**International Geothermal Days Romania 2012**

**Groundmed Workshop**

Paper n°8

# ENERGY EFFICIENCY IN FIXED, STAGED AND VARIABLE CAPACITY GROUND SOURCE HEAT PUMPS

C. Montagud*, J.M. Corberán**

** Instituto de Ingeniería Energética, Universitat Politècnica de València, Valencia, Spain*

*carmonmo@iie.upv.es*

*** Instituto de Ingeniería Energética, Universitat Politècnica de València, Valencia, Spain*

*corberan@ter.upv.es*

## SUMMARY/ABSTRACT

*This paper presents the results of a study carried out as part of a European project titled Ground-Med “ Advanced Ground Source Heat pumps in the Mediterranean climatwe” which aims at demonstrating the efficiency of ground source heat pumps (GSHP) in mild weather climates like the Mediterranean, obtaining a seasonal performance factor (SPF) for the integrated system greater than 5. In order to achieve this goal, it is necessary to design heat pumps as efficient as possible. The work presented in this paper consists in a theoretical study carried out for the development of a high efficiency ground source heat pump prototype for heating and cooling in one of the demo sites of the Ground-Med project, located in an institutional building at the ‘Universitat Politècnica de València’, Valencia, Spain. The design will be based in the optimal selection of components for the ground source heat pump application. Different compressor technologies available in the market will be evaluated: fixed, staged and variable capacity compressors, and the seasonal performance factor for both winter and summer will be evaluated for each configuration of compressor considered. Results show that the best solution corresponds to the use of two compressors in tandem.*

## INTRODUCTION

As a result of growing concerns about global warming, developed countries are pushed to reduce greenhouse gases emissions. According to data provided by IDEA ("Institute for Diversification and Energy Saving"), energy consumption in buildings in Spain represents 20% of final energy consumption, a percentage that also tends to increase. The use of geothermal heat pumps stands for a 40% savings in annual electricity consumption compared to air heat pumps conventional water [Urchueguía, JF et al. (2008)].

The state of the art to date on the control of GSHP has focused on issues of capacity control. The most common type of capacity control currently developed for commercial GSHP seems to consist of a control type on / off for the compressor, which is cycling. Control on / off for the compressor has been compared with the variable speed control of a brine-water heat pump [Fahlen, P and Karlsson, F, 2005] and has been observed that the main benefit of using a variable speed compressor is a reduction in the need for additional heating. In another study, it was concluded that heat pumps are usually designed with a size that matches the 60% of the total heating load [Karlsson, F and Fahlen, P, 2007].

This paper presents the results of a study conducted as part of a European project titled Ground-Med [Montagud C. and Corberan J.M., 2010]. Ground-Med Project, funded by the European Commission under the Seventh Framework Programme, shows the next generation of ground source heat pump (GSHP) systems for heating and cooling in 8 demonstration sites in southern Europe: Italy, France, Romania, Portugal, Slovenia, Spain (Valencia and Barcelona), and Greece. Ground-Med is a 5 years duration project and its consortium consists of 24 European organizations including research institutes, universities, manufacturers of heat pumps, national and European industry associations, energy consultants, builders and a center for dissemination of information from 12 European countries: Spain, Portugal, Greece, France, Germany, Italy, Romania, Austria, Slovenia, Sweden, Holland, Ireland.

The main objective of the project is the design, construction, installation, monitoring and evaluation of GSHP systems presenting a "SPF" (Seasonal Performance Factor) for heating and cooling above 5, with a return period investment of less than 7 years compared with a conventional boiler with natural gas for heating and an air to water heat pump for cooling and with a high durability of at least 20 years. For this it is necessary to design heat pumps as efficient as possible.

The ground source heat pump system considered in this paper is an existing facility at the *‘Universitat Politècnica de València’, Valencia, Spain*. The plant was constructed within the framework of the FP5 European project 'Geocool', funded by the European Union, whose main objective was to adapt the technology of geothermal heat pump to areas where cooling energy demand dominates the heating. The implementation of this experimental plant was completed at the end of 2004, starting in February 2005 the regular operation and monitoring of the system. Since then, the installation 'GeoCool' has been monitored by a sensor network by which it has been possible to characterize the most relevant parameters of its operation and the determination of energy efficiency as presented in *[Montagud, C. and Corberan, JM. 2011]*.]. In 2011, and since it is a demonstration facility in the Ground-Med Project, it was decided to replace the existing heat pump with a new design which has been conducted within the framework of the project and has been manufactured by Hiref, which is an Italian heat pump manufacturer involved in the project. The results of work undertaken to design the new heat pump are presented in this paper.

