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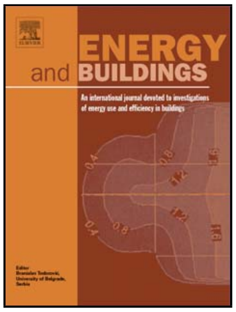

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Comparison between design and actual energy performance of a HVAC-Ground Coupled Heat Pump system in cooling and heating operation

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ABSTRACT

This work compares the experimental results obtained for the energy performance study of a ground coupled heat pump system with the design values predicted by means of standard methodology. The system energy performance of a monitored ground coupled heat pump system is calculated using the instantaneous measurements of temperature, flow and power consumption and these values are compared with the numerical predictions. These predictions are performed with the TRNSYS software tool following standard procedures taking the experimental thermal loads as input values. The main result of this work is that simulation results solely based on nominal heat pump capacities and performances overestimate the measured overall energy performance by a percentage between 15% and 20%. A sensitivity analysis of the simulation results to changes in percentage of its input parameters showed that the heat pump nominal coefficient of performance is the parameter that mostly affects the energy performance predictions. This analysis supports the idea that the discrepancies between experimental results and simulation outputs for this ground coupled system are mainly due to heat pump performance degradation for being used at partial load. An estimation of the impact of this effect in energy performance predictions reduces the discrepancies to values around 5%.

Keywords: Heating and cooling systems, ground coupled heat pump, energy efficiency, computer simulation, experiments.

1. INTRODUCTION

Ground coupled heat pumps are recognized by the U.S. Environmental Protection Agency [1] as being among the most efficient and comfortable heating and cooling systems available today. These pumps represent a good alternative system for heating and cooling buildings [2-10]. By comparison with standard technologies, these heat pumps offer competitive levels of comfort, reduced noise levels, lower greenhouse gas emissions and reasonable environmental safety. Their electrical consumption and maintenance requirements are lower than those required by conventional systems and, therefore, have lower annual operating cost [11-13].

The design of a ground coupled heat pump HVAC system is based on predictions coming from simulation tools. First step in a standard design procedure is the estimation of the thermal loads that the air-conditioned area is going to demand. Its value determines the capacity of the ground source air-conditioning system. From this value and a proper estimation of the ground thermal properties, the characteristics of the water to water heat pump and the required length and layout of the borehole heat exchangers are estimated.

The purpose of this work is to compare a standard design procedure based on a TRNSYS [14] simulation with the experimental results obtained on a monitored geothermal plant. The experimental validation of design models for thermal facilities is the subject of a considerable amount of research works [15-19]. In [15], an air-cooled reciprocating chiller is modeled and analyzed. In [16], for instance, a variable-refrigerant-volume air conditioning system is simulated using EnergyPlus and experimentally validated. In [17], a numerical model for heat storage with phase change materials is presented and experimentally checked. In [18], the cooling capacity of earth-air-pipe systems is modeled and evaluated. And in [19], the design and performance of solar powered absorption cooling systems is studied. Many other references can be included here, being the ones presented before just a representative sample of the strong activity in the area.

Some of the models describing the behaviour of thermal facilities have been implemented as modules for the TRNSYS software tool. Its experimental validation is also a field of strong research activity [20-23]. In [20], a comparison between measured and predicted long term performance of a grid connected photovoltaic system using TRNSYS is performed. In [21], thermal testing and numerical simulation with the TRNSYS software of a prototype cell using light wallboards coupling vacuum isolation panels and phase change materials is presented. In [22], experimental measurements and numerical modeling with TRNSYS of a green roof are compared. And in [23], the validation of a TRNSYS computer model for solar assisted district heating systems with seasonal hot water heat store is studied. More references can be added being the presented ones just a representative sample of the almost standard use of TRNSYS software as simulation tool of the dynamic behaviour of thermal facilities.

In the subject of the present work, the experimental validation of models for ground coupled heat pumps working in heating and cooling mode, the amount of research works is more limited. In Europe, research focused in this area has been performed in Turkey, with the objective of experimentally characterizing the system performance, and also in the development of models to predict this performance. In Hepbasli [24, 25] and Inalli and Esen [26, 27] the experimental characterization of ground coupled heat pump system performance working in both heating and cooling modes was tempted. There are also studies of ground coupled heat pump system performance when combined with thermal solar energy [28-31]. In Esen, Inalli et al [32, 33] models to predict ground coupled heat pump system performance are presented. These authors have also developed research in the subject of the present work. In [34] experimental measures of the energy performances of a horizontal ground coupled heat pump system are shown, which are used to validate a numerical model describing ground heat transfer. In [35] a study on modeling and performance assessment of a heat pump system for utilizing low temperature geothermal resources in buildings is presented. Finally, in [36-40], the recent research developed in China on the subject of ground coupled heat pumps working in refrigeration mode is described.

