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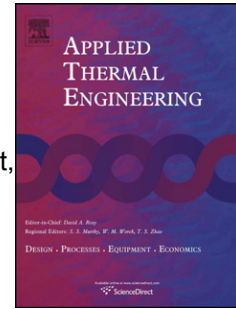
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> We modeled a ground source heat pump for residential application > Technical and economic analysis > Optimal arrangement of geothermal heat exchanger determined > Optimal operation studied > Winter heating and summer free-cooling are the best options

ACCEPTED MANUSCRIPT

# Feasibility study and numerical simulation of a Ground Source Heat Pump Plant, applied to a residential building

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**ABSTRACT:** To evaluate the efficiency of ground source heat pump (GSHP), an application is studied in a residential building divided in two independent apartments blocks.

The numerical results are produced by using the software tool TRNSYS 16. The results include the trends of the thermo-climatic variables and a comparison between conventional and geothermal systems analysing operation and capital costs.

The GSHP studied in this paper produces a thermal power output of 29 kW in winter and a cooling power output of 26 kW during summer time.

The numerical simulations have highlighted that with such a system, the GSHP has a winter coefficient of performance of 4.9, and a warm season coefficient of performance which reaches 2.3. The simulations show that the best plant configuration consists of a winter heating GSHP system and a summer free-cooling bypass. The thermo-climatic variables inside the building provide good comfort conditions.

Keywords: Ground source heat pump, renewable energy.

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**NOMENCLATURE**

$q$	Thermal energy [kWh]
$\dot{Q}$	Thermal power [W]
$\dot{Q}_{T,e}$	Thermal power lost through external walls [W]
$\dot{Q}_{T,x}$	Thermal power lost toward not heated rooms [W]
$\dot{Q}_{T,E}$	Thermal power lost to the ground [W]
$\dot{Q}_{a,i}$	Thermal power lost through ventilation [W]
$\dot{Q}_{p,i}$	Thermal power lost in the heating distribution [W]
$h_f$	Daily hours of cold[h]
$h_g$	Daily hours of heat[h]
NN	Heating days
NR	Cooling days
<b>Subscripts</b>	
GSHP	Ground source heat pump
COP	Coefficient of performance
$\Delta T_1$	Yearly temperature's variation
$\Delta T_{25}$	25 years temperature's variation
<b>Greeks</b>	
$\varepsilon$	Utilization factor

## 1. INTRODUCTION

The groundsource heat pump (GSHP) has historical origins in the European and American regions with unfavourable winter weather conditions where conventional systems need considerable amounts of thermal energy to comfortably heat buildings.

Geothermal systems have been used in northern Europe and America for a long time, but were not used on commercial scale in the USA until 1980s[1].

This explains the territorial diffusion and trend of the development of these kinds of systems.

The following figures indicate that there are over 150,000 groundwater and 250,000 ground closed loop, heat pump installations in the USA. The annual growth rate is estimated at 10%. GSHPs account for 63% of the geothermal direct use in the USA, amounting to 12,000 TJ (3,340 GWh) annually [1-2]. In Europe, the spread of the GSHPs is different in each country [3-7]. It is influenced by national politics, energy, weather and cultural situations.

Geothermal heat production has been increasing steadily with over 64 PJ in 2005 generated by approximately 7488 MW of installed capacity. GSHPs accounted for much of this heat (36.6 PJ) with an installed capacity of 5379 MW. However, the total contribution of geothermal heat has remained negligible at only 0.4% of 2006 total heat supply. Total geothermal heat production (direct use and low temperature<sup>1</sup>) in Hungary amounts at 690.2 MW, in Italy the installed capacity is 486.6 MW, while France had the highest levels of installed capacity in 2005 and a capacity of 291.9 MW. The highest contribution of heat from GSHP is largely in Sweden, Switzerland, France, Germany and Austria with respectively an installed capacity of 1700 MW, 880 MW, 549.5 MW, 632.6 MW, 611.5 MW.

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<sup>1</sup> Low temperature geothermal heat users are the GSHP plants

Most support for geothermal heat has been in the form of installation targets and support for research activities such as the Enhanced Geothermal Innovative Network for Europe and the Soultz-sous-Forêts pilot project involving France, Germany, Italy, Switzerland, Japan and the US [4].

Currently, 95% of new buildings for civil use in Sweden are designed to use GSHP systems. Other countries that have a high number of this type of heating plants are Germany and Austria. In Italy, geothermal energy could be a relevant resource: 11 of 20 regions have 30%-60% of soil with good characteristics both for industrial and domestic geothermal plants. At present in Italy there are 723 MW of geothermal plants both for power production and for direct use for residential utilization. Within 2020, CO<sub>2</sub> atmospheric emission could decrease by 8-10 millions of tons using geothermal energy [8]. In Italy the target is to improve the utilization of geothermal energy from 0.6% to 1.2%. Asian countries are also considering GSHPs to reduce energy consumption and reduce the environmental impact of air conditioning in urban areas [9-10]

The main advantage of GSHPs derives from their better performance in comparison with traditional systems [11-14], owing to a more stable temperature of the heat source during the whole year. Underground temperature is practically constant and close to 15 °C for most of the year.

The main drawback of this technology is the necessity of exchanging heat with the soil and this requires drilling a number of holes several meters deep, where the working fluid will be flowing. This increases considerably the overall capital costs of the installation.

There are two types of heat exchanger arrangements with the ground: vertical and horizontal heat exchangers [2]. The vertical heat exchangers are employed when enough surface is not available considering the heat transfer demand. This type of system is less influenced by the external weather conditions and by the sun radiation than horizontal heat exchangers, because the boreholes are drilled several meters below the surface.