First, a theoretical study devoted to the selection of the optimal compressor model to be installed in the new ground source heat pump to provide air conditioning in an office building in the Polytechnic University of Valencia, is performed. The prototype consists of a water to water reversible heat pump that is capable of delivering 16kW in cooling mode and 19 kW in heating mode. The working fluid is R410A and the evaporator and condenser heat exchangers consist of two alpha-laval AC70 brazed plate heat exchangers.

Second, the optimal number of plates as a compromise between the performance of the machine and the manufacturer's requirements will be determined.

Finally, a comparison of the energy performance of not only the heat pump but the integrated system formed by the heat pump and the external circulation pump is carried out by analyzing the seasonal performance factor SPF in winter, summer and throughout the year.

# 1 GROUND SOURCE HEAT PUMP SYSTEM LAYOUT

## 1.1 Description of the installation

The ground source heat pump system considered in this paper is located in an institutional building at the '*Universitat Politècnica de València', Valencia, Spain*. Figure 1a shows the wiring diagram, where it can be distinguished the heat pump, the external circuit formed by the external circulation pump and the ground source heat exchanger, consisting of 6 vertical single U boreholes of 50 m deep connected in parallel, in a rectangular grid of 2x3, with a 3 m borehole spacing; and the internal circuit, formed by the internal circulation pump, a storage tank and 12 fan coil units or terminal units connected in parallel whose distribution in the building is shown in figure 1b.

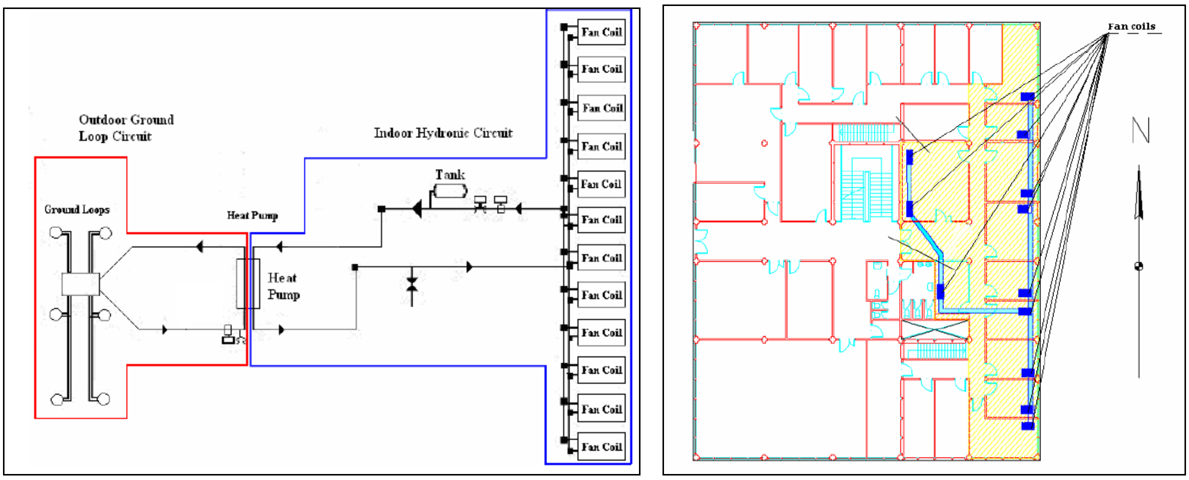

*Figure 1.: GSHP system: a) Hydraulic schematic diagram b) Fan-coils distribution in the building.*

The operation of the heat pump is governed by an electronic controller which, depending on the return water temperature of the internal circuit (fan-coils), switches on or off the heat pump compressor. The external circulation pump is also governed by the controller of the heat pump, which activates the external circulation pump 60 seconds before activation of the compressor and turns it off 60 seconds after the compressor. The system is fully monitored and there is a data acquisition system which, among other parameters, registers the values of the flow on the internal and external circuits, temperatures of the water at the inlet and outlet of the heat pump in both circuits, and electrical consumption of the compressor and other auxiliary equipment such as water circulation pumps and fan coils. The facility is in operation during the 5 days of the week for a total of 15 hours a day, ie from 7 o'clock in the morning at 22h in summer and from 6:00 am to 21h in winter.

