SEASONAL SYSTEM PERFORMANCES FOR VERTICAL GROUND-COUPLED HEAT EXCHANGERS AND COMPARISON WITH AIR-TO-WATER HEAT PUMP SYSTEMS

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ABSTRACT

To make a global assessment of a Ground Coupled Heat Exchanger (GCHE) it is not only important to understand the behaviour of the GCHE, but also to consider the system in which it will operate: its loads and utilization factors (as a function of climate conditions and application), efficiency (which also depends on the heat pump) and other system parameters, such as pumping requirements, long term soil heat imbalance, etc.

In the frame of the European project GEOCOOL, this paper shows the results of applying a methodology developed for the comparative study between a system combining a water-to-water reversible heat pump of commercial size with a vertical GCHE and an equivalent air-to-water heat pump system in typical conditions of the European Mediterranean rim (of great importance for Cooling). For this purpose, the seasonal system performance factors for heating and cooling, and the temperature profiles of the water in the GCHE for a 25 years period were calculated for different backfill materials. In addition, an extensive study of relevant climatological parameters of Valencia-Spain was made. These results were transformed in bin-hour data which are used for calculating the seasonal system performance factors for heating and cooling for the air-to-water heat pump system. The heat pump properties have been calculated using the IMST Group's ART software.

Finally, a comparison was made between GCHE-system and air-to-water heat pump showing the efficiency improvement obtained for various grouting materials.

Key Words: *cooling, heating, heat pump, ground coupled heat exchanger, performance*.

1   INTRODUCTION

Energy is one of the greatest supports for the human development and the improvement of life quality. Energy is not an aim itself, but the way of achieving the target of sustainable human development. At the middle of last century, it was thought that the economical prosperity had a parallel behaviour with energy consumption. Nevertheless, year by year several published reports have shown how higher economical growths are possible with lower energy consumptions. This situation has been demonstrated in many countries as United States, Japan and European Union.

If there is a rational use of the energy, then there will be many possibilities to have a society based on high technology with low energy consumption. The tendency shows that society is using more

efficiently goods and services. Energy savings have always been present in regulation, education, market profit, investigations and development.

Not only a new scenery which points at the efficient use of the energy is required for a development of the energetic systems, but also a higher compromise in the research of new and more efficient energy sources.

As it can be inferred from the last paragraphs, it is necessary to search for other ways of maintaining the nowadays standards of living, when referring to technical and energetic requirements, heating and cooling systems in buildings which are indeed a very important aspect of the energy consumption. Within this context, the energy storage by means of heat, ground storage of thermal energy (UTES-Under Ground Thermal Energy Storage), has had a growing development during the last years.

The use of the ground capability for air conditioning, cooling and heating, has reached a growing acceptance during the last years due to the economic benefits obtained comparing to other conventional systems. Depending on the geographic localization, heat pump systems with a ground heat exchanger can show an improvement in the efficiency of the system of 35% in heating mode compared to the conventional air source heat pumps systems. This value reaches up to 40-60% in cooling performance. As a consequence of this, the IMST[1] team of UPV has been developing this relatively new technology which contributes towards energy efficiency. The IMST works in association with other national and European institutions, and it is the main member in the European project GEOCOOL, whose main aim is to show the advantages of using Ground Heat Exchangers as an energy saving technology in Mediterranean Europe.

2   DESIGN PARAMETERS

Based on the energy demand of the GEOCOOL prototype facility at the UPV, the performance data of the heat pump and the local geology and geo-hydrology, have resulted in a design consisting of six boreholes in rectangular configuration 3X2, to a depth of 50 meters. When we consider the application of the GEOCOOL concept for the whole region, it is obvious that the size and construction of the borehole heat exchanger may differ between regions, e.g. due to the need to use an anti-freeze solution or due to legislation requiring specific backfill materials. To be able to define proper sizing rules and to extent the theoretical work it has been decided to construct different borehole configurations for the GEOCOOL experiment. The following alternative constructions will be implemented: backfill with course sand with spacers, backfill with fine sand without spacers, backfill with fine sand with spacers, backfill with bentonite10% in water with spacers and backfill with bentonite12% mixed with fine sand with spacers.

Other parameters used in the pre-design process are: GROUND (test IN-SITU): Ground thermal conductivity: 1.6 W/m·K, Volumetric heat capacity: 2.4MJ/m^3·K and Ground surface temperature: 18.5 °C. BOREHOLE: Configuration: 6: 3 x 2, rectangle, Borehole depth: 50 m, Borehole spacing: 3 m, Borehole diameter: 0.14 m, U-pipe diameter: Polyethylene PE100, DN 1 ¼", PN 10. HEAT CARRIER FLUID: Water, HEAT PUMP: IZE70 (CIATESA, 2001), THE CIRCULATION PUMP: CH 4-20 (Grundfos).