The main objective of this work is to compare long term energy performance experimental measurements of a monitored ground coupled heat pump system, with the predictions from a standard design procedure based in the TRNSYS simulation tool. One of the difficulties that appear when doing this comparison comes from the fact that the actual thermal loads differ significantly from the estimated ones. These estimations are based in predictions of building occupancies, weather data … being in many cases substantially different than the actual operation conditions. To avoid this difficulty, in this work the measured thermal loads are used as input value of the simulation design tool to evaluate the goodness of the models describing the ground coupled heat pump HVAC system.

In this study a comparison between the energy performance measured in GeoCool geothermal experimental plant and the energy performance predictions from a standard design procedure is performed. This procedure uses as input values the thermal loads measured along a whole year of measurements, nominal heat pump capacities and performances, and ground thermal properties. Numerical predictions and experimental results are compared, performing then an exhaustive analysis of the origin of the discrepancies between both.

This article is structured as follows. In Section 2, the experimental setup of GeoCool installation is described. Section 3 presents the procedure to calculate the system energy efficiency. Afterwards, in Section 4, the simulated system, its structure, inputs and outputs are explained. In Section 5, the results are presented and discussed. Finally, in section 6, the conclusions obtained from the presented results are summarized.

NOMENCLATURE

2. GEOTHERMAL EXPERIMENTAL PLANT

Geothermal experimental system was the result of a EU project (GeoCool) and air-conditions a set of spaces in the Department of Applied Thermodynamics at the Polytechnic University of Valencia, Spain, with a total surface of approximately 250 m^2. This area includes nine offices, a computer classroom, an auxiliary room and a corridor. All rooms, except the corridor, are equipped with fan coils supplied by the experimental system: an air to water heat pump and a ground coupled (geothermal) heat pump working alternately (Figure 1). The geothermal system consists of a reversible water to water heat pump (15.9 kW of nominal cooling capacity and 19.3 kW of nominal heating capacity), a vertical borehole heat exchanger and a hydraulic group. The water to water heat pump is a commercial unit (IZE-70 model manufactured by CIATESA) optimized using propane as refrigerant. As reported in GeoCool final publishable report [41], the coefficient of performance of the improved heat pump is 34% higher in cooling and 15% higher in heating operation.The vertical heat exchanger is made up of 6 boreholes of 50 m. depth in a rectangular configuration, with two boreholes in the short side of the rectangle and three in the large side, being 3 m. the shorter inter-borehole distance. All boreholes are filled with sand and finished with a bentonite layer at the top to avoid intrusion of pollutants in the aquifers.

FIGURE 1

The GeoCool plant was designed to allow a fair comparison between a ground source (geothermal) heat pump system and an air source heat pump system [2, 3], therefore a network of sensors was set up to allow monitoring the most relevant parameters of these systems (Figure 1). These sensors measure temperature, mass flow and power consumption. The temperature sensors are four wire PT100 with accuracy ±0.1 ºC. The mass flow meters are Danfoss Coriolli meters, model massflo MASS 6000 with signal converter Compact IP 67 and accuracy <0.1%. The power meters are multifunctional power meters from Gossen Metrawatt, model A2000 with accuracy ±0.5% of the nominal value. Data from this sensor network is collected by a data acquisition unit Agilent HP34970A with plug-in modules HP34901A.

The geothermal system is characterized by the heat that the ground can absorb or transfer. To record this value inlet and outlet fluid temperature of the water to water heat pump and circulating mass flow are measured. In addition inlet and outlet temperature in each borehole are measured too and in three of the boreholes the temperature at several depths is recorded to acquire ground temperatures. There is a power meter located on the right of figure 1 which has two functions: record the consumption of the air to water heat pump including the fan when the air system is working or record the consumption of the water to water heat pump plus the circulation pump when the geothermal system is working.

3. SYSTEM ENERGY EFFICIENCY

The system energy efficiency is calculated from the power consumption readings and the values of the internal thermal loads calculated from experimental measurements. These thermal loads are calculated with the values of T_{in}, T_{out} and $\dot{m}$ showed in Figure 1 (measured with four wire PT100 temperature sensors and a Coriolis meter). Instantaneous thermals loads are obtained by means of the following expression:

$$\dot{Q}(t) \,!\, \dot{h}_{out}(t) \,!\, \dot{h}_{in}(t) \quad , \qquad (1)$$

where,

$$\dot{h}_{in}(t) \,!\, \dot{m}CpT_{in}(t) \quad ; \quad \dot{h}_{out}(t) \,!\, \dot{m}CpT_{out}(t) \qquad (2)$$

are the input and output enthalpy flows at the circuit connecting the fan coils and the heat pump. Because of all the measures are taken in one minute intervals, the internal thermal load is defined as the integral of expression (1). It represents the cooling or heating load demanded by the building during the time period *Δt* starting at T_0 time.

$$Q = \int_{T_0}^{T_0+\Delta t} \dot{Q}(t)dt \qquad (3)$$

Likewise, the system energy consumption is calculated by integrating numerically the power consumption, $\dot{W}$, measured by the power meter located on the right of figure 1, corresponding to the consumption of the water to water heat pump, $\dot{W}_{ww}$, plus the consumption of the circulation pump, W_{cp} .

$$W \,!\, \underset{T_0}{\overset{T_0 ! ! t}{!}} \dot{W}(t)dt \quad ; \quad \dot{W}(t) \,!\, \dot{W}_{ww}(t) \,!\, \dot{W}_{cp}(t) \qquad (4)$$

The system energy efficiency is characterized by the energy performance factor, defined as the ratio between the thermal load and the electric energy consumption during a time interval:

$$PF = \frac{Q}{W} \tag{5}$$

Depending on the duration of the integration period the energy performance factor can be seasonal, monthly, daily, etc. The most representative one is the seasonal performance factor (SPF) that estimates the system performance in a working mode (heating or cooling).