## 2 COMPRESSOR SELECTION

One key component that represents a high consumption with respect to the global consumption of the system is the compressor. So, the first step to consider is the selection of one compressor model of high energy efficiency for this type of application (ground source heat pump).

With respect to the refrigerant, it is already set in the Ground-Med project objectives that the new heat pump designed for the installation at the '*Universitat Politècnica de València'* will work with R410A.

An assessment of the best compressor technology available in the market is carried out, and after a review of the leading manufacturers, the choice is limited to scroll type compressors. Different models are considered so that it is possible to analyze the performance of compressors with three types of control: variable speed compressor, compressor ON / OFF of larger displacement, and two compressors in tandem with a smaller displacement. Table 1 shows the compressor models considered in the study and its main features.

*Table 1: Compressors catalogue data.*

| COMPRESSOR MODEL | CAPACITY [kW] | [cm$^3$] | TEMP COND [ºC] | TEMP EVAP [ºC] |
|---|---|---|---|---|
| COPELAND ZP29KSE (2 tandem compresores ) | 6.2 | 27.6 | 50 | 5 |
| LG AR055VAD 30 Hz (variable capacity: low freq.) | 7.3 | 55.4 | 54.4 | 7.2 |
| LG AR055VAD 60 Hz (variable capacity: nominal freq.) | 14.2 | 55.4 | 54.4 | 7.2 |
| DANFOSS HLH061T4( ON/OFF) | 14.8 | 57.8 | 54.4 | 7.2 |
| DANFOSS HLH068T4 (ON/OFF) | 16.9 | 64.4 | 54.4 | 7.2 |

To carry out the analysis of the compressor performance and selecting the optimal model for the application, it is necessary to consider the pressure ratios in which the compressor will work throughout the year.

According to the Ground-Med project objectives, within which is framed the design of the heat pump presented in this paper, and in order to maximize the heat pump coefficient of performance COP, a minimum supply water temperature to the terminals of the building will be set to10 ° C in summer (cooling) and a maximum temperature of 40 ° C, will be set in winter (heating). These temperatures correspond to the cases of peak demand, ie the months of January and February for heating mode, and July and August for cooling mode. The rest of the year during the months of milder weather in Valencia, the water will be sent at a temperature up to 10 º C in summer and 35 º C in winter. With regard to the water temperature coming from the ground source heat exchanger, based on experimental measurements in the installation of the GSHP installation that exists in Valencia, four different values have been considered for the four seasons of the year in Valencia. As presented in [Montagud C.et al., 2011], the ground temperature has a sinusoidal evolution over the years, taking maximum values at the end of the summer, during which the condensing heat has been injected into the ground, and minimum values at the end of the winter, since the land has been cooled during the heating period. Thus, four different operating conditions are considered: summer, winter, extreme summer (end of cooling period) and extreme winter (end of heating period). Water temperatures at the inlet and outlet of the heat pump are considered for these four cases are shown in the table 2.

*Table 2: Water temperatures for the supply and return considered for the performance evaluation of the compressors: IC (building loop) and EC (ground loop).*

| WORKING MODE | Tsupply IC (ºC) | Treturn IC (ºC) | Tsupply EC [ºC] | Treturn EC (ºC) |
|---|---|---|---|---|
| Summer | 10 | 15 | 27 | 22 |
| Winter | 35 | 30 | 13 | 18 |
| Extreme summer | 10 | 15 | 35 | 30 |
| Extreme Winter | 40 | 35 | 5 | 10 |

Once the working temperatures for the application have been defined, and using the catalogue data ARI coefficients provided by the manufacturer of each compressor, the performance of each compressor will be compared in the simulation software IMST-ART [Corberán J.M., et al., 2002], for a given configuration of 42 plates and maintaining constant the superheat and subcooling values in the refrigerant cycle, and equal to 4K and 5K respectively. Results of the compressor efficiency comparison are shown in Figure 2, which shows the curves of compressor efficiency as a function of the pressure ratio.

