European Geothermal Congress 2013
Pisa, Italy, 3-7 June 2013

Optimization methodology for GSHP installations based on the circulation pumps frequency variation

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Keywords: Heating and cooling systems, ground source heat pump, energy efficiency.

ABSTRACT

Geocool plant was the result of a EU project whose main purpose was to adapt ground source heat pump technology to cooling dominated areas. The installation was finally built by the end of year 2004 and it has been monitored since February 2005 until nowadays. In April 2011, in the framework of another European project, Ground-med, the old heat pump (ON/OFF compressor) located at the Geocool plant was replaced by a new more efficient design with two compressors (ON/OFF) of the same capacity working in tandem. In order to optimize the energy performance of the installation, a new methodology based on the frequency variation of the circulation pumps was developed for both systems: Geocool (1 compressor ON/OFF) and Ground-med (2 compressor ON/OFF in tandem), so that it is possible to experimentally determine the optimal frequencies for the circulation pumps as a function of the thermal energy demand of the building along the day for both systems. This work presents the optimisation methodology for both systems.

Nomenclature

COP_1	Heat pump COP
COP_2	Heat pump and outdoor loop COP
COP_3	System COP
c_p	Specific heat at constant pressure
$\dot{Q}_{HP}$	Heat pump capacity
q	Building thermal demand
t_{ON}	On-cycle time
t_{OFF}	Off-cycle time
V	Volume
$\dot{W}_{comp}$	Compressor consumption
$\dot{W}_{ECP}$	External circulation pump consumption
$\dot{W}_{ICP}$	Internal circulation pump consumption
$\dot{W}_{par}$	Heat pump parasitic losses consumption
ΔT_{db}	Temperature deadband
α	Partial load ratio
α'	Partial load ratio for each compressor in Ground-med installation
η	Electrical efficiency of the ICP
ρ	Density
τ	Total cycle time

Acronyms

COP	Coefficient Of Performance
ECP	External Circulation pump
GSHP	Ground Source Heat Pump
GSHX	Ground Source Heat Exchanger
ICP	Internal Circulation Pump
PF	Performance Factor

1. INTRODUCTION

In the current context of global warming concern, renewable energies are spreading more and more. In recent years, geothermal energy has generated keen interest due to its already proved potential energy savings. Specifically, Ground Source Heat Pump systems (GSHP) (Lund 2001, Sanner et al 2003, Spitler 2005, Chua et al 2010), which take advantage of shallow geothermal energy, can lead to a 40% savings in annual electricity consumption compared to air to water conventional heat pumps (Urchueguía et al 2008).

Research to date on control of GSHP has focussed on capacity control issues and to a lesser extent on control of secondary loop working fluids. Control for on/off compressor has been compared with variable speed control for a brine-to-water heat pump (Fahlén and Karlsson 2003, Fahlén and Karlsson 2005) and it has been noted that the main benefit of using a variable speed compressor is a reduction in the need of supplementary heating.

However, several studies have pointed out the considerable amount of energy consumed by auxiliary equipment in air conditioning systems (Bernier and Bourret 1999, Brodrick and Westphalen 2001). This considerable consumption coming from auxiliary equipment is particularly relevant for GSHP systems in which two circulation pumps are required. In Granryd (2010) analytical expressions for possible optimum flow rates on the secondary loop are shown.

The present paper is focused on finding the optimal pump frequencies for a particular GSHP air conditioning facility as well as determining the way to identify this optimum by means of a simple and adequate experimental methodology. Moreover, this experimental methodology is explained for the same GSHP system but with different types of heat pump units working. As introduced in the abstract, the geothermal plant considered in the present work was built during the Geocool project in the year 2004 (Montagud et al 2011). A few years later, in April 2011, in the framework of Ground-med project (Montagud and Corberán 2010), the old heat pump located at the geothermal plant was replaced with a new one. Therefore, in the following, when referring to either Geocool installation or Ground-med installation, they will be the same geothermal facility (same ground heat exchanger) but with different heat pumps:

- Geocool installation: 1 compressor ON/OFF
- Ground-med installation: 2 compressors ON/OFF of the same capacity working in tandem

First of all, the experimental installation considered in this work is introduced. Once familiarized with the system, the impact of the flow rates on the system performance is studied and the energy efficiency parameters to assess the system performance are presented. After that, the experimental methodology itself is explained for both Geocool and Ground-med installation, and results are shown accompanied by the corresponding explanation. Finally some conclusions are drawn regarding the potential energy savings by applying the optimal frequencies.

2. METHODOLOGY

2.1 GSHP experimental plant

The experimental plant studied in this paper air-conditions a set of spaces in the Department of Applied Thermodynamics at the Universitat Politècnica de València, Spain, with a total surface of 250 m^2. All rooms are equipped with fan coils supplied by the GSHP system. The geothermal system consists of a reversible water to water heat pump, a vertical borehole heat exchanger and a hydraulic group (Fig. 1).

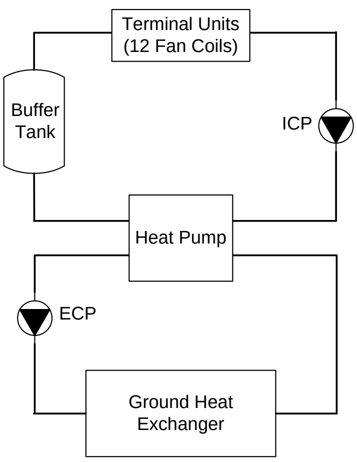

Figure 1: GSHP installation schematic diagram.

As it can be observed in Fig. 1, the system can be divided into two main circuits: an internal circuit which consists of a series of 12 parallel connected fan coils, an internal hydraulic loop and a water storage tank, and an external circuit which consists of the ground source heat exchanger (GSHX) which is coupled to the heat pump by an external hydraulic loop. Both circuits, internal and external, are provided with circulation pumps which make the water circulate towards the fan coil units (ICP) and the GSHX (ECP).

A network of sensors was set up in order to monitor the most relevant parameters. These sensors measure temperature, mass flow and power consumption. A timer controls the overall system operation, which was programmed to operate from 7am to 10pm, 5 days per week. Two frequency inverters, one for each circulation pump, were installed in order to vary the fan coil units and GSHX water flow rates.