The building is a set of spaces in the Departamento de Termodinámica Aplicada in UPV with a total surface of approximately 250 m^2. This area includes a corridor, nine offices, a computer classroom, and a room with copiers and a coffee dispenser. Apart from the corridor all spaces are equipped with one fan coil, the computer class with two fan coils. In the Table 1 below an overview is given of the maximum loads calculated for each one of these spaces, except for the corridor.

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[1] IMST- Investigación y Modelado de Sistemas Térmicos

The Figure 1 shows the configuration of the air-conditioned spaces and the location of the fan coils (black rectangles).

Table 1. Maximum loads calculated for the building studied

Space	Maximum heating load (kW)	Maximum Cooling load (kW)
Office 1 person	2,05	1,66
Office 2 persons	1,17	0,92
Office 1 person	1,18	0,99
Office 1 person	1,20	1,02
Office 2 persons	1,20	1,02
Office 1 person	1,36	1,20
Office 1 person	1,37	1,23
Office 1 person	1,09	0,87
Office 1 person	1,78	1,62
Computer Classroom	5,74	2,00
Copiers and Coffee dispenser	3,39	2,00

Additionally, a study of heating and cooling loads for the GEOCOOL building was done (Figure 2). It is important to realize that the load calculated for the month of August is zero corresponding to the vacation period of the university.
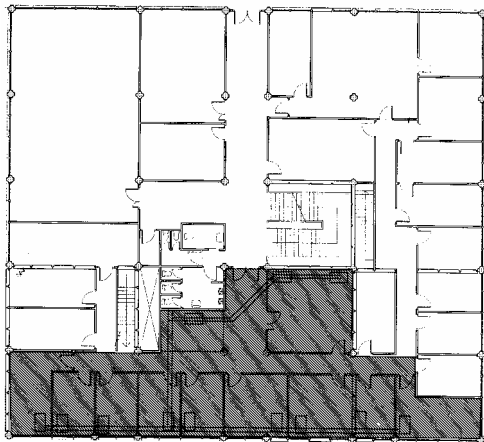

Fig. 1. Air-conditioned spaces and the location of the fan coils (black rectangles).
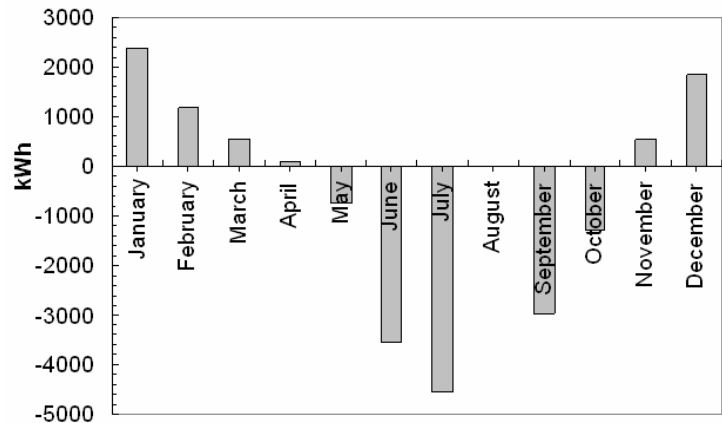

Fig. 2. Load Profile GEOCOOL Building

It can be seen in Figure 2 that the values of the thermal loads are given in kWh, being positive for heating requirements and negative for cooling requirements. The software used for the evaluation of the heating and cooling load profile is CALENER (2001), a package made to characterise building in Spain.

3   RESULTS AND DISCUSSION

3.1 Evolution of the water temperature in the GCHE

Figure 3 shows the average temperature profile of the water inside the ground heat exchanger for a 25 years period. . The design parameters given above are used to do this analysis, using 10% bentonite in water as the backfill material. Results shown here were obtained using two different software packages, EED (2000) and GLHEPRO (2000). It can be seen that the average temperature of the water over the years is increased by 2.6 ºC at the end of the 25 year period because the annual cooling load is higher than the annual heating load.
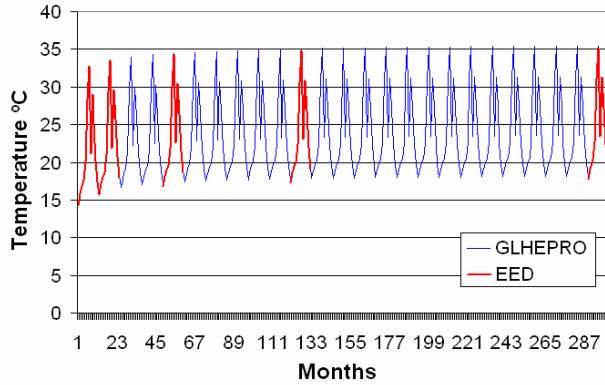
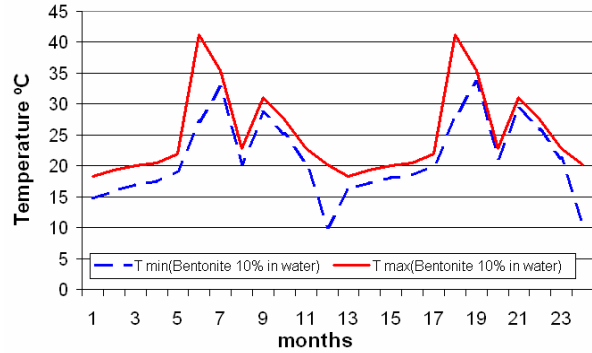**Fig. 3. Average temperature profile of the water in the GCHE**

