MODELLING AND ENERGY ANALYSIS OF A DUAL SOURCE HEAT PUMP SYSTEM IN AN OFFICE BUILDING

José Miguel CORBERÁN[1], Antonio CAZORLA[1], Javier MARCHANTE[1], Carla MONTAGUD[1], Ximo MASIP[1]

[1] *Instituto de Ingeniería Energética, Universitat Politècnica de València, Camino de vera s/n, 46022, Valencia, Spain, corberan@iie.upv.es*

Abstract: This paper presents the modelling and energy analysis of a dual source heat pump system for the production of heating, cooling and domestic hot water (DHW) in buildings.

The research work was carried out in the framework of the Geot€ch project 'GEOthermal Technology for economic Cooling and Heating'. The Geot€ch project, funded by the European commission within the H2020 program, is a four years' duration project which demonstrates the next generation of ground source heat pump systems with a high energy efficiency but also with lower system costs with respect to those already existing in the market. To this end, one of the objectives of the project is the design of an efficient and comparative low cost 'plug& play' system for providing the heating, cooling and DHW needs in three demonstration sites located in Italy, the United Kingdom and the Netherlands respectively.

In this context, an innovative dual source heat pump has been developed, which is capable of making optimal use of ground or air environmental heat sources according to operating and climate conditions. On the other hand, in order to reduce the costs of the installation, a new more efficient technology of coaxial borehole heat exchangers will be developed within the framework of the project. The project started on May 2015 and it is still ongoing.

This paper first describes the characteristics of the dual source heat pump designed in the project. Then, 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 (dual source heat pump, ground source heat exchanger, air conditioning and DHW hydraulic loops) is presented for the demo site located in the Netherlands. Finally, the paper presents an analysis of the system operation as well as a first energy assessment in order to identify key control strategies needed to optimize the seasonal energy performance of the system.

Keywords: heat pump, geothermal energy, energy efficiency

1. INTRODUCTION

According to the International Energy Agency, buildings account for almost a third of the final global energy consumption, and they are an equally important source of CO_2 emissions. In particular, heating, ventilating and air-conditioning systems (HVAC) account for roughly half of global energy consumption in buildings. The sector is expanding, which is bound to increase its energy consumption. Therefore, reduction of energy consumption and the use of energy from renewable sources in the buildings sector constitute important measures needed 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.

GSHP systems have proved to be more efficient than conventional air-to-water heat pumps, as demonstrated by (URCHUEGUÍA, et al., 2008, 2917) 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 ground source heat pumps is their high economic cost. Therefore, a reduction in both construction and operation costs is required for these systems to be successful, especially for Southern European countries where the market of GSHP systems has not taken off yet. Optimizing the energy consumption in these systems becomes thus of paramount importance.

A possible approach to saving 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 (BAKIRCI, et al., 2011). In the case of heating and cooling systems, a common practice is to combine GSHPs with thermal energy storage, mainly by means of phase change materials, as described in (LIU, et al., 2015). In fact, both a hybrid system and a thermal storage device can be combined as described in (PARDO, et al., 2010). However, this is not always possible and it contributes to increase the economic cost of the system.

The Geot€ch project 'Geothermal Technology for economic Cooling and Heating' (www.geotech-project.eu) is a 4 years' duration project (2015-2018) which demonstrates the next generation of ground source heat pump 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 and 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 will be designed and installed in 3 demo sites located in Italy, the United Kingdom and the Netherlands respectively. In this context, an innovative dual source heat pump was developed, which is capable of making optimal use of ground or air environmental heat sources according to operating and climate conditions. Thanks to the use of this innovative heat pump, the seasonal performance is optimized. Additionally, the total heat exchanged with the ground will decrease, also decreasing the ground source heat exchanger area needed with a consequent lower cost of the installation. Furthermore, in order to reduce the cost of the ground source heat exchanger, an innovative more efficient and easy to install borehole heat exchanger configuration will be developed, installed, and finally optimized in the project.