2.2 Impact of secondary loop flow rates on system performance

When optimizing the overall system performance, it is important to understand how the increase of the circulating water flow rate affects the COP of the heat pump and that of the entire system. In a given system, the higher the inverter frequency, the greater the circulating water flow rate. A higher water flow rate enhances the heat transfer coefficient through the heat exchanger of the heat pump and diminishes the water temperature variation across it; the same happens at the GSHX. On the heat pump side, the increase of the water flow rate helps to reduce the temperature difference between the water and the refrigerant and, as a result, the temperature lift that must overcome the compressor becomes lower and the heat pump COP increases (Corberán et al 2008).

This can be observed in Fig. 2 which shows experimental results of the effect of varying the water flow rates in both the heat pump COP and the system COP.

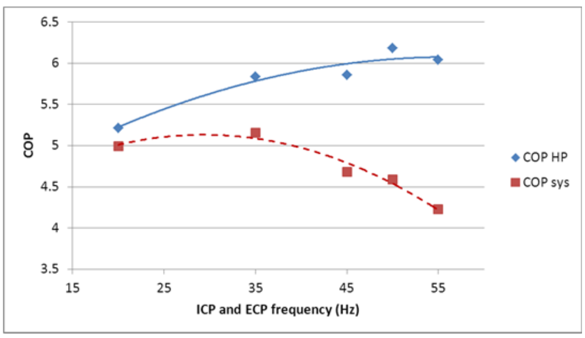

Figure 2: Influence of the water circulating flow rate on the heat pump and system COP.

It is clearly noted that the higher the flow rate at both the external and internal circuit, the better the COP of the heat pump (COP_{HP}). Nevertheless, when the whole system is considered, even though operating at maximum flow rates (maximum frequencies) leads to higher heat pump COP, it also deteriorates the COP of the system (COP_{system}) due to the circulation pumps consumption big influence.

In a few words, on one hand, increasing the water flow rate on both sides of the heat pump (evaporator and condenser) diminishes the compressor consumption but, on the other hand, increases the circulation pumps consumption. The fact that there are opposite trends on energy consumption for the variation of the circulation pumps frequencies means that there is an optimum frequency for each one of the water loops.

Therefore, this paper focuses on finding out which are the optimum circulation pumps frequencies leading to the minimum energy consumption of the whole system. For that purpose, an experimental methodology to carry out on site in a given installation is explained and applied in the present work.

2.3 Evaluation of energy efficiency parameters

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. Depending on the duration of the time interval, the energy performance factor can be seasonal, monthly, daily, etc.

However, for this study it has been decided that, in order to characterize the typical performance of the system, a characteristic cycle ON/OFF could be analyzed, and the energy performance factor of the day would correspond to the performance factor of one characteristic cycle ON/OFF. Moreover, when calculating the performance factor for a single ON/OFF cycle, if only the ON time is considered, the performance factor integrated for one cycle will be the same as the coefficient of performance (COP) at steady state conditions. In fact, it was proven in Corberán et al (2011 (1)) that water to water units have negligible startup losses, hence partialization losses only depend on the parasitic losses due to the electronics. During the ON time cycle, the unit works as in steady state conditions with condensing and evaporating temperatures gliding with the corresponding inlet water temperatures variation.

Returning to the energy performance factor, three different coefficients of performance will be employed in the following: heat pump (COP_1 or COP_{HP}), heat pump and outdoor loop (COP_2) and the whole system without considering the fan coils consumption (COP_3 or COP_{system}). These coefficients of performance will be calculated using expressions [1] to [3] during the ON time period for each cycle, and therefore they will correspond to the performances under quasi-steady state conditions.

$$COP_1 = \frac{\int_0^{t_{ON}} \dot{Q}_{HP}(t) \cdot dt}{\int_0^{t_{ON}} \left(\dot{W}_{comp}(t) + \dot{W}_{par}(t)\right) \cdot dt} \quad [1]$$

$$COP_2 = \frac{\int_0^{t_{ON}} \dot{Q}_{HP}(t) \cdot dt}{\int_0^{t_{ON}} \left(\dot{W}_{comp}(t) + \dot{W}_{par}(t) + \dot{W}_{ECP}(t)\right) \cdot dt} \quad [2]$$

$$COP_3 = \frac{\int_0^{t_{ON}} \left(\dot{Q}_{HP}(t) \pm \eta \cdot \dot{W}_{ICP}(t)\right) \cdot dt}{\int_0^{t_{ON}} \left(\dot{W}_{comp}(t) + \dot{W}_{par}(t) + \dot{W}_{ECP}(t) + \dot{W}_{ICP}(t)\right) \cdot dt} \quad [3]$$

The main difference between COP_1, COP_2 and COP_3 is that the first one just takes into account the energy consumption of the heat pump (parasitic losses and compressor consumption), whereas the second considers the heat pump and the external circulation pump consumption, and the third one takes into consideration both the external and internal circulation pumps consumption as well as the heat pump consumption.

When analyzing the system coefficient of performance (COP_{system}), which also includes the internal pump consumption, the heating/cooling capacity that should be considered is the one transferred to the building. As a matter of fact, the internal circulation pump heats up the water; for this reason, the internal circulation pump consumption needs to be added to the heat pump capacity during heating mode whereas must be subtracted during cooling mode. That is the reason for the term $\pm\eta \cdot \dot{W}_{ICP}$ in the numerator of equation [3]. Moreover, not all the electrical power consumed by the circulation pump is transferred to the water, but part of these losses goes to the surroundings. That is why the electrical efficiency appears in the term as well.

2.4 Proposed methodology: Geocool installation

The use of a detailed model for the estimation of the optimal frequencies is something which is not possible in most of the geothermal installations since it requires an accurate experimental characterization of the different components and an appropriate mathematical model of the installation. On the other hand, even with the best of the models, it is not possible to take into account the wide group of parameters which in practice affects the performance of a geothermal installation, as for instance, the users' daily activity, the ground thermal response, ambient temperature variations along the day…

Therefore, it was decided to try an experimental approach to get the optimums in a given installation under real operating conditions. Next, a new methodology for the in situ optimization of the frequency of each of the water circulation pumps of a geothermal system with ON/OFF regulation is proposed, consisting of the following 3 steps.

<u>Step 1: ON time operation characterization</u>

The first step consists of several experimental tests of pseudo-random sequence of frequency steps for both, internal and external circulation pumps, carried out during a single day. Table 1 shows an example of the possible test sequence for a variation of the frequencies of the circulation pumps from 20Hz to 60Hz.

