (\dot{Q}_{\mathrm{USER}} + \dot{Q}_{\mathrm{DHW}}) \cdot dt}{\int_0^t (\dot{W}_{\mathrm{HP}} + \dot{W}_{\mathrm{FAN}} + \dot{W}_{\mathrm{BHE}}) \cdot dt} \tag{7}$$

$$\mathrm{SPF3} = \frac{\int_0^t (\dot{Q}_{\mathrm{USER}} + \dot{Q}_{\mathrm{DHW}}) \cdot dt}{\int_0^t (\dot{W}_{\mathrm{HP}} + \dot{W}_{\mathrm{FAN}} + \dot{W}_{\mathrm{BHE}} + \dot{W}_{\mathrm{BACKUP}}) \cdot dt} \tag{8}$$

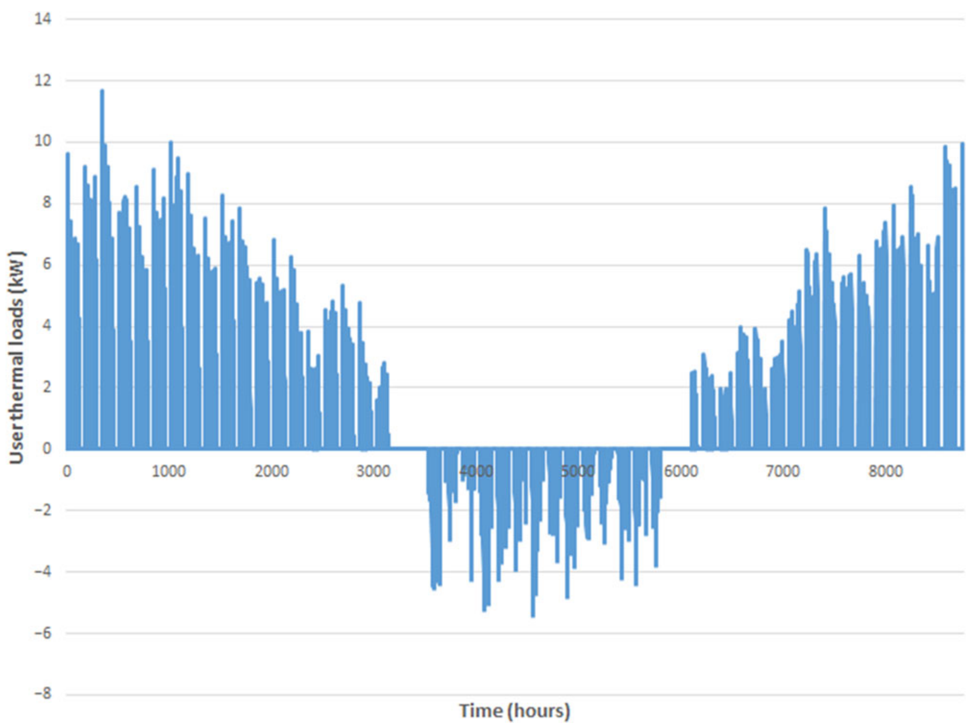

Figure 13. *Thermal demand load profile. Heating loads (>0) and cooling loads (<0).*

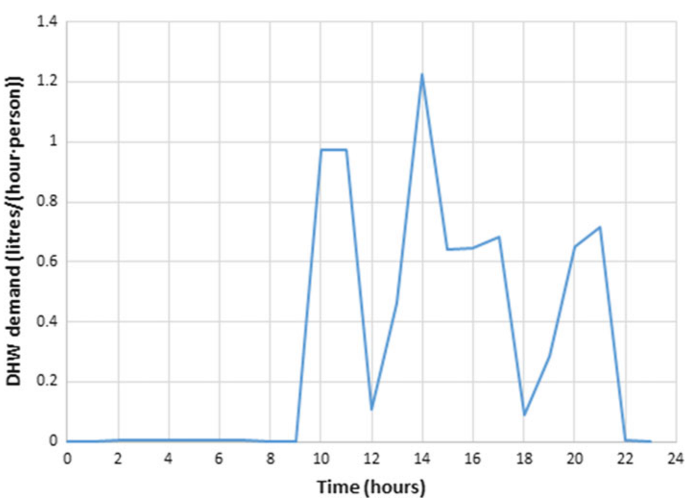

Figure 14. *Domestic Hot Water demand profile for an office building per person.*

