Purdue University
Purdue e-Pubs

International Refrigeration and Air Conditioning Conference
School of Mechanical Engineering

2012

Energy Efficiency in a Ground Source Heat Pump With Variable Speed Drives

Davide Del Col
davide.delcol@unipd.it

Giacomo Benassi

Mauro Mantovan

Marco Azzolin

Energy Efficiency in a Ground Source Heat Pump with Variable Speed Drives

Davide DEL COL[1]*, Giacomo BENASSI[2], Mauro MANTOVAN[2], Marco AZZOLIN[1]

[1]Università degli Studi di Padova, Dipartimento di Ingegneria Industriale,
Via Venezia 1 – 35134 Padova, Italy
Phone: 049 8276891, E-mail: davide.delcol@unipd.it

[2]Hiref SpA, Viale Spagna, 31/33 – 35020 Tribano (PD), Italy

* Corresponding Author

ABSTRACT

A variable capacity heat pump, with variable speed drives on the secondary circuits, needs an adequate control for gaining the maximum energy efficiency. In the present paper, a model of a ground-source heat pump with variable speed compressor, variable speed water pumps and variable speed fans in the coils is presented. This model is used to develop a control strategy which is implemented in the GSHP (ground source heat pump) system installed at Hiref SpA (Italy) in the framework of the European Project Ground-Med.

The goal is to maximize the overall coefficient of performance, accounting for energy inputs to the compressor and the auxiliary components. The control parameters of the model that can be varied are the followings: frequency of compressor, frequency of water pump to the borehole heat exchangers, frequency of the water pump to the user, velocity of the fans, water temperature to the user.

The present model allows to evaluate the operating conditions that lead to the maximum seasonal coefficient of performance of the system.

The present model is also the baseline for the development of the strategy control, with the final objective of maximizing the seasonal coefficient of performance of the system. In order to make the system operate at optimal conditions when the load varies, a primary and secondary control has been implemented. The temperature of the tank is used in a primary controller to vary the heat output of the heat pump, depending on the difference between the measured temperature of the tank and the setpoint value. The ambient temperature is used in a secondary controller to vary the heat capacity of the fancoil unit as a function of the difference between the measured and the ambient setpoint temperature.

1. INTRODUCTION

Heat pumps operate intermittently in the on/off mode in order to adjust the heating and cooling capacity to the load required by the building. When the heat pump operates in this way to provide the heat needed during the entire cycle it has to supply heat at higher temperature during the on cycle. On the contrary, if a variable speed compressor is used, the heat pump could simply follow the load, taking advantage at partial loads from the "oversized" heat exchangers and thus increasing its efficiency.

Regarding the best approach to match the output capacity of a ground source heat pump with the load required by the building, Zhao *et al*. (2003) proposed a theoretical and experimental analysis. They considered several methods for capacity control such as turning on/off the compressor, controlling intake and discharge valves on/off times, varying composition of the refrigerant mixture and varying the compressor speed. After the theoretical analysis they chose the latter method for the experimental analysis. Their tests focused on the influence of the frequency (compressor speed) on the COP at different water tank temperatures. While heat and cool capacity increased almost linearly by increasing frequency, COP presented different trends depending on water tank temperature.

Munari *et al*. (2011) compared the energy performance of variable-capacity geothermal heat pumps against the performance of on/off controlled equipment. The comparison between variable speed compressor and on/off controlled equipment was carried out by taking into account three major aspects: the temperature level of the heated water, the effect of the performance of the heat exchanger at partial loads, and the compressor global efficiency. The

electrical efficiency of asynchronous motors at variable speed was compared against the efficiency of permanent magnet brushless motors. The last two points (heat exchangers performance at part loads and compressor efficiency) were found as the major aspects in the comparison.
Recently Madani *et al.* (2011a) developed a model for capacity control in ground source heat pump systems. The model is composed of several sub-models: the variable speed compressor sub-model, the condenser and evaporators sub-models, the building and ground heat source sub-models. The model is developed in order to compare the seasonal performance of different control strategies at different systems layouts finding a balance between complexity and simplifications. For example, the variable speed compressor sub-model is modeled in detail by taking into account electromechanical losses, built-in volume ratio, internal mass leakage coefficient and inverter losses. They also performed a comparison of the model with experimental results from two variable speed GSHP systems, both located in Stockholm. This model was used in Madani *et al*. (2011b) to carry out simulations in order to make a comparison between the annual performance and COP of on/off controlled and variable capacity systems. They considered five ground source heat pumps, four on/off controlled and designed to cover different percentages of heat peak demand of the building, and one with variable capacity system. At different values of ambient temperature the five heat pumps showed different performance. This study indicates that if the on/off controlled systems are dimensioned to cover more than 65% of the building peak demand and approximately more than 95% of the annual heat demand, there is no considerable difference with the variable capacity system. If the on/off controlled heat pump is oversized (94% of the building peak heat demand), the annual energy consumption is the lowest but economic constraints prevent the system to be the best option; if the on/off controlled heat pump is dimensioned to cover only the 55% of the peak demand, the large use of electrical auxiliary heater makes the highest annual energy consumption and the SPF of the heat pump be lower than the variable speed one.
A steady-state model for variable speed heat pump for a wide range of cooling conditions and loads was developed by Zakula *et al.* (2011). This model is composed by three components (evaporator, compressor and condenser) sub-models within a main solver loop. An idealized expansion valve has also been assumed and variable heat transfer coefficients and pressure drops in the components and in suction and discharge lines have been taken into account. Simulations were compared within experimental data at different values of condenser inlet air temperature and evaporator inlet air temperature.
A comparison between on/off compressor control and variable speed control for a brine-to-water heat pump was conducted by Karlsson and Fahlen (2003). The main benefit that emerges from the study is that by using a variable speed compressor, the need for supplementary heating is reduced. Recently, the same authors (Karlsson and Fahlen, 2007) investigated the issue in domestic ground source heat pumps. In this study they compared three heat pumps, two capacity controlled heat pumps specifically designed for this project (one designed for variable speed operation) and a single speed compressor heat pump available on the market. According to their results, if the heat pump adopts an on/off control strategy, the supply temperature during the on-time (in heating mode) must be higher than the temperature than would be required if the system would operate continuously. For this reason, the condensing temperature level for the on/off control strategy is higher than the condensing temperature required by a variable-capacity heat pump, this leading to a lower energy efficiency.
In this paper, a model of a GSHP system with variable speed drives is discussed, taking into account the compressor efficiency, the performance of the heat exchangers at partial loads and the influence of the water supply temperature.