4. SIMULATED SYSTEM

The aim of this work is to compare a ground coupled heat pump design methodology with experimental results; therefore, GeoCool plant is modelled and simulated with TRNSYS software tool, commonly used by geothermal engineers.

TRNSYS [14] is a transient system simulation program with a modular structure that was designed to solve complex energy system problems by breaking the problem down into a series of smaller components (referred to as “Types”). TRNSYS Library includes the components commonly found in a geothermal system (ground heat exchanger, heat pump, circulation pump, etc) and the program allows to directly join the components implemented using other software (e.g. Matlab or Excel). In this case this feature is important because the simulation uses as input values the experimental thermal loads measured in GeoCool experimental plant, in form of an Excel file.

Figure 2 shows the TRNSYS model scheme used to simulate GeoCool plant. The model scheme consists of four components: water to water heat pump, circulation pump, vertical ground heat exchanger and loads. The first three components have been selected from TRNSYS library. And the last component, “Loads”, is an Excel file containing the experimental thermal loads. All components are described next.

FIGURE 2

4.1 Water to water heat pump (WWHP)

The water to water heat pump selected component is a reversible heat pump; it supplies the thermal loads absorbing energy from (heating mode) or rejecting energy to (cooling mode) the ground. This type is based on user-supplied data files containing catalogue data for the capacity and power draw, based on the entering load and source temperatures. These files (one for heating and one for cooling) are modified introducing the values of the GeoCool commercial unit (CIATESA IZE-70). The performance improvement coming from using propane as refrigerant instead of R-407c is also included (an increment of 34% for the Efficiency Energy Rate, EER, and an increment of 15% for the coefficient of performance, COP, as reported in GeoCool final report [41]). These corrections have been included by diminishing the value of the absorbed power by the compressor for the same amount of generated thermal power. The model is able to interpolate data within the range of input values specified in the data files but it isn't able to extrapolate beyond the data range.

The component works with two control signals: heating and cooling. When one of these signals are on, the model calls the corresponding data file and calculates the coefficient of performance (COP), the energy absorbed ($\dot{Q}_{absorbed}$) or rejected ($\dot{Q}_{rejected}$) and the outlet temperatures of the water in the internal (load) and external (source) circuits. In this case source represents the ground heat exchanger. These values are given by the following equations.

Heating mode

$$COP = \frac{\dot{Q}_{ww,heating}}{\dot{W}_{ww,heating}} \tag{6}$$

$$\dot{Q}_{absorbed} = \dot{Q}_{ww,heating} - \dot{W}_{ww,heating} \tag{7}$$

$$T_{source,out} = T_{source,in} - \frac{\dot{Q}_{absorbed}}{\dot{m}_{source}\, Cp_{source}} \tag{8}$$

$$T_{load,out} = T_{load,in} - \frac{\dot{Q}_{ww,heating}}{\dot{m}_{load}\, Cp_{load}} \tag{9}$$

Cooling mode

$$COP = \frac{\dot{Q}_{ww,cooling}}{\dot{W}_{ww,cooling}} \tag{10}$$

$$\dot{Q}_{rejected} \; ! \; \dot{Q}_{ww,cooling} \; ! \; \dot{W}_{ww,cooling} \tag{11}$$

$$T_{source,out} = T_{source,in} + \frac{\dot{Q}_{rejected}}{\dot{m}_{source}\, Cp_{source}} \tag{12}$$

$$T_{load,out} = T_{load,in} + \frac{\dot{Q}_{ww,cooling}}{\dot{m}_{load}\, Cp_{load}} \tag{13}$$

Where $\dot{Q}_{ww}$ represents the heat pump heating or cooling capacity at current conditions and $\dot{W}_{ww}$ means the power drawn by the heat pump in each mode.

4.2 Circulation pump (CP)

The circulation pump component is a simple-speed model which computes mass flow rate using a variable control function, which must have a value between 1 and 0 (*f*). The user can fix the maximum flow capacity, in the model established for the heat pump, and the pump power is calculated as a linear function of mass flow rate, defined in the following expression:

$$\dot{W}_{cp} = \dot{W}_{\max,cp}\frac{\dot{m}_{source}}{\dot{m}_{\max,source}} = \dot{W}_{\max,cp}\frac{\dot{m}_{\max,source} f}{\dot{m}_{\max,source}} = \dot{W}_{\max,cp} f \tag{14}$$

$\dot{W}_{\max,cp}$ and $\dot{m}_{\max,source}$ are the pump power consumption and the water mass flow when the pump is operating at full capacity, and $\dot{m}_{source}$ is the water mass flow through the pump in each time step, obtained by multiplying the maximum flow rate by the control signal.