The objective of this step is the characterization of the performance of the system during the ON time of the compressor at all possible combinations of frequencies. Therefore, each couple of frequencies must be kept constant during a complete ON cycle of the heat pump which should last, according to the authors' experience, ten minutes or longer.

As the obtained results may be influenced by the average ground temperature, and this changes along the year, as well as with the compensation of the setting temperature for the water return temperature, it is recommended to perform the tests during at least 4 days a year: two during the heating season and two during the cooling season, being one test at the middle of the season and one at the end of the season. In the Geocool installation, the optimum frequencies obtained from different days along one season are quite the same (Corberán et al 2011 (2)). The reason to propose to repeat the test along one season is exactly to assess the influence of the ground temperature, and confirm whether the optimum remains at the same zone or it requires some readjustment along the season.

Table 1: Example of a test sequence for the variation of the circulation pumps frequencies.

ICP \ ECP	20 Hz	30 Hz	40 Hz	50 Hz	60 Hz
60 Hz	●	●	●	●	●
50 Hz	●	●	●	●	●
40 Hz	●	●	●	●	●
30 Hz	●	●	●	●	●
20 Hz	●	●	●	●	●

Once the test is finished, the different coefficients of performance COP_1, COP_2 and COP_3 can be evaluated (as defined in section 2.3) for each ON cycle so that it will allow the construction of the performance maps of the unit as a function of both the external and internal circulation pumps frequency.

Step 2: Estimation of the heat pump and system COP maps

The second step consists of the analysis of the results obtained from step 1, the estimation of the heat pump COP (COP_1), the heat pump and outdoor loop COP (COP_2) and the system COP (COP_3), as defined by equations [1] to [3] respectively, and finally their representation in form of maps as a function of the circulation pump frequencies. Fig. 3 shows the COP maps obtained for a heating day in the Geocool installation.

As can be concluded from Fig. 3a, for the heat pump COP, the optimal frequency for both the external and internal pumps is the maximum working frequency, being in this case 60 Hz. As previously explained in section 2.2, a higher water flow rate (higher frequency) enhances the heat transfer coefficient through the heat exchanger and diminishes the water temperature variation across the heat exchanger. Therefore, the temperature difference between the water and the refrigerant tends to significantly decrease and, as a result, the temperature lift that the compressor must overcome is lower and the heat pump COP (COP_1) increases.

When the external circulation pump is taken into account (Fig. 3b), it can be seen how the optimum COP_2 moves to a lower external pump frequency, whereas the internal pump optimum remains at the maximum working frequency since its consumption has not been considered yet.

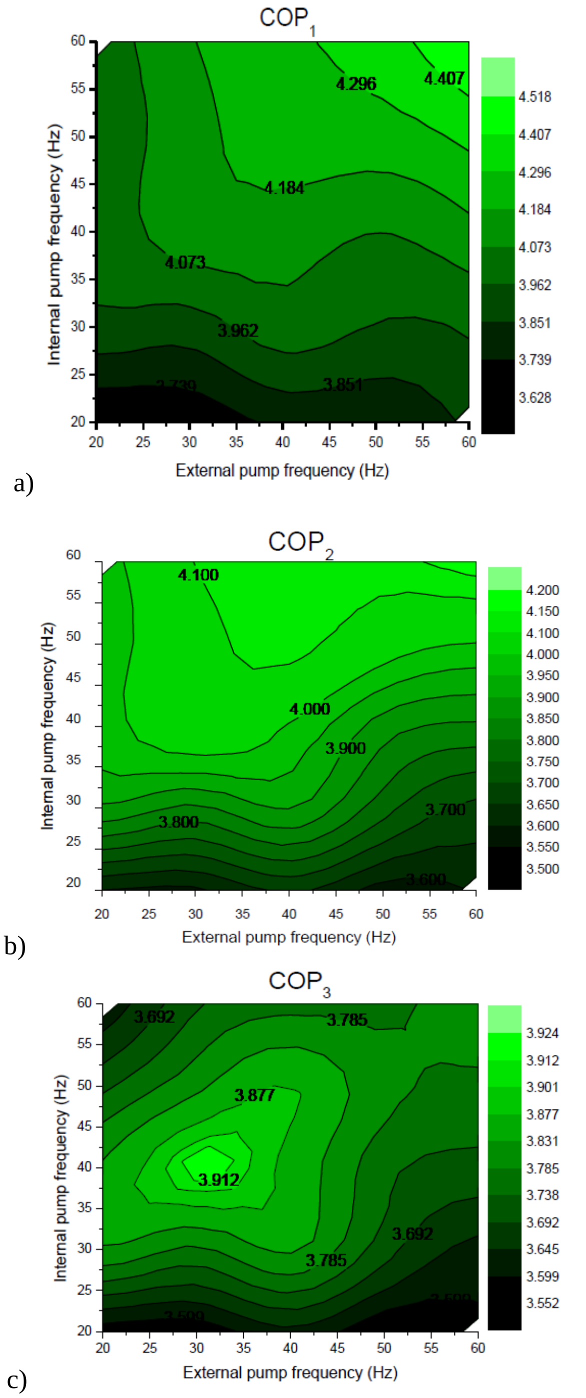

Figure 3: Quasi-steady state COP maps as a function of pump frequencies for heating mode (Geocool installation).

Finally, Fig. 3c shows that including both circulation pumps consumption in the performance evaluation makes the system COP (COP_3) move to lower pump frequencies. As it can be observed, the opposite effects of heat transfer improvement at the expense of energy consumption increase described in section 2.2 clearly leads to an optimum pair of frequencies which optimize the energy performance of the overall system. In this case of heating mode, that optimum corresponds to 32Hz for the external pump and 39Hz for the internal pump. Despite results are only shown for heating mode, for cooling mode the trend is similar but with slightly different optimum values.

The optimal pair of frequencies shown in Fig. 3 corresponds to quasi-steady state working conditions. In practice, steady state will only happen when the building thermal load equals the heat pump capacity, what should never happen since the heat pump is designed in order to be able to satisfy the maximum peak load for heating and cooling mode and the nominal capacity is always greater than the building load.

Several measurements at quasi-steady state conditions as explained above were carried out at different days for heating and cooling mode at the installation along more than one year. The maps of the different COPs were not exactly the same, with different absolute values which correlated well with different ground temperatures depending on the day and season. Nevertheless, the global trends and particularly the position of the optimal frequencies of the circulation pumps were very similar to those of the example shown for heating (Corberán et al 2011 (2)).

