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Ground Coupled Heat Pumps in Mixed Climate Areas: Design, Characterization and Optimization

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1. Introduction

This contribution reviews the research work developed by the authors on the design, characterization and optimization of ground coupled heat pumps in mixed climates areas.

The design of a ground coupled heat pump HVAC system starts with the estimation of the thermal loads that the air-conditioned area demands. The capacity of the ground source air-conditioning system is determined from this thermal load estimation. With 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.

Determining ground thermal properties is crucial for an accurate design of the air conditioning system. In situ thermal response tests are carried out to have a measurement at site of ground properties. These tests are based on the Kelvin infinite line source model of heat transfer by thermal conduction. Improvements of this technique can be pursued in different ways. One approach consists in refining the model describing the borehole heat exchanger to include effects not taken into account. Finite length effects can be incorporated in the analysis procedure of test in situ outputs as shown in Bandos et al (2008a), Bandos et al (2008b) and Bandos et al (2009a). A filtering technique of the undesired effect produced in temperature measurements by the ambient temperature can be used to improve the estimating of ground thermal properties (Bandos et al (2009b), Bandos et al (2009c), Bandos et al (2009d)).

Another approach to improve the in situ estimation of ground thermal properties is the development of new devices able to measure relevant quantities for the correct characterization of heat transfer between the fluid and the ground. This characterization could be done if the evolution of the fluid temperature along the heat exchanger is known. A sensor probe including a temperature sensor, an acquisition system, temporary storage and wireless communication has been developed to obtain these measurements (Martos et al (2008), Martos et al (2009), Martos et al (2010a), Martos et al (2010b), Martos et al (2010c)). With this new information it will be possible to infer some properties about the ground structure relevant for the design of the ground coupled system.

Once ground properties are estimated the length and layout of the heat exchanger is determined. After the execution of the ground source air conditioning system, will be desirable to validate the goodness of the design procedure to predict the energy performance of the ground coupled system. For this purpose, the experimental data presented in Urchueguía et al (2006) and Urchueguía et al (2008), characterizing the energy performances of a monitored ground coupled heat pump system, were used to compare with the predictions of a standard design procedure based of nominal heat pump capacities and performances. The achieved result was that this prediction overestimates the energy performances by a percentage between the 15% and the 20% (Magraner et al (2009), Magraner et al (2010a), Magraner et al 2010b)). A sensitivity analysis of the energy performance results to the design input parameters showed that heat pump nominal coefficient of performance was the parameter that mostly affects the prediction. The result of this analysis supports the idea that the differences between experimental results and design predictions, are mainly due to heat pump performance degradation for being used at partial load.

Optimizing the energy performance of a ground coupled air conditioning system can also be faced by managing its operation. Note that, in the standard design of an air conditioning system, the references taken to estimate the heating and cooling capacity of the heat pump to be installed are usually based on the coldest and the warmest day along the year. Therefore, the thermal energy required by the thermal load is under the design point of the air conditioning system during most part of the time. In this context, the development of strategies for the operation of the air conditioning system based on the ground coupled heat pumps, allowing adapting the thermal energy generated by the system with the thermal load is a good way to improve the system energy efficiency while satisfying the thermal comfort. In Pardo et al (2007), Pardo et al (2008) and Pardo et al (2009c), a new management strategy is designed to diminish the consumption of the system while keeping the comfort requirements. In Pardo et al (2009a), Pardo et al (2009b) and Pardo et al (2010), an approach based in combining the ground source system with other production system, and decoupling energy production from energy distribution using a thermal storage device was studied. In both cases substantial energy savings, of the order of the 30%, were achieved.

This chapter is organised as follows. Section 2, *determining ground thermal properties*, presents the results obtained by the authors on the improving of data analysis tools and experimental techniques to measure ground thermal properties. Section 3, *energy performances of a ground coupled heat pump system: experimental characterization*, reviews the results obtained for the performances of GeoCool plant. Section 4, *predictions of ground coupled heat pump energy performances from standard design procedures*, shows a comparison between the energy performances of GeoCool plant and predictions from standard design procedures. Section 5, *optimization of ground coupled heat pump systems*, studies two ways of improving the efficiency of this system. Finally, section 6 summarizes the conclusions.

2. Determining ground thermal properties

A ground coupled heat pump exchanges heat with the ground through a U-tube loop based in the earth. The knowledge of ground thermal properties is crucial for the design of ground coupled heat pump air conditioning systems. Field tests allow performing measurements of effective thermal conductivity in realistic conditions.

In situ tests are based on studying of the thermal response of the borehole heat exchanger (BHE) to a constant heat injection or extraction. The outputs of the TRT are the inlet and

outlet temperature of the heat carrier fluid as a function of time. From these experimental data, and with an appropriate model describing the heat transfer problem between the fluid and the ground, the thermal conductivity of the surroundings is inferred. The infinite line-source (ILS) model is the most widely used method for evaluation of response test data because of its simplicity and speed (Hellström, 1991; Witte et al, 2002).

This section presents improvements on the data analysis procedure and on the measurement technique of a standard in situ test. On the first subject, improving the analysis procedure, the results of a standard test are compared with the predictions from finite line-source (FLS) model for the average ground temperature with finite-length corrections accounting the actual borehole heat exchanger. In addition, this study also addresses the influence of the diurnal temperature on the estimation of the thermal conductivity of the ground, which was pointed repeatedly (Signorelli et al, 2007). And on the improvement of the measurement technique, a new instrument has been developed able to measure the evolution of the fluid temperature along its way through the borehole heat exchanger.

2.1 Improving data analysis tools

Within the infinite line-source (ILS) approach commonly applied for the estimates of thermal response test data, the ground is assumed to be a homogeneous infinite medium characterized by its thermal conductivity. In the surroundings of the borehole the results of the infinite line-source model for sufficiently large time values give:

$$T(r,t) = T_0 - \frac{Q_z}{4\pi\lambda} Ei(-\frac{r^2}{4\alpha t}) \approx \frac{Q_z}{4\pi\lambda} \{\ln\frac{4\alpha t}{r^2} - \gamma + O(\frac{r^2}{4\alpha t})\} + T_0 , \; for \; \frac{4\alpha t}{r^2} \gg 1 \qquad (2.1)$$

The function $Ei(u)$ denotes the exponential integral, γ is Euler´s constant, α is the ground thermal diffusivity, and T_0 is identified with the undisturbed ground temperature. It is usually assumed that the heat is released at a constant rate from the borehole heat exchanger (BHE), in the 'radial' direction orthogonal to it, and is transferred by the mechanism of thermal conductivity.

In the frame of the finite line-source (FLS) model in the semi-infinite region, the approximation of the ground temperature, averaged along the BHE, for the times corresponding to the TRT was proposed by Bandos et al. (2009a). The average ground temperature response for $t_z \gg t \gg t_r = r_b^2/\alpha$ is given by:

$$<T(r,z,t) - T_0> \approx \frac{Q_z}{4\pi\lambda} \{-\gamma - 2\ln\frac{r}{H} + \ln\frac{4t}{t_z} + \frac{3r}{H} - \frac{3}{\sqrt{\pi}}\sqrt{\frac{4t}{t_z}} - \frac{3}{\sqrt{\pi}}\frac{r^2}{H^2}\sqrt{\frac{t_z}{4t}}\}, \qquad (2.2)$$

Where $t_z = H^2/\alpha$ the characteristic axial time scale and H is the borehole depth.

The acquisition system used to monitor the ground thermal response was connected to the borehole heat exchanger by insulated 1-m long tubes. The experiments were performed on a borehole heat exchanger with radius r_b=0.15m and depth H = 25m. Figure 2.1a illustrates correlation between the changes of the measured fluid temperature and the ambient temperature. To provide the constant heat rate (of 1000 W as shown in Figure 2.1b), the difference between temperature of the circulating fluid on the input and output of the ground loop were hold constant as well as the volume flow rate of fluid, G. The test parameters were monitored every 3 minutes by a data logger. Figures 2.1 present the plots of 1418 readings after data averaging over 30 points. The following values of model

parameters in equations (2.1, 2.2) are set in the numerical calculations throughout the section: $\alpha = 1.21\ 10^{-6}\,m^2/s$, $T_0 = 14^oC$.
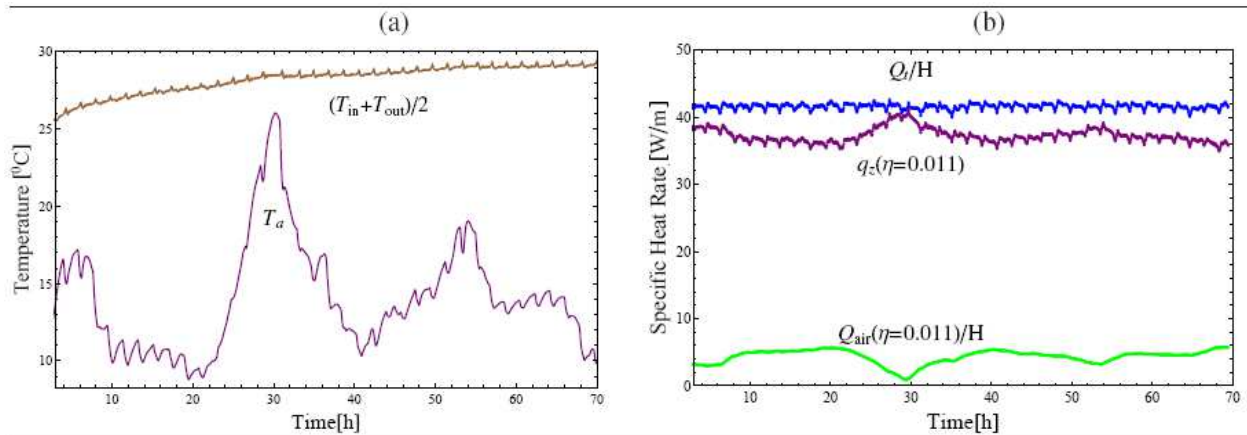