4.3 Vertical ground heat exchanger (VGHE)

A vertical ground heat exchanger model must analyze the thermal interaction between the duct system and the ground, including the local thermal process around a pipe and the global thermal process through the storage and the surrounding ground. GeoCool ground heat exchanger has been modelled using ‘Duct Ground Heat Storage Model’ [42]. This model assumes that the boreholes are placed uniformly within a cylindrical storage volume of ground. There is convective heat transfer within the pipes, and conductive heat transfer to the storage volume. The temperature in the ground is calculated from three components: global temperature, local solution and a steady flux solution. The global and local problems are solved with the use of an explicit finite difference method. The steady flux solution is obtained analytically. The temperature is then calculated using superposition methods.

The user can define ground thermal properties like thermal conductivity and heat capacity and also determine the main heat exchanger characteristics (depth, radius, number of boreholes, etc.). The parameters used in the simulation are shown in Table 1.

In order to evaluate the ground thermal properties at GeoCool site, laboratory experiments on soil samples were performed. The fill thermal conductivity considered is the average value for wet sand. Also U-tube pipe parameters correspond to the properties of polyethylene pipes DN 32 mm PE 100.

4.4 Loads

Loads module represents the thermal loads that the air conditioning area is demanding. Normally, these thermal loads are estimated from the building characteristics, occupancies, weather data… obtaining an a priori prediction of the hourly thermal loads that the air conditioning area is demanding. To make a better comparison between the usual design procedure to predict the energy performance of the system and the experimental data measured, the simulation uses the experimental thermal loads measured in GeoCool plant along a whole cooling season and a whole heating season as input values. An Excel file keeps these experimental thermal loads calculated from the experimental data using equation (3) with integration periods of one hour. Figure 3 shows the values obtained after performing this calculation.

FIGURE 3

At each time step of the simulation (one hour) loads module can supply the value of the corresponding hourly thermal load. Nevertheless, the quantity that the water to water heat pump module needs as one of its inputs is the inlet load temperature ($T_{load,in}$). To calculate this input value from the hourly thermal load, the internal circuit (hydraulic pipes that connect the heat pump with the fan coils) is considered as a control volume where the power balance can be evaluated as:

$$\dot{m}_{load}\, Cp_{load}\,(T_{load,in} - T_{load,out}) = \dot{Q}_{ww} - \dot{Q} \qquad (15)$$

where $\dot{Q}$ represents the hourly thermal loads and $\dot{Q}_{ww}$ is the heat pump capacity at current conditions. The initial conditions were assumed as 20 ºC of pipes water temperature and a pipes volume of 0.5 m^3. Equation (15) is programmed in the Excel file, so the excel file directly supplies the inlet load temperature to the water to water heat pump module. Loads component also supplies to the heat pump module the control signals for running or stopping ($T_{load,\ out}$ =45 ºC is fixed as stop temperature in heating and $T_{load,\ out}$=12 ºC is fixed as stop temperature in cooling).

4.5 Model outputs: energy performance factor

The TRNSYS model calculates the energy performance factors which are to be compared with the corresponding experimental values. The simulation program obtains this quantity following the same procedure outlined in section 3 to calculate the experimental value for the energy performance factor. Besides, the model plots the evolution of the main system parameters as ground and source temperatures, ground heat exchanger flow rate, control signals, devices power consumption, etc.

5. COMPARISON BETWEEN SIMULATION OUTPUTS AND EXPERIMENTAL RESULTS

In this section, a comparison between the measured energy performance factors of GeoCool plant and its predictions from the TRNSYS simulation is presented, as well as an exhaustive analysis of the discrepancies between both.

GeoCool experiment was designed to make a comparison between a ground source heat pump system and an air source heat pump system operating during a whole year. The conclusions of this comparison were presented in [2, 3]. The experimental measurements of GeoCool ground coupled system are used in this work to compare the energy performance factors of the plant with the ones predicted by a standard design procedure based in TRNSYS simulation. Experimental measures were acquired in one minute intervals during a whole heating season and a whole cooling season. Heating season experimental data are available for the periods from January 31, 2005 until May 6, 2005, and from October 17, 2005 until January 13, 2006. Cooling season experimental data are available for the periods from May 9, 2005 until July 31, 2005, and from September 1, 2005 until October 14, 2005. A simulation using TRNSYS software tool has been performed to exactly reproduce GeoCool experiment. The measured energy performances and the predicted ones are compared in figures 4 and 5.

