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Experimental analysis of optimal operation mode of a ground source heat pump system

Davide Del Col[a]*, Marco Azzolin[a], Giacomo Benassi[b], Mauro Mantovan[b]

[a] Università degli studi di Padova, Dipartimento di Ingegneria Industriale, Via Venezia 1, Padova - 35131, Italy
[b] Hiref Spa, Viale Spagna 31/33, Tribano (PD) - 35020, Italy

Abstract

This paper presents experimental data and modeling of a ground source heat pump (GSHP) with variable speed compressor, variable speed water pumps and variable speed fans in the coils, installed at Hiref Spa (Italy) in the framework of the European Project Ground-Med. The present model has been developed to evaluate the operating conditions that lead to the maximum seasonal coefficient of performance and to analyze the behavior of the system at partial loads since variable capacity heat pumps do not work at nominal power for most of the time. 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 water pump to the user, velocity of the fans and water temperature to the user. The model has been compared with experimental data taken during a heating season and it can be the baseline to develop a control strategy with the final objective of maximizing the seasonal coefficient of performance of the system.

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Keywords: GSHP, variable speed, coefficient of performance, experimental, model

1. Introduction

To adjust the heating and cooling capacities to the load required by the building, heat pumps usually operate in on/off mode but working in this way they have to supply heat at higher temperature during the on time as compared to the temperature level needed by terminal units. If a variable speed heat pump is considered, it can simply follow

* Corresponding author. Tel.: +39-049-827-6891; fax: +39-049-827-6896.
E-mail address: davide.delcol@unipd.it

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Nomenclature

BHE	borehole heat exchanger
COP	coefficient of performance
m	water flow rate [kg s^{-1}]
P	power [W]
Q	heat flow rate [W]
SCOP	seasonal coefficient of performance [/]
SWT	supply water temperature
T	temperature [°C]
VFC	velocity of fancoils [%]
Subscripts	
c	condenser
COMP	compressor
e	evaporator
ECP	external circuit pump
ex	exchange
FC	fancoils
geo	geothermal
HP	heat pump
ICP	internal circuit pump
NOM	nominal
w	water
Greeks	
α	load ratio [/]
ΔT	temperature difference [K]
β	ratio of compressor electrical power consumption to the electric power absorbed by the water pumps [/]

the load, taking advantages 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.* [1] proposed a theoretical and experimental analysis. They considered several capacity control methods 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. Their tests focused on the influence of the frequency of the compressor speed on the *COP* at different water tank temperatures. While heating and cooling capacities increased almost linearly by increasing the frequency, *COP* presented different trends depending on water tank temperatures. An energy performance comparison of a variable capacity geothermal heat pump against an on/off controlled equipment has been reported in Munari *et al.* [2]. In the same paper the electrical efficiency of an asynchronous motor compressor at variable speed was compared against the efficiency of permanent magnet brushless motor compressor. The heat exchangers performance and compressor efficiency at partial loads were found as the major aspects in the comparison. Karlsson and Fahlén [3] compared an on/off compressor control and a variable speed control in a brine-to-water heat pump highlighting that using a variable speed compressor, the need for supplementary heating was reduced. More recently the same authors [4], experimentally compared three GSHP with different layouts; two, specifically designed for their project, one with a standard two-cylinder piston compressor, one with a variable speed scroll compressor and lastly a commercial heat pump with a single speed compressor. They pointed out that to exploit the heat exchangers unloading at partial loads, efficiency of the compressor and efficiency and control of water pumps are important keypoints.