2. MODEL OF THE SYSTEM

2.1 Description of the system

In Figure 1 the schematic system of the ground source heat pump installed at Hiref S.p.A. in Tribano (PD, Italy) is depicted. The heat pump system is developed within the European Ground-Med project (Seventh Research Framework Program, www.groundmed.eu) and consists of a reversible water-to-water ground source heat pump which meets the heating and cooling demand of the office and provides hot water to the production department. The heat pump provides air-conditioning to an office room of 152 m^2, with five fancoils, and a meeting room of 16 m^2, with one fancoil.
In this heat pump system, the heat is transferred by circulating water through four vertical 80 m deep probes. They are two single U-pipe loops and two double U-pipe loops connected in parallel.
As one can see in Figure 1, the heat pump system has a large number of variable speed components:

- inverter-driven scroll compressor adopting an asynchronous electric motor;
- centrifugal pumps with brushless DC motors driven by inverters for the secondary loops;
- fancoils with brushless DC motors driven by inverters.

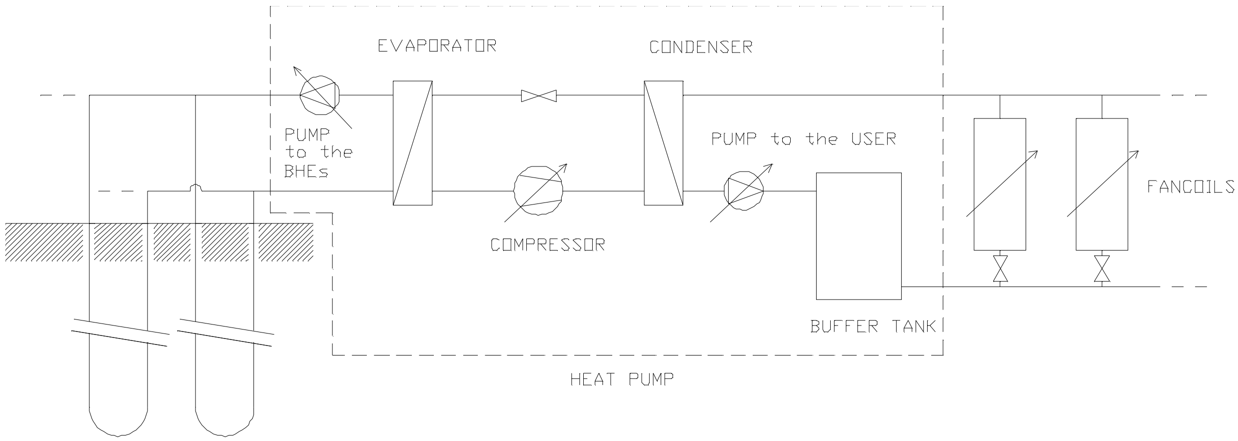

Figure 1: GSHP schematic system

In Figure 2, the thermal loads and heat peak demand of the building are shown; one can see that the heating demand is greater than the cooling demand (7954 kWh vs. 3075 kWh), but the peak loads are more or less the same (~ 14 kW). The possible thermal variation of the ground, caused by this energy unbalance between winter and summer, was analyzed during the design step through dynamic simulations of the behavior of the geothermal field.

2.2 Objectives and control strategy

As depicted in Figure 1, the parameters that can be varied for the heat pump system control are the followings: frequency of compressor, frequency of water pump to the borehole heat exchangers, frequency of the water pump to the user, velocity of the fans, water temperature to the user. In the present case, four COPs for the heat pump system can be defined, as follows:

$$COP1 = q_{HP} / P_{COMP} \tag{1}$$

$$COP2 = q_{HP} / (P_{COMP} + P_{ECP}) \tag{2}$$

$$COP3 = q_{HP} / (P_{COMP} + P_{ECP} + P_{ICP}) \tag{3}$$

$$COP4 = q_{HP} / (P_{COMP} + P_{ECP} + P_{ICP} + P_{FC}) \tag{4}$$

where q_{HP} is the thermal power provided by the heat pump, P_{COMP} is the electrical power input to the compressor, P_{ICP} is the power to the internal (building) loop pump, P_{ECP} to the external (borehole) circuit pump and P_{FC} to the fancoils.