Fig. 2.1. (a) Influence of varying ambient air temperature (purple line) on the average fluid (brown line) temperature in the injection mode. (b) Measured total heat rate Q_t (blue line), variable heat rate $q_z H$ transferred to ground (purple line), Q_{air} calculated with the fitting parameter η = 0.011 (green line) as a function of time.

Although pipes connecting the test rig with the borehole are insulated, undesirable correlation between the air temperature and the mean temperature of fluid is usually observed (as shown in figure 2.1a). The heat exchange between the air and the fluid flowing through the pipes changes the actual heat transferred to the BHE because of the heat dissipated to the ambient. Besides diurnal changes of the temperature of the air, the interior temperatures of the test rig affect the efficiency of the system operation. The total heat rate Q_t can be written as $Q_t = Q_{air} + Hq_z$, where Q_{air} is the heat rate transferred to the ambient air, and $q_z H$ is the actual heat transferred to the ground.

The fluid temperature changes with coordinate along the plastic pipes outside the borehole due to the heat exchange with the ambient air. At the quasi steady-state case, the heat transport by the fluid along the pipes, accompanied by the transverse heat flux to the air, is governed by the convection equations (Hellström, 1991). To account for this climatic influence, the dimensionless parameter η is used thereafter; it appears in the solution of the convection equations for T_{in} (T_{out}), the fluid input (output) temperature at the point of the external pipe on the ground surface level, and can be calculated using physical parameters of the fluid, flow rate of the heat carrier, the thermal resistance for the heat flow from fluid to the air, and the length of the pipe between the point of measurement of the fluid temperature at the test rig and the corresponding point, at which the flow upwards/downwards the subsurface. This parameter influences both the heat rate and mean fluid temperature as:

$$q_z(\eta,t)H = C_f G\left(2T_a(t)\sinh\eta - e^\eta T_{out}^* + e^{-\eta}T_{in}^*\right),$$

$$\overline{T_f}(\eta,t) = \left(T_{out}(t) + T_{in}(t)\right)/2 = T_a(t)(1-\cosh\eta) + \left(e^\eta T_{out}^* + e^{-\eta}T_{in}^*\right)/2 \qquad (2.3)$$

Where $T_{in}*$ ($T_{out}*$) corresponds to the fluid temperature at the point of measurement related to the input (output) of the borehole. Fig.1.b shows that $q_z(\eta;t)$ and $Q_{air}(\eta;t)$ change with

time, but Q_t remains the same by keeping the temperature difference $T_{in} - T_{out}$ constant during the TRT. Since $Q_{air}(\eta;t)$ is subtracted from the fluid outside of where the temperature measurements are made, the actual heat injection rate to the ground is reduced. To estimate the effective thermal conductivity, time-dependence of the ground temperature approximation is compared with time-dependence of the measured fluid temperature. The parameter λ of the ILS model depends on the η through the late time slope k of the experimental $T_f(\ln(t))$ curve and the heat- flux $q_z(\eta;t)$, equation (2.3), as:

$$\lambda = <q_z(\eta;t)>_t / (4\pi k(\eta)) \tag{2.4}$$

Where $\langle...\rangle_t$ denotes time averaging. Here k is the slope of linear approximation in the semi-logarithmic scale, i.e. $\overline{T}_f(t) = k \ln(t) + m$ for the time-dependence of the experimental data. A lower heat injection rate results in a lower estimate for thermal conductivity. Notice that if the connecting pipes were ideally insulated, i.e. $\eta = 0$, neither T_{in} nor T_{out} would depend on the air temperature. The question then arises: how to select parameter η for filtering the air temperature variation effect.

On the pre-processing step of the test analysis, model parameter η can be evaluated by matching the measured fluid data to equation (2.3). The energy rate balance and equation (2.3) set the correspondence between η and the p-th part of the total heat rate transmitted to the air ($Q_{air} = p Q_t$) as:

$$\left(p - 1 + \cosh\eta\right)\left\langle T_{in}^*(t) - T_{out}^*(t)\right\rangle_t = \left\langle T_{out}^*(t) + T_{in}^*(t) - 2T_{air}(t)\right\rangle_t \sinh\eta \tag{2.5}$$

If the heat loss to the ambient is a small part of the total injected heat rate ($p \ll 1$), the series expansion for η in equation (2.3) gives:

$$\eta = p \frac{\left\langle T_{in}^*(t) - T_{out}^*(t)\right\rangle_t}{\left\langle T_{out}^*(t) + T_{in}^*(t) - 2T_a(t)\right\rangle_t} \tag{2.6}$$

Then, the values of η turn out to be small as far as $p \ll 1$. One can use the above equation to select η. There is some arbitrariness involved in the exact quantity of the total measured heat rate transferred to the ground. The idea behind the proposed method of pre-processing TRT data is to use the freedom in choosing η to suppress the influence of the air temperature oscillations, i.e. to select such a value of fitting parameter η that the fluid temperature of outward flow from the borehole $T_{out}(t)$ can be approximated by a sufficiently smooth function of time. In the non ideal test the fluid is exposed to a certain climatic influence when flowing from the test rig to the borehole at the same time as the heat conduction into the ground attenuates $T_{in}(t)$ oscillations when fluid passing through the borehole as shown in figure 2.2a. The effect of the air temperature variation is mitigated if the fitting parameter η is set equal to 0.011 given $p = 0.1$ and below it is applied to the parameter estimation analysis of the results of the test.

On the next stage, when fitting $T_b = T(r_b;t)$ to the experimental data of $\overline{T}_f$, the thermal resistance R_b between the borehole wall and the fluid must be taken into account, $R_b C_f G(T_{in} - T_{out}) = R_b q_z = \overline{T}_f(\eta,t) - T_b(t)$, where C_f is the volumetric heat capacity of fluid. Notice that the borehole temperature $T_b(t)$ is influenced by the measured fluid temperatures and $T_a(t)$ through the $Q_{air}(t)$ as:

$$Q_{air}(t) \approx 2\eta C_f G((T_{in}^* + T_{out}^*)/2 - T_a(t)) \tag{2.7}$$

For small values of η. The parameter R_b is also obtained from the TRT. Below the proposed formulas for subtraction of climatic influence on the TRT are applied to the parameter estimation analysis.

The assessment of the late-time TRT data allows to estimate, and to obtain a extrapolated value for T_0 (Hellström, 1991). The data obtained from the 71-h TRT were evaluated and compared by making use of the ILS and FLS models along with the introduced method of account for the heat transfer to the ambient characterized by η. To find suitable model parameters, equations (2.1) and (2.2) are matched to the heat carrier fluid temperature data by using regression; the average heat rate, $<q_z(t)>_t$, is applied when the heat rate variations $q_z(t)$ are caused by $T_a(t)$ fluctuations as described before. Figure 2.2b compares the estimated values from both the ILS and FLS models on different length of data interval of the same test, for $\eta = 0$ and $\eta = 0.011$.

Figure 2.2b reveals noticeable oscillations of λ caused by small fluid temperature variations due to the air temperature changes (Signorelli et al, 2007). Small changes in the mean fluid temperature curve are significantly amplified for thermal conductivity λ that is inversely proportional to the time derivative of the temperature. Figure 2.2b also demonstrates that these fluctuations are lowered for $\eta \neq 0$, $R_b = 0$ and almost disappear for $\eta \neq 0$, $R_b \neq 0$. Therefore, the developed method successfully removes cyclic distortions caused by the diurnal temperature cycle.