Several models of heat pump systems have been developed in literature. Madani *et al.* [5] developed a model for ground source heat pump systems in order to compare the seasonal performance of different control strategies. The model was composed of several sub-models representing the components of the system. This model was used in Madani *et al.* [6] 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 four on/off controlled GSHPs designed to cover different percentages of building heat peak demand and one variable capacity system with variable speed compressor and single speed water pumps. This study indicated that if the on/off controlled systems were dimensioned to cover more than the 65% of the building peak demand, there was no considerable difference with the variable capacity system. At both ends, if the on/off controlled heat pump was oversized, thus covers more than 94% of the building peak load, economic constraints prevented the option to be the best solution and if it was dimensioned to cover the 55% of the peak demand, the electrical auxiliary heaters consumption would be extremely high. In the variable speed heat pump due to almost continuous operations the energy consumptions of the water pumps were 5-30% higher than the on/off controlled heat pumps. A model and a year round analysis of a GSHP fitted with a variable speed compressor was presented in Lee [7] and three different locations with different loading profiles and weather data were considered. Three control schemes for the partial load control were applied and compared. With a frequency control a reduction of compressor energy input and an overall energy saving were achieved. Kinab *et al.* [8] developed and successfully compared with experimental data a system model of an air to water reversible heat pump including detailed sub-models of various components. Modeling the compressor, on/off controlled, multiple stage compression and variable speed compressor were taken into account. A mathematical model for quasi-steady state performance of a water-to-water reversible GSHP working with propane with on/off control system has been presented in Corberan *et al.* [9].

The present paper regards on experimental investigation carried out on the heat pump installed at Hiref Spa in Tribano (PD): it is one of the eight GSHP systems developed and monitored within the European Ground-Med project (Seventh Research Framework Program), aimed at studying the sustainability of GSHP systems in the Mediterranean Countries. In the project, different types of heat pump systems have been installed to compare different solutions and various components. Because the building peak load is required only for few hours in a year, heat pumps usually work at partial loads, thus it is necessary to adjust the heating and cooling capacities to the load required by the building. Therefore, in order to increase the energy efficiency, it is important to consider the behaviour of the system at partial loads. To gain the maximum seasonal energy efficiency in a ground source heat pump with variable speed drives, a satisfactory regulation, which allows to achieve the maximum coefficient of performance when varying the load, is needed.

2. Description of the heat pump facility

In Fig. 1 the schematic system layout is depicted 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. The heat exchange with the ground is made through four vertical 80 m deep probes, arranged in square with 7 m sides. The geometrical data of the two single U-pipe loops and of the two double U-pipe loops connected in parallel are reported in Tab.1.

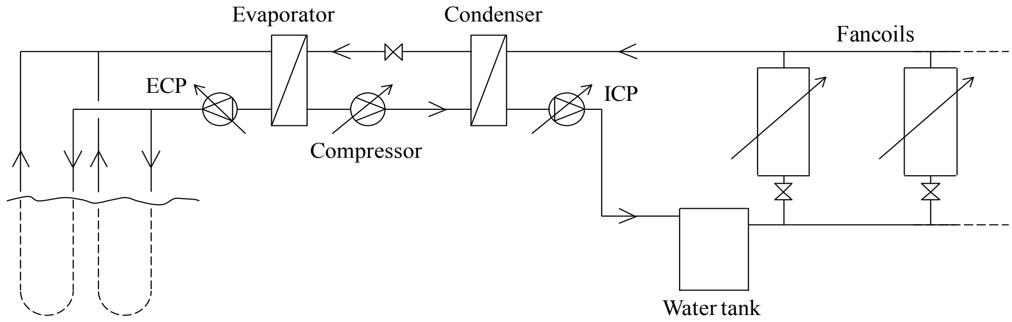

Fig. 1. Layout of the ground source heat pump system.

Table 1. Geometrical data of the borehole heat exchangers.

	Single U-pipe	Double U-pipe
Borehole diameter [mm]	140	140
Internal pipe diameter [mm]	26.6	32.6
Pipe thickness [mm]	2.9	3.7
Pipe distance [mm]	85	70

The heat pump system has a large number of variable speed components: inverter-driven scroll compressor, centrifugal pumps with brushless DC motors driven by inverters and fancoils with brushless DC motors driven by inverters. The compressor is driven by an asynchronous electric motor controlled by converter, which consists of a three-phase bridge rectifier on network side and an IGBT inverter on the motor side with stabilizer capacitor in the middle. The expansion valve is electronically commanded and allows large capacity modulation. The building and the borehole heat exchangers pumps are driven by inverter control single-phase synchronous permanent magnet electric motors, instead the sanitary water pump is driven by a single-phase synchronous permanent magnet electric motor with a three-step velocity modulation. The six fancoil terminals installed are equipped with inverter driven brushless electric motors and every terminal has its own microprocessor to measure the temperature and humidity of the air and the water temperature at battery inlet. The heat exchangers are three equal braze-welded 40 plate heat exchangers. The power consumption of the hydronic pump and the heat exchange on the sanitary loop were not considered in this analysis.