During cooling mode the definitions remain the same and allow to calculate EER1, EER2, EER3, EER4 by replacing the condenser heat flow rate with the evaporator cooling power. The seasonal efficiency (SCOP1, SCOP2, SCOP3, SCOP4 – SEER1, SEER2, SEER3, SEER4) could be calculated using the corresponding thermal energy, which means by integrating the thermal and electric powers on the whole heating/cooling seasons. The parameters to be varied (frequency of compressor, frequency of water pump to the borehole heat exchangers, frequency of the water pump to the user, velocity of the fans, water temperature to the user) are chosen for each load condition to obtain the maximum seasonal performance factor SCOP4, defined as:

$$SCOP4 = \frac{\text{Thermal energy required}}{\text{El.energy}_{COMP} + \text{El.energy}_{ECP} + \text{El.energy}_{ICP} + \text{El.energy}_{FC}} \tag{5}$$

To achieve the maximum seasonal performance, the present problem has been approached in two steps.

1. In the first step, the curve of the thermal power is divided in n intervals of load ratio α, where α is defined as q_{HP}/q_{BLD} , being q_{HP} the heat flow rate provided by the heat pump and q_{BLD} the heat flow rate required by the building. For a certain value of the load ratio α, by adopting a fixed supply water temperature to the building,

through a numerical model simulation of the thermal system, it is possible to calculate the values of the remaining four variables (compressor speed, water pump frequency in internal and external circuit, fans speed) that allow to produce the desired thermal power with the maximum system efficiency. By repeating this procedure for the different values of load ratio α, the seasonal performance factor SCOP4 can be calculated according to equation 5, where the denominator is equal to

$$\text{El.energy}_{COMP} + \text{El.energy}_{ECP} + \text{El.energy}_{ICP} + \text{El.energy}_{FC} = \sum_{i=1}^{n} \frac{E(\alpha_i)}{COP4_{MAX}(\alpha_i)} \quad (6)$$

where $E(\alpha_i)$ is the fraction of the total seasonal thermal energy demand corresponding to the i-th load ratio, $COP4_{MAX}(\alpha_i)$ is the maximum value of COP4 obtained for the i-th value of load ratio and n is the total number of load ratio intervals.

2. By repeating the steps carried out at point 1 for a different value of the supply water temperature, it is possible to compare the SCOP4 at varying supply water temperature.

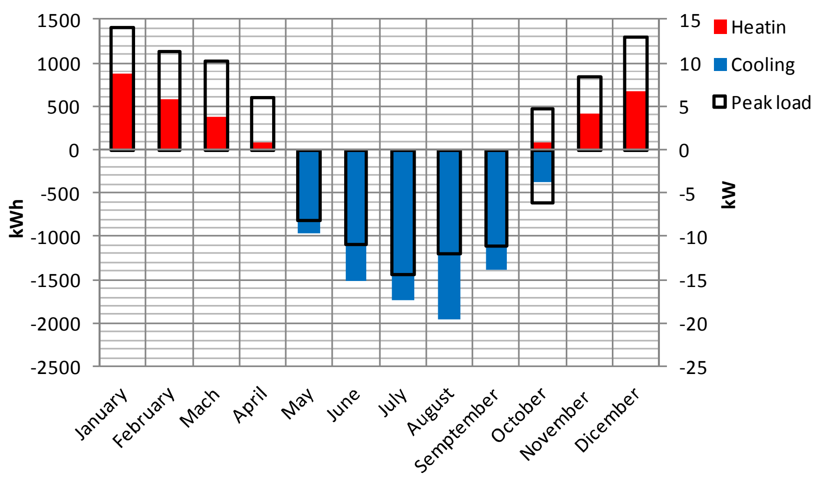

Figure 2: Thermal loads of the present air-conditioned ambient

The numerical model implemented at point 1 considers the main components of the system, which are modeled with ad-hoc lookups tables (LUT) as it is shown in Figure 3. These components considered in the model are: the evaporator, the geothermal field, the compressor, the condenser and the fan coils.

- Evaporator and borehole heat exchangers: the LUT is represented by a two-dimensional matrix, containing the evaporating temperature (T_e) as a function of water flow rate to the geothermal probes ($m_{w,ex,geo}$), and thermal power exchanged with the ground ($q_{ex,geo}$). This matrix was assembled simulating the behavior of the boreholes with the commercial software EED, and the behavior of the evaporator with the commercial software distributed by the heat exchangers manufacturer. The superheating in the suction was about 5 K, with a minimum approach between refrigerant and water of 1 K.
- Compressor: the LUTs are represented by two three-dimensional matrixes, containing the cooling and heating capacities of the compressor as a function of the evaporating temperature (T_e), condensation temperature (T_c), and the electrical frequency (Hz) of the electric motor. These matrixes are obtained from the data declared by the compressor manufacturer and express indirectly the refrigerant flow rate that the compressor is able to develop as a function of T_e, T_c, and frequency, with about 5 K superheat and about 5 K subcooling.
- Condenser: the LUT is represented by a two-dimensional matrix containing the condensing temperature (Tc) as a function of condenser water flow ($m_{w,ex,user}$) and of the thermal power transferred to the building ($q_{ex,user}$). This matrix has been assembled by simulating the behavior of the heat exchanger with the software provided by the manufacturer, with 80°C inlet gas temperature, 5 K subcooling, a minimum approach between refrigerant and water of 1 K, and a constant outlet water temperature ($T_{w,out}$). For each value of water temperature ($T_{w,out}$) considered in the search of the maximum SCOP4, it is necessary to build up a new condenser matrix.
- Fancoils: the LUT is made of a two-dimensional matrix containing the thermal power transferred from the six fancoils of system ($q_{ex,user}$) as a function of the total water flow rate ($m_{w,ex,user}$) and of the fan speed ($\%\ VFC$). The LUT of the terminal units is constructed by considering a constant inlet water temperature ($T_{w,in}$) and a constant