This paper first describes the characteristics of the dual source heat pump designed in the project. Then, 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 (dual source heat pump, ground source heat exchanger, air conditioning and DHW hydraulic loops) is presented for the demo site located in the Netherlands. Finally, the paper presents an analysis of the system operation as well as a first energy assessment in order to identify key control strategies needed to optimize the seasonal energy performance of the system.

2. 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 (dual source heat pump, 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 1.

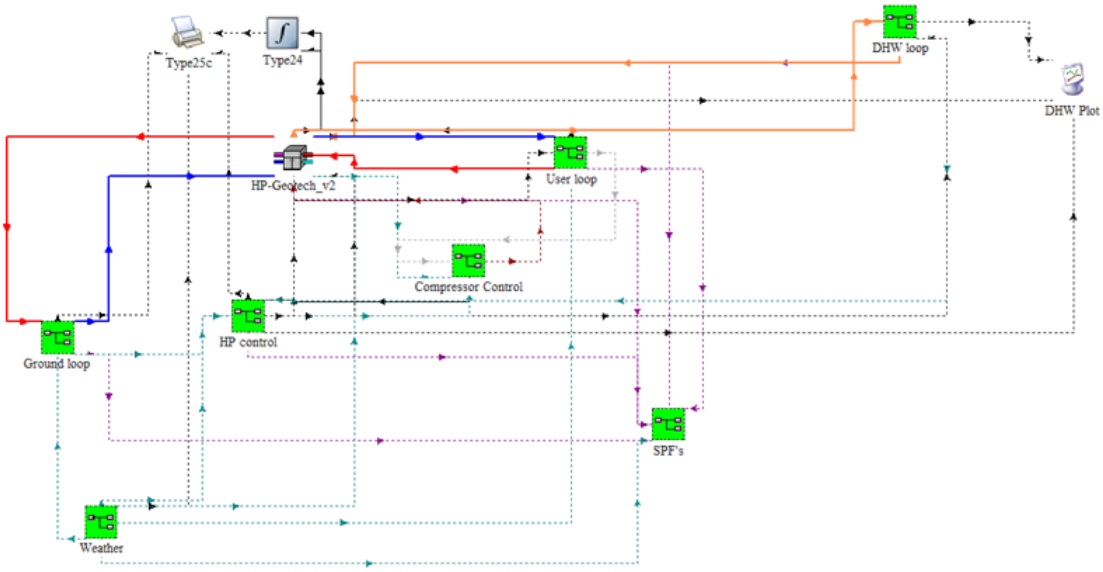

Figure 1: Integrated system model in TRNSYS layout

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 litres, but the demand of DHW in the office is 22.5 litres/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's 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).

2.1. Model of the innovative dual source heat pump

The concept of the innovative heat pump (HP) described below is to use a dual source air and ground to minimize the use of the ground source in order to reduce the required size of the borehole heat exchangers. Furthermore, this concept will allow using the air source instead of the ground source whenever it is more convenient from an efficiency point of view, leading to a superior seasonal performance compared with current technology.

This model of HP operates with R32 as refrigerant with a nominal heating capacity of 8 kW and a constructive solution 'plug&play'. It has to satisfy all the heating and cooling demand of the residence where it will be installed and 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. To achieve it, from a constructional point of view, it is necessary to provide to the HP with four different types of heat exchangers. One of these for the heat transfer with the air source and another for the ground source. The last two will provide DHW and User (heating and cooling) demands.

Figure 2 shows a simple layout of the dual source It is possible to identify three hydraulic loops for the User, Ground and DHW heat exchangers respectively. This system uses a brazed plate heat exchanger (BPHE) technology for these loops, flowing water as secondary fluid for the User and DHW loops and glycol for the Ground loop in order to protect the equipment from freezing. The plate size for the User and Ground heat exchanger is the same (526 x 119 mm), being smaller for the DHW heat exchanger (376 x 119 mm).