For either procedure of evaluating, when energy loss in the air is absent, the values of thermal conductivity, calculated from both ILS and FLS models are higher than the ones evaluated with the proposed method of decoupling climatic influence from the TRT data; this is the effect of the heat transfer to the ambient.
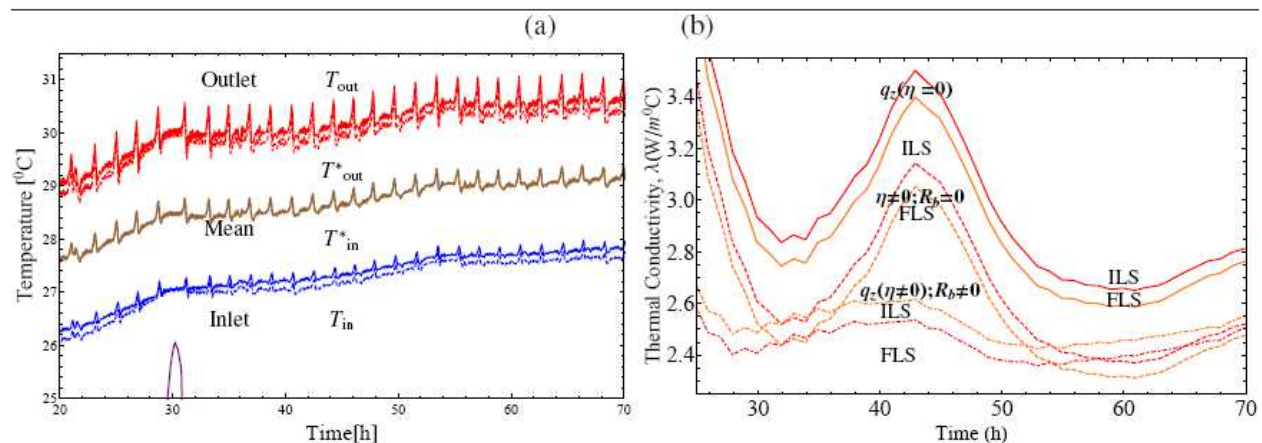

Fig. 2.2. (a) Comparison between measured inlet, T_{in}^*, outlet, T_{out}^*, temperatures (dashed red, blue lines, respectively) and calculated T_{in}; T_{out} with $\eta = 0.011$ (red, blue lines, respectively). Brown line indicates the mean fluid temperature. (b) Comparison between dependence of thermal conductivity from the ILS (red line) and FLS (orange line) model on the time estimation interval for the same test data. Estimates of thermal conductivity based on the ILS and FLS models when the end of the evaluation interval is fixed, while starting time increases: (i) without heat loss to the ambient ($\eta = 0$)- solid lines (ii) with heat loss to the ambient ($\eta = 0.011$): dashed for $R_b = 0$, dot-dashed for $R_b \neq 0$ lines, respectively.

2.2 Improving experimental measurements

To improve the measurement technique, it has been developed a new measurement instrument that allows obtaining the evolution of the fluid temperature along its way through the borehole heat exchanger (Martos et al (2008), Martos et al (2009), Martos et al (2010a), Martos et al (2010b), Martos et al (2010c)). The characteristics of this equipment are the following: small volume and easy to transport; it is very easy to locate temperature sensor in the borehole head measuring only the heat transfer between the pipes and the ground; allow new methodologies for TRT because gives new information for the fluid temperature evolution.

The aim of this instrument is to determine the spatial and temporal behaviour of the fluid temperature along the BHE, so it is necessary to measure the temperature of the fluid flowing through the pipe along the entire length. To do so, miniature temperature probes are inserted to measure the temperature at specified intervals. The instrument has been divided in three parts: Autonomous sensors, Software for recording and analysis, and the Hydraulic system.

Figure 2.3 presents the logic diagram of the instrument; the hydraulic system comprises a water tank, a circulation pump and two valves for the insertion and extraction of the autonomous temperature probes. A laptop runs the program for thermal response test configuration, acquisition and analysis of the values of measured temperature. Finally, a set of small 25 mm diameter balls contains the electronic circuitry of the autonomous temperature probes. Also, a set of sensors monitor several variables during the running of thermal response test, such as the flow of water that circulates, the inlet and outlet water temperature, the temperature of the tank, as well as the pressure in the pipes.

The autonomous sensors are key components of the instrument. They are devices that measure the thermal evolution of an elementary volume of water along the BHE pipe. Its sizes must be as small as possible so they can move easily through the pipes carried by the water flow, and at the same time contain an acquisition system, temporary storage and unloading of temperature data. To achieve these functions and capabilities, a circuit has been designed based on the CC1010 transceiver that allows including it in a sphere with diameter smaller than 25 mm. It has been designed a 4-layer PCBs for mounting all the necessary components. The characteristics of each autonomous sensor are: temperature range: 0-40 °C; resolution temperature :< 0.05 °C; accuracy temperature :< 0.05 °C rank sampling: 0.1-25 s; capacity sampling: 1000 samples.

The mode of operation of the autonomous sensors is as follows: the control system selects an available probe and puts it in the status of test run; transfer the parameters of sampling; insert the probe into the BHE water flow; the probe starts the process of acquiring, storing temperatures at fixed intervals; after the tour, the temperature data are downloaded to the control system; it's going to low-power mode.

The hydraulic subsystem has as many autonomous sensors as necessary to ensure that they are inserted to the programmed rate. The key factor of success in using these autonomous probes lies in the duration of the power supply. Each probe carries a battery type button 3 volts and 230 mAh, which guarantees the proper functioning of the probes during more than a year for a duty cycle which they perform. The hydraulic circuit comprises a water tank, as buffer for the thermal fluid, an electronically controlled circulation pump, a flow meter and two valves, one for inserting probes and another for their extraction. The water temperature can be set through an electric heater controlled by the program that runs on the PC, which also controls the flow of water that is injected into the pipe of BHE. The insertion of the probes is performed with selected time intervals in terms of realizing the TRT,

controlled by the PC. When extracted, the probe is situated at the point of data discharge and, once it is completed, the data contained in the probe is deleted and is prepared for next insertion.

To verify the operation of the instrument a laboratory installation has been built to simulate a borehole heat exchanger. A 5 meters U-pipe has been arranged in spiral layout to perform two tests: transit time verification and response time of the Pt100 sensor. A key point to the proper location on the pipe is ensuring that the sensor is carried by the water flow at the same speed. If the density of the sphere that constitutes the probe is close to the density of the thermal fluid, will be carried both in vertical as horizontal configurations. To verify this, a set of measures of transit time has been completed with a set of spheres throughout the interior of a pipe with a length of 10 m. The result was that this is a technique with small error (smaller than 2%) allowing to accurately deducing the sensor position. Another relevant point is the response time of the element used as Pt100 temperature sensor. This time has to be precisely known to estimate the accuracy of the sensor measurement with the actual value of the temperature at the measuring point. This response time has been characterized in laboratory, being 0.5 s the time needed to reach 66% of the actual temperature and 1.5 s to reach 90%. These values for the response time allow obtaining a signal that appropriately filtered reproduces the temperature evolution of the fluid along its way through the borehole heat exchanger.
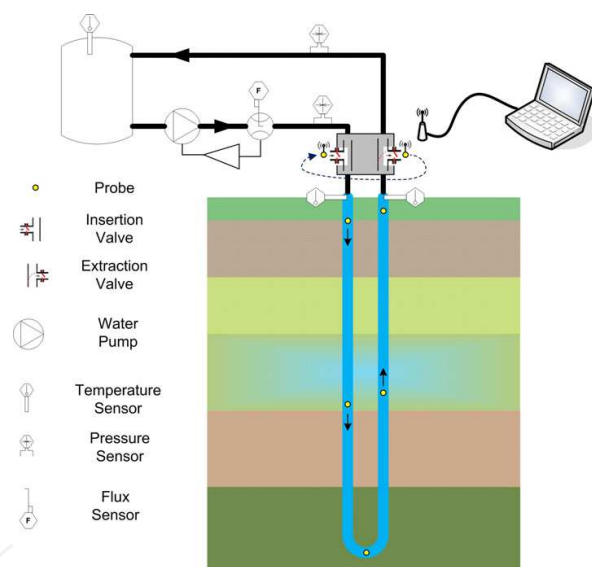

Fig. 2.3. Logic diagram of the instrument

3. Energy performances of a Ground Coupled Heat Pump: experimental characterization

This section presents the energy performance characterization of a monitored ground coupled heat pump system (Urchueguía et al (2006), Urchueguía et al (2008)). This plant 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 m². 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 heat pump working alternately (Figure 3.1).

The ground coupled 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 (GeoCool 2006), 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.