FIGURE 4

In figure 4, the accumulated (long term) energy performances for both seasons are depicted. Dash-dotted lines correspond to the values obtained from the simulation and solid lines correspond to experimental measured values. The quantity shown in the vertical axis is the performance factor (PF) defined in equation (5), for an integration period starting the first day of each season and ending at the day in the abscissa. From this figure it can be seen that the simulation outputs overestimate the experimental measures by a percentage between 15% and 20%. Errors for these energy performance experimental measures were calculated from the accuracy of measurement sensors following standard linear propagation of errors, being estimated in the range between 15% and 20% [2]. Taking into account these experimental uncertainties in principle it can be concluded that in most cases experimental values and simulation estimations are compatible. Nevertheless, the tendencies observed are very similar when comparing both curves, pointing to the fact that there may be systematic discrepancies.

FIGURE 5

In figure 5, the energy performance factor calculated for each day of operation (daily performance factor, DPF) is shown. Dash-dotted lines correspond to the values obtained from the simulation and solid lines correspond to experimental measured values. The quantity shown in the vertical axis is the daily (short term) performance factor, which corresponds to the performance factor defined in equation (5) for an integration period starting at the beginning of each day and finishing at the end of it. The quantity shown in the horizontal axis corresponds to the considered day. Errors for these daily performances were also calculated from the accuracy of measurement sensors following standard linear propagation of errors [2]. In figure 5 two dash-dot-dotted lines are included to indicate the error bandwidth corresponding to each value of daily performance factor, for each date, the distance between the top dash-dot-dotted line and the bottom dash-dot-dotted line represents the error bandwidth of the DPF value for this date. Estimated and experimental data show a similar behaviour to the one observed for the accumulated value of the performance factor presented in figure 4. The main difference is a higher discrepancy between both values when the heating or cooling demand is very low (close to the dates in which the system changes operation from heating to cooling mode).

To understand the origin of the discrepancies between simulated and experimental data, a sensitivity analysis of the energy performance simulation results to changes of its input parameter values was performed. Important design parameters to take into account here are those describing the soil. Nevertheless, the simulated energy performance results are rather insensitive to changes in the main ground parameters such as conductivity and diffusivity. The reason for this insensitivity lays in the design of the GeoCool borehole heat exchanger itself, which was done taking into account a large uncertainty in the available soil conductivity estimations. Regarding the parameters describing the heat pump, a high sensitivity of the simulated results to the heat pump nominal coefficient of performance is observed.

As the heat pump coefficients used in the simulation are well established steady test data, the idea was at hand that the differences between experimental and simulated data could be explained as degradation of the heat pump performance for being used at partial load, i.e., at conditions where the capacity is higher than the actual thermal demand. To take into account this effect in the simulation outputs a suitable parameterization for the instantaneous coefficient of performance degradation factor (CDF) is needed. The ARI standard [43] suggests a generalized use of the following equation to calculate CDF:

$$CDF = 1 - \alpha(1 - PLR) \tag{16}$$

Being PLR the partial load ratio and α the coefficient of performance degradation parameter. PLR variable is defined as the quotient between the actual thermal demand and the capacity at current conditions (maximum energy which could be supplied in the case of continuous working at full capacity). And α parameter is a degradation coefficient characterizing the diminishing of the heat pump coefficient of performance for being used at partial load. This parameter is specified by the heat pump manufacturer. In case of lack of information about it, ARI standard [43] suggests to use α=0.25as a default value. As shown in [44], this simple correlation between partial load ratios and heat pump performance degradation is sometimes an unacceptable approximation. Nevertheless, as there is no available parameterization for the instantaneous coefficient of performance degradation factor of the GeoCool water to water heat pump, the ARI suggestion was followed to make an estimation of the impact of heat pump performance degradation in the energy performance results.

To include this effect in the simulation results, an average degradation coefficient is estimated just integrating equation (16) for a time period. The hourly results for the PLR (calculated as the ratio between the hourly thermal loads and the heat pump capacity at current conditions for the same time period) and the default value α=0.25are used to calculate it. Then, energy performances estimated for a time period are multiplied by the average degradation coefficient corresponding to the same period. Accumulated energy performances (Figure 4) and daily energy performances (Figure 5) have been corrected following this procedure.

Dotted lines in figure 4 correspond to the accumulated energy performance values obtained from the simulation multiplied by the corresponding average degradation coefficient. With this correction included simulation results match very well with the tendencies described by the experimental measured values, being the differences between both always smaller than 3% for data belonging to the heating season and always smaller than 7% for data belonging to the cooling season. Dotted lines in figure 5 correspond to the daily energy performance values obtained from the simulation for this quantity multiplied by the corresponding average degradation coefficient. These corrected values also match well with experimental results, giving support to the hypothesis that the discrepancies between experimental measures of energy performance and simulation predictions are mainly due to the degradation of the heat pump performance for being used at partial load.