$$\text{SPF4} = \frac{\int_0^t (\dot{Q}_{\text{USER}} + \dot{Q}_{\text{DHW}}) \cdot dt}{\int_0^t (\dot{W}_{\text{HP}} + \dot{W}_{\text{FAN}} + \dot{W}_{\text{BHE}} + \dot{W}_{\text{BACKUP}} + \dot{W}_{\text{USER}} + \dot{W}_{\text{DHW}}) \cdot dt} \quad (9)$$

where, $\dot{Q}$ is the useful heat in the user loop ($\dot{Q}_{USER}$) or DHW loop ($\dot{Q}_{DHW}$), and $\dot{W}$ is the power consumption of each of the components existing in the system (heat pump $\dot{W}_{HP}$, fan $\dot{W}_{FAN}$, BHE circulation pump $\dot{W}_{BHE}$, user circulation pump $\dot{W}_{USER}$ and DHW circulation pump $\dot{W}_{DHW}$), including the back-up system in case that there is any $\dot{W}_{BACKUP}$.

4.3 Comparison of the two scenarios

One of the main advantages of a DSHP system (air/ground) against a GSHP system (only ground) mainly consists of a reduction in the BHE required length thanks to a lower use of the ground. This means that the investment of the installation will be significantly reduced, as the BHE field is one of the most expensive parts in a GSHP system. In order to analyse this advantage, two scenarios were considered using the TRNSYS system model developed. In the first scenario, a GSHP system is analysed. Therefore, the heat pump will work using only the ground as a source/sink. In the second scenario (DSHP system), the heat pump will work using either the air or the ground as a source/sink, depending on the temperature of each one as it was detailed in Section 3.2.

It should be noted that the BHEs field used in the DSHP system is half the size of the one used in the ground source system (three BHEs with a depth of 70 m in the ground source system and three BHEs of 35 m in the dual source system). The efficiency of each system is assessed by calculating the SPFs. As there is no back-up heater considered in none of the systems, SPF_2 and SPF_3 will get the same values. Figure 15 presents the results for the assessment of the systems energy performance over a whole year of operation.

As it can be observed, the resulting SPFs are quite similar in both cases, with a yearly SPF_4 of 3.58. So, it can be concluded that both systems are able to work with a similar efficiency, this means

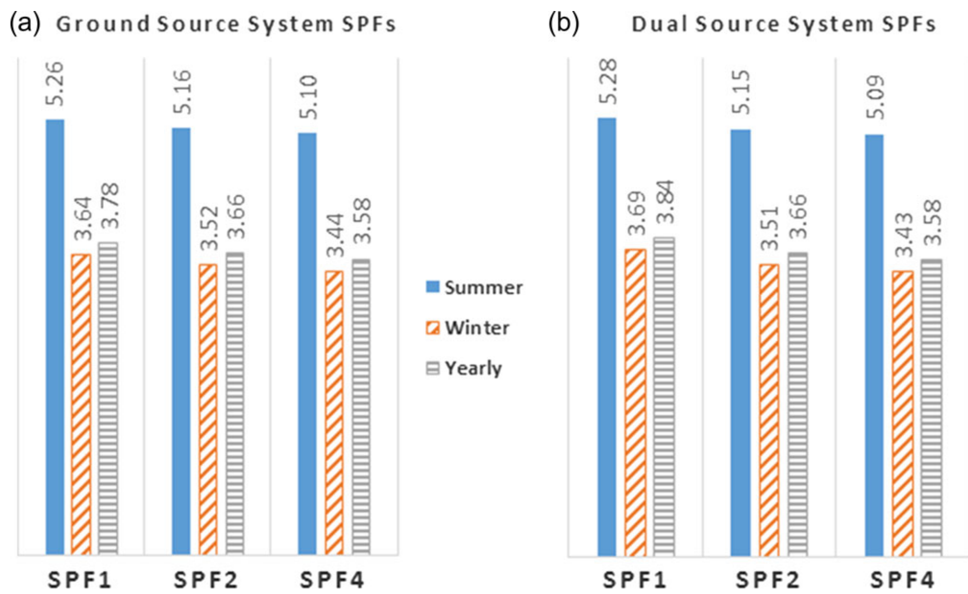

Figure 15. *Seasonal performance factors for a whole year of operation in the two scenarios: (**a**) ground source heat pump system (first scenario) and (**b**) dual source heat pump system (second scenario).*

that they can provide the same amount of energy to the system with the same power consumption. However, as the DSHP system needs half the length of BHEs compared to that of the GSHP system, the investment required will be lower and therefore the payback time for the DSHP system.