A network of sensors allows monitoring the most relevant parameters of these air conditioning systems (Figure 3.1). These sensors measure temperature, mass flow and power consumption. The temperature sensors are four wire PT100 with accuracy ±0.1 °C. 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 3.1 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.
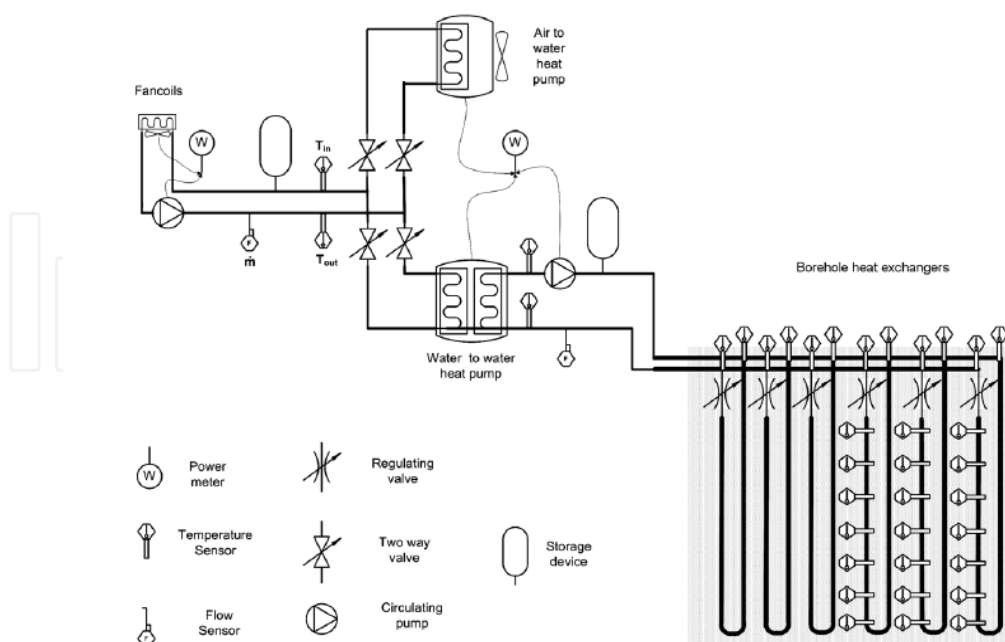

Fig. 3.1. GeoCool schematic diagram. The air to water heat pump and the ground source heat pump are linked in parallel to the internal hydraulic group that transfers the energy to fan-coils.

The system energy efficiency is calculated from the power consumption readings and the values of the internal thermal loads calculated from experimental measurements. Instantaneous thermals loads are obtained by means of the following expression:

$$\dot{Q}(t) = \dot{h}_{out}(t) - \dot{h}_{in}(t) \tag{3.1}$$

Where:

$$\dot{h}_{in}(t) = \dot{m}CpT_{in}(t); \dot{h}_{out}(t) = \dot{m}CpT_{out}(t) \tag{3.2}$$

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 (3.1). It represents the cooling or heating load demanded by the building during the time period Δt starting at T_0 time.

$$Q = \int_{T_0}^{T_0 + \Delta t} \dot{Q}(t)dt \tag{3.3}$$

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 3.1, corresponding to the consumption of the water to water heat pump, $\dot{W}_{ww}$, plus the consumption of the circulation pump, $\dot{W}_{cp}$.

$$W = \int_{T_0}^{T_0 + \Delta t} \dot{W}(t)dt \; ; \dot{W}(t) = \dot{W}_{ww}(t) + \dot{W}_{cp}(t) \tag{3.4}$$

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:

$$PF = \frac{Q}{W} \tag{3.5}$$

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).
From the performance factor values it is possible to estimate the energy savings that may be obtained when switching to GCHP technology assuming that the heating/cooling load are the same for both systems. Under this hypothesis, the energy savings can be evaluated as:

$$\frac{W_{aw} - W_{ww}}{W_{aw}} \times 100 = \left(1 - \frac{SPF_{aw}}{SPF_{ww}}\right) \times 100 \tag{3.6}$$

Figures 3.2 present the results of the seasonal and daily evolution the performance factor. The value of the performance factor at the end of each season, defined as the seasonal performance factor (SPF), gives an estimation of the total energy performance. As stated above, from this quantity it can be estimated the energy savings that may be obtained when

switching to GCHP technology. The performance factor evolution for both AC systems and both operating modes, heating and cooling, is shown in figures 3.2. These numbers are calculated as stated previously, being T_0 the starting point of each season and δT the period of time covered from this starting point to the day considered. The top solid line corresponds to the ground source heat pump system and the bottom solid line to the air source heat pump system. At the end of the heating season the value for the performance factor was SPF=3.5±0.6 for the geothermal system and SPF=2.0±0.3 for the conventional system. This result represents an averaged efficiency improvement of 73% in heating mode. At the end of the cooling season the final value of the performance factor was SPF=4.3±0.6 for the geothermal system and SPF=2.7±0.4 for the conventional system, with a cooling efficiency improvement of 60%. The conclusion achieved was that the system based on ground coupled heat exchangers uses 43±7% less electric energy than the conventional one for the same heating requirements. In cooling mode, the ground coupled heat exchanger based system requires an average of 37±18% less energy than the conventional one. These results show that the use of the ground as a heat source allows achieving considerable energy savings in heating mode, and its use as a heat sink also allows to achieve important energy savings in cooling mode.
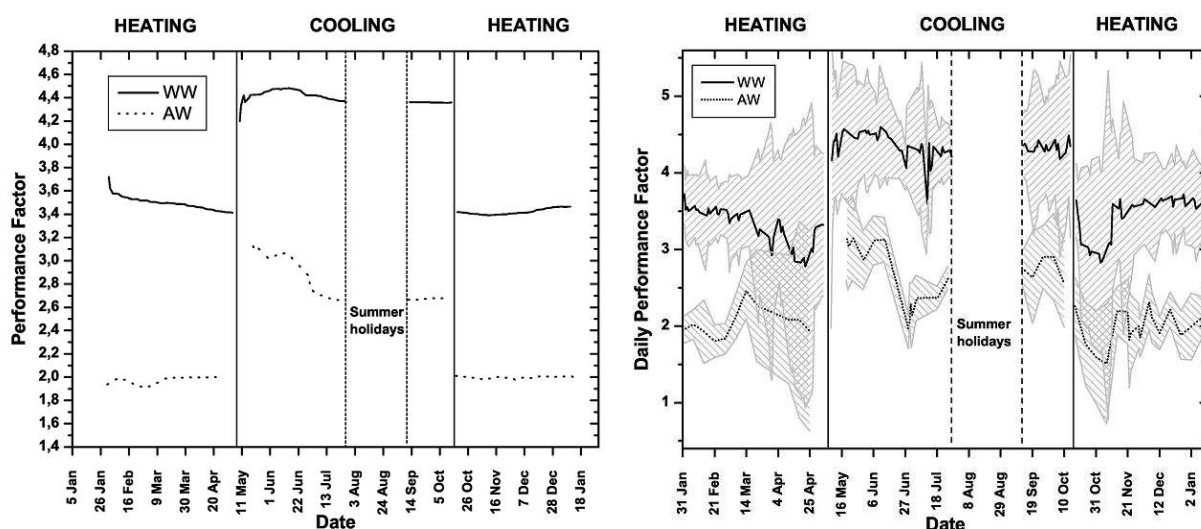

Fig. 3.2. Performance Factor evolution (left figure) for both AC systems and both operating modes, heating and cooling. The top solid lines correspond to the water to water system (WW); the bottom broken lines correspond to the air to water system (AW). Daily Performance Factor is shown on the right figure for both AC systems, and for both operating modes. The top solid lines correspond to the water to water system (WW); the bottom broken lines correspond to the air to water system (AW). Each value for Daily performance Factor is shown together with its error bandwidth.