In the horizontal arrangement, pipes are buried in trenches spaced a minimum of 1.5m apart and from 1.2 to 1.8m deep. This spacing is the minimum distance to avoid thermal interference between

pipes. In this type of heat exchanger, the performance may be greatly influenced by the local climatic conditions [18].

The operating conditions of the GSHP and its components, evaporator, compressor, condenser and expansion valve were simulated by using the TRNSYS 16 software tool [15].

The parts of the system that are implemented in TRNSYS are:

- the residential building;
- the GSHP whose operational characteristics were introduced by using performance maps describing the values of thermal power that during summer and winter are provided and absorbed, when evaporator and condenser temperatures change.
- the geothermal heat exchanger.
- free-cooling bypass;
- storage volume, dimensional features of the heat exchangers and thermal properties of the ground.

The final result of the calculations are the trends of the global variables with time, and the optimisation process of the energy performance of the building, heat pump, ground heat exchanger, ground.

The numerical simulations of underground behaviour when a GSHP is installed is not straightforward. The environmental impact due to the geothermal heat exchangers is commonly studied with the aim to verify the environmental sustainability of this type of plant.

When the plant sizing is well done, a steady state heat balance between the undisturbed natural soil conditions and the new condition where the energy is extracted and transferred to the heat exchangers is reached after a few years. The thermal behaviour near the geothermal heat exchangers is complex and it depends by many overlaying processes:

1. a substantial soil cooling near the heat exchangers (at a distance shorter than 10 cm) when the GSHP is operated in heating mode ;



2. a soil cooling effect that radially decreases from the borehole to a few meters of distance, creating a typical funnel temperature trend during a seasonal cycle (few months);
3. on the long term, at least a radius of 10 m around the boreholes of cooled soil during the all plant's life cycle (25-30 years).

The short period effect is due to the thermal load of the ground heat exchanger: the lower the soil temperature near the borehole, the higher the power extracted around the heat exchanger. The long period effect is due to the annual transferred energy from and to the soil. The higher is the soil temperature difference between the initial and the final conditions (after 25-30 years), the higher is the energy amount yearly extracted by the heat exchangers from the total storage volume of the ground.

Usually GSHPs are sized to minimize the drilling cost, i.e. to minimize the overall heat exchangers length. Furthermore, a suitable sizing avoids a quick soil temperature drop, because this may cause a performance decrease of the GSHP. It is acceptable that the soil temperature does not change more than 2 °C in the long term (25-30 years). This temperature change is considered as reasonable by VDI4640 code [16] and the ground heat storage software EED by Buildingphysics [17], which is one of the leading tools to calculate heat transfer in ground coupled heat exchangers.

In this paper several different plant configurations were studied with the aim of comparing the results as a function of the number of boreholes and depth, of serial and parallel pipe arrangement, of the kind of soil. The aim of this paper is to define the best technical and economic plant configuration, with reference to a real case study.

## **2. BUILDING FEATURES**

To evaluate the efficiency of the GSHPs, a plant was studied for a residential building. The residential building which is chosen for this simulation is located in Central Italy, near the city of Perugia. It is a house divided into two apartments distributed on two floors. The ground floor

surface is 174 m<sup>2</sup> and the first floor has a surface of 248 m<sup>2</sup>. The volume, excluding the building structures, is approximately 1200 m<sup>3</sup>. The walls facing north and south have an area of 117.75 m<sup>2</sup>. The walls facing east and west have an area of 135.30 m<sup>2</sup>. Heat and cooling is distributed internally by means of radiant panels under the floor, which allow using a lower temperature delivery in the winter and a higher in the summer.

To calculate the heat and cooling demand, a simplified method is used which does not consider the solar input. The result of the calculations provides a winter nominal heat demand of about 26.5 kW, divided among the different contributions as follows:

$$\dot{Q}_{\text{requested}} = \dot{Q}_{T,e} + \dot{Q}_{T,x} + \dot{Q}_{T,E} + \dot{Q}_{a,i} + \dot{Q}_{p,i} = 26.5 \text{ kW} \quad (1)$$

$$\dot{Q}_{T,e} = 8.64 \text{ kW}$$

$$\dot{Q}_{T,x} = 2.09 \text{ kW}$$

$$\dot{Q}_{T,E} = 0.60 \text{ kW}$$

$$\dot{Q}_{a,i} = 11.71 \text{ kW}$$

$$\dot{Q}_{p,i} = 3.46 \text{ kW}$$

From this data it was possible to size the geothermal heat pump.

The building structure and the heat distribution plant have been implemented in TRNSYS 16. The building is divided into three different parts, called thermal zones: “Z1”, “Z2”, “NOT HEATED”.

A thermal zone is a part of the building where the climatic variables can be considered uniform.

### 3. THE SELECTION OF THE GROUND SOURCE HEAT PUMP

The ground source heat pump was selected considering the winter heat load of about 26.5 kW. The selected heat pump, has a winter thermal power of about 29 kW and a summer thermal power of about 26.1 kW, in nominal conditions. These figures refer to:

- summer conditions: evaporator water temperature:  $12^{\circ}\text{C}$ – $7^{\circ}\text{C}$ ; water temperature to the condenser:  $15^{\circ}\text{C}$ – $30^{\circ}\text{C}$ ;
- winter conditions: condenser water temperature:  $40^{\circ}\text{C}$  –  $45^{\circ}\text{C}$ ; water temperature to the evaporator:  $15^{\circ}\text{C}$ . [6]

This configuration has been tested and verified by numerical simulation of all possible operating conditions. This is possible by using a special routine of TRNSYS 16, named Type 668, which allows inserting the performance map of the device. The performance map includes the values of thermal power that during summer and winter are provided and absorbed, when evaporator and condenser temperature change.