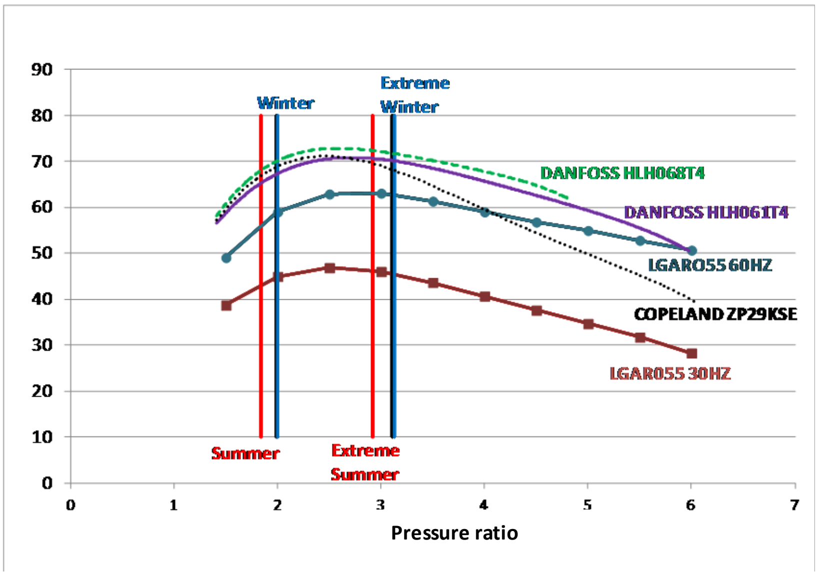

*Figure 2. : Compressor efficiency comparison.*

As seen in Figure 2, the energy efficiency of the variable-speed compressor LG when operating at a frequency of 60 Hz is much lower than other alternatives within the pressure ratio corresponding to the application under consideration, and this difference worsens at low frequencies around 30 Hz, therefore, the alternative of variable-speed compressor is discarded in this first study. With respect to the other compressor models, it is observed that there is an optimal compressor which has the efficiency curve more focused on the different working pressure ratios of the compressor and it corresponds to the model HLH068T4 Danfoss, followed by the model HLH061T4 Danfoss compressor. With respect to compressor ZP29KSE Copeland, it presents an intermediate efficiency with respect to the Danfoss compressors, except in the case of considering a pressure ratio corresponding to extreme winter conditions, in which ZP29KSE Copeland compressor would present a slightly lower efficiency.

In order to analyze the energy efficiency of the heat pump during the year, a comparison of the different compressors is carried out, fixing the temperature of the water returning from the fan coil units at 35 ° C for heating and 15 ° C for cooling, and making a parametric study where the return water temperature from the borehole heat exchanger varies according to the values experimentally measured in the UPV throughout the year.

Comparative results are shown in Figures 3 and 4 respectively for cooling and heating mode.

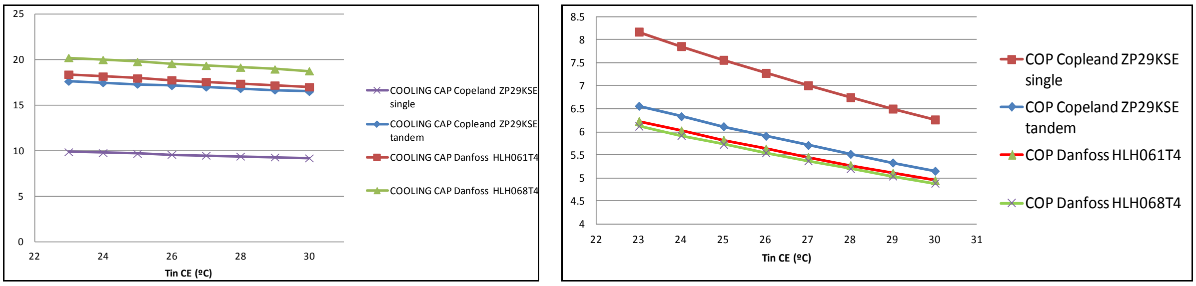

*Figure 3.: Influence of the outlet water temperature from the ground: a) Cooling capacity b) hea pump COP in cooling mode.*

As seen in Figure 3, in cooling mode, the COP of the compressors working in tandem with only one stage is much greater than the rest, and could take values above 8 at the beginning of the season just when the ground is cooler as it has been cooled down during the winter and the return water temperature of the ground loop shall, therefore, present a lower value. When comparing the two Copeland compressors operating in tandem, the COP decreases but remains higher than in the case of any type of compressors ON / OFF manufactured by Danfoss. The analysis is similar for heating mode, being the results presented in Figure 4.