Figures 3.3 present the results of the monthly electrical consumption of both AC systems. The contribution corresponding to the energy transport components (fan coils and internal circulation pump) are separated from the energy production ones (water to water heat pump and external circulation pump for the geothermal system, and air to water heat pump for the conventional system) in order to evaluate the impact of these secondary components in the energy balance. From this figures it can be seen that in the case of the geothermal system the consumption of the secondary components exceeds 30% of the total

consumption. In heating mode and during warmer periods, when the system use is low, their contribution to the total electrical consumption increases to 50%. For the air to water heat pump system these percentages are lower due to the higher electrical consumption of the heat pump itself, but in any case they are larger than 20%. This analysis points to the need of more intelligent control strategies to attenuate the energy demand of the secondary components.
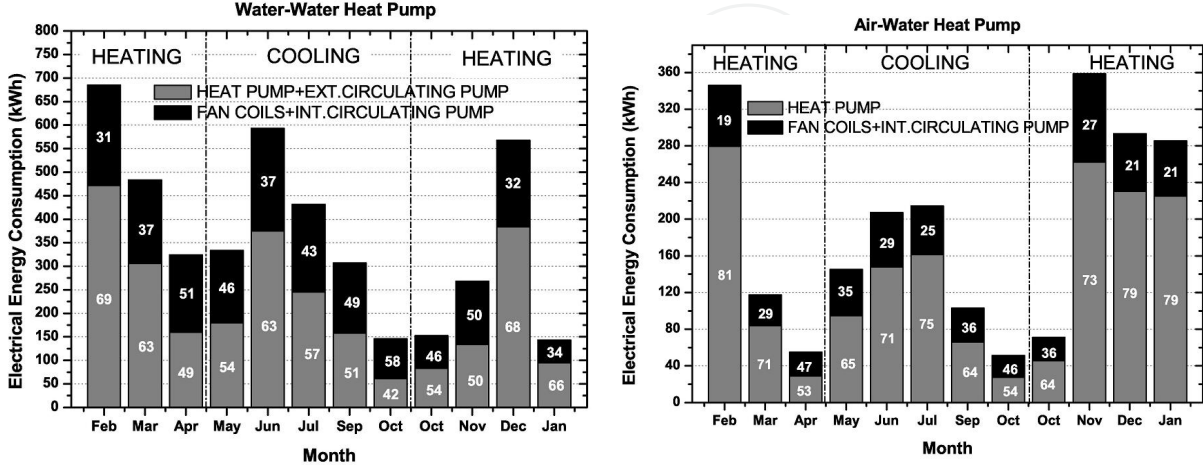

Fig. 3.3. Electrical energy consumption for the water-water system (left figure) and for the air-water system (right figure).

4. Predictions of Ground Coupled Heat Pump energy performances from standard design procedures

This section deals with the comparison of energy performances obtained from ground coupled heat pump design methodology based in TRNSYS with the experimental results just presented in previous section (Magraner et al (2009), (2010a), (2010b)). TRNSYS 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).

Figure 4.1 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.
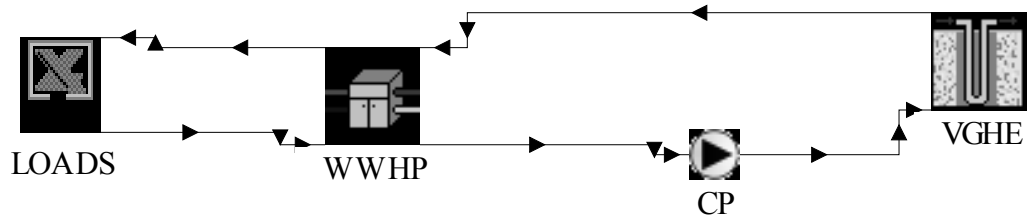

Fig. 4.1. TRNSYS model scheme used to simulate GeoCool plant.

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 (GeoCool (2006))).

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. 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.

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' (Hellström (1991)). 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 4.1. 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.

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.

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/m³/K

Table 4.1. Description parameters of the ground and of the borehole heat exchanger.

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. The measured energy performances and the predicted ones are compared in figures 4.2.

In figures 4.2, the accumulated and daily energy performances for both seasons are depicted. In both cases 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 the corresponding integration period. From these figures it can be seen that the simulation outputs overestimate the experimental measures by a percentage between 15% and 20%. 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. The main difference between both figures 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).
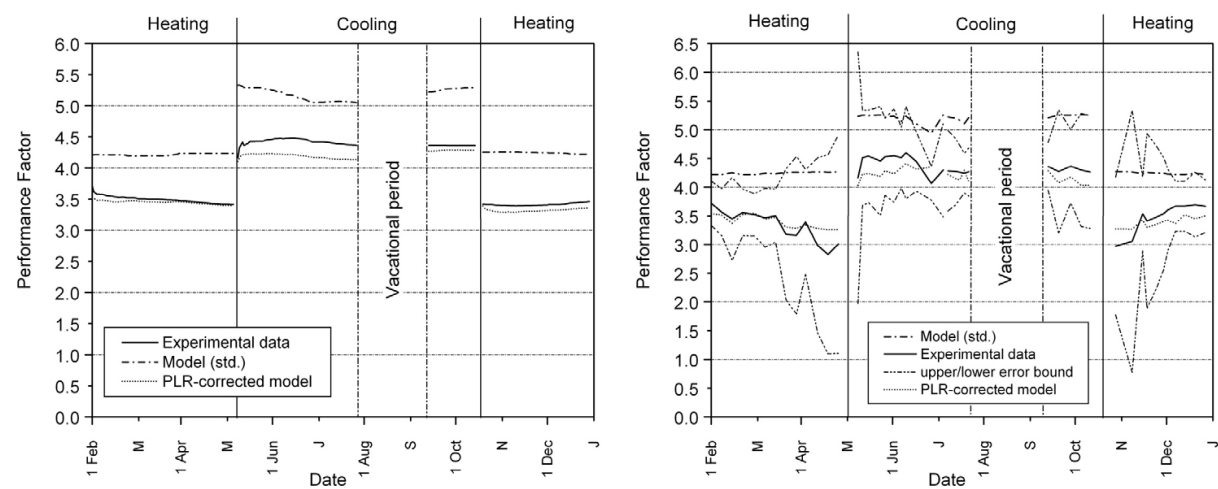

Fig. 4.2. On the left, comparison between experimental measurements (solid lines) and numerical predictions (dash-dotted lines and dotted lines) for the accumulated performance factor of GeoCool plant. On the right, comparison between experimental measurements (solid lines) and numerical predictions (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).

The discrepancies between simulated and experimental data have been investigated with a sensitivity analysis of the energy performance simulation results to changes of its input parameter values. This analysis showed that the simulated energy performance results are rather insensitive to changes in the main ground parameters such as conductivity and diffusivity. Regarding the heat pump description, a high sensitivity of the simulated results to the heat pump nominal coefficient of performance is observed. The differences between experimental and simulated data could be explained as degradation of the heat pump performance for being used at partial load. ARI recommendations have been followed to

make an estimation of the impact of heat pump performance degradation in the energy performance results, and an average degradation coefficient is estimated. Energy performances estimated for a time period are multiplied by the average degradation coefficient corresponding to the same period. Accumulated energy performances and daily energy performances have been corrected following this procedure. Dotted lines in figures 3.2 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.

5. Optimization of Ground Coupled Heat Pumps systems

In the standard design of an air conditioning system, the references taken to estimate the heating and cooling capacity of the heat pump to be installed are usually based on the coldest and the warmest day along the year. Therefore, the thermal energy required by the thermal load is under the design point of the air conditioning system during most part of the time. In this context, the development of strategies for the system based on the ground coupled heat pumps, allowing adapting the thermal energy generated by the system with the thermal load is a good way to improve the system energy efficiency while satisfying the thermal comfort.

In subsection 5.1 it is studied the efficiency improvements that a new management strategy can produce in a HVAC system composed by a ground coupled heat pump and a central fan coil linked to a standard office space (Pardo et al (2007), Pardo et al (2008) and Pardo et al (2009c)). In this new management strategy, the air mass flow in the fan, the water mass flow in the internal and external hydraulic system and the set point temperature in the heat pump, usually fixed in conventional strategies, have the possibility of a continuous regulation that allows us to design a more efficient way to achieve the desired thermal comfort. This new management strategy is based on five capacity levels developed from the total electrical power equation of the HVAC system. In our particular case, this equation indicates that to achieve energy savings is desirable to work with low water flows. For this reason, the new management strategy tries to achieve a steady state in which the water mass flow is maintained as low as possible, using first all other possibilities to supply energy.

In subsection 5.2 the efficiency improvements which can be achieved by combining a ground coupled heat pump system with other HVAC system, such as an air to water heat pump is studied. In addition, the impact of incorporating a thermal storage device in an air conditioning system to decouple energy generation from energy distribution is investigated (Pardo et al (2009a), Pardo et al (2009b) and Pardo et al (2010)). This possibility produces two important advantages, it allows a size reduction of the heat pump, and diminishes the effects of the thermal load peaks generating thermal energy when the environmental conditions are more favourable. The purpose of this work is to evaluate the electrical energy consumption and the energy efficiency of several air conditioning layouts in a cooling dominated building, implementing both ideas together. The procedure to evaluate the energy efficiency of the designed HVAC system is as follows. First step in the procedure is the evaluation of the electrical energy consumptions of the air conditioning system when is driven by a pure air to water heat pump (AWHP) system or by a pure ground coupled heat

pump (GCHP) system. These values are used as a reference for comparison with the consumptions of the implemented layouts. Then, we present an air conditioning configuration composed by a GCHP combined with an AWHP to study the behaviour of both systems working together. Afterwards, we combine a stratified water tank as thermal storage device with a GCHP or with an AWHP to study the behaviour when energy generation and distribution is decoupled. Finally, we present a hybrid configuration which combines both heat pumps and a stratified water tank and we evaluate the efficiency improvement of this combined alternative.