In this studied case, the reduction in the investment cost could reach values up to 30%. Since, the BHE field length would be reduced to the half (although it is also necessary to pay the cost of the drilling equipment), but the cost of the heat pump would be around a 20% higher. The increase in the heat pump cost is due to the fact that it is a prototype, which includes a non-conventional compressor and a more complex thermo-hydraulic system (including a dedicated BPHE for DHW and two heat exchangers for the use of ground or air as a source/sink). Focusing on the DSHP system (Figure 15b) shows that the SPF_1 values are quite higher during the summer, getting values around 5.28, whereas in winter it is a 30% lower, getting values around 3.69. Then, as the system is mainly operating in heating mode during most part of the year (shown in Figure 16), the yearly value of SPF_1 is 3.84 approximately, which is more similar to that obtained in winter season. Regarding the rest of the SPFs of the system, it can be concluded that SPF_2 and SPF_3 get values which are very close to SPF_1 (only 5% lower in the yearly value) which means that the auxiliaries consumption for the ground loop circulation pump and the fan do not have a great impact in the energy consumption of the system. Analogously, when considering the circulation pumps existing in the DHW loop and the user loop (for heating and cooling), SPF_4 decreases slightly (around 2%), being the yearly performance factor of the system around 3.58.

In general, it can be concluded that the SPFs obtained during the summer are quite higher than the ones obtained during the winter due to the use of free-cooling and natural ventilation. This happens because during the operation in these modes, the heat pump is switched off and there is no consumption of the compressor, only the consumption of the ground and user loops circulation pumps in the case of the free-cooling and the parasitic losses of the heat pump in both modes.

Figure 16 shows the amount of time in which the DSHP works in each mode. It is possible to check that most of the year it is working in heating mode (mostly using the ground as a source). During the summer, the heat pump works just a few hours in cooling mode, being the free-cooling mode the most used as expected. The rest of the cooling demand is met by natural ventilation and the inertia of the building (midseason mode).

4.4 Dynamic performance during 1 week (dual source system)

This section presents a preliminary analysis of the system energy performance during 1 week of operation in order to describe the dynamic behaviour of the dual source system and the selection of working mode for 1 typical week in heating mode. Figure 17 shows the selection of working mode during 1 typical week in autumn. It is possible to see that, as the air and ground temperature change, the heat pump will select one source or the other, selecting the most favourable, as it was already explained in Section 3.2 of the present paper. When the air temperature is higher than the ground temperature, the heat pump will select the air as source (in heating mode as well as in DHW mode). Analogously, when the air temperature becomes lower than the ground temperature (considering the hysteresis in the control), the source will change from air to ground. On the other hand, if the air temperature is lower than the ground temperature and then, the air temperature becomes higher, the source will change from ground to air.