Fig. 4. Maximum and minimum temperature profile of the water in the GCHE (years:1-2 for Tmin & 24-25 for Tmax)

Figure 4 shows the minimum temperature profile of the water (for the first and second year of the period) and the maximum temperature profile (for the 24th and 25th years). Results shown are obtained considering an 8 hours peak load (common/normal period of work at the university) and bentonite10% in water as backfill material. However, the same calculations were made considering different backfill materials (mentioned before) and similar results were obtained. Maximum temperatures obtained with other backfill materials are lower than those obtained with bentonite10% in water as well as minimum temperatures are higher in the case of other backfill materials than using bentonite10% in water, being this last material the one which offers/shows the higher thermal resistance to the heat exchange with the soil.
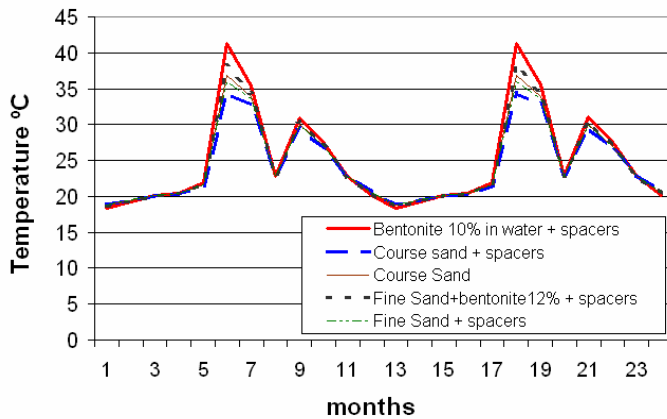

Fig. 5. Peak temperature profile for the water in the GCHE for 5 kinds of backfill materials(year 24 & 25)

It is also observed that intermediate values among those mentioned above are obtained with other backfill materials for the same design conditions. Another very important factor is the use of spacers; for example, the maximum temperature reaches 36.86°C for coarse sand without spacers in the 25th year, which is 2.54°C higher than the temperature obtained with the same backfill material but using spacers (Shank Spacing = 0.083m).

Figure 5 shows the peak temperature profile for the 5 different backfill materials considered for the 24 and 25th year, when the highest temperatures are reached. The Figure 6 also shows that 10% bentonite in water is the backfill material which results in the highest temperatures, reaching 41.27°C in the 25th year supposing an 8 hour peak load. The analysis shows that the best behaviour with respect to the highest temperature of the water is achieved with coarse sand with spacers (SS=0.083m) as a backfill material where the temperature reaches the maximum of 34.22°C in the 25th year. This demonstrates the importance of the thermal conductivity of the backfill material in the installation design. The thermal conductivity of the 10% bentonite in water is 0.7 W/mK while this value rises to 2.1 W/mK for the sand.

4

3.2 Estimation of seasonal system performances for the GCHE of the GEOCOOL system

The Seasonal Performance Factor is the ratio between the thermal energy provided to/extracted from the building (heating/cooling thermal load) and the supplied electrical energy used for heating or cooling in a determined season. We distinguish between the HSPF - Heating Seasonal Performance Factor, as the performance factor of the system in winter, and the CSPF - Cooling Seasonal Performance Factor for the summer.

In the calculation of the SPF all energy inputs of the system must be taken into account, such as heat pump consumption, circulation pumps, blowers, etc. In this section the results obtained for the GEOCOOL project will be described, and HSPF and CSPF will be calculated for different borehole configurations and for different backfill materials. Moreover the study will compare the performance of the GCHE system with an equivalent air-to-water heat pump system, calculating HSPF and CSPF and showing the enhancements of the GCHE system compared to the air-to-water heat pump system.

A sensitivity analysis of the different options in the design of the GEOCOOL GCHE system has been done with the help of CALENDER, EED, GLHEPRO, ART (2000), and other software tools. The parameters used for this analysis are those given in section 2. Also the heat pump and circulation pump properties are taken in account.