5.1 Efficiency improvement from energy management

An office space in a cooling dominated area is modelled to evaluate in these conditions the energy performance of the ground coupled heat pump HVAC system when it is managed by the new management strategy and by a conventional one. The thermal comfort criterion which has to be satisfied is the Predicted Mean Vote index (PMV). The annual electrical energy consumption of the HVAC system when it is managed by our new management strategy and by a conventional one are compared. For both systems, two aspects are fixed: the working period and the heating and cooling seasons. The working period in the office is fixed from 9:00 to 18:00, which is the period when is switched on the air conditioning system. The heating season is considered from November to December and from January to March and the cooling season from April to October.

The objective of the new management strategy is to improve the energy efficiency of the air conditioning system by adapting properly its generated energy to the actual thermal demand in the office. To maintain neutral comfort conditions in the office the PMV value should be zero in the Thermal Comfort Scale. The deviation of the PMV from zero indicates a variation of the thermal demand in the office area. Notice that, to satisfy this thermal demand, there are several configurations of the management variables which are suitable to compensate this PMV deviation. Our management strategy is based on the choice of a particular configuration of these variables. This choice classifies the air conditioning system capacity in five capacity levels given by the steady state value of the management variables. In figure 5.1, we show a diagram illustrating this classification. We now describe each capacity level.

- First capacity level; in steady state conditions the fan capacity is between the 0% and the 50% of its maximum capacity and the other devices are switched off. In this level the blown air by the fan modify the convective factor and homogenize the temperature in the office to achieve the thermal comfort.
- Second capacity level; the fan, the hydraulic pumps and the heat pump are switched on. In steady state conditions the air blown by the fan is fixed to the 50% of its maximum capacity, the water mass flows of the internal and external hydraulic pumps are between the 0% and the 10% of its maximum allowed value, and the set point temperature is 0%, which indicates the lowest or highest set point temperature for heating or cooling mode respectively.
- Third capacity level; in steady state conditions the air blown by the fan is fixed to the 50% of its maximum capacity, the water mass flows of the internal and external hydraulic pumps are fixed to the 10% of its maximum allowed value and the set point temperature is between 0% and 100%, meaning that the set point temperature can be any value of its range in heating or cooling mode.

- Fourth capacity level; in steady state conditions the air blown by the fan is fixed to the 50% of its maximum capacity, the water mass flows of the internal and external hydraulic pumps are between the 10% and the 100% of its maximum allowed value and the set point temperature is 100% which indicates the highest or lowest set point temperature for heating or cooling mode respectively.
- Fifth capacity level; in steady state conditions the fan capacity is between the 50% and the 100% of its maximum capacity, the water mass flows of the internal and external hydraulic pumps are fixed to the 100% of its maximum allowed value and the set point temperature is 100%.

If the energy supplied by the air conditioning system when all the active devices are given the 100% of its capacities is not enough to maintain the thermal comfort conditions, the PMV diverts from zero and the thermal demand is not satisfied.

The choice of these five capacity levels is based on a study of the system energy consumption as a function of the operational variables. The conclusion of this study was that to achieve energy savings is desirable to work with low water flows in the air conditioning system. For this reason, the new management strategy tries to achieve a steady state in which the water mass flow is maintained at the lower level that guarantees the production of enough thermal energy to satisfy the thermal demand. A suitable way to implement this new management strategy is through a cascade control structure because allows to quickly achieve the thermal comfort state and, after this, drive the management variables to the steady state of the capacity levels described before without comfort penalties.
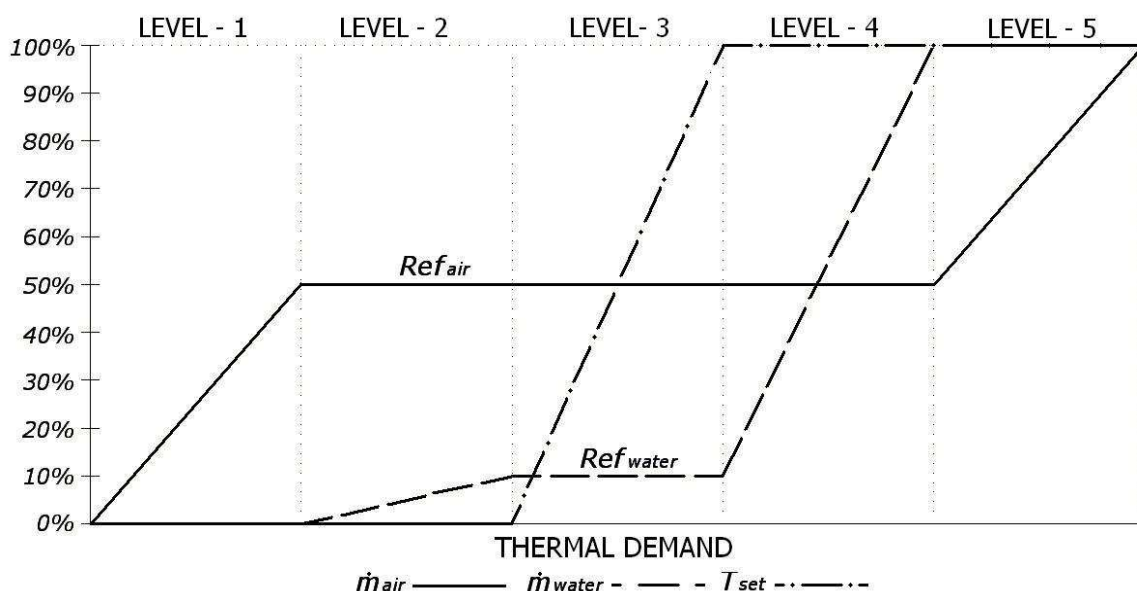

Fig. 5.1. Diagram illustrating the classification in capacity levels of the system capacity given by the steady state values of the management variables. For a given thermal demand there is a unique choice for the operational point of the air conditioning system.

The conventional strategy uses on/off regulators to manage the air conditioning system. This kind of regulators has only two possible operation points. They give its maximum capacity when switched on and nothing when switched off. The on/off regulators are installed in active elements: the heat pump, the electric motor of the fan coil and the internal

and external water pumps. In this strategy, three aspects are important. First, the difference between the set point temperature and the inlet temperature of fluid in the heat pump from the office area indicates the connection or disconnection of the generator energy system, composed by the heat pump, the external water pump and the ground heat exchanger. Therefore, the internal water pump is always switched on to have a measure of this difference during the working period. Second, to maintain the comfort conditions, the PMV is located in a comfort band between 0.5 and -0.5, in order to try to avoid an excessive number of connections and disconnections of the system which, in an actual situation, could damage the actuators of the fan coil. Finally, the set point temperature in the heat pump is constant, for the heating season is fixed to 45°C and for the cooling season to 7°C.

In heating mode the conventional management strategy works as follows. When the value of the PMV variable is below the lower limit of the comfort band, PMV equal to -0.5, the electric motor of the fan is switched on. Then, the air goes through the coil, where a constant water mass flow is pumped from the internal circulation pump. This heat exchange produces a variation of the temperature of the water in the internal circuit which is detected by the heat pump; immediately this device and the external water pump are switched on to provide the necessary energy to fix the value of the PMV to the upper limit of the comfort band, PMV equal to 0.5. After achieving this value, the heat pump, the external water pump and the electric motor of the fan are switched off. In cooling mode the conventional management strategy is the same as the previous one except that the references of the comfort band are inverted. The electric motor of the fan is connected when the PMV is above the upper limit, PMV equal to 0.5, and the air conditioning system stops supplying energy when the PMV arrives to -0.5.
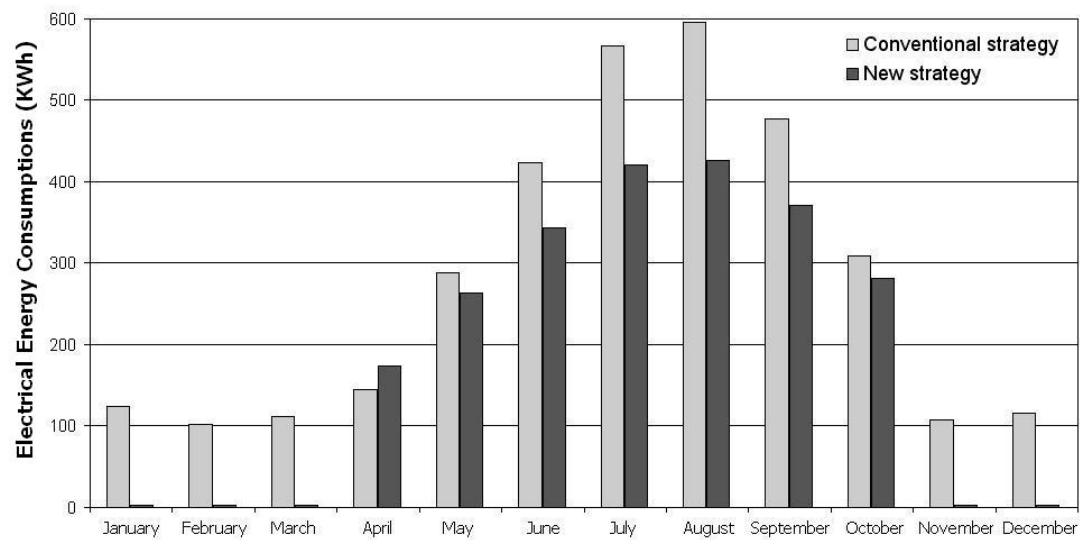