Furthermore, Air source uses a Round Tube and Plate Fin (RTPF) heat exchanger technology with the following dimensions: 1125 x 750 mm.

This unit incorporates a variable speed compressor based on scroll technology in order to be able to adapt the performance to the wide range of operating conditions and to minimize part load operation losses.

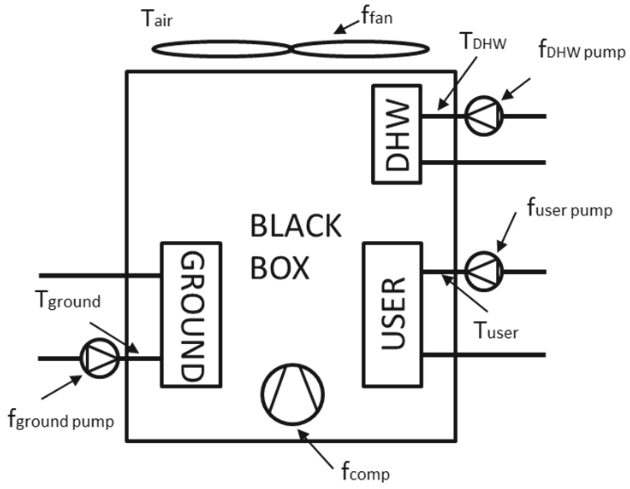

Figure 2: Black box model with the main components and inputs of the heat pump

The dual source heat pump must operate 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. Regarding the working mode MS-Midseason indicated in Table 1, it corresponds to those moments of the year in which the external ambient temperature gets very mild values in the range of 21±1.5ºC in cooling mode and 19±1.5ºC in heating mode. For those conditions, there is no need for the heat pump to be switched on, and the system can take advantage of the thermal inertia of the building.

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

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 for each operating mode. These polynomial correlations will be obtained for the condenser capacity, evaporator capacity and compressor power input.

The process followed for obtaining these polynomial correlations consists of the definition and adjustment of a previous model of the HP by the use of the IMST-ART software (www.imst-art.com) which is a powerful tool for the design, modelling, analysis and optimization of vapour compression cycles.

The first step of the heat pump modelling consists of introducing the data from the heat pump manufacturer corresponding to the characteristics of the different components (compressor, heat exchangers, refrigerant circuit, expansion device, etc). Then, the IMST- ART's simulation results are compared to the experimental measurements for the design points and the IMST-ART model is adjusted in case that it is needed. Then, by running parametric studies in IMST-ART, it is possible to obtain the heat pump performance at different working conditions (different source and distribution temperatures, water/brine flow rates and different compressor frequencies). Finally, for a given cloud of performance points (640 points considered in total) obtained as a result of the parametric studies' simulations in IMST-ART, the fitted polynomial correlations can be obtained with a good precision and they can be programmed as a black box in a model. Additionally, in case that there are experimental measurements available of the unit working at different conditions, the coefficients of the polynomial correlations can be updated if needed.

An example of the correlations is presented in the following. Equations (1), (2) and (3) correspond to Winter Ground operating mode (M5); and Equations (4) and (5), correspond to Winter Air operating mode (M4). The final coefficients appearing in the correlations were finally adjusted after a comparison with the experimental results obtained during the first experimental campaign carried out in the framework of the GEOT€CH project.