#### **4. GEOTHERMAL HEAT TRANSFER**

The main parameters, which influence the geothermal heat transfer are the following:

- The kind of underground soil;
- The amount of heat to transfer;
- The type of geothermal heat exchanger.

In the present study many different kinds of underground soil and type of geothermal heat exchanger are investigated varying the scheme of a reference plant configuration.

##### **4.1 Features of the underground soil**

The type of soil influences heavily both the sizes and the number of the pipes of the heat exchanger.

The two main characteristics of the soil are:

- thermal conductivity;
- thermal capacity.

As it was not possible to find detailed information about the kind of soil where the building is located, it was assumed, from information of near excavations, that the underground is mostly

composed by clayey orogenesis deposits<sup>2</sup>. Furthermore two different types of soil are considered with the aim to compare the results relating different soil characteristics. The characteristics of the three types of underground are reported in Tab. 1.

INSERT TABLE 1

#### 4.2 Amount of heat to exchange

The geothermal boreholes must be able to absorb from the ground, during the winter, the required thermal energy to heat the building. Conversely, during the summer, they must be able to dissipate the heat in order to cool the building. The amount of these transferred energy flows is based on the heat pump power output. As the power of the device is larger than the power required by the building, the heat exchanger size has been calculated by the procedure reported in Tab. 2 and Tab. 3.

INSERT TABLE 2

INSERT TABLE 3

The annual energy load for heating is calculated with the following equation:

$$Q_{\text{heating}} = Q_{\text{requested}} \cdot \varepsilon \cdot NN \cdot h_g \quad (2)$$

where  $Q_{\text{requested}}$  is the power of the heat pump in heating mode. The annual energy load for heating amounts at 52,009 kWh. Likewise, the annual energy load for cooling is calculated with equation 3:

$$Q_{\text{cooling}} = Q_{\text{requested}} \cdot \varepsilon \cdot NN \cdot h_f \quad (3)$$

where  $Q_{\text{requested}}$  is the power of the heat pump in cooling mode. The annual energy load for cooling

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<sup>2</sup> From Greek.  $\omega\rho\omicron\varsigma$  = elevation +  $\gamma\acute{\epsilon}\nu\epsilon\sigma\iota\varsigma$  = origin. An orogenesis deposit is made by rocks that was be subjected to tectonic deformation due to a side push that createa chain of mountains.

amounts at 22,289 kWh.

### 4.3 Heat exchanger configuration

Even though there is enough surface around the building to accommodate a horizontal heat exchanger, the vertical type was selected to have a higher efficiency, less sensitivity of climatic conditions and a more flexible utilization of the garden space.

The features of the reference plant configuration boreholes (defined for the case of study) are reported in Tab. 4 .

INSERT TABLE 4

Different heat exchanger arrangements are simulated. In particular, the influence of the number of the vertical drillings and their depth on the soil temperature is studied by varying the scheme of the reference plant configuration. On the contrary, the influence of the heat exchangers arrangement (serial or parallel) and the distance between the boreholes are investigated.

Finally, the reference plant configuration was simulated with a geothermal heat exchanger consisting of nine boreholes arranged in strings of 3 pipes. Each string is connected in parallel with the others in a reverse mode (the first string at inlet is the last one at outlet). The depth of each borehole (reference case) is 80 m. The linear distance between boreholes is 8 m

## 5. OPERATION OF THE PLANT

The operational mode of the plant is determined by the heat and cooling demand of the user. In this case, considering the particular weather conditions of the site, it was decided that this plant was to work with three different modes in winter and summer.

The first mode is for winter time. From 10:00 pm to 05:30 am the plant will keep the temperature of the heated spaces at 16 °C, whereas from 05:30 am to 10:00 pm, the plant will keep the temperature of the heated space (zones Z1 and Z2) at 20°C.

The second (A) and third (B) modes are employed in the summer and the results are compared with each other.

- Mode A: if the temperature of the cooled area exceeds 26°C, the GSHP will start operating in cooling mode.
- Mode B: the use of free cooling is considered. In this mode a bypass system consisting of a pump and a plate heat exchanger cools the fluid of the radiant floor directly using the fluid circulating in the geothermal heat exchangers loop.

## 6. INFLUENCE OF THE SOIL TYPE

The underground minimum soil temperatures reached in the long term (25-30 years) are investigated with the aim to define the best soil type where to build a GSHP. The numerical simulations are done considering three underground soil types.

In case 1 the soil consists of clay-silt with low thermal conductivity and low thermal capacity.

In case 2 the soil is the most common type in the region where the house is built, and mainly consists of clay orogenesis deposits with a medium moisture. Such a soil has average thermal characteristics.

Case 3 is on a sedimentary rocks soil with high thermal characteristics values.

The numerical simulation shows how the best type of soil is in the third case, because the temperature drop in the long term does not exceed 0.52 °C. The soil type of the case study is the

second one. The temperature drop after 25 years of GSHP operation reaches  $1.63^{\circ}\text{C}$ , value that can be still considered acceptable.

The worst soil type is in the first case, because of low thermal characteristics values. The results are shown in Fig. 1 and Fig. 2. The Fig. 2 shows the interpolation of the minimum temperature reached in 25 years.

INSERT FIGURE 1

INSERT FIGURE 2

The results show how sedimentary rocks ground are the best suited for GSHPs, while the worst are the clay-silt types and the sandy-gravel soils. The geological characteristics of the soil in the area, where the GSHP will be build, are good.