Fig. 5.2. Electrical energy consumptions for both management strategies.

The energy consumption of the air conditioning system for both management strategies are now presented. In the simulation conditions, the weather database employed models the Mediterranean coast weather, which is characterized to have hot summers and warm winters and the working period coincides when the external ambient temperature and the solar radiation is the highest throughout the day. The combination of these two factors

produces that the energy demand in cooling mode is much higher than in heating mode. In figure 5.2 the monthly electrical energy consumptions of the air conditioning system for the two management strategies are presented. From this figure we can see that in all months, except April, the system consumes less electrical energy when it is managed by the new management strategy.

In winter season, our simulation results show that the influence of the external temperature and the solar radiation in the office area during the working period is enough to maintain the thermal comfort in it during most days of the winter season, therefore the energy provided by the air conditioning system is very low. A remarkable difference between both management strategies exists because the conventional strategy employs the temperature change of the water in the internal hydraulic circuit to communicate the energy demand in the office room to the generator system. As a consequence, the internal water pump is always switched on during the working time and, therefore, consuming electrical energy independently of the air conditioning necessities. Whereas, in the new management strategy, the air conditioning system only consumes electrical energy according with the thermal demand in the office area. Then, the unnecessary electrical consumptions, which can suppose a great energy loss throughout the year, are avoided.

In summer season, the influence of the external temperature and the solar radiation which were an advantage during the winter season are now a disadvantage. Therefore, both management strategies are more active to provide the necessary high cooling power to achieve the thermal comfort. In these conditions is when the new management strategy has more chances to manage the air conditioning system to reduce its electrical consumption. In figure 5.2, the electrical consumption for both strategies grows up in function of the cooling demand, being the maximum electrical consumptions during the warmest months, July and August. In all months during the cooling season, except April, the electrical consumptions of the air conditioning system when is employed the new one are significantly smaller than the ones achieved when using the conventional one. Furthermore, the new strategy improves the efficiency of the air conditioning system as the cooling demand increases.

In April, the electrical consumption of the air conditioning system is smaller when the conventional management strategy is used. This behaviour is because in the first day of this month the air conditioning system changes from heating mode to cooling mode. When the conventional management strategy manages the air conditioning system the value of the PMV index is between 0.0 and 0.5 most of the time belonging to the working periods. In these conditions, this strategy keeps switched off the fan, the heat pump and the external water pump. Meanwhile, the new strategy activates these three devices to supply energy to the office to compensate the small deviation of the PMV from the neutral thermal comfort state.

Finally, the annual electrical consumption of the air conditioning system when it is managed by the conventional management strategy is estimated in 3359 kWh, and in 2289 kWh when managed by the new one. Then, the annual energy savings achieved are estimated in a value around the 30%.

5.2 Efficiency improvement of Ground Coupled Heat Pumps when combined with air source heat pump and thermal storage

An office building, located in the Mediterranean coast area, is modelled to obtain a thermal load in order to evaluate in these conditions the energy performance of the air conditioning

configurations to be studied. The thermal comfort criteria employed to obtain an estimation of the thermal loads demanded by the office building is the Predicted Mean Vote (PMV) index. This study is focused in the cooling dominated area of the Mediterranean coast. To estimate the thermal loads demanded by the office building it is considered that the heating season comprises the period from January to March and from November to December, and the cooling season the period from April to October. Daily load in heating or in cooling has to satisfy neutral thermal comfort conditions. The weather database employed describes an average season representing the Mediterranean coast weather for the city of Valencia (Spain).

The model of the office building adopted for this study is chosen to be representative of this kind of buildings in our area. It features three floors with a length of 30 m, a width of 20 m and a height of 3 m with eight thermal zones per floor. External walls are defined as ventilated façades composed by four elements: perforated brick, 5 cm of insulation, air chamber and a Naturex plate cover; its global conductivity is 0.51 W/m²K. The window fraction is approximately 22% in each façade; the windows are composed by a glass, with solar radiation transmissivity equal to 0.837 and conductivity equal to 5.74 W/m²K, dedicating a 15% of this area to the frame surface with conductivity equal to 0.588 W/m²K. The internal and external shadow factor for these windows is estimated at 0.7. Gains due to occupancy and lights are as follows. Peak building occupancy is 11 m²/person. Each office worker contributes 132 W of internal gain, where 54% are assumed to be sensible and 46% latent. The working schedule starts at 7:00 and at 9:30 the occupancy arrives to its peak. The lunch time is from 13:00 to 15:00 in this period the occupancy decrease until a fifty per cent. Finally, people start to leave the office at 18:00 and the office is empty at 20:00. The light system is switched on at 7:00 in the morning and increases progressively until its peak at 8:00. The value of the peak lighting density is 20 W/m². This value is constant until 19:00; in this moment starts to decrease and all the lights are switched off at 20:00. Taking into account all these considerations, the daily loads in heating season and in cooling season are calculated.

The air conditioning configurations coupled to the above described office buildings are described in the following paragraphs.

- *Air to water heat pump configuration ('Air')*, composed by an AWHP, an internal water pump and an air fan. Thermal energy is generated and supplied by the air to water heat pump when it is demanded by the building.

- *Ground coupled heat pump configuration ('GCHP')*, composed by a water to water heat pump (WWHP), a ground heat exchanger and an internal and an external water pump. Thermal energy is generated and supplied by the ground coupled heat pump when it is demanded by the building.

- *Ground coupled heat pump and air to water heat pump configuration ('GCHP + Air')*. This configuration combines a GCHP and an AWHP. Thermal energy is generated and supplied to the thermal load by the GCHP and, if this element has not enough capacity to supply the demanded thermal energy, the AWHP is switched on and supplies thermal energy up to satisfy the thermal demand. The GCHP system is switched on first because its coefficient of performance is the higher of both.

- *Air to water heat pump with thermal storage device configuration ('Air + S')*. This configuration combines an AWHP system with a thermal storage device. A period in the night is dedicated to store thermal energy. This energy is generated by the AWHP

system. The procedure to supply thermal energy to the load in cooling mode is as follows. The thermal demand is first satisfied by the thermal energy previously stored during the night. If the thermal storage device has not enough capacity, this element is supported by the AWHP until the thermal demand is covered. Finally, the thermal storage device is by-passed when its water outlet temperature is higher than its water inlet temperature. At these conditions only the AWHP supplies thermal energy to the load.

- *Ground coupled heat pump with thermal storage device configuration ('GCHP + S'):* This configuration combines a GCHP system with a thermal storage device. A period in the night is dedicated to store thermal energy. This energy is generated by the GCHP system.

- *Hybrid configuration ('HC'):* This configuration comprises a GCHP, an AWHP and a thermal storage device, implementing both ideas, combining two heat pumps and decoupling thermal generation from thermal distribution. The GCHP is used to store thermal energy during the night. The procedure to supply thermal energy to the load in cooling mode is as follows. The energy stored during the night in the storage device is used to satisfy the thermal demand. If the thermal storage device has not enough capacity, the GCHP is switched on to support it. If these two elements are still not enough, the air to water heat pump is switched on. Finally, the thermal storage device is by-passed when its water outlet temperature is higher than its water inlet temperature and only the GCHP and the AWHP supply thermal energy to the load.