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

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

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

$$\dot{Q}_{cond} = f_{comp} \cdot (C_0 + C_1 \cdot Te_i + C_2 \cdot Tc_i + C_3 \cdot Te_i \cdot Tc_i + C_4 \cdot Te_i^{\,2} + C_5 \cdot f_{fan} + C_6 \cdot dTc + C_7 \cdot Te_i \cdot f_{fan} + C_8 \cdot Tc_i \cdot dTc) \quad (4)$$

$$\begin{aligned}\dot{W}_{comp} = {} & f_{comp} \cdot (C_0 + C_1 \cdot Te_i^{\,2} + C_2 \cdot Tc_i^{\,2} + C_3 \cdot f_{fan} + C_4 \cdot dTc + C_5 \cdot Te_i \cdot f_{fan} + C_6 \cdot f_{comp} \cdot Te_i + C_7 \cdot f_{comp} \cdot Tc_i + \\ & + C_8 \cdot f_{comp}^{2} \cdot Te_i \cdot Tc_i + C_9 \cdot f_{comp} \cdot f_{fan} + C_{10} \cdot \frac{1}{f_{comp}})\end{aligned} \quad (5)$$

where $\dot{Q}_{cond}$ is the condenser capacity (W]), f_{fan} is the fan frequency (Hz), Te_i is the secondary fluid inlet temperature in the evaporator (K), $\dot{Q}_{evap}$ is the evaporator capacity (W), dTe is the secondary fluid temperature difference in the evaporator (K), Tc_i is the secondary fluid inlet temperature in the condenser (K), $\dot{W}_{comp}$ is the compressor power input (W), f_{comp} is the compressor's frequency (Hz), and dTc is the secondary fluid temperature difference in the condenser (K).

Finally, Figure 3 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% for both cases. Regarding the rest of the working modes, the maximum deviation found is lower than 5% in most of the cases.

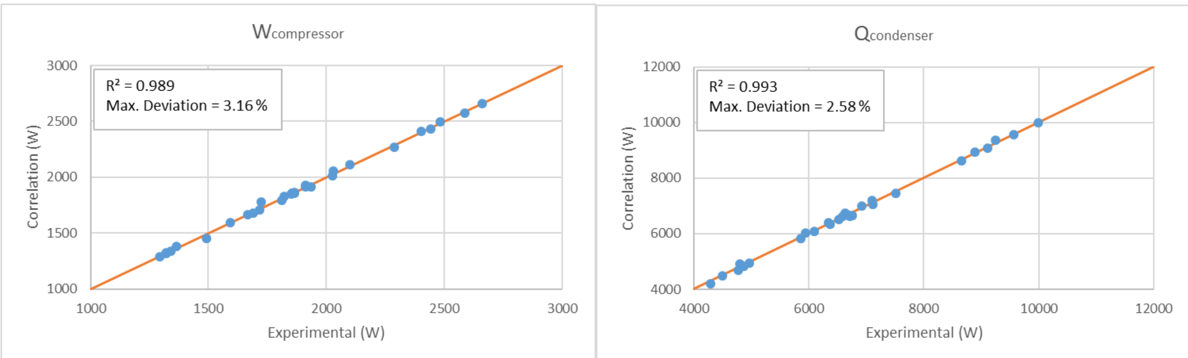

Figure 3: Experimental measurements versus polynomial correlations: a) Compressor consumption. b) Condenser capacity

2.2. Model of the new coaxial BHE

The BHE used in the system model is an innovative coaxial-spiral BHE that was developed by Geothex BV (http://geothex.nl). 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 (WITTE, 2012,10).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 (RUIZ-CALVO et al., 2015) and (DE ROSA et al., 2015).

The model reproduces the short-term behavior 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. The thermal network consists of different nodes connected via thermal resistances. Each node represents a part of the BHE with its thermal capacity and the thermal resistances represent the conductive and convective heat transfer between the different parts. These nodes represent the fluid in the inner pipe (T_i), the fluid in the outer pipe (T_o), the grout (T_b), and the surrounding ground (T_{g1} and T_{g2}). This thermal network is shown in Figure (a). Furthermore, the BHE is vertically discretized in n 2D thermal networks. The vertical advection of the fluid inside the pipes is considered, as well as the conductive heat transfer between the grout and ground nodes that are vertically adjacent. This can be shown in Figure 4 (b).