Figure 6 shows the interaction between the different software packages that were used to model the global GEOCOOL system.
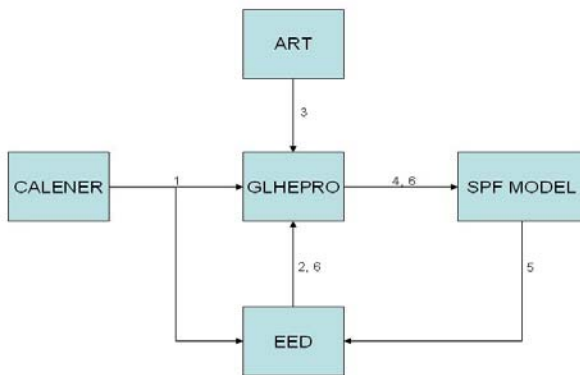

Figure 6 shows that the first step in the modelling process is to calculate the GEOCOOL building's load profile (1). To make this calculation the software package CALENER was used, entering the thermal load data of the building and the design parameters mentioned in section 2. EED calculates the effective thermal resistance of the borehole (2), and ART is used to model the heat pump (3).

Fig. 6. Interaction between the different software packages

The electricity consumption is calculated using GLHEPRO (4), and finally the SPF model calculates the SPF both for heating and for cooling for a period of 25 years (5). This result is introduced into EED again as one of the design parameters, in order to recalculate the effective thermal resistance of the borehole; this iterative process leads to a higher precision in the final result. EED and GLHEPRO calculate the water temperature in the GCHE for a period of 25 years (6).

3.2.1. The Heat Pump (IZE70)

As was mentioned before, the selected heat pump is CIATESA's IZE70, which is a reversible water-to-water heat pump equipped with a scroll compressor and using R-407c refrigerant. Thermal capacity and power curves are shown below.

Data used in the curves below (see figures 7, 8, 9 and 10) are taken from the manufacturer's catalogue. The team of researchers also made their own measurements of these properties and checked

them with the catalogue information. Additional data were calculated with ART (Advanced Refrigeration Technologies, of IMST-group, IIE, UPV) software.
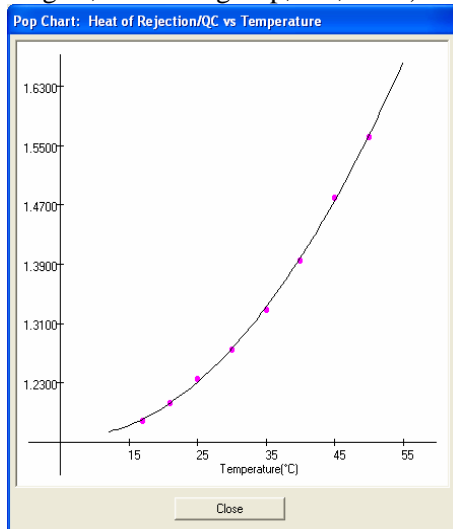

Fig. 7. Cooling capacity of the HP/Cooling Load vs entering water temperature at the HP.
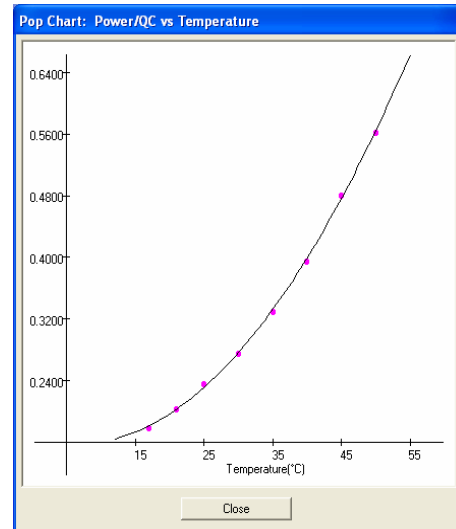

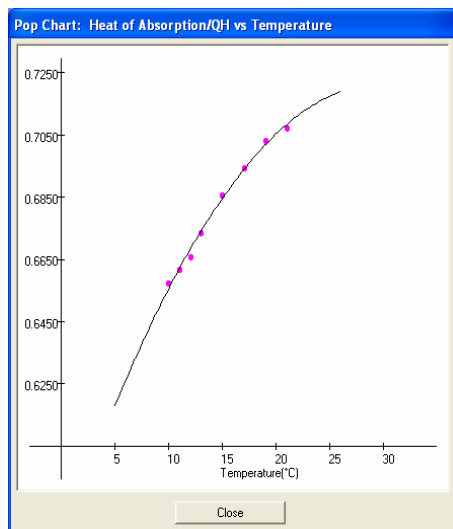**Fig. 8. Electrical power consumed by the HP/Cooling Load vs entering temperature at the HP.**