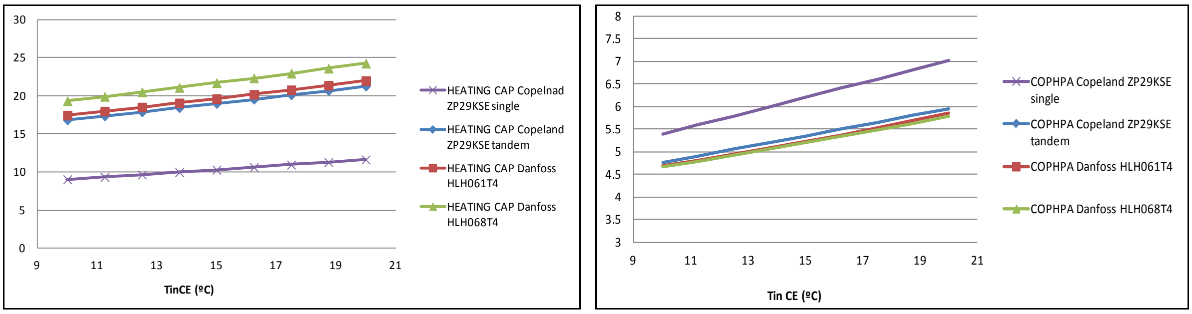

*Figure 4.:Influence of the outlet water temperature from the ground: a) Heating capacity b) hea pump COP in heating mode.*

According to the results presented in Figures 3 and 4, the optimal configuration of compressors would correspond to two tandem compressors manufactured by Copeland, model ZP29KSE.

## 3 DESIGN OF THE BRAZED PLATE HEAT EXCHANGERS: NUMBER OF PLATES

After selecting the optimal compressor model, the optimal number of plates for the condenser and the evaporator will be determined through a parametric study carried out in the IMST-ART software, setting the values of subcooling at 5K and superheating at 4K. The return water temperature from the fan coil units is 15 º C and the ground return temperature is 23 º C. As it is a water to water reversible heat pump, the number of plates must be the same in both heat exchangers. Results are shown below in Figure 5 for the selected compressor Copeland ZP29KSE in cooling mode.

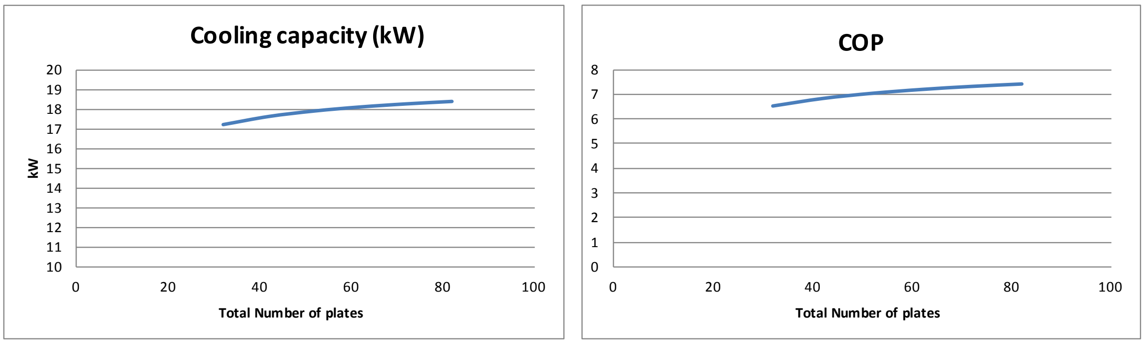

*Figure 5.: Total number of plates influence: a) Cooling capacity b) heat pump COP in cooling mode.*

As seen in Figure 5, the gain in both the cooling capacity and the COP of the unit is compensated by the increase in the number of plates up to 62 approximately, since the increase in performance shows an exponential tendency with the number of plates. From this moment, the greater investment in the number of plates is no longer justified by the increase in COP, because the tendency is more linear. Finally, in order to find a compromise with the requirements of the manufacturer of the heat pump (Hiref), the recommended optimal number of plates is 52.