temperature of 21°C for the inlet air ($T_{air,in}$). For each $T_{w,in}$ considered in the search for the maximum SCOP4, it is necessary to construct a matrix of the fancoils.
The electric power absorbed by the compressor, P_{COMP}, was calculated as the difference between the condenser and the evaporator capacity declared by the compressor manufacturer, while the electrical power absorbed by the pumps P_{ICP}, P_{ECP}, and by the fancoils P_{FC} were experimentally determined.
In the model the thermal system dissipation of each component to the surrounding environment has been neglected; this assumption allows to set $T_{w,in}$=$T_{w,out}$, so the temperature of the water coming out from the condenser is equal to the temperature at the inlet of fancoils.

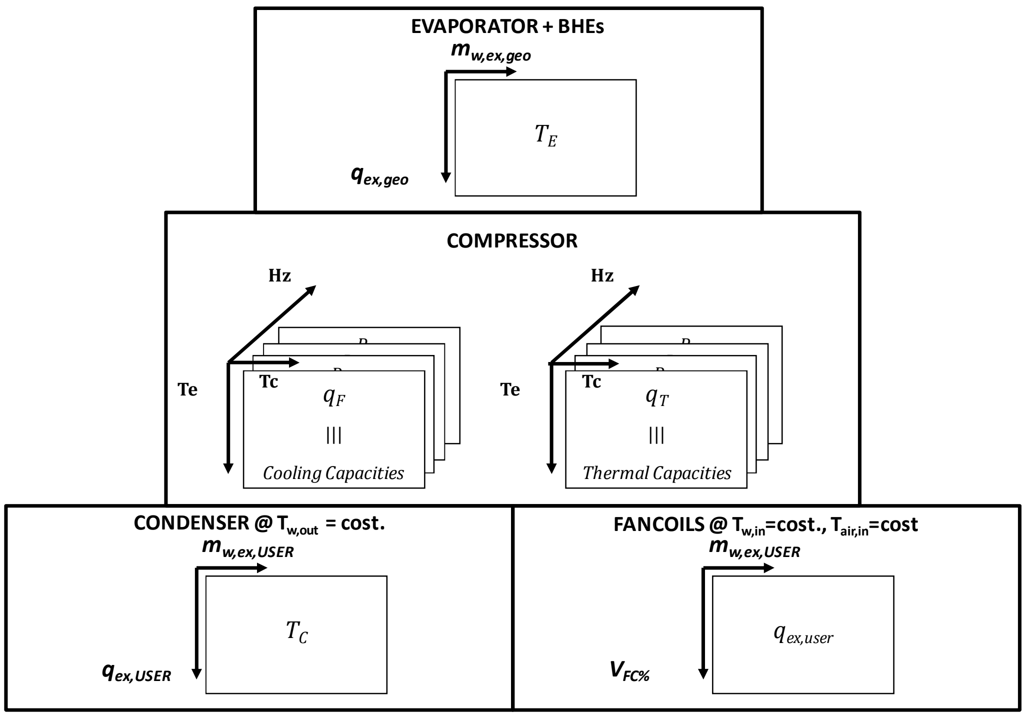

Figure 3: Look-up tables of the heat pump system model

3. OPTIMAL SYSTEM SEASONAL PERFORMANCE

3.1 Curves of COP

The present GSHP system has been simulated and solved for three different values of the supply water temperature $T_{w,\,in}$ (34, 37, 40°C). The results obtained from this model are shown in the figures below. In Figures 4 and 5 the values of the COP1 and COP3 are depicted as a function of water flow rate in the internal (user) and external (probes) circuits, for α=1 and for a supply water temperature to the fancoils of 37°C, while Figures 6 and 7 show the values of the COP1 and COP3 for the same water temperature but with α=0.5.
From these figures one can see the penalizing effects of electrical consumption in the water pumps on the performance of the heat pump system, especially at partial loads. If α=1 is considered, the maximum COP1 is achieved with a mass flow rate to the borehole heat exchangers equal to 2500 L/h, and it remains constant with the water flow rate to the building because, in the condenser, the dominant thermal resistance is on the refrigerant side.
Referring to Figure 5, one can see that the probes flow rate that maximizes the COP3 is in the range 1500÷2000 L/h, that is lower than the value that maximizes the COP1, while the user flow rate is approximately equal to the value corresponding to the minimum velocity of the circulation pump. This result is due to the fact that the heat transfer in the fan coils is not considered in the COP3 analysis and also to the controlling thermal resistance in the condenser. It results that the optimal size of the water pump and its speed depends also on the heat transfer area in the fan coils.
If α=0.5 (Figures 6 and 7), the maximum value of COP1 is achieved with a probes flow rate in the range 1000÷1500 L/h, next to the flow rate obtained at the minimum speed of the pump. This is due to the excess of heat transfer area in the evaporator and in the geothermal probes. This can also be seen from the probes flow rate and users flow rate

that maximize the COP3; at this load ratio, the optimal values of flow rate remain unchanged as compared to the graph of COP1.