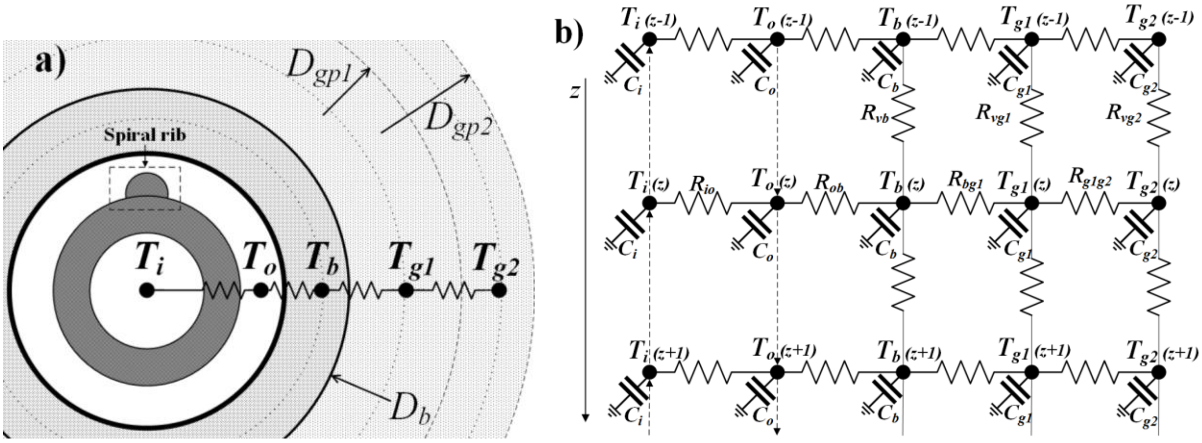

Figure 4: Thermal network of the BHE model: a) Borehole layout. b) Vertical discretization (CAZORLA-MARIN et al., 2017)

In order to take into account the spiral flow path along the outer pipe, it is considered an equivalent section and an equivalent hydraulic diameter in the calculation of the hydraulic and thermodynamic properties. A more detailed explanation of the model can be found in (CAZORLA-MARIN et al., 2017).

2.3 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 5 shows the operation of this control for (a) heating mode and (b) cooling mode.

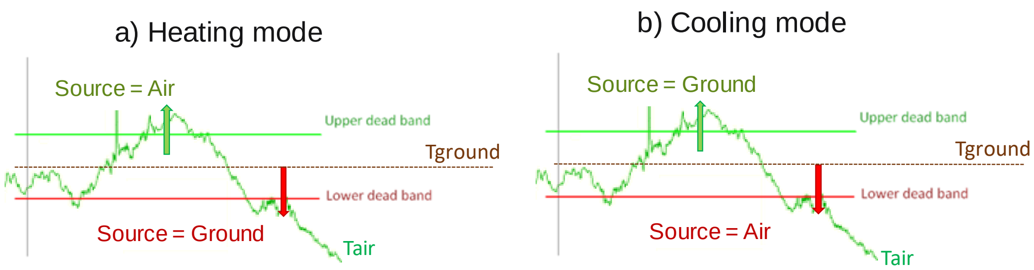

Figure 5: Selection of the source depending on the air and ground temperature; a) Heating mode. b) Cooling mode

In order to program 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.

3. PRELIMINARY ANALYSIS OF THE SYSTEM ENERGY PERFORMANCE

The system developed in TRNSYS was used to make a preliminary assessment of the system performance.

3.1. Dynamic performance during one week

This section presents a preliminary analysis of the system energy performance during one week of operation in order to describe the dynamic behaviour of the system and the selection of working mode for one typical week in heating mode. Figure 6 shows the selection of working mode during one 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 2.3 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.