## **7. INFLUENCE OF THE HEAT EXCHANGER BOREHOLE NUMBER AND DEPTH**

With the aim of understanding the influence of the borehole number, being equal the depth of each drilling, the arrangement of the piping, and the plant and soil characteristics, many simulations were done considering 6, 9, 12 drills. Nine heat exchangers is the number selected for the reference case with a total length of 720 m, because it is the number calculated by using the procedure described section 4.2. Both the configuration with 12 and 9 heat exchangers are acceptable because the temperature differences between the lie in the optimal range. The configuration with 12 boreholes is the best from a thermodynamic point of view, but the one with 9 boreholes is less expensive.

Figure 3 shows the mean temperature of the underground soil where the heat exchangers are located. Fig. 4 shows the interpolation of the minimum temperature reached during 25 years of simulations.

INSERT FIGURE 3

INSERT FIGURE 4

In the case of 6 heat exchangers with a total length of 480 m, the annual temperature's oscillation ( $\Delta T_1$ ) is high: the underground heat storage is subject to an annual temperature variation that reaches 1.95 °C. High temperature variations affect the GSHP performance and could represent a stress factor for the soil. The long term effect is an underground temperature drop ( $\Delta T_{25}$ ) of 2.1°C. The base case configuration (9 heat exchangers) produces a  $\Delta T_{25}=1.63^\circ\text{C}$  and  $\Delta T_1=0.81^\circ\text{C}$ .

The best case is with 12 heat exchangers with a total length of 960 m: the annual variation of temperature is 0.63 °C with a temperature drop after 25 years equal to 1.27 °C.

The above simulations are influenced by the total length of the drills. To avoid this problem new simulations were done maintaining constant the heat exchanger overall length (720 m) and considering 6 drills (one drill length is 120 m), 9 drills (L=80 m), 12 drills (L=60 m).

The simulation results show that the underground soil temperature is higher if the borehole is deeper. The best configuration is with 6 boreholes, even if the temperature difference of 1.67 °C is the highest of the three arrangements, meaning an undersized number of boreholes.

INSERT FIGURE 5

Comparing the simulation of figs. 3 and 5 is possible to state that large annual temperature variations occur when the total heat exchangers length is undersized. A large long term temperature drop indicates an undersizing of both the boreholes number and overall heat exchanger length.



## 8 INFLUENCE OF THE DISTANCE BETWEEN BOREHOLES

An important factor that has to be considered when designing an underground heat exchanger for GSHP applications is the thermal interference between the boreholes.

Indeed, when a vertical heat exchanger is in the region of thermal influence of others drills, the heat exchanger performance changes. The thermal interference effect is difficult to predict during the plant design phase, but it strongly depends on the centre to centre distance of boreholes. In particular in large heat exchanger fields, used in large GSHPs, the correct borehole arrangement is essential. The common state of the art distance between boreholes is 8-10 m.

Several simulations were done using TRNSYS 16, with the aim of studying the boreholes distance effect on the soil temperature and the heat exchanger fluid temperature. The simulations are done considering 9 drills with a length of 80 m each and distances equal to 4 m, 8 m, 12 m.

The calculations show a  $\Delta T_1$  and  $\Delta T_{25}$  dependence on the boreholes distance.

When the boreholes are too close ( $D=4\text{m}$ ) the annual temperature variation and the long term temperature drop are too high. These values are smaller as when the boreholes distance is larger.

## 9 INFLUENCE OF THE HEAT EXCHANGER ARRANGEMENT

The soil temperature trend in the long period is influenced by the heat exchanger arrangement too.

Using again TRNSYS 16, it was possible to define the optimal arrangement for the available ground surface.

Several simulations were done considering 12 heat exchangers ( $L=80\text{ m}$ , borehole distance  $D=8\text{m}$ ) with the following arrangement:

- 6 series of 2 drills;
- 4 series of 3 drills;
- 3 series of 4 drills;
- 2 series of 2 drills.

Since no appreciable difference in the underground average temperature was found, the heat exchangers fluid temperatures were studied. The worst plant configuration is the one with 6 series of 2 boreholes, while the best one is with 1 series of 12 boreholes. In fig. 6 the curves of the minimum fluid temperature reached every year are plotted. To reduce costs and to use larger groundsurfaces, the arrangements with 3 series of 3 drills, 3 series of 4drills and 4 series of 3drills (with rectangular/square layout ) are preferred. If the configuration with 9 boreholes is selected, the best arrangement is with 3 series of 3 drills, since it has the lowest cost and smallest use of space.

INSERT FIGURE 6

## 10 COMPARATIVE SIMULATIONS

A comparative simulation of several plant configurations was done to define the acceptable range of  $\Delta T_1$  and  $\Delta T_{25}$  for a correct heat exchangers sizing.

In figs. 7 and 8 the underground temperature trends and the annual minimum temperature during 25 years of plant operation are reported for the following heat exchanger arrangements:

1. 12 heat exchangers, L=120, boreholes distance D=12 (code: 12.120.D12);
2. 9 heat exchangers, L=80, boreholes distance D=8 (code: 9.80.D8);
3. 9 heat exchangers, L=100, boreholes distance D=8 (code: 9.100.D8);
4. 9 heat exchangers, L=120, boreholes distance D=10 (code: 9.120.D10);
5. 6 heat exchangers, L=80, boreholes distance D=8 (code: 6.80.D8).

INSERT FIGURE 7

INSERT FIGURE 8

The results show the correct sizing of the reference case plant (9.80.D8). The simulation are done considering that the GSHP operates in cooling mode in the summer. To avoid a long term temperature drop during summer free-cooling operation, the 9.100.D8 or the 9.100.D10 plant configurations are better than the others.