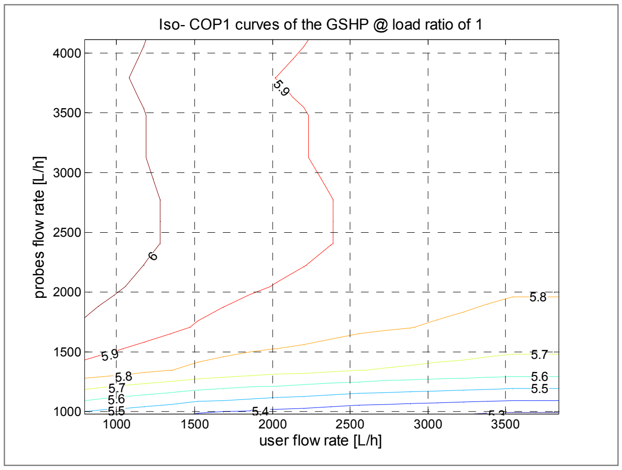

Figure 4: COP1 for α=1 and $T_{w,in}$ =37°C

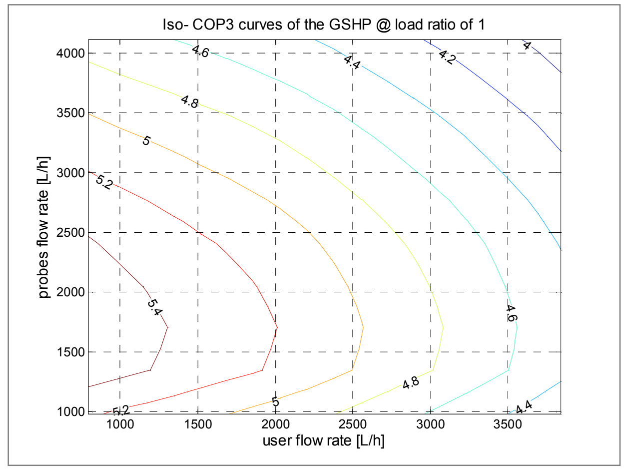

Figure 5: COP3 for α=1 and $T_{w,in}$ =37°C

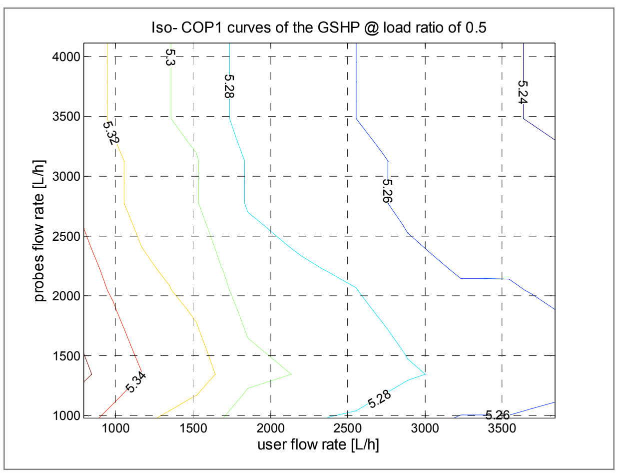

Figure 6: COP1 for α=0.5 and $T_{w,in}$ =37°C

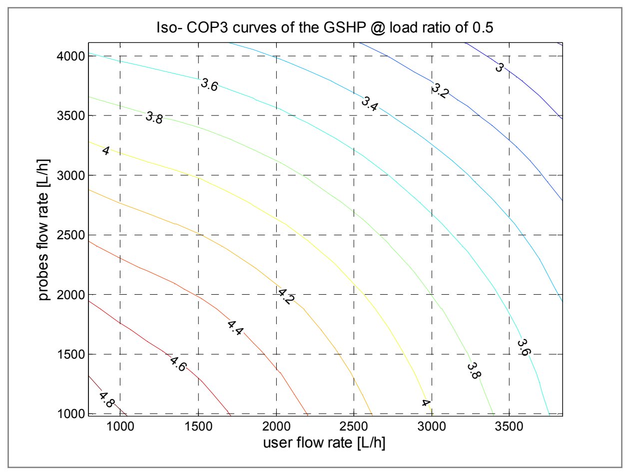

Figure 7: COP3 for α=0.5 and $T_{w,in}$ =37°C

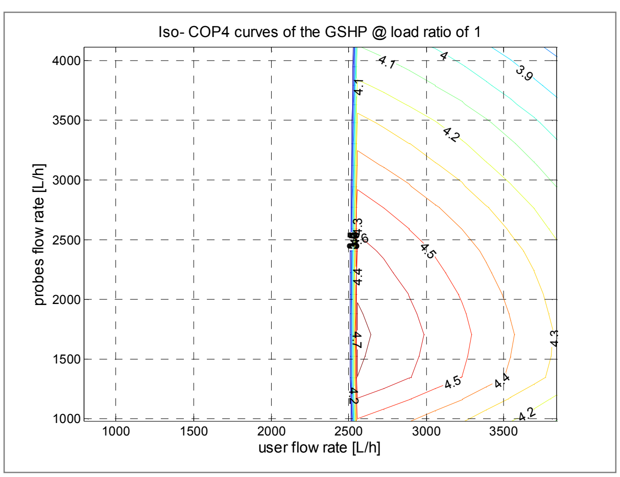

Figure 8: COP4 for α=1 and $T_{w,in}$ =37°C

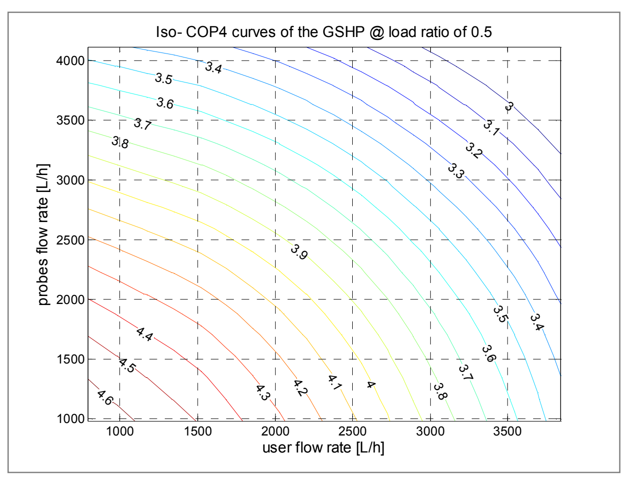

Figure 9: COP4 for α=0.5 and $T_{w,in}$ =37°C

Figures 8 and 9 report the *COP4* (of the heat pump system) referred to α=1÷0, for $T_{w,in}$ = *37°C*. From Figure 8, one can see that the minimum user flow rate, to meet the load required, is slightly higher than 2500 L/h, and it is a function of supply temperature, heat transfer surfaces and air flow rate in the fancoils. The *COP4* is still a function of the probes flow rate, and it is maximum in the range 1500÷2000 L/h.

When α=0.5 is considered, the results obtained for *COP4* are not much different from the ones of *COP3*. From Figure 9 it can be noticed that the maximum value of the COP4 is obtained at the minimum users and probes flow rate, as previously obtained in Figure 7 for the *COP3*. This is because, with the present fan coils, at 0.5 load ratio, the heat transfer area and the air flow rate are high enough to provide the required thermal power with the minimum user flow rate: . At this partial load, due to the excess of heat transfer area in the borehole heat exchangers and in the evaporator it is possible to exchange the required heat at the minimum probes flow rate. Comparing the data of Figure 7 and Figure 9, it can be noticed that the values of COP4 are lower than the values of COP3, with the same dependence on the users and probes flow rate, because of fancoils power consumption.