Due to heat gains during the cooling season, the cooling demand is more than two times the heating demand, but peak loads are more or less the same, as reported in Fig. 2.

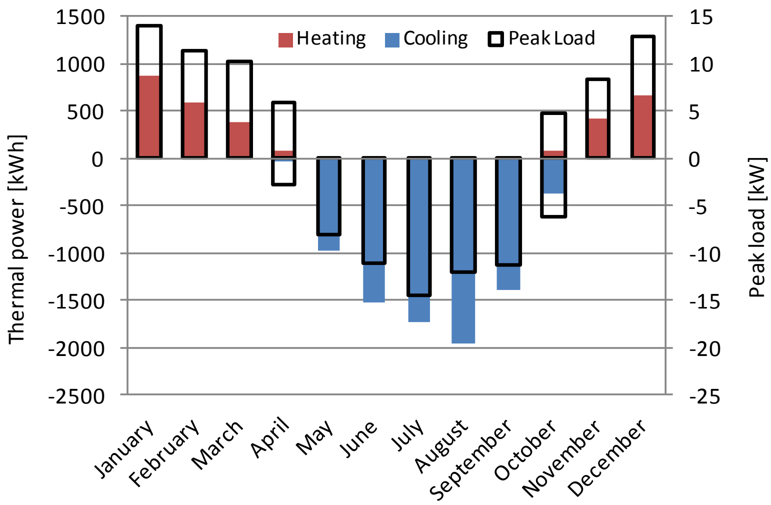

Fig. 2. Thermal power and peak loads required by the building (in figure negative values are used to represent cooling).

3. Experimental results

Considering the variable components of the heat pump system four definitions of *COP* can be given:

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

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

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

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

where q_{HP} is the thermal power provided by the heat pump, P_{COMP} is the compressor electrical power consumption, P_{ECP}, P_{ICP} and P_{FC} are the power consumption of the external (borehole) loop pump, of the internal (building) circuit pump and of the fancoils, respectively. It is useful to define the load ratio α as the ratio of thermal power provided by the heat pump to the nominal power q_{HP}/q_{NOM}.

During the heating season of the winter 2012-2013 a measurement campaign was carried on and experimental data were taken for two different supply water temperatures (37°C and 40°C) and for three different values of load ratio (0.4, 0.6, 0.8) varying the water flow rate to the building in the range 1060 L/h to 4000 L/h and to the borehole heat exchangers between 1250 L/h and 3500 L/h. Once the system has reached steady-state conditions and all parameters were fixed, data were recorded. The compressor and the hydronic pumps power consumptions were measured using three-phase electrical energy meters with class 1 accuracy according to European standards EN62053-21. The heat exchangers were equipped with thermal energy meters on the water side. Every thermal energy meter consists of 2-wired paired platinum sensors PT500 and a volume meter tube, based on Faraday's magnetic induction principle. The meters were approved according to European standard EN 1434 accuracy class 2.

For a load ratio α=0.77 and a supply water temperature to the fancoils equal to 40°C the experimental *COP3* curves are reported in Fig. 3 as a function of the water temperature difference to the building (x-axis) and the water temperature difference to the borehole heat exchangers (y-axis). The water temperature difference at the borehole heat exchangers that maximizes the *COP3* is in the range 2.4 ÷ 3.1 K, while the water temperature difference to the building is in the range 4.2 ÷ 5.4 K.