The acceptable range of  $\Delta T_1$  and  $\Delta T_{25}$  for a correct heat exchangers sizing is:

$$0.50 \ll \Delta T_1 \ll 1 \text{ and } 1 \ll \Delta T_{25} \ll 1.65$$

The geothermal heat exchanger field is undersized when:

$$\Delta T_1 > 1 \text{ and } \Delta T_{25} > 1.65$$

The geothermal heat exchanger field is oversized when:

$$\Delta T_1 < 0.50 \text{ and } \Delta T_{25} < 1$$

All the above parameters are deduced using the data obtained by the simulations previously shown and will be tested, compared and validated as soon as possible with more GSHP plant projects and simulations in future research activities.

The installation of an oversized heat exchanger field, such as the 12.120.D12, leads to an extra cost of 36,000 € against a yearly saving of 25 € of electrical energy: which is not economically convenient.

Finally a comparative simulation between the two following plant configurations was done:

1. 9 heat exchangers, L=80, boreholes distance D=8 (code: 9.80.D8 Freecooling) using a GSHP for winter heating and a free-cooling bypass for summer cooling;
2. 9 heat exchangers, L=80, boreholes distance D=8 (code: 9.80.D8) using a reversible GSHP for winter heating and summer cooling.

Fig. 9 shows the interpolation of the yearly minimum temperatures of the soil. The results show that the plant configuration with a ground heat exchanger field composed by 9 boreholes, 80 m deep and with a borehole distance of 8 m is well sized for a GSHP used for winter heating and summer free-cooling. Using the free-cooling operation there is no efficiency drop of the GSHP, because the  $\Delta T_1$

and  $\Delta T_{25}$  values are in the optimal range. The free-cooling operation also leads to an optimal thermal regeneration of the soil with a temperature drop after 25 years that is lower than  $1.09\text{ }^{\circ}\text{C}$ , and also lower than the  $1.63\text{ }^{\circ}\text{C}$  temperature drop in the case of a reversible GSHP plant installation. In fact the case of reversible GSHP in cooling mode, the fluid flowing in the radiant floors quickly reaches the working temperature to cool the rooms and so the control system will quickly stop the compressor and the recirculation pumps, especially the ground heat exchangers. This will cause some on/off cycles during the day. In the case of free-cooling, a more constant and continuous fluid recirculation in the ground heat exchangers is necessary, causing an optimal thermal regeneration of the ground.

INSERT FIGURE 9

## 11. CASE OF STUDY SIMULATION RESULTS

The case of study will use the reference case plant configuration with 9 ground heat exchangers ( $L=80\text{ m}$ ,  $D=8\text{ m}$ ) arranged in 3 strings of 3 boreholes (code: 9.80.D8).

Figure 10 shows the trend of air temperature in the thermal zones “Z1”, “Z2”, “NOT HEATED”, during the year.

INSERT FIGURE 10

In winter, the air temperature of the heated thermal zones (“Z1”, “Z2”) is between  $19^{\circ}\text{C}$  and  $23^{\circ}\text{C}$ .

The air temperature of the zone “NOT HEATED” and the temperature of the air outside the building follow the same trends.

If the air temperature of the heated zones is between  $20^{\circ}\text{C}$  and  $26^{\circ}\text{C}$ , the plant will be switched off.

### 11.1 Trend of the plant variables in January

The simulation in January allows to analyse the behaviour of the plant variables, in one of the coldest month of the year. Every variable oscillates with a period of a day.

It is possible to make several considerations:

- the difference between the lowest and the highest temperature values of the thermal zone “Z1”, is about 2.5°C;
- the difference between the lowest and the highest temperature value of the thermal zone “Z2”, is about 2°C;

These considerations are shown in Fig. 11

INSERT FIGURE 11

A very important aspect is the time of operation of the plant. The simulation, referred to the three coldest days of the year, that were 28<sup>th</sup>, 29<sup>th</sup>, 30<sup>th</sup> January, highlighted the following aspects:

- in each of the considered days, the plant switches on at 05:30 am and it works for about 8 hours;
- the start-up is shorter than the shut down;
- during the period in which the plant is switched off, both in the boreholes and in the radiant panels, the fluid does not cool.

These three points mean a high thermal inertia of the system. Such factors produce two positive aspects:

- the decrease of the installation costs. It is due to the thermal accumulation which allows using a heat pump with less nominal power;
- the decrease of operating costs, because in the evening hours the electrical energy is cheaper than in during the day.

## 11.2 Trend of the plant variables in summer

In summer the temperature trend of mode A and mode B plant configurations are compared.

In summertime (mode A, with the use of a GSHP in cooling mode), the mean air temperature of cooled thermal zone (“Z1”, “Z2”) is between 24.8°C and 25.3°C. The maximum temperature reached in zone 1 and zone 2 is 26.63°C and 26.49°C respectively.

Considering a free-cooling operation (mode B), the fluid circulating in the radiant floor is chilled directly by the cold fluid flowing on geothermal heat exchangers piping. In this case the heat pump is not on and it is possible to operate the system from 8:00 a.m. to 10:00 p.m. with a very low electric power consumption.

Using the free-cooling operation, the mean air temperature of cooled thermal zone (“Z1”, “Z2”) is between 23.54°C and 24.07°C. The maximum temperature reached in zone 1 and zone 2 is 26.13°C and 25.68°C respectively (Fig.12).

INSERT FIGURE 12

Comparing the temperature range of the two cooling plant modes of operation, the use of free-cooling appears better than the use of the GSHP in cooling mode because:

1. using the free-cooling plant operation the temperatures in the cooled zone are lower than using the GSHP in the cooling mode and inside the temperature comfort range.
2. The temperature standard deviation in mode A is 2.50°C and 2.05°C for zone 1 and 2 respectively, the temperature standard deviation in mode B is 1.113°C and 0.965°C for zone 1 and 2 respectively. These data mean a smaller temperature variation with the free-cooling operation than with the cooling mode. The use of free-cooling leads to improved comfort conditions.