Fig. 9. Heating capacity of the HP/Heating Load vs entering water temperature at the HP.
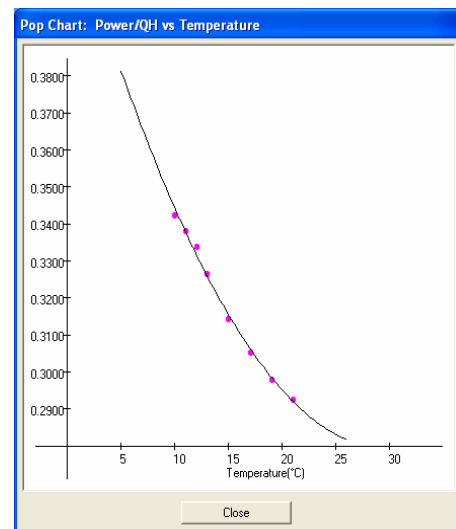

Fig. 10. Electrical power consumed by the HP/Heating Load vs entering temperature at the HP.

The characteristic curves for the HP are given for a water flow rate of 2.9 m^3/h and specific temperature conditions in the interior circuit in the building, therefore:

For summer (cooling):
Hot Temperature = load side entering water temperature at heat pump = 12 ºC
Cold Temperature = load side temperature of the water leaving the heat pump = 7 ºC

For winter (heating):
Hot Temperature = load side temperature of the water leaving the heat pump = 50 ºC
Cold Temperature = load side entering water temperature at heat pump = 45 ºC

3.2.2. The Circulation Pump

For the design of the GCHE system a Grundfos' CH 4-20 centrifugal circulating pump was selected based on a hydraulic study of the system.

The electrical power consumption of the pump for a 2.9 m³/h water flow is 0.479 kW, which must be added to the electrical consumption of the heat pump in order to obtain the total consumption of the system.

3.2.3. SPF calculation for the GCHE system

Once the design parameters are defined and introduced in the corresponding software EED and GLHEPRO, the average, minimum and maximum temperature profiles of the water in the GCHE for a 25 years period are obtained, as well as the monthly average value of the heating/cooling energy (winter/summer) and the consumed energy of the system (heat pump + circulating pump) for the same period of time.

The Heating Seasonal Performance Factor, HSPF [kWh/kWh], is defined as:

$$HSPF = \frac{\sum_{i=1}^{n} \dot{Q}H_i}{\sum_{i=1}^{n} \dot{E}H_i} \qquad [1]$$

where: $\dot{Q}H_i$ is the heating thermal load for the month i (in kWh)

n is the amount of heating months per year

$\dot{E}H_i$ is the electrical power consumed by the system in the month i (in kWh)

Similarly, the Cooling Seasonal Performance Factor, CSPF [kWh/kWh], is determined.

Figure 11 shows the results obtained for the HSPF and CSPF for a GCHE system with a rectangular borehole configuration 3x2, using spacers (SS=0.083m) and using bentonite10% in water as backfill material.
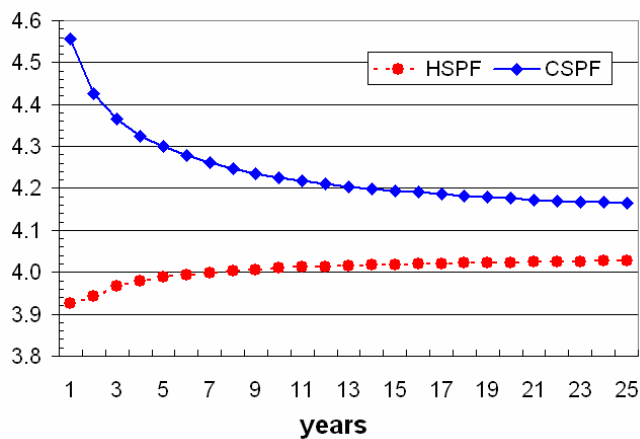

Fig. 11. HSPF and CSPF for the GCHE system with bentonite10% and rectangular configuration 3x2.

According to the model the HSPF value remains almost constant from the 6th year on, with a value close to 4.01, depending on the heat pump and circulating pump behaviour. These devices must be selected in such a way that the maximum thermal yield is obtained with a minimum electrical consumption for the selected design conditions. It can be seen how the HSPF raises in time due to the warming up of the soil, caused by the bigger load in cooling than in heating. The expected value for the HSPF reaches its minimum of 4.16 after 25 years.

The values of CSPF are higher than those of HSPF, this is a consequence of the fact that the heat pump in heating mode works in hot water conditions of 45/50ºC

3.2.4. Modelled values for HSPF and CSPF for a system with a GCHE with different grouting materials

In this section we will comment on the HSPF and CSPF values found for the GEOCOOL concept; we will compare these parameters for 5 different grouting materials, in order to show which grout is more favourable. (Total borehole length = 300 m)

Figure 12 shows that 10% bentonite in water gives the lowest HSPF, while coarse sand with spacers gives the best. The difference between both is 0.0154 after 25 years. Figure 13 also shows that 10% bentonite in water gives the lowest CSPF again, while coarse sand with spacers gives the best. The difference between both is more pronounced than in the case of the HSPF, being 0.1541 after 25 years.