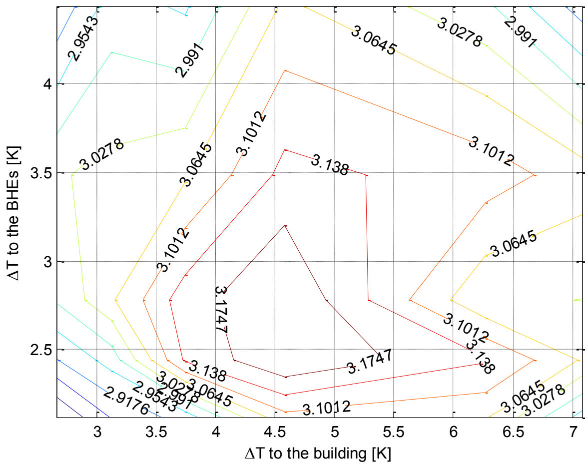

Fig. 3. Experimental COP3 curves for α=0.77 and 40°C SWT

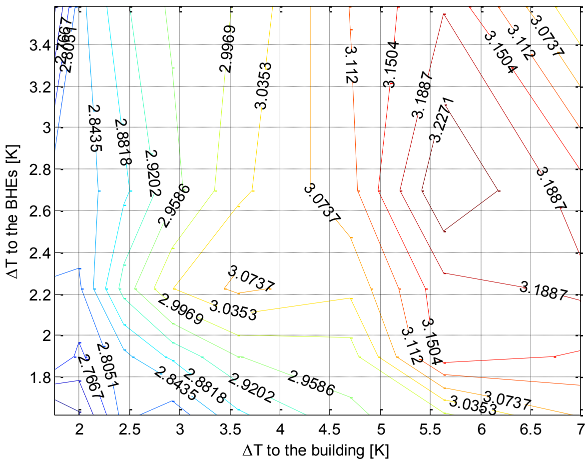

Fig. 4. Experimental COP3 curves for α=0.59 and 40°C SWT

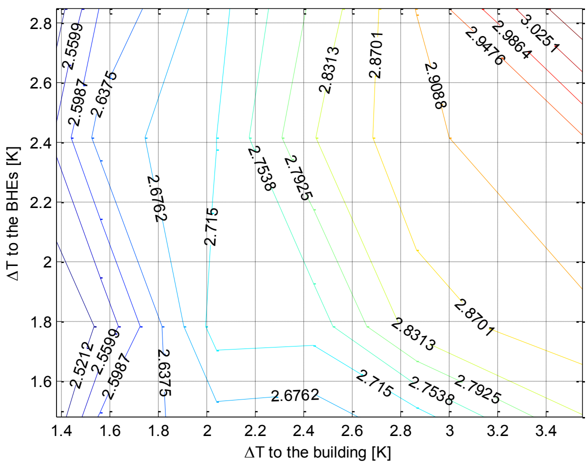

Fig. 5. Experimental COP3 curves for α=0.40 and 40°C SWT.

When the load ratio is decreased, maximum values of *COP3* are achieved for a temperature difference at the borehole heat exchangers between 2.5 K and 3.1 K and close to the minimum speed of the pump respectively for α=0.59 and α=0.40. If α=0.59 is considered, it can be noticed that the maximum *COP3* is reached for values of the water temperature difference to the building in the range 5.4 ÷ 6 K and for α=0.40 at about the minimum speed of the water pumps.

In Fig. 6 the values of the compressor and water pumps electrical power consumptions at maximum *COP3* conditions are reported as a function of load ratio. Interesting information can be obtained by looking at the ratio of compressor electric power consumption to the electric power absorbed by the water pumps. The ratio is hereafter called as β parameter and it is defined in Eq. 5:

$$\beta = \frac{P_{COMP}}{P_{ICP} + P_{ECP}} \tag{5}$$

At partial loads, when the electric power at the compressor decreases, the consumption of the water pumps becomes relatively higher and therefore, to achieve the maximum COP3 the frequency of water pumps must be properly controlled. The β parameter, when varying the load ratio, is almost constant showing the opportunity of using this parameter as a control variable in the regulation strategy. For the present ground source heat pump the values of β parameter are about 8.5 except for the point at 37°C supply water temperature and load ratio equal to 0.8, because in this test condition the heat flow rate required could not be provided for water flow rate lower than 1500 L/h.