Step 3: Estimation of the system performance maps for any thermal load

The quasi-steady state performance maps of the unit were obtained during the ON cycle time. However, during the OFF time period there is power consumption, which comes from the internal pump (since it works continuously during the 15 hours of system operation) and the parasitic losses, which can significantly degrade the daily performance factor of the system. This third step allows taking into account this influence and calculating, from the quasi-steady state performance maps obtained in the step 2 of the methodology, the optimal frequencies as a function of the partial load ratio.

The ratio between the total thermal load that the system copes with and the heat pump capacity is known as the load ratio or partial load ratio, and can be evaluated by the following expression:

$$\alpha = \frac{\dot{q} \pm \eta \cdot \dot{W}_{ICP}}{\dot{Q}_{HP}} \quad [4]$$

Where the heat generated by the internal circulation pump ($\eta \cdot \dot{W}_{ICP}$) must be considered continuously since it is always kept ON, and must be added to the building thermal load during the cooling season and subtracted during the winter season (heats are considered in absolute value so they have the same sign for heating and cooling).

When ON/OFF regulation is employed, the ratio between the total thermal load to the system and the capacity of the compressor results in the end in the cycling of the compressor, and the partial load ratio can be evaluated as the relationship between the ON time of the compressor and the total time duration for each cycle:

$$\alpha = \frac{t_{ON}}{t_{ON}+t_{OFF}} \quad [5]$$

Where t_{ON} and t_{OFF} are the ON and OFF operational times for one cycle respectively.

The performance maps obtained in step 2 correspond to the quasi-steady state performance of the system and they do not take into consideration the thermal load of the building, but assume it equals the heat pump capacity. However, when considering the performance of the system during the whole day and not only during the ON cycle period, that is when considering both the ON and OFF cycle periods, the optimum values for the pumps frequencies are strongly dependent on the load ratio. Therefore, a simple analytical methodology can be employed to obtain the system performance factor (PF_{system}) as a function of the partial load ratio as it will be described in the following.

Considering a single whole cycle, the system performance factor can be obtained dividing the useful heat (for both the on-cycle and the off-cycle periods) by the energy consumed (for both the on-cycle and the off-cycle periods) as represented on equation [6].

$$PF_{system} = \frac{(\dot{Q}_{HP} \pm \eta \cdot \dot{W}_{ICP}) \cdot t_{ON} \pm \eta \cdot \dot{W}_{ICP} \cdot t_{OFF}}{\sum \dot{W} \cdot t_{ON} + (\dot{W}_{ICP} + \dot{W}_{par}) \cdot t_{OFF} + \dot{W}_{ECP} \cdot \frac{t_{extra}}{t_{ON}} \cdot t_{ON}} \quad [6]$$

Where $\sum \dot{W}$ stands for the total system consumption including both the internal and external circulation pumps, the compressor consumption and the heat pump electrical parasitic losses consumption, but not the fan coils ($\sum \dot{W} = \dot{W}_{ICP} + \dot{W}_{ECP} + \dot{W}_{comp} + \dot{W}_{par}$). Notice that, since the internal pump is continuously running during the whole day, it both generates heat and consumes power during the off time. The parasitic losses are also computed during the off-cycle time. Finally, the external pump follows the compressor operation, that is to say, when the compressor stops the external pump stops too. However, there is an extra time term (t_{extra}) in equation [6] which comes from a two minutes delay between the startup and stop of the external pump and that of the compressor due to operative requirements in the Geocool heat pump (more information in Montagud et al (unpublished results)). This does not happen in Ground-med installation, where the heat pump and the external pump starts up and stops exactly at the same time.

Using equation [5] and defining the total time τ ($\tau = t_{ON} + t_{OFF}$), the following relations can be obtained:

$$\frac{t_{ON}}{\tau} = \alpha \qquad \frac{t_{OFF}}{\tau} = 1 - \alpha \quad [7]$$

Multiplying equation [6] by $\frac{\tau}{\tau}$ and using expressions in [7]:

$$PF_{sys} = \frac{\alpha \cdot (\dot{Q}_{HP} \pm \eta \cdot \dot{W}_{ICP}) \pm (1-\alpha) \cdot \eta \cdot \dot{W}_{ICP}}{\alpha \cdot \sum \dot{W} + (\dot{W}_{ICP} + \dot{W}_{par}) \cdot (1-\alpha) + \dot{W}_{ECP} \cdot \frac{t_{extra}}{t_{ON}} \cdot \alpha} \qquad [8]$$

Dividing all the terms in equation [8] by the expression $\alpha \cdot \sum \dot{W}$ and considering the expression $t_{on} = \frac{\rho \cdot V \cdot c_p \cdot \Delta T_{db}}{\dot{Q}_{HP} \cdot (1-\alpha)}$, it is possible to obtain the system performance factor as a function of the coefficient of performance of the system (COP_{sys}) at quasi-steady state conditions, at different partial load ratios α:

$$PF_{sys} = \frac{COP_{sys} \pm \frac{(1-\alpha)}{\alpha} \cdot \frac{\eta \cdot \dot{W}_{ICP}}{\sum \dot{W}}}{1 + \frac{(1-\alpha)}{\alpha} \cdot \frac{(\dot{W}_{ICP} + \dot{W}_{par})}{\sum \dot{W}} + \left[COP_{sys} \mp \frac{\eta \cdot \dot{W}_{ICP}}{\sum \dot{W}}\right] \cdot \frac{\dot{W}_{ECP} \cdot (1-\alpha)}{\rho \cdot V \cdot c_p \cdot \Delta T_{db}} \cdot t_{extra}} \qquad [9]$$

Where COP_{sys} is identical to COP_3 obtained in step 2, as defined in [3]. The second term in the numerator of expression [9] will get higher or smaller values depending on the ratio between the internal circulation pump consumption and the total consumption of the system and it will have a greater influence on the system performance factor for low partial load ratios. Regarding the second term in the denominator, its influence will depend on the proportion of the added consumption of the internal circulation pump and parasitic losses to the total consumption of the system and, again, will have a stronger influence at low loads.

The information required in expression [9] comes from step 1, where the experimental measurements were carried out, and step 2, where the performance maps of the unit at quasi-steady state conditions were built. The third step of the proposed methodology would therefore consist in using expression [9] to extrapolate the performance maps characterization at quasi-steady state conditions obtained in step 2 to any partial load ratio. Looking at expression [9] it can be noticed that, when the partial load ratio α equals one, the system performance factor (PF_{system}) takes the same value as in quasi-steady state conditions, (COP_{system}).