## 4 SEASONAL PERFORMANCE FACTOR EVALUATION

The energy efficiency of a system is characterized by the performance factor, defined as the ratio of useful heat supplied to the building and the energy consumption during a time interval, dt. Depending on the integration period, the performance factor may be seasonal, monthly, daily ... etc. The most representative one is the seasonal performance factor (SPF) that evaluates the energy performance of the system during each season (winter or summer).

$$PF = \frac{\int_{t}^{t_0+\Delta t} \dot{Q}_{HP} \cdot dt}{\int_{t_0}^{t_0+\Delta t} \dot{W} \cdot dt} \quad (1)$$

Where

$\dot{Q}_{HP}$ is the heat pump capacity

$\dot{W}$ is the electrical power consumption considered

Figure 6 shows the results of calculating the average heaingt demand of the building to be air conditioned for each day in which experimental data were available at the facility during the year 2008-2009. Measurements are presented from September 2008 to September 2009. As shown in Figure 6, there are some days in which no representative data was available due to maintenance operations or because energy optimization strategies were carried out [Montagud, C., et.al., 2011] during those days.

The values of the thermal demand of the building have been calculated according to expression (2) along the day.

$$\int_{0}^{24hr} \dot{m} \cdot c_p \cdot \left(T_{outCI}(t) - T_{inCI}(t)\right) \cdot dt \quad (2)$$

Where,

$\dot{m}$ is the internal circuit flow rate measured by means of a coriolis flow meter

$c_p$ is the spacific heat of the water,4180 J/kg·K.

$T_{outCI}$ is the outlet water temperature at the internal circuit (building loop)

$T_{inCI}$ is the inlet water temperature at the internal circuit (building loop)

Taking into account the running time of the installation (15 hours) it is possible to determine the average daily heat demand as the ratio between the thermal load in kWh calculated by expression (2) along the day and the number of operating hours of the installation. These average values of thermal demand are presented in Figure 6.

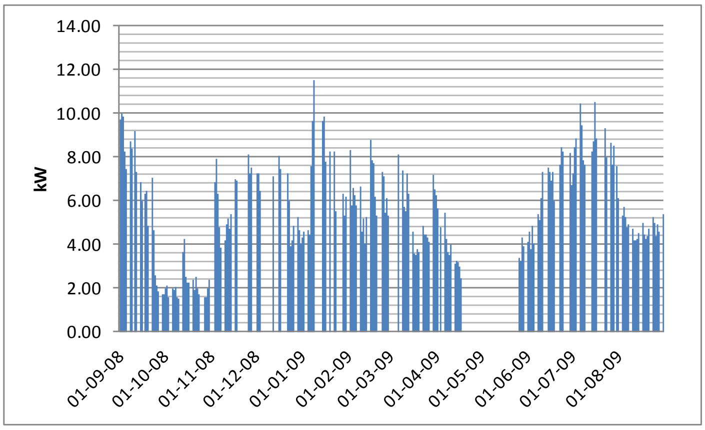

*Figure 6.: Thermal energy demand in the building: average values.*

From the data of thermal demand of the building to be air-conditioned shown in Figure 6, and setting temperature values for the inlet of the heat pump at both the internal and external circuits based on experimental measurements as shown in Table 3, it is possible to estimate the seasonal performance factor of the heat pump $SPF_1$ (only takes into account the consumption of the compressor) and the outer loop $SPF_2$ (considering the consumption of the compressor and the external circulation pump which sends water to the borehole exchanger), and calculate them in winter, summer and throughout the year for each compressor configuration considered. The results are shown in Figure 7.

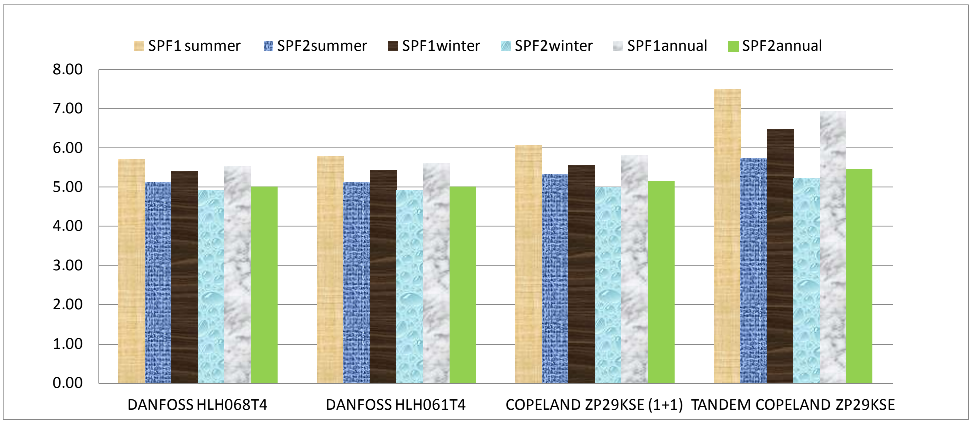

*Figure 7.: Evaluation and comparison of the seasonal performance factor for each configuration.*

It has been considered in the comparison that there are two possibilities to control the tandem compressors Copeland ZP29KSE. Thus, it is considered a first configuration wherein both compressors are running in parallel at a time as if it were an ON / OFF compressor (Copeland ZP29KSE (1 +1)), and a second configuration in which the two compressors operate in tandem.