The choice of the parameters for each air conditioning configuration considers the nominal coefficient of performance for the WWHP in cooling mode to be 4.5, and the nominal coefficient of performance for the AWHP in cooling mode to be 2.5. For the configuration design it is assumed that the thermal storage device supplies 40% of the thermal energy demanded by the load and that the water inlet temperature to the thermal load is fixed to the value T=7ºC (a typical distribution temperature for these type of systems). To estimate the size of the ground heat exchanger we use the method proposed in (Canada (2005)). The obtained length is normalized to boreholes of one hundred meters depth which are connected in parallel. We consider a ground thermal conductivity and a pipe thermal conductivity of 2 W/mK and 0.42 W/mK respectively and a borehole radius of 0.1016 m.

	P_{WWHP} (kW)	P_{AWHP} (Kw)	P_{AF} (Kw)	m_{AF} (m³/h)	P_{EWP} (Kw)	m_{EWP} (Kg/h)	P_{IWP} (Kw)	m_{IWP} (Kg/h)	Number Boreholes	TSD (m³)
Air		45.0	4.30	18000			1.70	9000		
GCHP	45.0	------	-------	------	0.85	9000	1.70	9000	13	
GCHP+Air	35.0	10.0	0.74	4100	0.76	7000	1.42	7000	12	
Air+S	------	35.0	3.27	14000	------	------	1.42	7000	------	40
GCHP+S	35.0	------	-------	------	0.76	7000	1.42	7000	8	40
HC	24.6	9.0	0.65	3700	0.67	5000	1.14	5000	7	27

Table 5.1. Parameters of the different devices for the air conditioning configurations.

Following these considerations and design procedures we have chosen the parameters for each air conditioning configuration. Table 5.1 shows the values of these parameters. For

each configuration the nominal cooling capacity of the WWHP (PWWHP), the nominal cooling capacity of the AWHP (PAWHP), the air fan power consumption (PAF), the air volume flow (mAF), the external water pump power consumption (PEWP), the water mass flow in the external circuit (mEWP), the internal water pump power consumption (PIWP), the water mass flow in the internal circuit (mIWP), the number of boreholes and the size of the thermal storage device (TSD), are included. The values of the parameters presented in table 5.1 have been chosen after performing a study of the efficiency of the combined system for different capacities of the heat pumps and for different sizes of the thermal storage device. The most efficient configurations are the ones presented in table 5.1.

In figure 5.3 the total electrical energy consumption of each air conditioning layout during cooling season evaluated using TRNSYS software tool is presented. The total electrical energy consumption is composed by three parts: the electrical energy consumption of the heat pumps (AWHP and WWHP), the electrical energy consumption of the water pumps (IWP and EWP) and the electrical energy consumption of the air fan (AF). We evaluate the energy efficiency for each layout with the Cooling Mode Performance Factor, CMPF, defined as the ratio between the total cooling thermal load and the total electrical energy consumption in cooling mode:
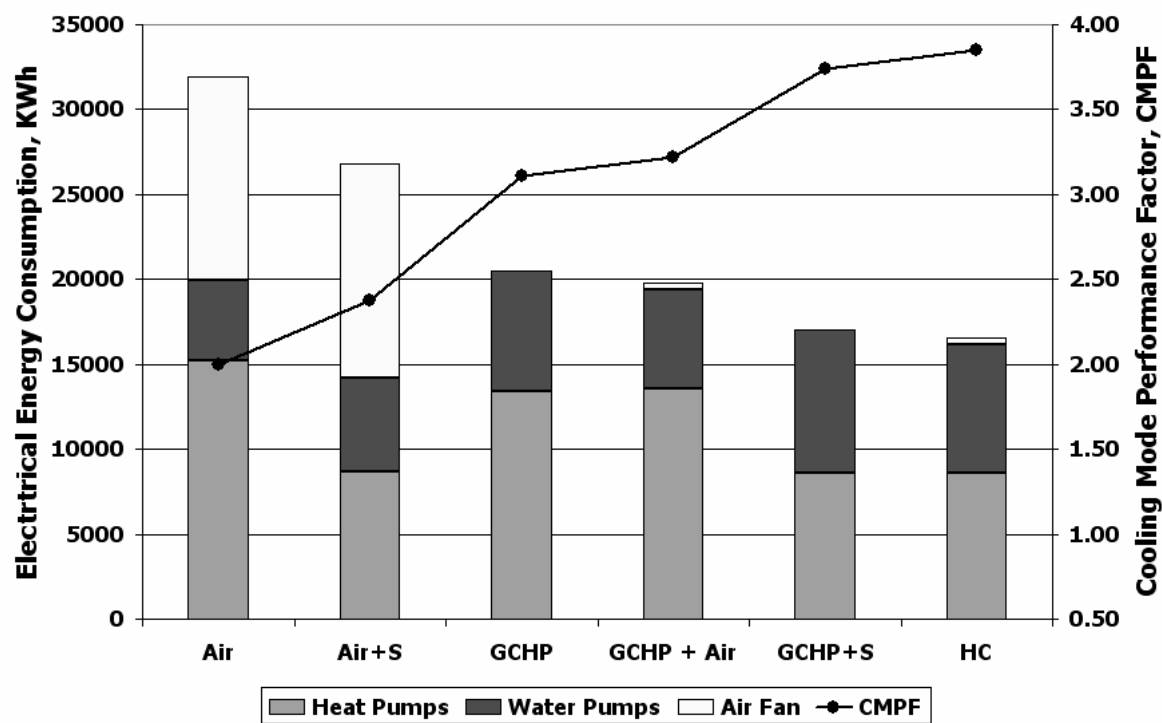
$$CMPF = \frac{Q_{load,cool}}{W_{elec,cool}}$$

(5.1)

Fig. 5.3. Total electrical energy consumption in cooling mode and Cooling Mode Performance Factor (CMPF) for the air conditioning configurations.

Looking at figure 5.3 we observe the following behaviour of the studied layouts:

- Combination of both heat pumps ('GCHP+Air' configuration). In figure 5.3 it can be seen that the 'GCHP + Air' configuration is more efficient than the sole ground source

system 'GCHP' because the electrical energy savings achieved by reducing the sizes of the water to water heat pump and water pumps are higher than the electrical energy consumption of the auxiliary system.

- One heat pump and thermal storage ('GCHP+S'or 'Air+S' configuration). An improvement in efficiency is produced in all air conditioning configurations which store cooled water during the night. In these configurations the electrical energy consumption of the heat pumps is reduced in spite of the increase in the electrical energy consumption of the air fan and the water pumps.
- Both heat pumps and thermal storage ('HC' configuration). This layout is the one that takes most advantage of the combination of both heat pumps and the decoupling of energy generation from energy distribution.

6. Conclusion

This contribution collects the recent research work of the authors on the subject of design, characterization and optimization of ground coupled heat pump systems. On the area of determining ground thermal properties, the effect of finite size corrections on the evaluation of ground thermal conductivity has been studied on a data set obtained from a standard thermal response test, and a method of subtraction of the influence of outside perturbations has been developed. In addition, an instrument able to measure the temperature evolution of the fluid along its way through the borehole heat exchanger has been designed.

The energy efficiency of an air conditioning system based on ground coupled heat exchangers (GeoCool plant) has been compared to a conventional air source system in heating and cooling modes. The obtained results have demonstrated that a ground source heat pump system is a viable and energy efficient alternative to conventional systems for heating and cooling applications in the South European regions. Predictions of energy performances based in standard design procedures based in TRNSYS have been compared with the actual performances of GeoCool plant. This comparison has allowed to estimate the accuracy of this design procedure.

A new management strategy for the operation of a ground coupled heat pump system has been designed and tested by means of TRNSYS simulation. The conclusion achieved was that management strategies able to adapt the production of thermal energy to the actual thermal demand can produce substantial energy savings (of the order of 30% in our study). Finally, the energy efficiency improvement of ground coupled heat pump systems when combined with thermal storage and supported by an air to water heat pump has been investigated. The obtained results showed that decoupling thermal energy production from thermal energy distribution, by means of a storage element, and properly combining a ground coupled heat pump with an air to water heat pump, are possible ways to reduce the electrical energy consumption of the system.

7. Acknowledgements

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Paths to Sustainable Energy

Edited by Dr Artie Ng

The world's reliance on existing sources of energy and their associated detrimental impacts on the environment- whether related to poor air or water quality or scarcity, impacts on sensitive ecosystems and forests and land use - have been well documented and articulated over the last three decades. What is needed by the world is a set of credible energy solutions that would lead us to a balance between economic growth and a sustainable environment. This book provides an open platform to establish and share knowledge developed by scholars, scientists and engineers from all over the world about various viable paths to a future of sustainable energy. It has collected a number of intellectually stimulating articles that address issues ranging from public policy formulation to technological innovations for enhancing the development of sustainable energy systems. It will appeal to stakeholders seeking guidance to pursue the paths to sustainable energy.

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