3. It is possible to use a wide range of free-cooling plant operation time with a very low consumption of electric power.

The air temperature of the zone “NOT HEATED” and the temperature of the air outside the building follows the same trends in each plant mode A and B.

## 12. COP YEARLY TRENDS

The winter COP is slightly over 4.9 and the summer one is about 2.3.

The winter value is much higher than the summer value for the following reasons:

- the ground source heat pump is optimized for winter operation;
- the geometry of the boreholes system.

COP in cooling mode is always lower than in heating mode, but this difference can have a higher or a lower value depending on the type of heat pump. It is possible to choose a ground source heat pump designed for a prevalent winter use (with very high COP in heating mode and low COP in cooling mode) or for a prevalent summer use (with middle value of COP in winter and summer mode). So the choice of the machine is strictly correlated to the prevailing type of yearly operation mode. In this paper it is supposed to have a prevailing winter use mode.

## 13. ECONOMICAL CONSIDERATIONS

The evaluation of the capital costs of the plant was done in order to compare a conventional heating-cooling system with a GSHP. Some costs of the elements which are common in the two systems are neglected, such as circulation pumps and other auxiliary. The evaluation of the operating costs was done for both systems.

### 13.1 Installation costs

The costs of different type of heating-cooling plants are assumed and compared:

1. GSHP for winter heating and summer cooling;
2. GSHP for winter heating and free-cooling bypass for summer cooling;
3. natural gas boiler for winter heating and air-water refrigerator for summer cooling;

It is supposed that the two main cost items of the GSHP installation are the following:

- thermal power plant;
- geothermal heat exchanger.

The first one has an estimated cost (based on market analysis that consisted in contacting several Italian societies, operating in GSHP plants' business, that quoted the installation of the plants as specified by the current study) of about 33,000 € and the second one has an estimated cost of about 36,000 €.

So the overall cost of the plant is approximately 66,000 €.

The cost for configuration 2) is about 20% less than the first one: 53,000 €.

Finally the installation cost of the third plant type is supposed composed by the following costs:

- thermal power plant composed by a natural gas boiler for winter heating: 5,100 €;
- cooling plant composed by air-water refrigerator for summer cooling: 12,900€.

The overall cost of this plant is approximately 18,000 €.

Comparing the capital cost of the three different kinds of plants only, the third one is the cheapest.

However a meaningful cost comparison of the three different types of heating-cooling plants, also has to include the analysis of the operating costs and the calculation of the pay-back.

### **13.2 Comparative evaluation of winter operating costs**

The selected GSHP has a power output for winter heating of 29 kW which is consistent with the winter heat demand of 26.5 kW. Such a system can supply the necessary winter demand, that in the worst case of 14 hours of operation per day, gives 52 MWh/year of heating load.

Based on the GSHP power output it was possible to size the geothermal heat exchangers. The numerical simulations have highlighted that the real winter energy load of the building is much



lower than the supposed one in the preliminary sizing. It is possible to observe this from the daily operation hours of the heat pump, that is approximately 8 hours, much shorter than the preliminary assumption of 14 hours operation.

The values of energy demand are very different at each value of time step used for the simulation. The correct value is established and the above simulations are done with a time step equal to 30 minutes, because the energy demand simulated is close to the mean value and because the on/off cycles of the GSHP compressor can be at least 2 per hour, due to the high thermal inertia of the system.

The winter operating costs of GSHP are based on the electrical energy used and the COP of the device.

The COP considered in winter time is 4.9. Therefore, the annual consumption of electrical energy is about 4,527 kWh<sub>e</sub> giving a winter operating cost of 750 €. This calculation is based on the simulation of the energy demand (22,182 kWh/year calculated with TRNSYS 16 at time step 30 minutes) and on the electrical energy cost of 0.165 €/kWh as shown in Tab. 5.

INSERT TABLE 5

To satisfy the same heat demand using natural gas boilers (the combustion efficiency is close to 0.9), about 3,400 Nm<sup>3</sup> of natural gas are required. The annual maintenance costs are approximately 50 € and the natural gas cost is 0.61 €/Nm<sup>3</sup>, so it is possible to say that the yearly operating cost of this system is approximately 2,100 €.

### **13.3 Comparative evaluation of summer operating costs**

In summer three scenarios were compared.

The total thermal energy for cooling is 15 MWh per year. Using a reversible GHP that works in cooling mode with a COP equal to 2.3, the annual electrical energy consumption is 6,700 kWh<sub>e</sub> giving a summer operating cost of 1,100 € per year.

If a boiler for winter heating and a compression chiller for summer cooling are used the annual consumption of electrical energy is about 5,660 kWh<sub>e</sub> giving a summer operating cost of 930 €. The calculation is done considering the total thermal energy that has been removed from the building to cool it (15 MWh) and a COP equal to 2.65

Finally, planning a heating-cooling plant composed by a GSHP for winter heating and a free-cooling bypass, the operating costs are due to the free-cooling recirculation pump's energy demand only: i.e. 50€ per year.

#### **13.4 Comparative evaluation of pay-back period**

To evaluate the pay back period, the GSHP configurations are compared to the base case, which is the plant with a natural gas boiler and a compression refrigerator system.

The Italian government promotes the building energy saving giving a grant that amounts at 55% of the capital plant cost for high efficiency boilers, GSHP systems, solar panels installation.

The base case's installation costs are 18,000 € and the yearly operation costs are 3,030 €. The total cost using the government grant is lowered to 8,100 €.