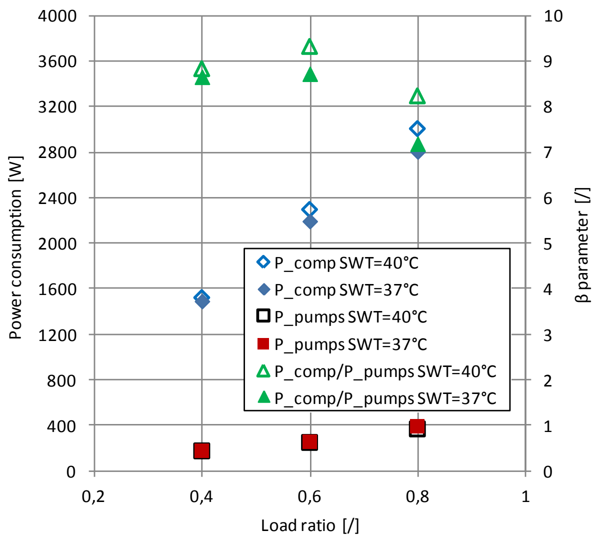

Fig. 6. Values of compressor electrical power consumption and β parameter at different load ratio.

4. Numerical model of the system

The numerical model is composed of proper lookup tables (LUTs) modeling the main components of the system, as reported in Fig. 7. The LUTs are two-dimensional or three-dimensional matrices depending on the number of variables of the component considered. For example, the evaporator and borehole heat exchangers LUT is represented by a two-dimensional matrix, containing the evaporating temperature (T_e) as a function of water flow rate to the borehole heat exchangers ($m_{w,ex,geo}$), and thermal power exchanged with the ground ($q_{ex,geo}$). In this matrix the behavior of the boreholes has been simulated with the commercial software EED and the software distributed by

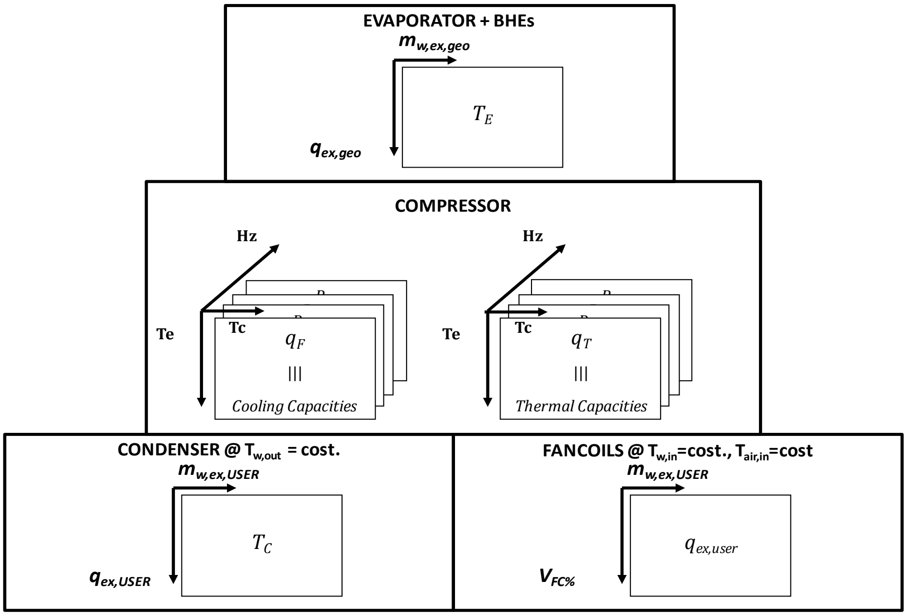

Fig. 7. Schematic representation of the look-up tables of the GSHP system model.

the heat exchangers manufacturer was used for the behavior of the evaporator. Instead three dimensional matrices are used to represent the compressor, containing the cooling and heating capacities as a function of the evaporating temperature, condensing temperature and frequency of the electric motor. These matrices have been obtained from the data declared by the compressor manufacturer and express indirectly the refrigerant flow rate that the compressor is able to develop.