The values of CSPF are higher than those of HSPF, this is a consequence of the fact that the heat pump in heating mode works in hot water conditions of 45/50ºC.
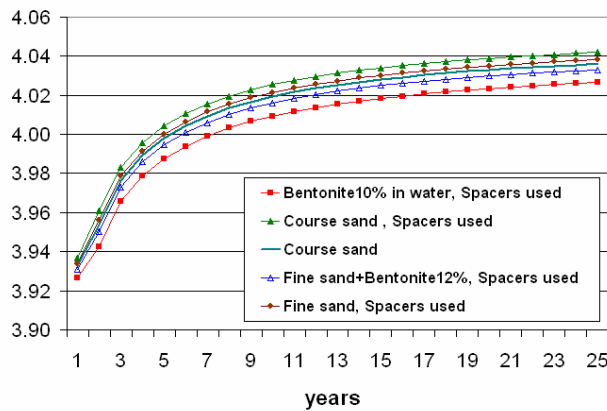

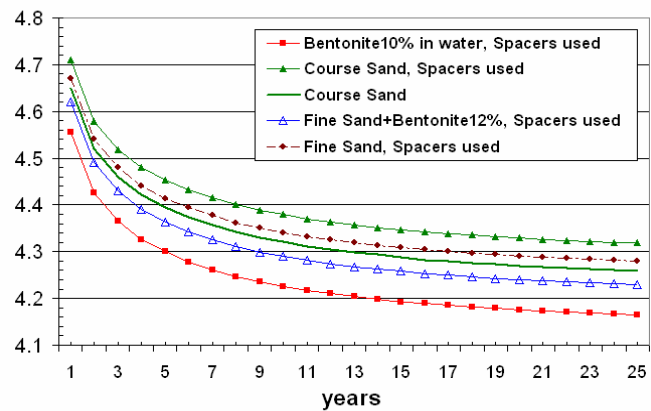**Fig. 12. HSPF for different grouting materials in a rectangular configuration (3x2)**

Fig. 13. CSPF for different grouting materials in a rectangular configuration (2x3)

3.3 Seasonal Performance Factor (SPF) for the Air-to-Water Heat Pump

The Seasonal Performance Factor for the air to water heat pump is the ratio between the thermal energy supplied to or extracted from the building (heating or cooling load), and the electrical energy consumed, in a certain season. In this case the devices that use energy are the heat pump's compressor and the axial fan.

In this section we will describe the different elements in the air to water heat pump, the computing methodology and the results obtained for the GEOCOOL project.

3.3.1 The air to water heat pump

The selected heat pump is a reversible IWD 80s, manufactured by CIATESA, featuring a scroll compressor and using R-407c as refrigerant with a cooling capacity of 15.9 kW and heating capacity of 18 kW. The calculations below are based on data from the manufacturer's catalogue, which have been checked using ART software and found reliable. (Table 2)

Table 2. Nominal characteristics of CIATESA´s IWD 80s heat pump

Series IWD	80s
evaporator capacity (1) (kW)	15.9
electrical power demand (C)(3) (kW)	6.9
condenser capacity (2) (kW)	18
electrical power demand (H)(4) (kW)	6.6

(1) evaporator capacity for water leaving the heat pump at 7 ºC and an exterior air temperature of 35 ºC
(2) condenser capacity for water leaving the heat pump at 50 ºC and an exterior air temperature of 6 ºC
(3) compressor's and fan's joint electrical power demand in nominal cooling conditions
(4) compressor's and fan's joint electrical power demand in nominal heating conditions

3.3.2 Climate data

A method to represent the temperature profile for a certain area in a certain period is the bin-hours method. It consists of adding up the number of hours that the outside temperature lies within a certain range, and repeating this calculation for all temperature ranges that may occur. In this way a temperature histogram is obtained. The information needed to make this histogram is a database of hourly temperature observations in the study area. In the case of Valencia, the Instituto Nacional de Meteorología (Spanish National Meteorological Institute) supplied the raw data for the years 2000, 2001 and 2002.

Documentation

The data furnished by the INM (1996) were compared with data gathered in the Climatic Atlas of Valencia (Atlas Climático de la Comunidad Valenciana). This book contains tables with absolute minimum temperatures, means of minima, means, absolute maximum temperatures and means of maximums for each month in the period 1961-1990. A period of 30 years is considered as a statistically representative period, therefore these values can be considered as the expected values for Valencia. On the other hand data were available for the INM website (http://www.inm.es), where means of minima, means and means of maximums are shown for the period 1971-2000, for each month.