The overview of the results obtained from the previous simulations is shown in Figure 10: the values of the maximum *COP4* are reported vs. load ratio at three supply water temperatures $T_{w,in}$. Each point plotted on the graph corresponds to a specific set of values of probes flow rate, users flow rate, compressor electric frequency, fancoils speed (in percentage) for the present GSHP system.

From figure 10 it can be seen that a temperature of $T_{w,in}$ = 34°C does not allow to provide the required heat at a load ratio greater than 0.85, due to heat transfer limitation in the fancoils. In the range *α*=0.5÷0.75 the supply temperature $T_{w,in}$ = 34°C allows to provide the required heat with the maximum value of *COP4*, while for the range *α*=0.75÷1 the optimum temperature is 37 °C. The transition from the optimum temperature of 34°C to 37°C is mainly due to the water flow rate required to supply the thermal power at $T_{w,in}$ = 34 °C, which penalizes significantly the *COP4*. For a $T_{w,in}$ = 40°C, the *COP4* is always lower than in the previous two cases, with the exception of very low and very high load ratios (α=0.45 and α=0.95÷1) where it is comparable with those obtained at $T_{w,in}$ = 37 °C. Even if in this case the condensation temperature is lower, and this would bring some advantage to the efficiency, the need for an increase of the user flow rate and fancoils speed may cancel the previous advantage in terms of overall efficiency.

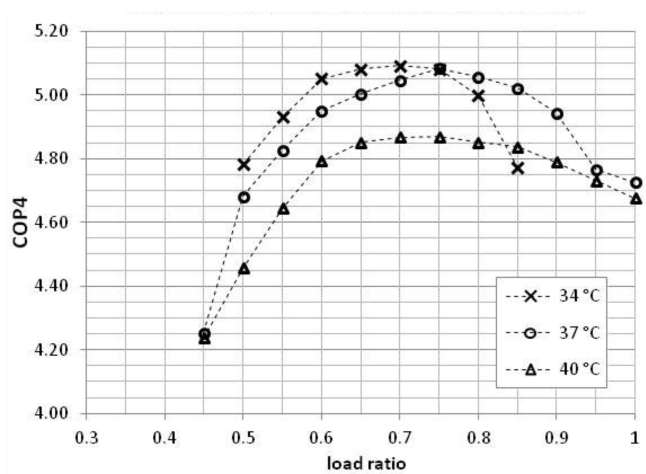

Figure 10: maximum *COP4* as a function of load ratio for different values of supply temperature

3.2 Calculation of maximum seasonal performance

By applying equations 5 and 6, and considering the maximum values of *COP4*, one can calculate the maximum seasonal performance factor *SCOP4*. In figure 11 the values of *SCOP4* at three different water temperatures are depicted.