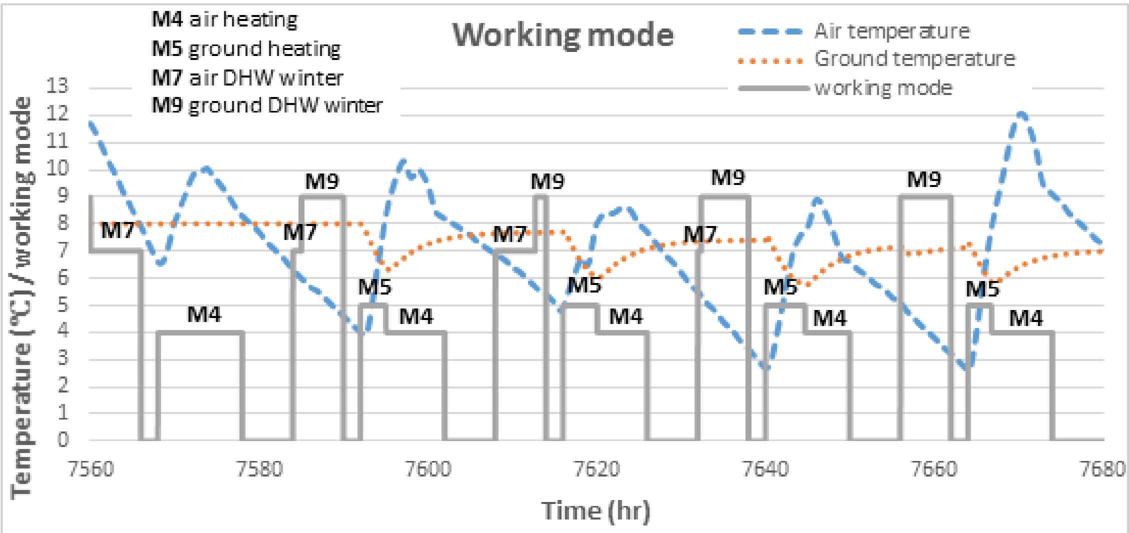

Figure 6: Change of working mode during one week in autumn

3.2. Working mode analysis

Figure 7 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 21ºC during most of the time. This is the reason why the thermal loads in the summer are very small leading to a high percentage (83%) 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. In the case that the air is hotter than 22.5ºC, then the system would work using the ground loop as a sink (M2-Summer Ground), during a 9% of the time. When the temperature of the fluid coming back from the ground loop and entering the heat pump is lower than 10ºC, the system will use the ground for free-cooling (M10- Free cooling) purposes. This latter situation takes place during a 1% 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. It should be pointed out that, during a 4% of the time, the system was working in heating mode using the air as a source (M4-Winter Air). This is because there was a slight heating thermal demand during the summer time in September that had to be covered by the heat pump.