It is worth to stress that with the quite simple correlation between partial load ratios and heat pump performance degradation described by equation (16) and using the default value α=0.25 (not corresponding to any experimental characterization of GeoCool water to water heat pump) discrepancies between energy performances experimental measures and PLR-corrected numerical predictions are substantially reduced to values around 5%. Experimental measures and PLR-corrected numerical predictions also show very similar tendencies confirming the idea that the partial load ratio is the variable needed to improve the agreement between both. To even reduce these differences an experimental characterization of GeoCool water to water heat pump working at partial loads is needed. Looking at figure 4, where accumulated energy performances are included, it can be guessed that equation (16) is a reasonable description of GeoCool heap pump performance degradation, being α=0.25 a good choice in heating mode while in cooling mode a slightly lower value will be a better choice. Looking at figure 5, where daily energy performances are included -more sensitive to variations in thermal demands-, it can be seen that the linear correlation shown in equation (16) is not so good for low values of partial load ratios, corresponding to dates in which the system changes operation from heating mode to cooling mode. A different description of GeoCool heat pump performance degradation should be given for low partial loads ratios.

Finally, notice that the previous estimation can be performed just because energy performances are not sensible to changes in ground thermal properties. A large influence of ground properties in energy performance predictions would have made it difficult to distinguish which effect is the responsible of the discrepancies between experimental measures and numerical predictions.

6. CONCLUSIONS

In this article a comparison between experimental and simulation results for the energy performance of a ground coupled heat pump system in mixed climate conditions is presented. The main conclusion of this work is that simulation results solely based on nominal heat pump capacities and performances substantially overestimate the measured energy performance of the ground coupled system by a percentage between 15% and 20%. The relevance of this comparison relies in the fact that the performed simulation is based on a standard design procedure for ground coupled heat pump systems, using as input values parameters that are usually available for the engineer in charge of the system design.

A sensitivity analysis of the energy performance simulation predictions to changes on the input parameters has been performed. This analysis shows that the heat pump coefficient of performance is the parameter that highly affects energy performance simulation predictions, pointing to the idea that discrepancies between experimental results and simulation outputs are mainly due to degradation of heat pump performance for being used at partial load. An estimation of the impact of this effect using a simple correlation between heat pump performance degradation and partial load ratio reduce the discrepancies to values around 5%.

A better description of GeoCool heat pump, based on an experimental characterization, will be needed to diminish the discrepancies to even smaller values. Furthermore, if other quantities, like ground heat transfer or instantaneous fluid temperatures, are to be predicted by the simulation tool and compared with experimental measurements, also a better model describing the borehole heat exchanger will be needed. Nevertheless, these further refinements are out of the scope of this work, whose objective was evaluating the goodness of a standard design procedure based on TRNSYS to predict the long term energy performance of a monitored ground coupled heat pump system.

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FIGURE CAPTIONS

Figure 1. GeoCool schematic diagram. The air to water heat pump and the ground source heat pump are linked in parallel to the internal hydraulic group that transfers the energy to fan-coils.

Figure 2. TRNSYS model scheme used to simulate GeoCool plant.

Figure 3. Hourly thermal loads calculated from the experimental data measured during one year of operation of GeoCool plant. Positive values are associated to heating demand and negative values to cooling demand.

Figure 4. Comparison between experimental measurements (solid lines) and numerical predictions (dash-dotted lines and dotted lines) for the accumulated performance factor of GeoCool plant.

Figure 5. Comparison between experimental measurements (solid lines) and numerical predictions (dash-dotted lines and dotted lines) for the daily performance factor of GeoCool plant. Errors for the experimental values of daily performance factor are represented by the distance between the top dash-dot-dotted line and the bottom dash-dot-dotted line (upper/lower error bound).

Nomenclature

CDF	Coefficient of performance degradation factor
COP	Coefficient of performance
Cp	Specific heat at constant pressure
cp	Circulation pump
DPF	Daily performance factor
f	Circulation pump variable control function
$\dot{h}$	Enthalpy flow
$\dot{m}$	Mass flow
PF	Performance factor
PLR	Partial load ratio
Q	Thermal loads
$\dot{Q}$	Instantaneous thermal loads
SPF	Seasonal performance factor
T	Temperature
W	Energy consumption
$\dot{W}$	Power consumption

Greek letters

α	Coefficient of performance degradation parameter

Subscripts

in	Input
max	Maximum
out	Output
ww	Water to water heat pump

Borehole heat exchanger parameters	**Value**
Number of boreholes	6
Borehole depth	50 m
Borehole radius	0.120 m
Outer radius of u-tube pipe	0.016 m
Inner radius of u-tube pipe	0.0131 m
Center to center half distance	0.035 m
Fill thermal conductivity	2.0 W/m K
Pipe thermal conductivity	0.42 W/m K
Ground parameters	**Value**
Undisturbed ground temperature	291.15 K
Storage thermal conductivity	1.43 W/m K
Storage Heat Capacity	2400 kJ/m^3/K

Table 1. Description parameters of the ground and of the borehole heat exchanger.