The configuration with the reversible GSHP for winter heating and summer cooling has an installation cost of 66,000 €, and a cost using the government's grant of 29,700 €. The operation costs are 1,850 €/y.

A heating-cooling plant consisting of a GSHP for winter heating and a free-cooling bypass costs 53,000 € (23,850€ using the government's grant) and the operation costs are as low as 800 €/y.

The pay-back period of reversible GSHP plant is 40 years without taking advantage of government incentives and 18 years with government's grant use respectively.

The pay-back period of GSHP and natural cooling plant is 16 years without the government grant use and 7 with them respectively.

Comparing the pay-back periods it is possible to state that the GSHP with freecooling is better than the reversible GSHP. In particular using the Italian government grant this last configuration appears even more convenient than using a separate heating and cooling system (natural gas boiler and a compression refrigerator).

#### 14. CONCLUSION

In this paper the software TRNSYS 16 is used to study different variations of a base case GSHP scheme. In particular the average ground temperature trend in the long period were studied varying the design of the underground heat exchangers and type of soil: number, total length and distance between the boreholes. Analyzing the annual temperature variation of the soil and the long term temperature difference, it was possible to select the most appropriate heat exchanger arrangement and to define the temperature range  $\Delta T_1$  and  $\Delta T_{25}$  to size them. Moreover it was possible to check the optimal sizing of the reference case plant used.

Residential buildings heating and cooling systems, consisting of a GSHP with radiant panels as thermal distribution, can be installed in central Italy with good results both for the comfort inside the building and for the costs. In particular the high thermal inertia of the system produces two positive aspects: the decrease of the installation costs and the decrease of operating costs satisfying the comfort requirements.

The operating costs necessary to heat the building with the GSHP, are lower than the operating costs necessary to heat the building with a natural gas power plant. If we consider a conventional cooling system, a similar saving is possible in summer too.

The environment benefits are evident, because 2,300 kg/year of CO<sub>2</sub> (considering a CO<sub>2</sub> emission of 0.58 kg per kWh<sub>e</sub> and a CO<sub>2</sub> emission of 0.20 kg per kWh<sub>t</sub> burning natural gas) and other polluting emissions to the atmosphere are reduced. The installation costs are still very high, but it has to be considered that the GSHP also provides cooling during summer.

Comparing the three plant configurations capital costs and pay-back periods (1-natural gas boiler for winter heating and compression refrigerator for summer cooling, 2-reversible GSHP for winter

heating and summer cooling, 3-GSHP for winter heating and free-cooling in the summer) the third plant configuration appears the best solution.

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Fig. 1: Underground average temperature (varying the soil type).

Fig. 2: Underground min temperature (varying the soil type).

Fig. 3: Underground average temperature (varying the number of boreholes).

Fig. 4: Underground min temperature (varying the number of boreholes).

Fig. 5: Underground average temperature (varying the number of boreholes, total length 720 m).

Fig. 6: Fluid min temperature (varying the heat exchanger disposition).

Fig.7: Underground average temperature.

Fig. 8: Underground min temperature.

Fig. 9: GHP cooling Vs. Natural cooling.

Fig. 10Trend of the air temperature.

Fig.11: Trend of the plant variable in January.

Fig. 12:Trend of the air temperature in summer season using a freecooling plant configuration.

Tab. 1: Underground features.

Tab. 2: Winter energy request.

Tab. 3:Summer energy request.

Tab. 4:Feature of the borehole.

Tab. 5: Winter operating costs of ground source heat pump.

Type of ground 1	Clay-silt (dry)	Value	Unit
Thermal Conductivity		1.8	KJ/hmK
		0.5	W/mK
Thermal Capacity		1.5	MJ/m <sup>3</sup> K
Type of ground 2	Clay (medium moisture)		
Thermal Conductivity		6.12	KJ/hmK
		1.7	W/mK
Thermal Capacity		2.5	MJ/m <sup>3</sup> K
Type of ground 3	Sedimentary rocks		
Thermal Conductivity		12.6	KJ/hmK
		3.5	W/mK
Thermal Capacity		2.16	MJ/m <sup>3</sup> K

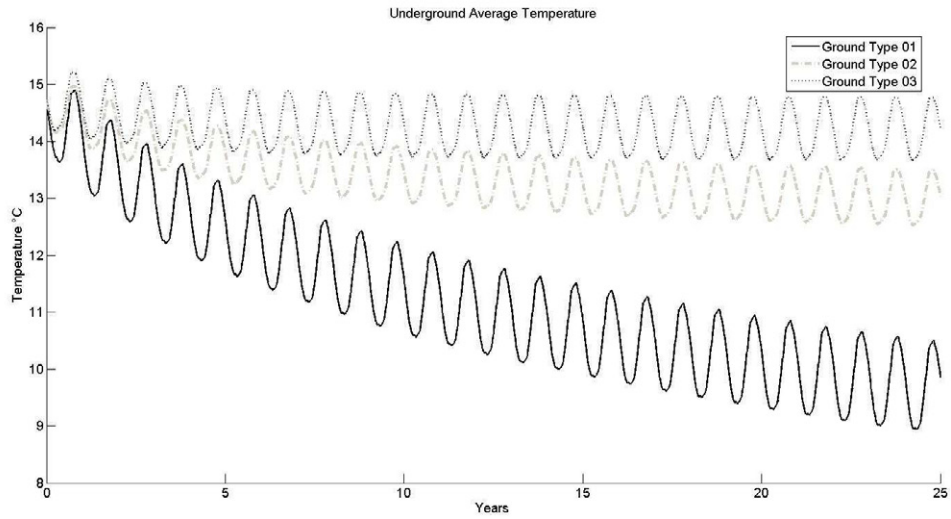
Parameter in Eq. (2)	Value	Unit
$\epsilon$	0.7	
NN	183	days
$h_g$	14	h
$Q_{\text{requested}}$	29	kW