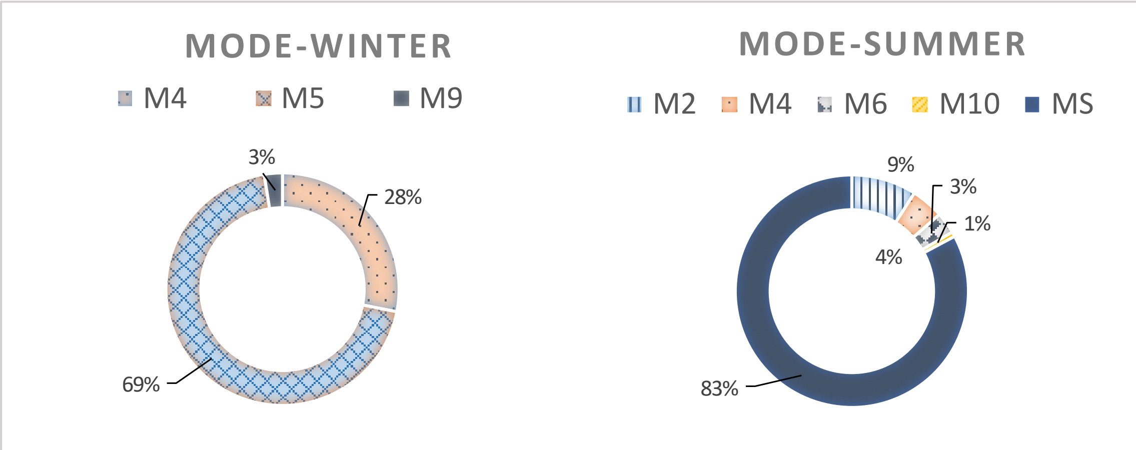

Figure 7: Working mode analysis for one year of operation.

Regarding the winter mode, as it can be observed in Figure 7, the system is working during most of the time (69%) in heating mode using the ground as a source (M5-Winter Ground). The rest of the time is mainly working in heating mode (28%) using the air as a source (M4-Winter Air), and a 3% of the time is working for the production of DHW using the ground as a source (M9-DHW Ground) . This is mainly because the schedule considered in the control board of the system for DHW production is fixed to take place at night. Therefore, during the winter, the ground will take higher values than the ambient air, which makes the ground a better source in this case than the air. 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 there is practically no heat injected into the ground. This can let the ground thermally recover during the summer. 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. So, 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.

3.3. Seasonal Performance Factor of the system

Finally, this section presents a first assessment of the energy performance carried out by means of quantifying the seasonal performance factor (SPF) of the system for a whole year of operation. The expressions used for the SPFs of the system (equations from 1 to 4) were defined according to the SEPEMO-build "SEasonal PErformance factor and MOnitoring for heat pump systems in the building sector (SEPEMO-Build)" project’s definition. 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 Seasonal Performance Factor (SPF) including a monitoring programme for 46 heat pump installations in six European countries. These expressions are presented in the following.

$$SPF_1 = \frac{\int_0^t \left(\dot{Q}_{USER} + \dot{Q}_{DHW}\right) dt}{\int_0^t \left(\dot{W}_{HP}\right) dt} \tag{6}$$

$$SPF_2 = \frac{\int_0^t \left(\dot{Q}_{USER} + \dot{Q}_{DHW}\right) dt}{\int_0^t \left(\dot{W}_{HP} + \dot{W}_{FAN} + \dot{W}_{BHE}\right) dt} \tag{7}$$

$$SPF_3 = \frac{\int_0^t \left(\dot{Q}_{USER} + \dot{Q}_{DHW}\right) dt}{\int_0^t \left(\dot{W}_{HP} + \dot{W}_{FAN} + \dot{W}_{BHE} + \dot{W}_{BACKUP}\right) dt} \tag{8}$$

$$SPF_4 = \frac{\int_0^t \left(\dot{Q}_{USER} + \dot{Q}_{DHW}\right) dt}{\int_0^t \left(\dot{W}_{HP} + \dot{W}_{FAN} + \dot{W}_{BHE} + \dot{W}_{BACKUP} + \dot{W}_{USER} + \dot{W}_{DHW}\right) dt} \tag{9}$$

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 $\dot{W}_{BACKUP}$.

It should be pointed out that, in the system considered for the analysis, there is no back-up heater. Therefore, SPF_2 and SPF_3 will get the same values. Figure 8 presents the results of the preliminary assessment of the system energy performance over a whole year of operation. As it can be observed, the seasonal performance SPF_1 is very high during the summer getting values around 8, whereas in winter it is much lower getting values around 3.5. This is mainly because during the summer, the system is working during most of the time in free-cooling mode using the ground loop or the ambient air as it was presented in section 3.2. Then, as the system is mainly operating in heating mode during most part of the year, the yearly value of SPF_1 takes a value of 3.8 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 which means that the auxilliarie's consumption for the ground loop circulation pump and the fan don't have a great impact in the energy consumption of the system. However, when considering the circulation pumps existing in the DHW loop and the user loop (for heating and cooling), SPF_4 decreases a 15% during the summer, getting a yearly value of 6.8 approximately, which makes the yearly performance factor of the system go down to 3.6. This is mainly due, on one hand to the very low thermal energy demand during the summer, which makes the user circulation pump's consumption have a great impact in the SPF as it is switched on during 10 hours a day, no matter the thermal energy demand of the building. This highlights the need of developing optimization strategies for the operation of the system especially during the summer, and then increase the global energy efficiency of the system.