An example of the performance maps obtained applying expression [9] is shown in Fig. 4. It can be observed that low partial load ratios degrade the performance of the system leading to lower optimal frequencies, because the auxiliary consumption (external and internal circulation pumps) has a great influence in the system performance factor, as it was predicted by expression [9]. On the contrary, the higher the partial load ratio, the more similar the optimal frequencies are to those corresponding to quasi-steady state conditions represented in Fig. 3, because the influence of the auxiliary consumptions turns out to be practically negligible.

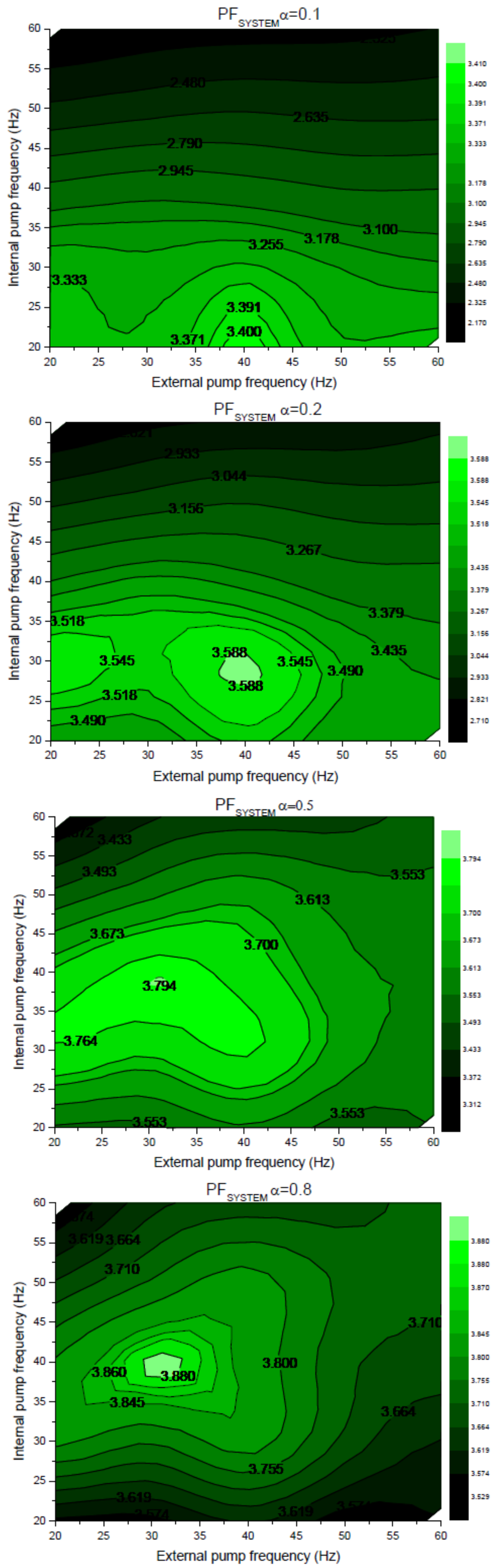

Figure 4: PF_{system} maps as a function of pump frequencies for heating mode (Geocool installation).

2.4 Proposed methodology: Ground-med installation

As introduced in section 1, under the framework of Ground-med project the former heat pump was replaced with a new one consisting of two ON/OFF controlled compressors working in tandem. Therefore, the methodology for the in situ optimization of the circulation pumps frequencies must be modified in order to consider a new heat pump with a new operation. The changes required on each one of the steps are explained in the following.

Step 1: ON time operation characterization

The first step consisted of applying a pseudo-random sequence of frequency steps for both circulation pumps during a single day. In the case of the Ground-med heat pump, since it consists of two compressors, the performance will be different whether there is one or there are two compressors working in tandem. Therefore, step 1 must be repeated for one compressor working on/off and for one compressor continuously running while the second compressor is cycling on/off.

It has been experimentally confirmed that the optimum frequencies when both compressors are running are located in higher values and hence there is no need of testing frequencies lower than 30Hz. Table 2 shows how, for the experimental tests carried out with two compressors running, it is possible to remove the points corresponding to 20 Hz, reducing the number of experimental points needed from 25 to 16 and hence reducing the required time to carry out the in situ experimental tests.

Table 2: Example of a test sequence for a variation of the circulation pumps frequencies.

60 Hz	--	●	●	●	●
50 Hz	--	●	●	●	●
40 Hz	--	●	●	●	●
30 Hz	--	●	●	●	●
20 Hz	--	--	--	--	--
ICP / ECP	20 Hz	30 Hz	40 Hz	50 Hz	60 Hz

As well as in the Geocool installation, in the Ground-med installation different experimental tests were carried out at different times along the year in order to confirm that the optimum remains at the same point.

Step 2: Estimation of the heat pump and system COP maps

In the same way as for Geocool installation, now the COP maps for Ground-med installation are obtained and an example of the results is shown in Fig. 5.

The maps in Fig. 5 represent the performance of the system when the heat pump is working with just one compressor. As we can see, results are similar to those for Geocool installation (Fig. 3). The higher the frequencies (and hence the flow rate), the higher the value of the heat pump coefficient of performance (COP_1). When the external pump is considered (COP_2), it penalizes the value of the COP as well as decreases the optimum external pump frequency. Finally, the system COP finds its optimum at 31HZ for the internal pump and 39Hz for the external pump.