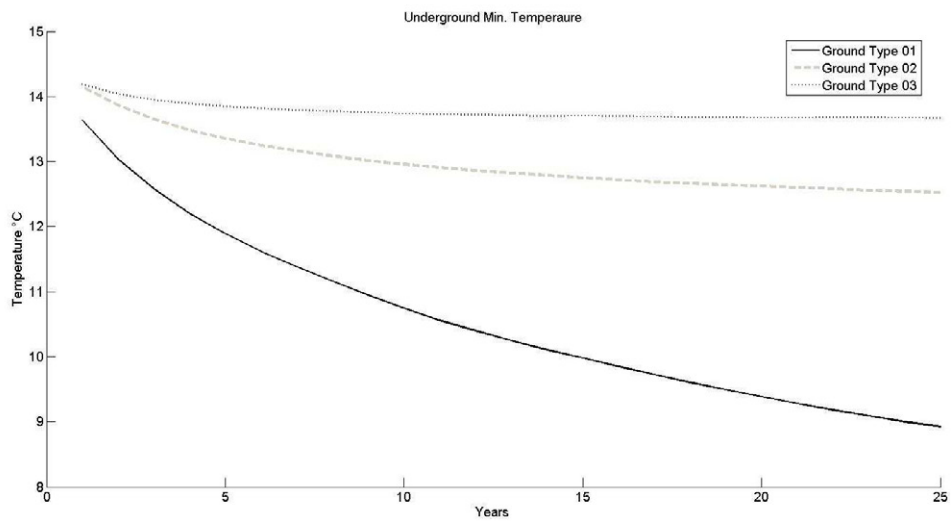


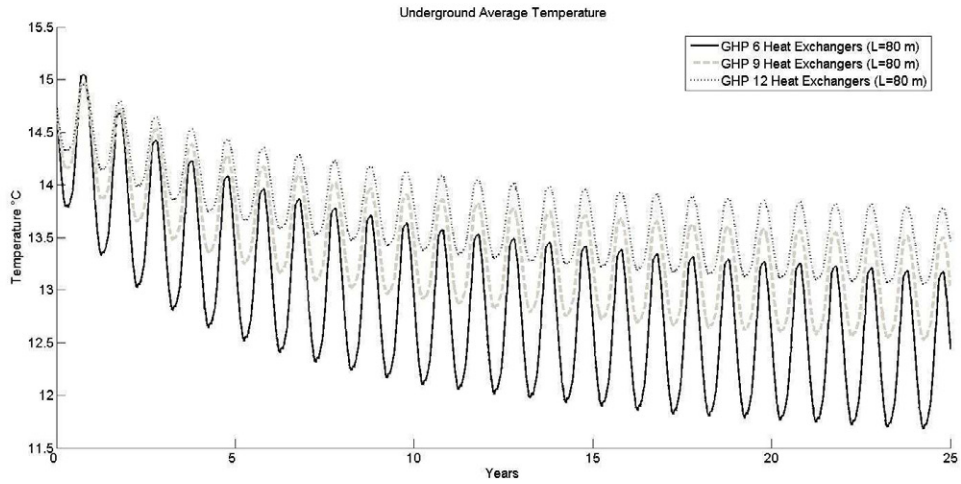
Parameter in Eq. (3)	Value	Unit
$\epsilon$	0.7	
NR	122	days
$h_f$	10	h
$Q_{\text{requested}}$	26.1	kW

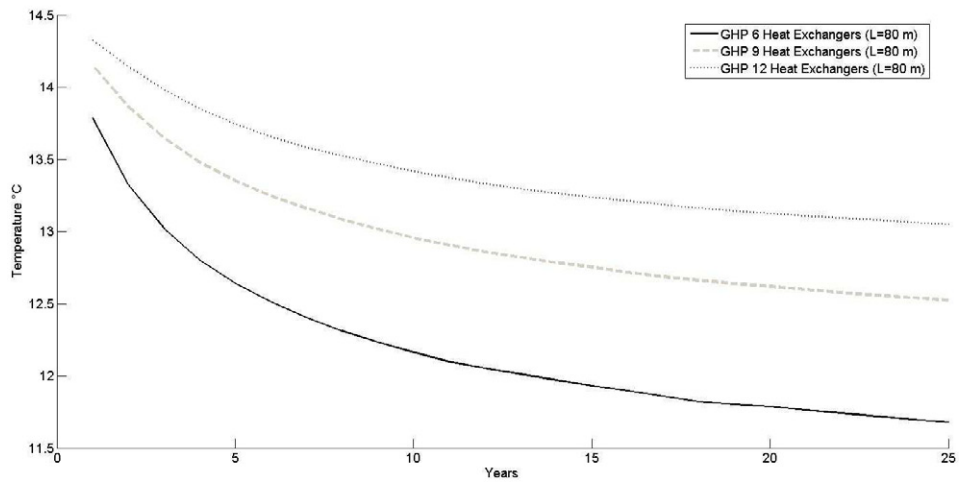
Features of boreholes	Value	Unit
Type of borehole	Vertical closed loop	
Borehole configuration	Single “U” pipe	
Piping material	Polyethylene	
Infilling	Bentonite-Concrete mix	
U-pipe external	0.032	m
Borehole diameter	0.15	m
Distance among boreholes	8	m
Borehole depth	80	m

Winter Operating Costs of GHP Systems		
Electrical Energy Demand	Electrical Energy Cost	Winter Operating Costs
kWh	€/kWh	€
4257	0.165	747









ACCEPTED MANUSCRIPT