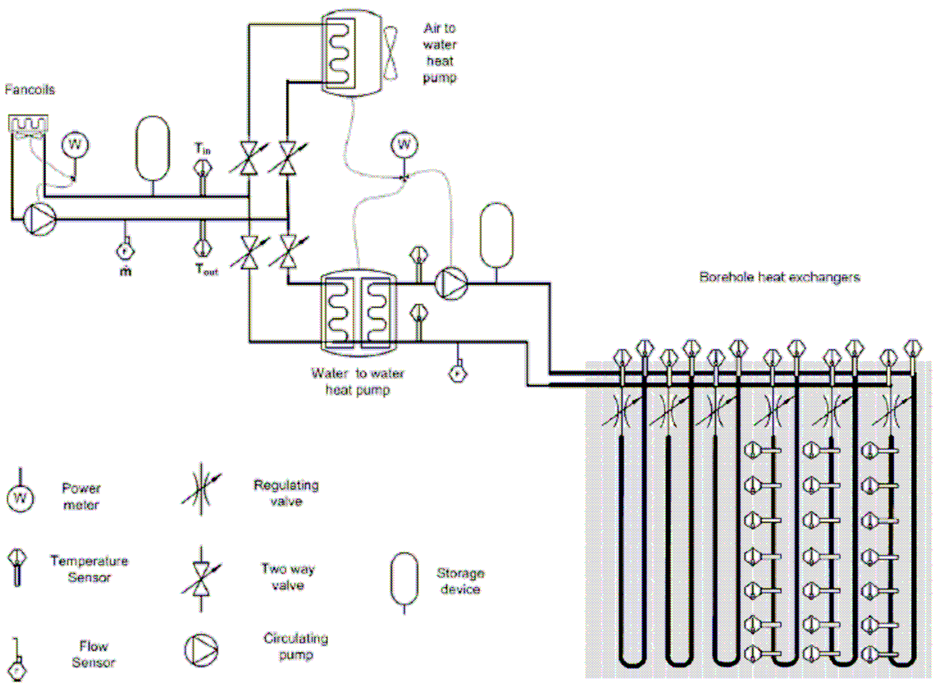
Air to water heat pump
Fancoils
W
T_in
W
ṁ
T_out
Borehole heat exchangers
Water to water heat pump
W
Power meter
Regulating valve
Temperature Sensor
Two way valve
Storage device
Flow Sensor
Circulating pump

FIGURE 1

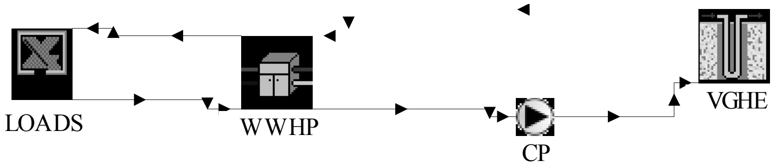

FIGURE 2

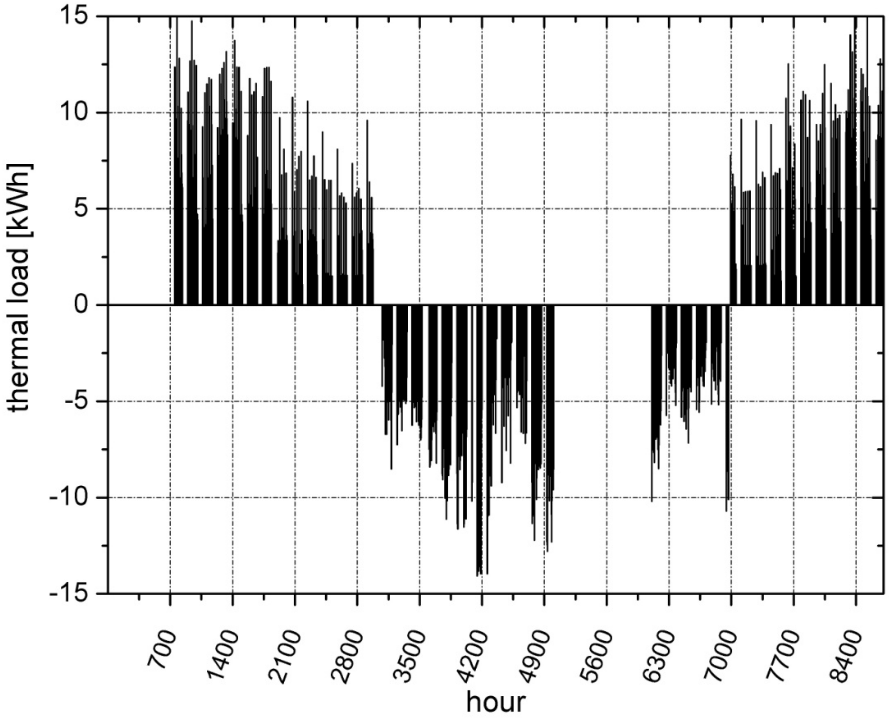
15
10
5
0
-5
-10
-15
thermal load [kWh]
700
1400
2100
2800
3500
4200
4900
5600
6300
7000
7700
8400
hour