All the values of *SCOP4* shown in Figure 11 have been calculated in the range of load ratio up to α=0.85, because in the case of 34°C water temperature it is not possible to satisfy higher values of load ratio. At higher load ratio, higher values of supply water temperature should be set (37 or 40°C). As it can be seen in Figure 11, in the range α=0÷0.85, the minimum *SCOP4* is obtained for 34°C water temperature, while the values of *SCOP4* remain pretty the same at 37°C or 40°C temperature. The distribution of the thermal energy required during the winter season is plotted as a function of the heat power in Figure 12. It is clear that approximately 75% of the seasonal thermal energy is supplied with a load ratio lower than 0.5 (7 kW). For this load ratio the maximum *COP4* that can be obtained for the supply water temperature of 37°C and 40°C is almost the same, as shown in Figure 10, and this is the reason why the *SCOP4* remains roughly the same with 37°C or 40°C supply water temperature.

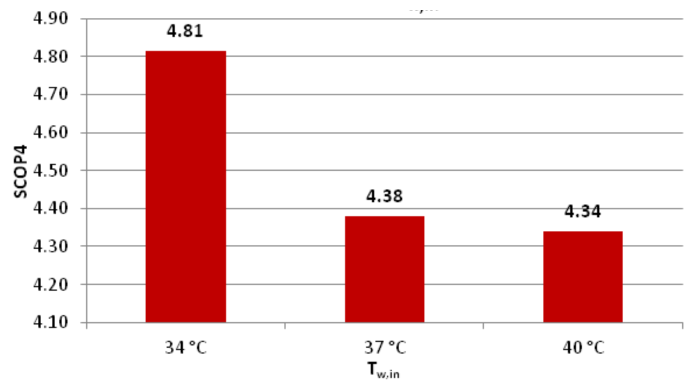

Figure 11: Maximum *SCOP4* as a function of the water temperature to the building $T_{w,in}$

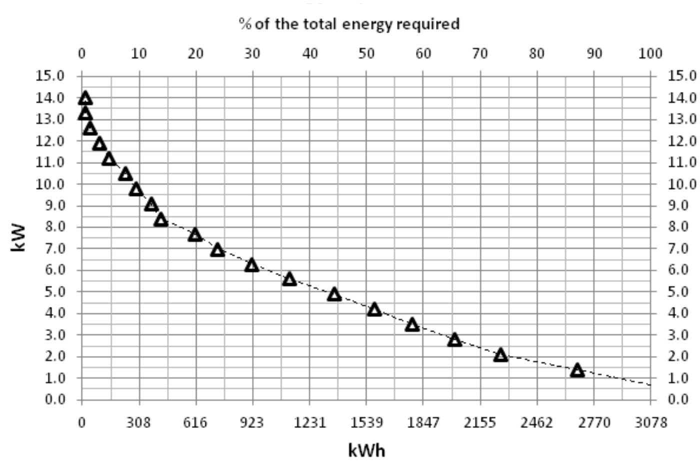

Figure 12: Duration curve of the winter energy demand: thermal power as a function of the required heat.

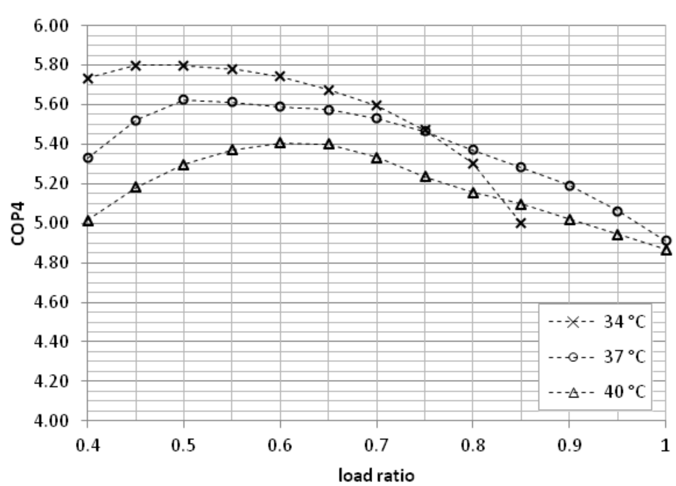

Figure 13: Maximum *COP4* as a function of load ratio for different $T_{w,in}$ using the brushless compressor

The numerical model allows to replace easily one of the components of the look-up tables (in Figure 3) and to analyze the different performance in terms of *SCOP4*. For example, it is possible to compare the performance of the heat pump system using an inverter-driven scroll compressor, as previously mentioned, with a brushless DC (BLDC) motor compressor; in this case it is needed to insert in the model the look-up table of new compressor. Figure 13 reports the maximum *COP4* as a function of load ratio for different $T_{w,in}$ when using the brushless DC motor compressor, while Figure 14 shows a comparison between the maximum *SCOP4* as a function of $T_{w,in}$ for the BLDC compressor and the asynchronous compressor.
From the results shown in Figure 14, one can see that the adoption of a brushless compressor, as compared to the use of an asynchronous compressor, allows to increase the overall efficiency of the heat pump system at all the values of supply temperature considered in the present study (+18.6%, + 23% and +17.3% @ 34, 37 and 40°C, respectively). The difference between the *SCOP4* values calculated with the two compressors is enhanced by the need to operate mostly at partial load (see Figure 12), where the efficiency of the asynchronous compressor is significantly penalized as compared to the brushless compressor.