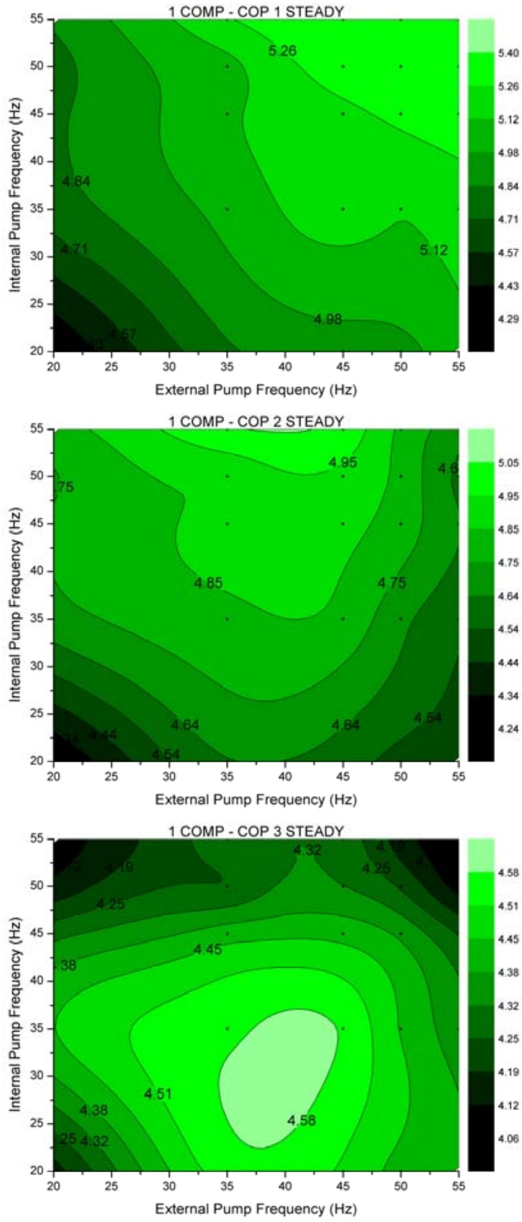

Figure 5: Quasi-steady state COP maps as a function of pump frequencies for heating (Ground-med installation, 1 compressor).

As it can be observed in Fig. 5, the influence of the internal pump is higher than that of the external pump, since the latter presents a higher value of the optimal frequency. Results are shown for heating mode but, in the same way as for Geocool installation, several tests were carried out along the year during each working mode and results confirmed similar trends and similar locations for the optimums. It can also be noticed the enhancement with respect to the Geocool heat pump, with an optimum system COP of 3.91whereas the new

Ground-med heat pump presents a value of 4.58 when one compressor is working.

However, as stated above, the Ground-med heat pump comprises two compressors working in tandem, so different maps are obtained for the heat pump working with two compressors. Results are shown in Fig. 6.

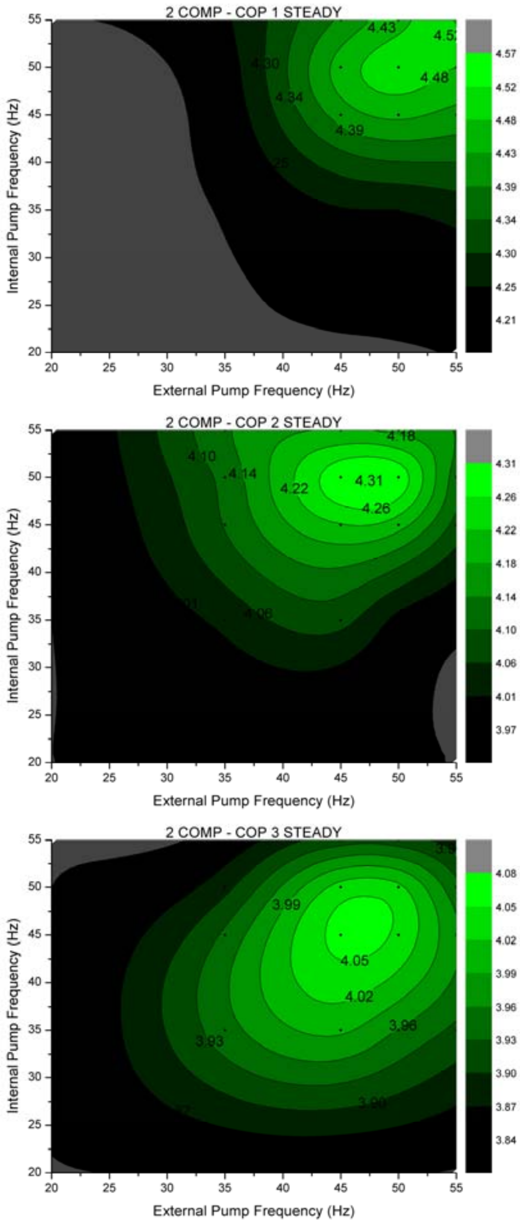

Figure 6: Quasi-steady state COP maps as a function of pump frequencies for heating (Ground-med installation, 2 compressors).

As in Fig. 3 and Fig. 5, the maximum heat pump COP (COP_1) corresponds to the maximum operating frequencies for both the external and internal pumps, since the higher the flow rate, the better the heat transfer. Nevertheless, when it comes to COP_2 and COP_3, it can be observed that the influence of the circulation pumps is smaller, since the heat pump consumption is higher when both compressors are running, and the optimum frequencies come to higher values.

Moreover, as introduced above in step 1, since the optimums come to higher frequencies, the points corresponding to 20Hz have been removed when carrying out the tests. That is why there is a dark zone for lower frequencies, because the program the maps have been obtained with is extrapolating without having the 20Hz points. Still the optimums remain at the same location.

<u>Step 3: Estimation of the system performance maps for any thermal load</u>

In order to calculate the optimal frequencies as a function of the partial load ratio from the quasi-steady state performance maps obtained in step 2 of the methodology, some changes must be introduced in the analytical expressions deduced for Geocool installation.

As it has been observed in the previous step, the performance of the installation is different whether one or two compressors are working. Therefore, a new variable, called "n" must be introduced in order to take into consideration the state at which the heat pump is working, whether one compressor cycling ON/OFF ($n = 1$) or one compressor continuously running and the second one cycling ON/OFF ($n = 2$). Fig. 7 depicts the two possibilities.

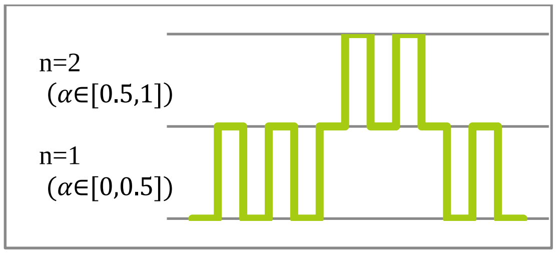

Figure 7: Heat pump working state for Ground-med installation.