FIGURE 3

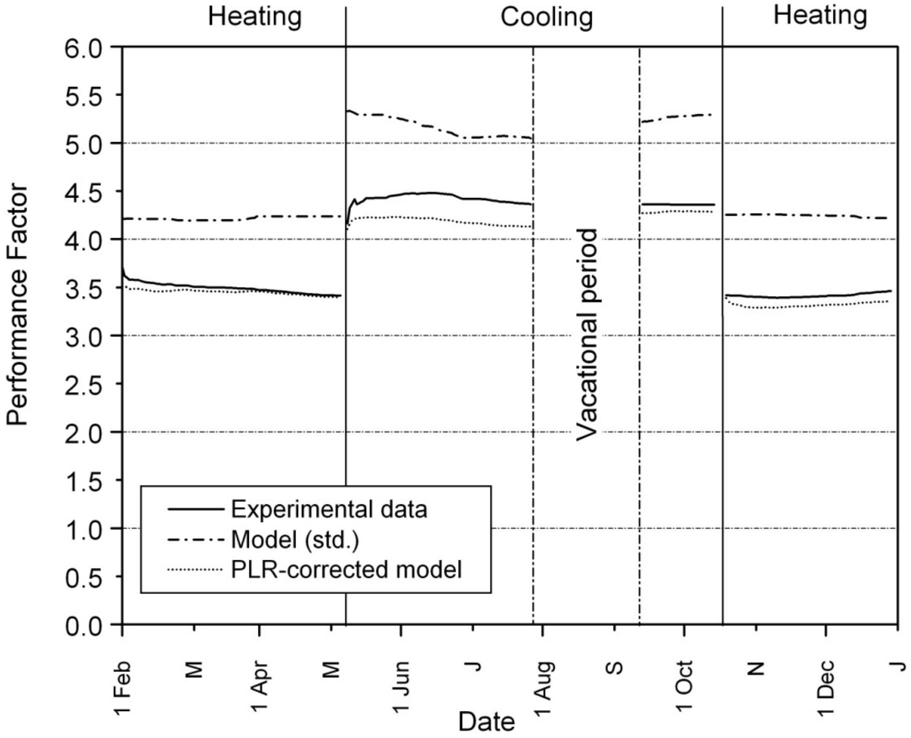
Heating
Cooling
Heating
Performance Factor
6.0
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
Vacational period
Experimental data
Model (std.)
PLR-corrected model
1 Feb
M
1 Apr
M
1 Jun
J
1 Aug
S
1 Oct
N
1 Dec
J
Date

FIGURE 4

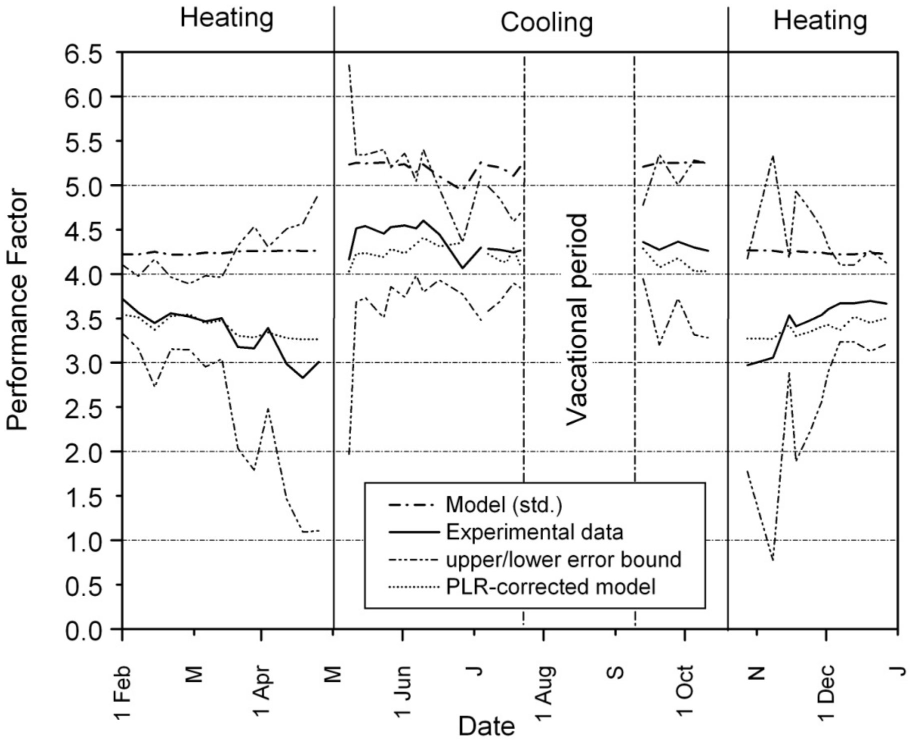
Heating
Cooling
Heating
Performance Factor
6.5
6.0
5.5
5.0
4.5
4.0
3.5
3.0
2.5
2.0
1.5
1.0
0.5
0.0
Vacational period
Model (std.)
Experimental data
upper/lower error bound
PLR-corrected model
1 Feb
M
1 Apr
M
1 Jun
J
1 Aug
S
1 Oct
N
1 Dec
J
Date

FIGURE 5