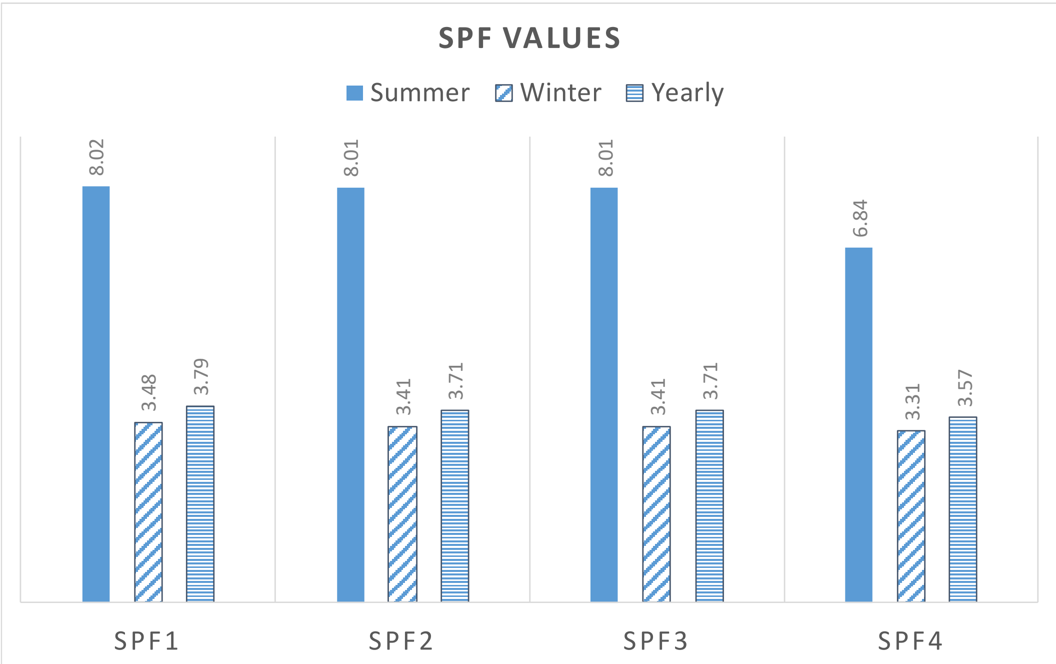

Figure 8: Seasonal performance factors for a whole year of operation.

4. CONCLUSIONS

This paper presents a first preliminary assessment of the dynamic behaviour and system energy performance of a dual source heat pump system for heating, cooling and DHW production in an office building located in the Netherlands. For that purpose, a model of the integrated system was developed in TRNSYS including all the system components (dual source heat pump, ground source heat exchanger, air conditioning and DHW hydraulic loops). 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 69% of the time, while it uses the air during 28% of the time in winter season. In contrast to this, during the 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 (9%) using the ground as a sink, mainly in free-cooling mode as expected. Regarding the system energy performance, the SPFs factors according to the SEPEMO-build project definition were obtained for a one year operation period. The system had a yearly performance factor SPF_4 of 3.6, whereas during the summer it took values around 6.8. A high influence in the system energy performance was observed for the auxiliaries' consumption, especially during the summer (15% reduction in the SPF). It was concluded the need for developing key control strategies to optimize the seasonal energy performance and also to provide a thermal energy balance in the ground.

5. ACKNOWLEDGEMENTS

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