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Using Ground-Source Heat Pump Systems for Heating/ Cooling of Buildings

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Additional information is available at the end of the chapter

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Abstract

This chapter mainly presents a detailed theoretical study and experimental investigations of ground-source heat pump (GSHP) technology, concentrating on the ground-coupled heat pump (GCHP) systems. A general introduction on the GSHPs and its development, and a description of the surface water (SWHP), ground-water (GWHP), and ground-coupled heat pumps are briefly performed. The most typical simulation and ground thermal response test models for the vertical ground heat exchangers (GHEs) currently available are summarized. Also, a new GWHP using a heat exchanger with special construction, tested in laboratory, is well presented. The second objective of the chapter is to compare the main performance parameters (energy efficiency and CO₂ emissions) of radiator and radiant floor heating systems connected to a GCHP. These performances were obtained with site measurements in an office room. Furthermore, the thermal comfort for these systems is compared using the ASHRAE Thermal Comfort program. Additionally, two numerical simulation models of useful thermal energy and the system coefficient of performance (COP_{sys}) in heating mode are developed using the TRNSYS (Transient Systems Simulation) software. Finally, the simulations obtained in TRNSYS program are analysed and compared to experimental measurements.

Keywords: Geothermal energy, heat pump, ground heat exchanger, energy efficiency, radiator heating, radiant floor heating, experimental measurements, system performance, simulation models

1. Introduction

An economical strategy of a sustainable development imposes certainly to promote efficiency and a rational energy use in buildings as the major energy consumer in Romania and the other member states of the European Union (EU). Energy consumption patterns EU reveal that

buildings are the greatest energy consumer, consuming 41% of energy, followed by industry and transportation consuming approximately 30% [1]. Buildings represent the biggest and most cost-effective potential for energy savings. Also, studies have shown that saving energy is the most cost-effective method to reduce greenhouse gas (GHG) emissions. At present, heat use is responsible for almost 80% of the energy demand in houses and utility buildings for space heating and hot-water generation, whereas the energy demand for cooling is growing year after year.

In order to realise the ambitious goals for the reduction of fossil primary energy consumption and the related CO₂ emissions to reach the targets of the Kyoto-protocol besides improved energy efficiency, the use of renewable energy in the existing building stock have to be addressed in the near future [2].

On 23 April 2009, the European Parliament and the Council adopted the Renewable Energy Directive 2009/28/EC. It establishes a common framework for the promotion of energy from renewable sources. This directive opens up a major opportunity for further use of heat pumps for heating and cooling of new and existing buildings. Heat pumps enable the use of ambient heat at useful temperature level need electricity or other energy form to function [2]. Furthermore, EU member states must stimulate the transformation of existing buildings undergoing renovation into nearly zero-energy buildings (nZEBs). Conversion to heating and cooling systems based on ground-source heat pumps and air-to-water heat pumps is a well-proven measure to approach nZEB requirements.

Ground-source heat pump (GSHP) systems use the ground as a heat source/sink to provide space heating and cooling as well as domestic hot-water. The GSHP technology can offer higher energy efficiency for air-conditioning compared to conventional air-conditioning (A/C) systems because the underground environment provides higher temperature for heating and lower temperature for cooling and experiences less temperature fluctuation than ambient air temperature change [3]. To date, the GSHP systems have been widely used in both residential and commercial buildings. It is estimated that the GSHP system installations have grown continuously on a global basis with the range from 10 to 30% annually in recent years [4].

A ground-coupled heat pump (GCHP) system consists of a conventional heat pump coupled with a ground heat exchanger (GHE) where water or a water-antifreeze solution exchanges heat with the ground. The GHE may be a simple pipe system buried in the ground; it may also comprise a horizontal collector or, more commonly, borehole heat exchanger (BHE) drilled to a depth between 20 and 300 m with a diameter of 100–200 mm [5].

The widespread distribution of heat pumps as single generators in heating systems has mainly been in new, rather isolated buildings, thus having limited unit loads. This has enabled the use of low-temperature terminal units, such as fan coil units and, especially, radiant systems [6]. After the introduction of plastic piping water-based radiant heating and cooling with pipes embedded in room surfaces (floor, wall, and ceiling), the application increased significantly worldwide. Due to the large surfaces needed for heat transfer, the systems work with low water temperature for heating and high water temperature for cooling. However, in order to extend the use of these types of generators and benefit from their energy efficiency to reach

the targets of 20-20-20, it is also compulsory to work with radiators, which were the most commonly used terminal units in heating systems in the past.