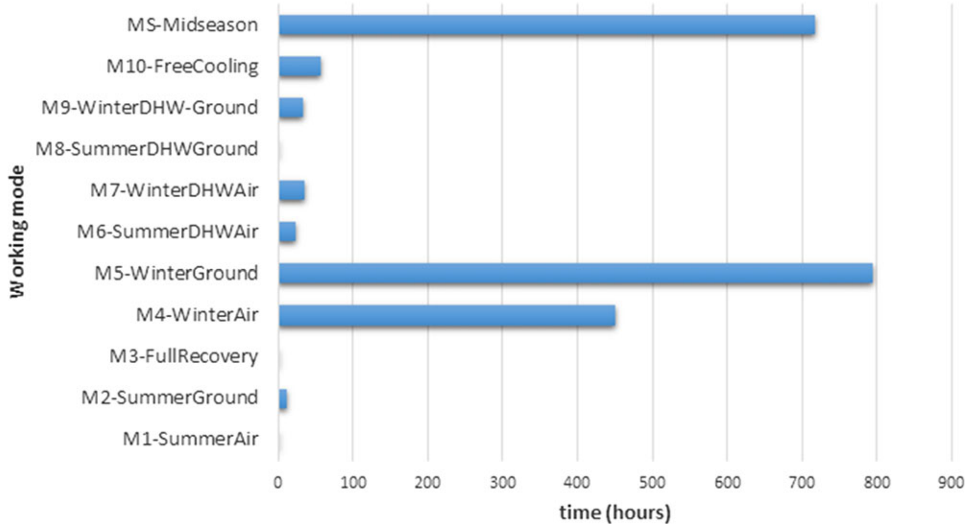

Figure 16. *Dual source system operating modes analysis.*

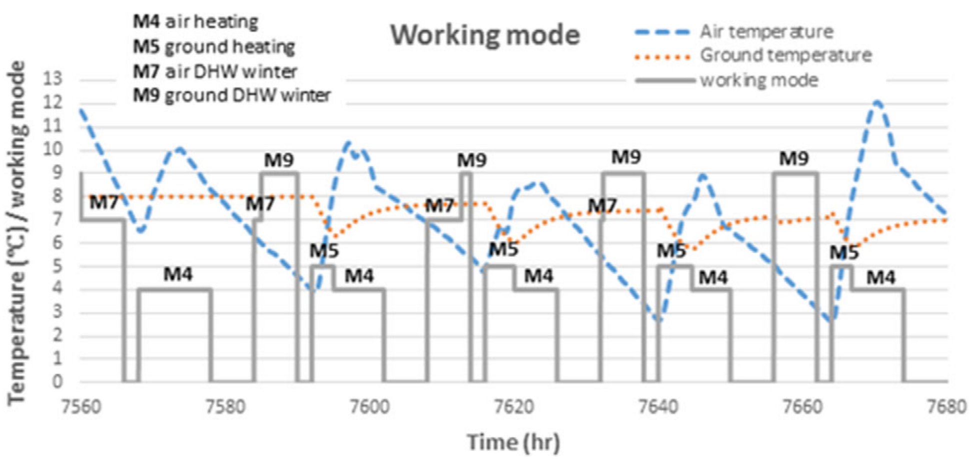

Figure 17. *Change of working mode during 1 week in autumn.*

4.5 Working mode analysis (dual source system)

Figure 18 presents the percentage of time during which the system has been working in each operation mode over the year. It should be pointed out that these percentages are extremely influenced by the weather and the thermal demand of the demo site existing in the Netherlands where temperatures are really mild during the summer (June to September) taking values lower than 21°C during most of the time. This is the reason why the thermal loads in summer are very small leading to a high percentage (89%) of the time working in MS—midseason, which means that the heat pump is off, and the small cooling demand or air renovation needs are satisfied just by opening the windows (natural ventilation). In the case that the air is hotter than 22.5°C, then the system would work using the ground loop as a sink (M2—Summer Ground), during a 1% of the time, which is very low and could lead to an annual thermal unbalance in the ground. When the temperature of the fluid coming back from the ground loop and entering the heat pump is lower than 10°C, the system will use the ground for free-cooling (M10—Free cooling) purposes, this would happen during a 7% of the time. The rest of the operating modes during the summer mainly correspond to DHW production (M6—DHW Air) which uses the air as a source, during a 3% of the time.

Regarding the winter mode, as it can be observed in Figure 18, the system is working approximately during a 60% of the time in heating mode using the ground as a source (M5—Winter Ground). The rest of the time is mainly working in heating mode (34%) using the air as a source (M4—Winter Air). It can also be observed that the system only works a 6% of the time for DHW production (3% of the time using the ground as a source (M9—DHW Ground), and another 3% using the air as a source (M7—DHW Air)). This percentage is much lower than that of the heating mode operation, mainly due to the low DHW demand, which is practically zero during the night. On the other hand, as the system will be working in heating mode extracting heat from the ground during most part of the year, special attention should be paid to the summer where the heat injected into the ground is very low. This