Comparison with data gathered in the Climatic Atlas of Valencia

The "Atlas Climático de la Comunidad Valenciana" unfortunately doesn't give data variance. It does however give information about the mean values, but the mean is defined as follows:

"The annual mean is normally calculated on basis of daily means, which are the average values of the daily maximums and minimums. Other methods to estimate the mean daily temperature exist as well, based on a continuous record of temperatures during the day or regular temperature measurements during the day."

This way of calculating the mean daily temperature, and therefore the mean monthly temperature, does not correspond to the method explained in the previous chapter. Since the histograms are not symmetrical, the average of the minimum and maximum temperatures T_{med} will be higher than the mean temperature based on 24 daily observations. Indeed, for the period 2000-2001-2002 we find important differences, especially in the months of January.

The question is: Do T_{med}, $\overline{T}_{min}$ and $\overline{T}_{max}$ for the period 2000-2002 correspond to the equivalent values for the period 1961-1990?

According to the atlas T_{med} for January is 11.5ºC. The mean minimum and maximum temperatures are respectively 7 and 15.9ºC. The average value of the T_{med} for 2000, 2001 and 2002 is the value that comes closer to 11.5ºC than any T_{med} in these three years. Therefore the best way to model the temperatures in January in Valencia is taking the average values for these three years.

According to the atlas T_{med} for July is 24.3ºC. The mean minimum and maximum temperatures are respectively 20.5 and 28.7 ºC. However the months of July during 2000, 2001 and 2002 were all warmer; the most similar year was 2002. Therefore the best way to model the temperatures in July in Valencia is taking the values of the year 2002.

3.3.3 SPF Calculation for an Air to Water Heat Pump

The American Refrigeration Institute (ARI, 2003) has determined that the frequency distribution of temperatures over the summer cooling season is roughly the same across the country. However, in warmer, southern climates, there are more "cooling load hours", which are defined as the hours when the outdoor temperature is above 65ºF(18.3ºC)[2], per year than in cooler climates. In Atlanta, for example, the number of cooling load hours is approximately 1300 hours per year, while it is only about 700 hours per year in Cleveland, OH. Of these hours, the outside temperature will be between 80 ºF(26.66 ºC) and 84 ºF(28.88 ºC) approximately 16.1% of the cooling season in either city. Table 3 shows the distribution of the cooling load hours.

Table 3. Distribution of cooling load hours

Bin Number	Temperature Range(ºF)	Representative Temperature (ºF)	Fraction of total temperature hours
1	65 – 69	67 (19.44ºC)	0.214
2	70 – 74	72 (22.22ºC)	0.231
3	75 – 79	77 (25ºC)	0.216
4	80 – 84	82 (27.77ºC)	0.161
5	85- 89	87 (30.55ºC)	0.104
6	90 – 94	92 (33.33ºC)	0.052
7	95 – 99	97 (36.11ºC)	0.018
8	100 - 104	102 (38.88ºC)	0.004

The COP changes with the outside air temperature and the SPF, for an air conditioner depends on the temperatures at which the appliance runs over an entire year. According to the ANSI/ASHRAE standard (1995)[3].

The Cooling Seasonal Performance Factor, CSPF [kW/kW], is determined by:

$$CSPF = \frac{\sum_{i=1}^{8} \dot{Q}C_{(T_i)}}{\sum_{i=1}^{8} \dot{E}_T C_{(T_i)}} \qquad [2]$$

$$\dot{E}_T C_{(T_i)} = \dot{E}_c C_{(T_i)} + \dot{E}_f C_{(T_i)} \qquad [3]$$

where: $\dot{Q}C_{(T_i)}$ adjusted evaporator capacity at ambient temperature Ti

$\dot{E}_T C_{(T_i)}$ adjusted electrical power demand (compressor + fan) at ambient temperature Ti

$\dot{E}_c C_{(T_i)}$ adjusted electrical power demand (compressor) at ambient temperature Ti

$\dot{E}_f C_{(T_i)}$

[2] ARI Standard 210/240-2003, Unitary Air-Conditioning and Air-Source Heat Pump Equipment. Air-Conditioning & refrigeration Institute. Arlington, Virginia, sect. 5.1, p20-27.

[3] ANSI/ASHRAE 116-1995, Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps. American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc. Atlanta, GA, 1995 p23-35.

adjusted electrical power demand (fan) at ambient temperature Ti

$( )_i$ values corresponding to temperature bin i (from Table 3)

Similarly, the Heating Seasonal Performance Factor, HSPF [kW/kW], is determined.