Therefore, two different expressions must be deduced in order to extrapolate the performance maps characterization at quasi-steady state conditions obtained in step 2 to any partial load ratio. Considering α as the building load ratio, a new variable called α' will be the partial load ratio for each one of the states, and it will be calculated from the operating times:

$$\alpha' = \frac{t_{ON}}{t_{ON}+t_{OFF}} \quad [10]$$

The expression which calculates the system COP must be redefined in order to consider both operation states:

$$COP_{system(n-1)} = \frac{\dot{Q}_{HP(n-1)} \pm \eta \cdot \dot{W}_{ICP}}{\dot{W}_{par}+\dot{W}_{comp(n-1)}+\dot{W}_{ECP}+\dot{W}_{ICP}} \quad [11]$$

$$COP_{system(n)} = \frac{\dot{Q}_{HP(n)} \pm \eta \cdot \dot{W}_{ICP}}{\dot{W}_{par}+\dot{W}_{comp(n)}+\dot{W}_{ECP}+\dot{W}_{ICP}} \quad [12]$$

Where the denominators can be written as $\sum \dot{W}_{(n-1)}$ and $\sum \dot{W}_{(n)}$ respectively. Now, considering a single whole cycle, the system performance factor can be obtained dividing the useful heat (for both the on-cycle and the off-cycle periods) by the energy consumed (for both the on-cycle and the off-cycle periods) as represented on equation [13] for $n = 1$ and [14] for $n = 2$.

$$PF_{3(n=1)} = \frac{\dot{Q}_{HP(n)} \cdot t_{ON} \pm \eta \cdot \dot{W}_{ICP} \cdot \tau}{\dot{W}_{comp(n)} \cdot t_{ON} + (\dot{W}_{par} + \dot{W}_{ECP} + \dot{W}_{ICP}) \cdot \tau} \quad [13]$$

$$PF_{3(n>1)} = \frac{\dot{Q}_{HP(n-1)} \cdot t_{OFF} + \dot{Q}_{HP(n)} \cdot t_{ON} \pm \eta \cdot \dot{W}_{ICP} \cdot \tau}{\dot{W}_{comp(n-1)} \cdot t_{OFF} + \dot{W}_{comp(n)} \cdot t_{ON} + (\dot{W}_{par} + \dot{W}_{ECP} + \dot{W}_{ICP}) \cdot \tau} \quad [14]$$

Using the definition of α' in [10] and doing some more calculations, the final expressions to extrapolate the performance maps characterization at quasi-steady state conditions to any load ratio are obtained.

$$PF_{system(n=1)} = \frac{COP_{sys\,(n)} \pm \frac{\eta \cdot \dot{W}_{ICP}}{\sum \dot{W}_{(n)}} \cdot \frac{(1-\alpha')}{\alpha'}}{1 + \frac{(\dot{W}_{par} + \dot{W}_{ICP})}{\sum \dot{W}_{(n)}} \cdot \frac{(1-\alpha')}{\alpha'}} \quad [15]$$

$$PF_{system(n>1)} = \frac{COP_{sys(n-1)} \cdot \frac{(1-\alpha')}{\alpha'} \cdot \frac{\sum \dot{W}_{(n-1)}}{\sum \dot{W}_{(n)}} + COP_{sys\,(n)}}{\frac{\sum \dot{W}_{(n-1)}}{\sum \dot{W}_{(n)}} \cdot \frac{(1-\alpha')}{\alpha'} + 1} \quad [16]$$

Where $COP_{sys(n-1)}$ and $COP_{sys(n)}$ would be as defined in equations [11] and [12] respectively. An example of the system performance maps obtained applying expressions [15] and [16] for heating mode is shown in Fig. 8. It must be noticed that, for load ratios greater than 0.5, which means one compressor is continuously running and the second one is cycling ON/OFF ($n = 2$), expression [16] is applied and 20Hz points can be removed. As in Fig. 4, it can be observed that low partial load ratios degrade the performance of the system because the auxiliary consumption (external and internal circulation pumps) has a great influence in the system performance factor.

On the contrary, the higher the partial load ratio, the more similar the optimal frequencies are to those corresponding to quasi-steady state conditions represented in Fig. 5, for values of α up to 0.5, or in Fig. 6, for values of α greater than 0.5.

3. CONCLUSIONS

A methodology for the in situ optimization of the frequency of each of the water circulation pumps of a geothermal system with ON/OFF regulation was proposed. The methodology was applied to the geothermal installation located in an institutional building at the Universitst Politècnica de València, working with two different kinds of heat pumps: Geocool installation, whose heat pump consisted of one compressor ON/OFF, and Ground-med installation, whose heat pump comprises two compressors ON/OFF working in tandem. In both cases, analytical expressions giving the optimal circulation pump frequencies as a function of the partial load ratio and the mode (heating or cooling) can be obtained from the performance factor maps created in step 3 of the methodology.

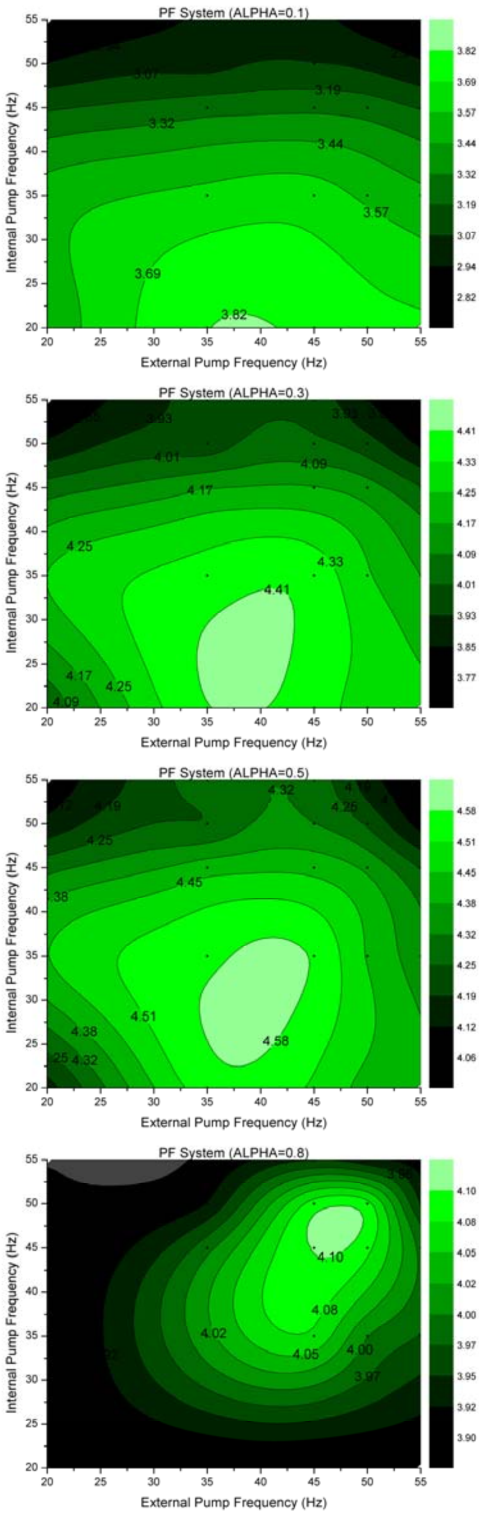

Figure 8: PF_{system} maps as a function of pump frequencies for heating mode (Ground-med installation).