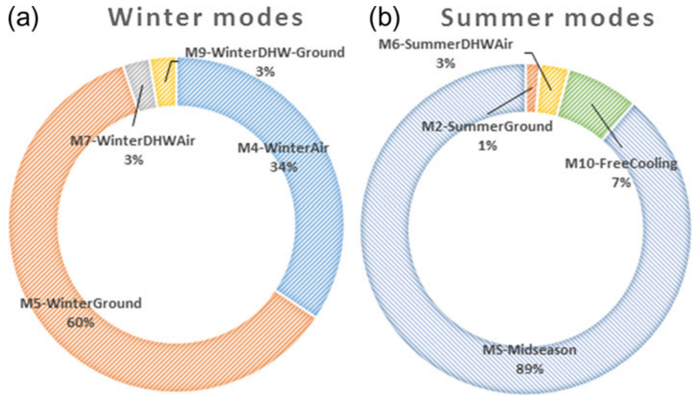

Figure 18. *Working mode time ratio for 1 year of operation of the dual source system: (**a**) Winter season and (**b**) Summer season.*

could let the ground thermally recover during the summer, but it may not be enough in order to reach a thermal balance in the ground, making it necessary to inject heat in the ground during the summer. This highlights the need for developing energy optimization and control operation strategies to avoid this situation and make the system work under its optimal operation point. By using the TRNSYS model developed in this research work, it will be possible to analyse the energy performance of the system and its impact in the ground temperature evolution over the years. Therefore, the model developed can be an assisting tool for the optimal design and operation of the system and also to assess the system energy performance and its suitability for other European countries with higher cooling thermal loads and lower heating thermal loads than the analysed case. For instance, a previous research work was carried out by the authors in Ref. [19], where the feasibility of the DSHP system was assessed for an installation located in Valencia, Spain. It was concluded that this type of DSHP system is even more convenient for Mediterranean climates (mostly cooling dominated), not only leading to a reduction in the size of the ground source heat exchanger needed, but also presenting a higher yearly SPF (SPF_4 equal to 4.62). This is mainly due to the higher efficiency of the DSHP prototype in cooling mode than in heating mode. Therefore, it can be concluded that the greater the cooling thermal demand of the location, the higher the annual SPF_4.

5 CONCLUSIONS

This paper presents the characteristics and energy performance of an innovative DSHP developed in the framework of a H2020 European project with title Geot€ch. The heat pump is able to employ either the air or the brine coming from the ground as heat sources in winter and provide hot water for heating the building. The unit is reversible, so it can also provide cooling during summer using the air or brine as a sink. Besides, it provides DHW all along the year, and in summer conditions, it can use the condensing waste heat to produce DHW. The heat pump is an outdoor unit, very similar to an air–water heat pump unit. All components, including circulation pumps are embedded in the same unit box, which only needs the connection of the heating/cooling water loop, the brine loop and the DHW loop. This way the heat pump can be considered as a 'plug&play' solution.

The DSHP unit has been designed, built and fully tested at the laboratory located at the IUIIE premises, at Universitat Politècnica de València. Furthermore, in the framework of the Geot€ch project it will be very soon tested in several demosites in the project. The unit has turned to be fully reliable with a smooth simple and full automatic operation. The DSHP works with R32 refrigerant and includes a variable speed compressor which give full capabilities for an efficient modulating operation. The unit has been fully tested at the laboratory with very accurate instrumentation and this has allowed the characterization of its performance with simple polynomials. The article includes a brief summary of its performance, including a small study about the influence of some important operating parameters on the system performance.

In order to assess the energy performance of the heat pump during 1 year of operation, an integrated system model has been developed in TRNSYS. The assessment consisted of two different steps.

The first step consisted of a comparison between a DSHP system (air/ground) and a GSHP system (only ground) for heating, cooling and DHW production in an office building located in the Netherlands. It was concluded that the DSHP system would be able to reach a similar efficiency than the GSHP system (yearly SPF_4 around 3.6) with half the length of BHEs. Therefore, the DSHP system would be a profitable option against a GSHP system, as the initial investment could be significantly reduced (up to a 30%) with a similar energy efficiency.

Then, the second step consisted of an analysis of the DSHP system operation and energy performance along 1 year. It was concluded that, for the type of weather considered (low cooling loads and high thermal loads) the system operates mainly in heating mode using the ground as a source during a 60% of the time, while it uses the air during a 34% of the time in winter season. In contrast to this, in summer season, the system is switched off during most of the time due to the mild summer existing in the Netherlands, and it only works during a low percentage of time (8%) using the ground as a sink (1% of mechanical cooling and 7% of free-cooling). Regarding the system energy performance, the SPFs factors according to the SEPEMO-build project definition were obtained for a 1 year operation period. The system presented a yearly performance factor SPF_4 around 3.58, whereas during the summer it took values considerably higher (around 5.09) due to the use of free-cooling and natural ventilation. It was concluded the need for developing key control strategies to optimize the seasonal energy performance of this type of system.

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