3.3.4 Calculated HSPF and CSPF for the IWD 80s heat pump in the GEOCOOL concept

After having analysed all climatic information, calculated the bin-hours for Valencia, modelled the IWD 80s heat pump with ART, calculated the heating and cooling load for the building of the GEOCOOL project and applying the methodology proposed in the Standard ANSI/ASHRAE 116-1995 and ARI Standard 210/240-2003, the following results were obtained (see Table 4):

Table 4. SPF values for CIATESA´s IWD 80s heat pump

Year	HSPF	CSPF
2000	2.96	2.81
2001	2.96	2.81
2002	2.96	2.83
Average	2.96	2.82

The HSPF values are very similar for the three years, having an average value of 2.97. The CSPF the 2002 has a slightly higher value than the other two years. According to the climatic study of Valencia, 2002 was the most representative year if you compare it with the last 25 years, while the most representative value for HSPF is the average of all three years.

3.4 Comparison between GCHE-system and Air-to-Water Heat Pump

This study compares two equivalent systems: the GCHE-system and an Air-to-Water Heat Pump with similar capacities in heating (18 kW) and cooling (16 kW). The heat pumps where selected because they use similar technology, both have a vapour compression cycle using R-407c as a refrigerant, a Scroll compressor and plate heat exchangers at the application side. The main difference between the two heat pumps is the normal working temperature of the outside circuit. In summer, the water in the external circuit will have a lower temperature than the air that cools the air-to-water heat pump, while in winter it is warmer than the air used to carry heat to the air-to water heat pump. Therefore the COP of the water-to-water heat pump will be higher in both modes of operation.

With the SPF calculated for the GCHE system (see 3.2.4.) and for the Air-to-water Heat Pump (see 3.3.4.), the results can be compared to quantify the efficiency gain obtained compared to a conventional air-to-water system.

Efficiency improvement obtained for various grouting materials
In order to quantify the improvement of the GCHE versus a conventional air-to-water heat pump, heating and cooling SPF values of several years were calculated. The results are shown below:
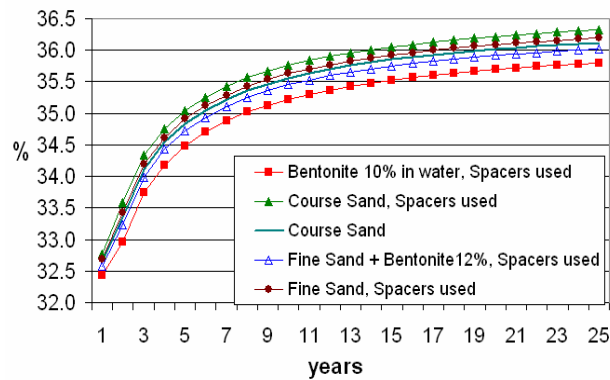
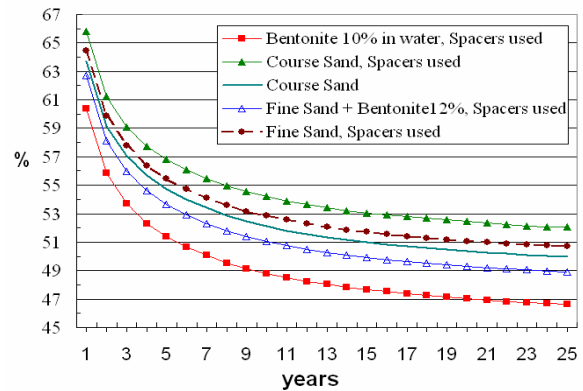**Fig. 14. Heating Efficiency Improvement GCHE vs. air-to-water Heat Pump for various Grouting Materials**

Fig. 15. Cooling Efficiency Improvement GCHE vs. air-to-water Heat Pump for various Grouting Materials

Figure 14 shows the percentage efficiency gain in heating mode of a GCHE system vs. an Air-to-Water Heat Pump, for various grouting materials. The grout that gives the highest improvement is coarse sand, followed by fine sand mixed with 12% bentonite; while 10% bentonite10% in water gives the least improvement. The average improvement over time of the GEOCOOL concept using coarse sand with spacers as grouting material is 35.4%.

Figure 15 shows the percentage efficiency gain in cooling mode of a GCHE system vs. an Air-to-Water Heat Pump, for various grouting materials. The grout that gives the highest improvement is coarse sand, followed by fine sand mixed with 12% bentonite; while 10% bentonite in water gives the least improvement. The average improvement over time of the GEOCOOL concept using coarse sand with spacers as grouting material is 52.6%.

4   CONCLUSIONS

The advantages of the use of ground coupled heat pumps compared to conventional air source heat pumps were shown to be an energy saving technology in the European Mediterranean area.

In particular the theoretical improvement in the seasonal coefficient of performance for the heating season has been shown to be about 32-36% and the improvement in the seasonal coefficient of performance for the cooling season to be 50-60% over a 25 year period of operation.

Other advantages of GCHE system compared to air source in a costal region are primarily that the ground source heat pumps will not be affected by salt corrosion, the lower noise level, the lower maintenance cost because the heat pump is placed indoors, the lower visual impact and the reduction in peak electrical requirements.

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