Focusing on the current installation, that is Ground-med installation, Fig. 9 shows, for the internal circulation pump and for both, heating and cooling mode, analytical expressions which give the optimal frequency to be applied in the pump frequency inverter as a function of the load ratio. Similar expressions were obtained for the external pump and all of them were programmed in the control board of the system.

In both cases results confirm that the lower the load ratio, the lower the optimal frequency, hence decreasing the pumps consumption and optimizing the energy consumption of the system. Some preliminary experimental results have been analyzed and energy savings of 16% for heating mode and 22% for cooling mode have been obtained.

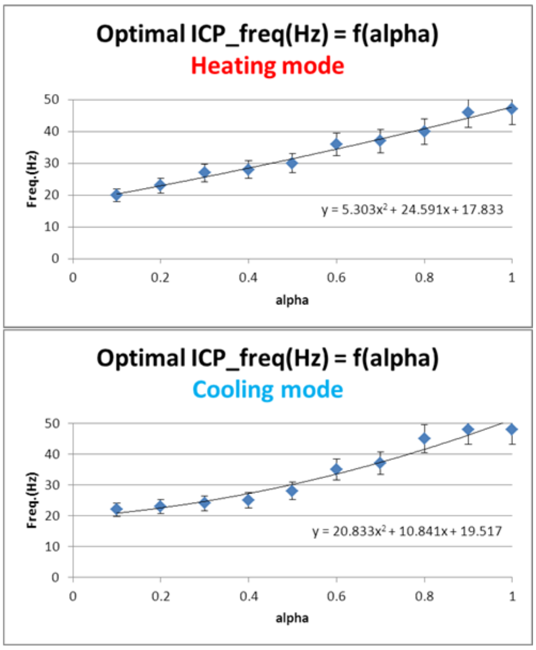

Figure 9: Internal pump optimal frequency as a function of the load ratio for both heating and cooling modes (Ground-med installation).

REFERENCES

Lund, J.W.: Geothermal heat pumps – an overview, *Geo-Heat Center Bulletin*, 22**(1)**, (March, 2001), 1-2.

Sanner, B., Karytsas, C., Mendrinos D. and Rybach, L.: Current status of ground source heat pumps and underground thermal storage in Europe, *Geothermics*, **32(4-6)**, (2003), 579-588.

Spitler, J.D.: Ground-Source Heat Pump System Research – Past, Present and Future, Europe, *HVAC&R Research*, **11(2)**, (2005), 165-167.

Chua, K.J., Chou, S.K. and Yang, W.M.: Advances in heat pump systems: A review, *Applied Energy*, **87(12)**, (2010), 3611-3624.

Urchueguía, J.F., Zacarés, M., Corberán, J.M., Montero, Á., Martos, J. and Witte, H.: Comparison between the energy performance of a group coupled water to water heat pump system and an air to water heat pump system for heating and cooling in typical conditions of the European Mediterranean Coast, *Energy Conversion and Management*, **49(10)**, (2008), 2917-2923.

Fahlén, P. and Karlsson, F.: Improving efficiency of hydronic heat pump heating systems, *21st International Congress of Refrigeration*, Washington, USA, (2003).

Fahlén, P. and Karlsson, F.: Optimizing and Controlling Media Flows In Heat Pump Systems, *8th IEA Heat Pump Conference*, Las Vegas, USA, (2005).

Bernier, M. and Bourret, B.: Pumping Energy and Variable Frequency Drives, *ASHRAE Journal*, **41(12)**, (December, 1999), 37-40.

Brodrick, J.R. and Westphalen, D.: Uncovering Auxiliary Energy Use, *ASHRAE Journal*, **43(2)**, (February, 2001), 58-61.

Granryd, E.: Analytical expressions for optimum flow rates in evaporators and condensers of heat pumping systems, *International Journal of Refrigeration*, **33**, (2010), 1211-1220.

Montagud, C., Corberán, J.M., Montero, Á. and Urchueguía, J.F.: Analysis of the energy performance of a ground source heat pump system after five years of operation, *Energy and Buildings*, **43(12)**, (2011), 3618-3626.

Montagud, C. and Corberán, J.M.: Ground-med project: advanced ground source heat pump systems for heating and cooling in Mediterranean climate, *Proceedings of the "II Congreso de Energía Geotérmica en la Edificación y la Industria (GEOENER 2010)" (Spanish)*, Madrid, Spain, (2010), 293-299.

Corberán, J.M., Radulescu, C. and Gonzalvez Macia, J.: Performance characterization of a reversible water to water heat pump, 9th *IEA Heat Pump Conference*, Zurich, Switzerland, (2008).

Corberán, J.M., Donadello, D., Martínez-Galván, I. and Montagud, C.: Partialization losses of on/off operation of water-to-water refrigeration/heat pump units, *International Congress of Refrigeration*, Prague, Czech Republic, (2011).

Corberán, J.M., Montagud, C. and Heselhaus, F.: Energy optimization of a ground source heat pump system for heating and cooling in an office building, *ISHPC11 Sources/Sinks alternative to the outside Air for Heat Pump and Air-Conditioning Techniques (Alternative Sources - AS), IIR International Conferences*, Padua, Italy, (2011), 93-102.

Montagud, C., Corberán, J.M. and Montero, Á.: Optimization methodology for the water circulation pumps frequency of ground source heat pump systems, *(Unpublished results)*.

Acknowledgements

This work was supported by the "Programa de Ayudas de Investigación y Desarrollo (PAID)" of the Universitat Politècnica de València.

This work was also supported under the FP7 programme "Advanced ground source heat pump systems for heating and cooling in Mediterranean climates" (Ground-med).