As shown in Figure 7, values are highest in summer $SPF_1$ and minimum in winter. This is because the difference in temperature between the hot and cold source in the heat pump is higher in winter [40 ° C, 18 ° C] than in summer [25 ° C, 10 ° C] and, therefore, the compressor must overcome a higher pressure ratio in winter (heating mode), being therefore, the $SPF_1$ lower than in summer.

*Table 3: Water temperatures coming out of the ground source heat exchanger considered along the year.*

| Month | Treturn EC (ground loop) (ºC) |
|---|---|
| January | 15 |
| February | 16.25 |
| March | 16.25 |
| April | 17.5 |
| May | 23 |
| June | 25 |
| July | 26 |
| August | 24 |
| September | 27 |
| October | 23 |
| November | 18.75 |
| December | 17.5 |

When comparing the results presented in Figure 7 for the $SPF_2$ with $SPF_1$ values, it can be observed that including the external circulation pump consumption, causes a decrease of 10% average both for heating and cooling mode in the case of an ON / OFF compressor regulation; taking even greater values of the order of 20% decrease in the case of the configuration of compressors running in tandem. This is because, when the two compressors operate in tandem, the thermal demand of the building can be supplied with a single compressor stage during most of the time.

The result is a longer number of cycles and therefore a longer time during which the external circulation pump is turned on. Furthermore, when operating the second stage of the tandem compressors, the external circulation pump does not cycle and remains switched on during all the time, resulting in an increased energy consumption during the day and, therefore, the impact of its consumption in the seasonal performance factor is higher than in the case of the ON / OFF control. The latter is justified in Figure 8, which shows the comparison between the (%) of operating time for the two configurations considered ZP29KSE Copeland compressor, two compressors operating in parallel as a single compressor ON / OFF and two compressors in tandem for each one of the days considered.

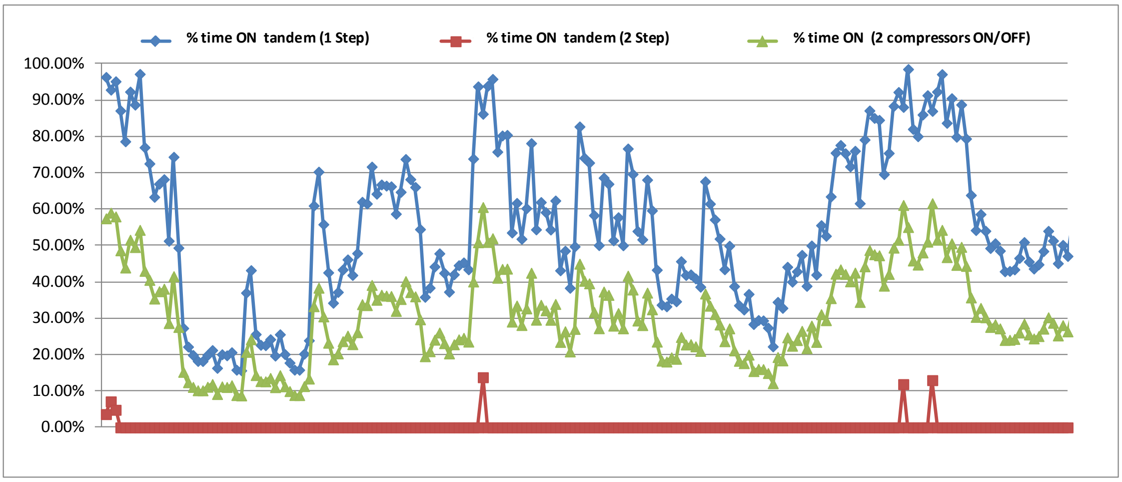

*Figure 8.: Comparison of the % time ON of the Copeland compressor ZP29KSE working with different types of control.*