The present GSHP model has been calibrated with the experimental data taken from the heat pump installed. The iso-*COP3* curves at α=0.77, α=0.59 and α=0.40 calculated with the model are depicted in Figs. 8-10. The model is able to predict well the data of *COP3* but there are some differences in the curve trend. These few differences can be due to some simplifications in the modeling LUTs. The model has been used to compare the experimental values of *COP3*, which were predicted within ±10% error band and it can be the baseline to develop a control strategy with the final objective of maximizing the seasonal coefficient of performance of the system.

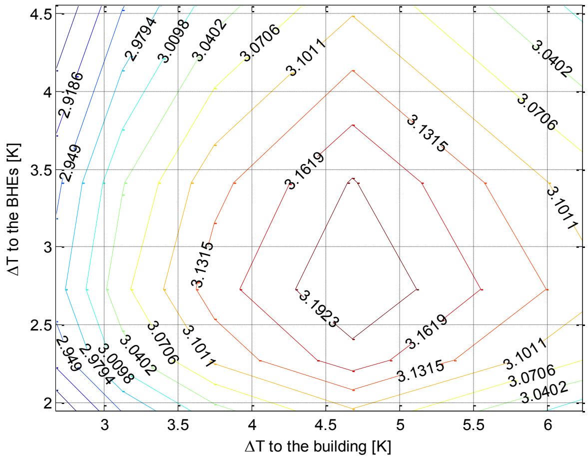

Fig. 8. Calculated COP3 curves for α=0.77 and 40°C SWT

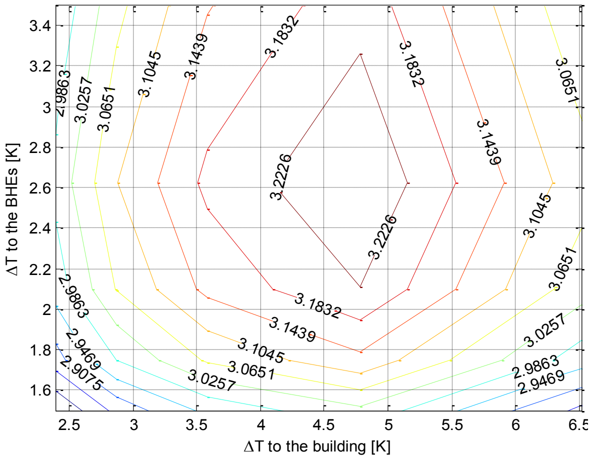

Fig. 9. Calculated COP3 curves for α=0.59 and 40°C SWT

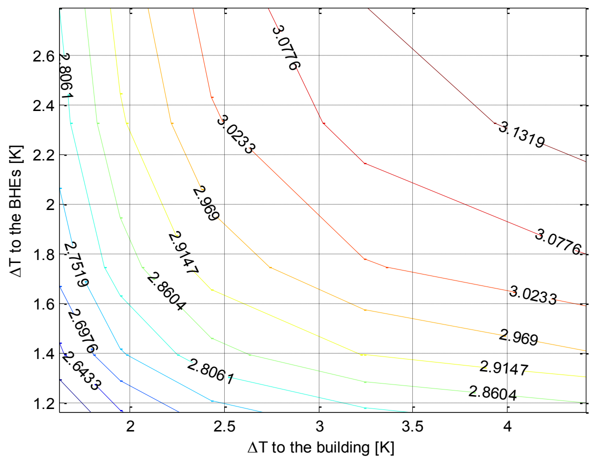

Fig. 10. Calculated COP3 curves for α=0.40 and 40°C SWT

In Fig. 11 the maximum experimental values of *COP3* and *COP1* at different load ratio are reported and compared with the values calculated from the model. It can be seen that in the range 0.4< α <0.9 the supply temperature $T_{w,in}$=37°C allows to provide the required heat with the maximum values of *COP3*.

The numerical model allows to replace the LUT of one component and to analyze the performance with different system layouts. For example, it is possible to compare the performance of the heat pump system using an inverter-driven asynchronous motor compressor, as previously mentioned, with a brushless DC (BLDC) motor compressor by modifying in the model the compressor LUT with the new one. The BLDC motor compressor allows to increase the *COP1* mainly at partial loads (+27%, +33% and +43% at α=0.8, 0.60 and 0.40, respectively) due to its better efficiency if compared with the asynchronous one.