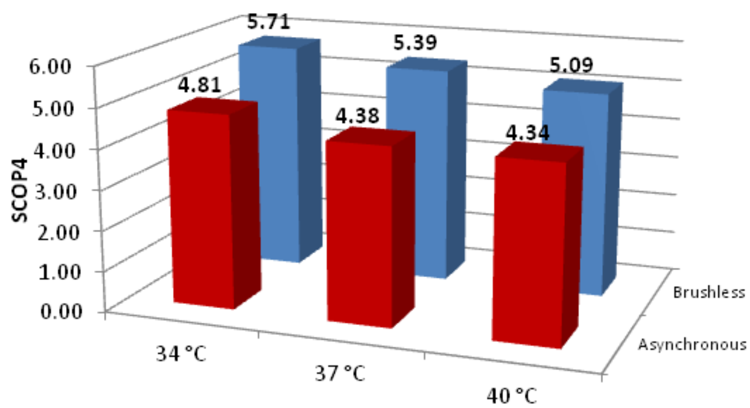

Figure14: Comparison between the maximum *SCOP4* as a function of $T_{w,in}$ for two different types of compressors

3.3 Control model

In order to adjust the heating and cooling capacity to the load requirement and maintain the maximum value of *SCOP4*, an appropriate control strategy is needed.
The control strategy can be implemented by means of a double regulation:

- primary regulation: the frequency of the compressor is varied with a proportional integral action as a function of the difference between the set-point value of the supply water temperature ($T_{w,in}$) and the water temperature in the tank, which is placed in series to the internal circuit, as depicted in Figure 1. The frequency of the water pump in the external circuit comes from the link (see above model) with the compressor speed.
- secondary regulation: the water flow rate in the internal (user) circuit and the fancoils speed is varied with a proportional integral action as a function of the difference between the set-point indoor temperature and the actual measured temperature in the ambient.

From the previous sections, it is now possible to determine the values of water flow rate in the internal and external circuits and fan speed as a function of the compressor frequency at the maximum *COP4* for different water temperature supply. Therefore, in the primary and secondary regulation the four controlling parameters (compressor speed, frequency of water pumps, fancoils speed) are varied according to the links that allow maximum seasonal COP in the system.

4. CONCLUSIONS

In this paper a model of a ground source heat pump with variable speed compressor, variable speed water pumps and variable speed fans in the terminal units is presented. The model is aimed at the evaluation of operating conditions which provide the maximum seasonal COP of the system. The parameters that are varied for the heat pump system control are: electric frequency of compressor, frequency of water pump in the external circuit, frequency of the water pump in the user circuit, velocity of the fans, supply water temperature to the user.

COP curves at constant load ratio are plotted as functions of flow rate in the borehole heat exchangers and flow rate to the building. When decreasing the load ratio, the maximum value of the system COP (*COP4*) is obtained at the minimum possible values of flow rate to the borehole heat exchangers and to the building because at partial loads the heat exchangers are somehow oversized, therefore reducing the energy consumption of the water pumps become important. The present model allows to predict the system behavior when replacing one of the HP components by modifying the corresponding lookup table: a comparison between performances with a brushless DC motor compressor and with an asynchronous motor compressor was presented. The results show that the adoption of the brushless DC motor allows to increase the values of *SCOP4* by 23% due to the higher efficiency of the brushless compressor at partial loads.
Finally, the present model can be used to develop a control strategy that allow to maximize the seasonal system performance.

NOMENCLATURE

α	load ratio		(–)
COP	coefficient of performance		(–)
EER	energy efficiency ratio		(–)
LUT	look-up table		
m	water flow rate		(kg s^{-1})
P	power		(W)
q	heat flow rate		(W)
SCOP	seasonal coefficient of performance		(–)
SEER	seasonal energy efficiency ratio		(–)
T	temperature		(°C, K)
Subscript			
BLD	building	ECP	external circuit pump
c	condenser	FC	fancoils
COMP	compressor	HP	heat pump
e	evaporator	ICP	internal circuit pump

REFERENCES

Karlsson, F., Fahlen, P., 2007, Capacity-controlled ground source heat pumps in hydronic heating systems, *International Journal of Refrigeration,* vol. 30, p. 221-229.

Karlsson F., Fahlen P., 2003, Improving efficiency of hydronic heat pump heating systems, *International Congress of Refrigeration*, Washington D.C, USA.

Madani, H., Claesson, J., Lundqvist, P., 2011, Capacity control in ground source heat pump systems part I: modeling and simulation, *International Journal of Refrigeration,* vol. 34, p. 1338-1347.

Madani, H., Claesson, J., Lundqvist, P., 2011, Capacity control in ground source heat pump systems part II: comparative analysis between on/off controlled and variable capacity systems, *International Journal of Refrigeration,* vol. 34, p. 1934-1942.

Munari P, Da Riva E, Del Col D, Mantovan M, 2011, Energy efficiency of variable capacity ground source heat pumps. *Profc. of AS - Alternative Sources / ISPHC11 - International Sorption Heat Pump Conference*. Padua, 5-7 april 2011.

Zakula, T., Gayeski, N. T., Armstrong, P. R., Norford, L. K., 2011, Variable-speed heat pump model for a wide range of cooling conditions and loads, *HVAC&R Research,* vol. 17:5, p. 670-691.

Zhao, L., Zhao, L. L., Zhang, Q., Ding, G. L., 2003, Theoretical and basic experimental analysis on load adjustment of geothermal heat pump systems, *Energy Conversion and Management,* vol. 44, p. 1-9.

ACKNOWLEDGEMENT

The authors would like to acknowledge the support of the European project GROUND-MED (www.groundmed.eu), co-funded by the European Commission within the Seventh Framework Programme.