As shown in Figure 8, in the case of the configuration of two compressors in tandem, a single compressor would be working during most of the time, because the thermal demand is less than or equal to the capacity of a single stage compressor. However, when operating the two compressors in parallel as a single compressor with an ON / OFF control, the time during which the heat pump would be turned on would take values of 60% at most, being therefore, the number of cycles throughout the day greater for the tandem compressors configuration and therefore, the consumption of the external circulation pump would be greater in this case, having a greater impact on the determination of seasonal performance factor $SPF_2$. Despite this fact, because the heat pump is more efficient in the configuration of two compressors in tandem, with the annual $SPF_1$ close to 7, the seasonal performance factor at the outer loop (including the external circulation pump consumption) is higher in this configuration.

Since the objective of Ground-Med project is to obtain a seasonal performance factor greater than 5 for the entire system, including all auxiliary consumption (compressor, internal and external circulation pump and fan coil units), it must be selected an alternative presenting an annual $SPF_2$ as high as possible considering the fact that the rest of auxiliary consumption of the installation will reduce the annual SPF entire system. One can conclude therefore that the best solution corresponds to the Copeland compressor ZP29KSE tandem.

## 5 CONCLUSIONS

The main objective of the Ground-Med project is to demonstrate the efficiency of ground source heat pumps (GSHP) in mild climates like the Mediterranean, with a seasonal performance factor (SPF) of the system greater than 5. In order to achieve this, it is necessary to design heat pumps as efficient as possible. A theoretical study based on the optimal selection of components for the design of a high efficiency GSHP was carried out.

A performance comparison of different models of compressors on the market was carried out using the simulation software IMST-ART. It is concluded that the optimum configuration for working pressure rates for the application under study (geothermal heat pump) is the model with two compressors Copeland ZP29KSE in tandem. Secondly, it is determined that the optimum number of plates in order to reach a compromise between performance and the heat pump manufacturer restrictions equals 52.

Finally, to assess the energy efficiency of the system over a year of operation, two coefficients are defined: seasonal performance factor of heat pump and $SPF_1$ and seasonal performance factor of the outer loop $SPF_2$. The difference between them is that the first considers only the consumption of the compressor, while the second considers both the consumption of the compressor and the external circulation pump. Based on the data of thermal energy demand of the building and taking into consideration the experimental measurements for the water return temperature from the ground loop in Valencia, the seasonal energy efficiency is determined for both winter and summer for each compressor configuration considered. Results show that the best solution corresponds to the use of two Copeland ZP29KSE compressors in tandem.

## 6 ACKNOWLEDGEMENTS

This work was developed in the framework of a European FP7 project “Advanced Ground Source Heat Pump systems for heating and cooling in Mediterranean climate (Ground-Med)”. The authors would like to thank the funding for the project by the European Commission.

## 7 REFERENCES.

Corberán J.M, Gonzálvez J., Montes P., Blasco R., 2002,"ART" A computer code to assist the design of refrigeration and A/C equipment. Werner Soedel, Eckhard A.Groll; PURDUE CONFERENCES: 94-95. http://www.imst-art.com/.

Fahlén P,Karlsson F, 2005:”Optimizing And Controlling Media Flows In Heat Pump Systems”, 8th IEA Heat Pump Conference, Las Vegas, USA.

Karlsson F, Fahlen P,2007: “Capacity Controlled Ground Source Heat Pumps in Hydronic Heating Systems”, Int. Journal of Refrigeration, 30(2):221-229.

Montagud C., Corberan J.M., 2010. : “Ground-Med project: Advanced Ground Source Heat Pump systems for heating and cooling in Mediterranean climate”. GEOENER 2010, II Congreso de Energía Geotérmica en la Edificación y la Industria.

Montagud C., Corberan J.M., 2010.: “Optimización energética de una instalación de bomba de calor acoplada al terreno para calefacción y refrigeración en un edificio de viviendas”. GEOENER 2010, II Congreso de Energía Geotérmica en la Edificación y la Industria.

Montagud C., Corberan J.M, Montero Á., Urchueguía J.F., 2011: “Analysis of the energy performance of a Ground Source Heat Pump system after five years of operation”. Energy & Buildings, 43:3618-3626.

Urchueguía J.F., Zacarés M., Corberán J.M., Montero A., Martos J., Witte H., 2008: “Comparison between the energy performance of a ground 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, 2917-2923.

Project GROUNDMED is supported by the European Union
within the 7th framework program