The seasonal performance factor *SCOP3* is defined in Eq. 6 and the present model allows to evaluate the operating conditions (frequency of compressor, frequency of water pump to the borehole heat exchangers, frequency of the water pump to the user) that lead to the maximum energy efficiency at each load ratio.

$$SCOP3 = \frac{\text{Thermal energy required}}{\text{El.energy}_{COMP} + \text{El.energy}_{ECP} + \text{El.energy}_{ICP}} \tag{6}$$

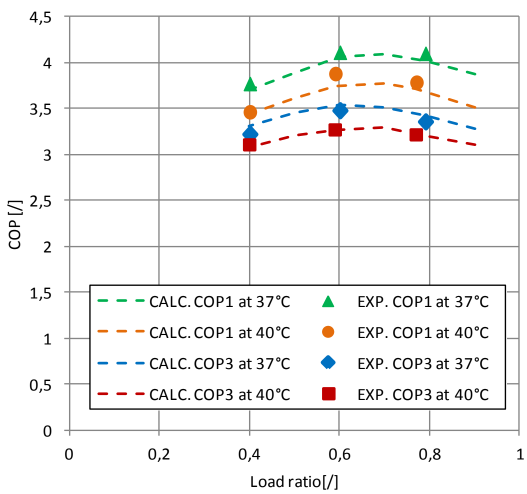

Fig. 11. Comparison between experimental and calculated maximum values of COP1 and COP3 at different test conditions.

To achieve the maximum seasonal performance, the present problem has been approached in two steps. In the first step, the curve of the thermal power is divided in *n* intervals. 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 variables that allow to produce the desired thermal power with the maximum system efficiency. By repeating this procedure for the different values of load ratio, the seasonal performance factor *SCOP3* can be calculated according to Eq. (6), where the denominator (electric consumption) is equal to:

$$\text{El.energy}_{COMP} + \text{El.energy}_{ECP} + \text{El.energy}_{ICP} = \sum_{i=1}^{n} \frac{E(\alpha_i)}{COP3_{MAX}(\alpha_i)} \tag{7}$$

with $E(\alpha_i)$ being the fraction of the seasonal thermal energy demand corresponding to the i-th load ratio, $COP3_{MAX}(\alpha_i)$ the maximum value of *COP3* obtained for the i-th value of load ratio and *n* the total number of load ratio intervals. In the second step, the procedure is repeated for a different value of the supply water temperature and it is possible to compare the *SCOP3* at different supply water temperatures. Using the distribution of the thermal energy and applying Eqs. 6 and 7, the maximum seasonal performance factor *SCOP3* during the winter season can be calculated. Comparing the *SCOP3* calculated data, at the same operating conditions, an increase of about 30% could be achieved using a BLDC motor compressor instead of the the inverter-driven asynchronous motor compressor.

5. Conclusions

In this paper a model and experimental values of *COP* of a ground source heat pump system with variable speed drives are presented. The parameters that can be varied for the heat pump system control are: electric frequency of compressor, frequency of water pumps in the external circuit, frequency of the water pump in the internal circuit, velocity of the fans and supply water temperature to the fancoils. During field tests, by varying the water flow rate in the heat exchangers, experimental data of *COP* at variable supply water temperatures and different values of load ratio were taken.

When decreasing the load ratio, the maximum value of the system *COP* (*COP3*) is obtained at the minimum values of water 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 becomes important as it is demonstrated when keeping constant the β parameter (Eq. 5). At higher load ratio the maximum *COP3* is obtained for a water temperature difference at the condenser of about 5 K.

The present model allows to predict the system behavior: values of *COP3* are predicted in the range ±10%, so it can be used as the baseline to develop a proper control strategy.

In the model the lookup tables of any components can be easily replaced, in this way, different system layouts can be compared. Substituting the asynchronous motor compressor LUT with a BLDC motor compressor LUT, the possible *SCOP3* increase is about 30%, due to higher efficiency of the BLDC at partial loads.

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