This chapter mainly presents a detailed theoretical study and experimental investigations of GSHP technology, concentrating on the GCHP systems. Initially, the operation principles of a heat pump are described and their energy, economic and environmental performances are defined, showing the opportunity to implement the heat pump in a heating/cooling system. Then, a general introduction on the GSHPs and its development, and a description of the surface water (SWHP), ground-water (GWHP), and ground-coupled (GCHP) heat pumps are briefly performed. The most typical simulation and ground thermal response test models for the vertical GHEs currently available are summarized, including the heat transfer processes outside and inside the boreholes. Additionally, a new GWHP using a heat exchanger with special construction, tested in laboratory, is well presented. The second objective of the chapter is to compare the main performance parameters (energy efficiency and CO₂ emissions) of radiator and radiant floor heating systems connected to a GCHP. These performances were obtained with site measurements in an office room. Furthermore, the thermal comfort for these systems is compared using the ASHRAE Thermal Comfort program. Additionally, two numerical simulation models of useful thermal energy and the system coefficient of performance (COP_{sys}) in heating mode are developed using the TRNSYS (Transient Systems Simulation) software. Finally, the simulations obtained in TRNSYS program are analysed and compared to experimental measurements.

2. Operation Principle of a Heat Pump

A heat pump (HP) is a thermal installation that is based on a reverse Carnot thermodynamic cycle (consumes drive energy and produces a thermal effect). Any HP moves (pumps) heat E_s from a source with low temperature t_s to a source with a high temperature t_w , consuming the drive energy E_D . A heat source can be:

- a gas or air (outdoor air, warm air from ventilation, hot gases from industrial processes);
- a liquid called generic water: surface water (river, lake, or sea), ground-water, or discharged hot-water (domestic, technologic, or recirculated in cooling towers); or
- ground, with the advantage of accessibility.
- *Heat consumer*. The heat pump yields thermal energy at a higher temperature, depending on the application of the heat consumer. This energy can be used for:
 - space heating, which is related to low temperature heating systems: radiant panels (floor, wall, ceiling, or floor-ceiling), warm air, or convective systems; or
 - water heating (pools, domestic or technologic hot-water);

The heat consumer is recommended to be associated with a cold consumer. This can be performed with either a reversible (heating-cooling) or a double effect system. In cooling mode, a heat pump operates exactly like central air-conditioning.

- *Drive energy*. Heat pumps can be used to drive different energy forms:
- electrical energy (electro-compressor);
- mechanical energy (mechanical compression with expansion turbines);
- thermo-mechanical energy (steam ejector system);
- thermal energy (absorption cycle); or
- thermo-electrical energy (Peltier effect).

The GSHPs are those with electro-compressor. The process of elevating low temperature heat to over 38°C and transferring it indoors involves a cycle of evaporation, compression, condensation, and expansion (Figure 1). A non-CFC refrigerant is used as the heat-transfer medium, which circulates within the heat pump [7].

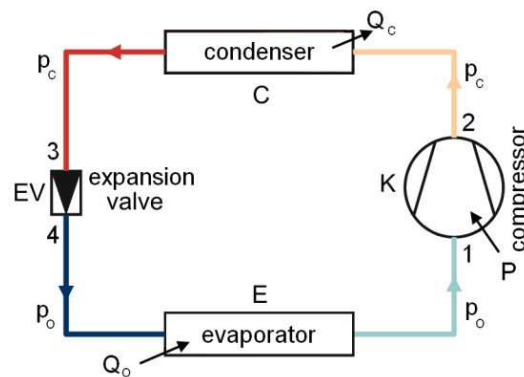


Figure 1. Schematic of single-stage compression refrigeration system

3. Performances and CO₂ emissions of a heat pump

The opportunity to implement a HP in a heating/cooling system results on the basis of energy indicators and economic analysis.

3.1. Energy efficiency

3.1.1. Coefficient of performance

The operation of a heat pump is characterised by the coefficient of performance (COP) defined as the ratio between useful thermal energy E_t and electrical energy consumption E_{el} :

$$\text{COP} = \frac{E_t}{E_{el}} \quad (1)$$

If both usable energy and consumed energy are summed during a season (year) is obtained by Eq. (1) seasonal coefficient of performance (COP_{seasonal}) or average COP over a heating (cooling) season, which is often indicated as seasonal performance factor (SPF) or annual efficiency.

In the heating operate mode, the heat pump COP is defined by equation:

$$COP_{hp} = \frac{Q_{HP}}{P_e} \quad (2)$$

where Q_{HP} is the thermal power (capacity) of heat pump, in W; P_e is the electric power consumed by the compressor of heat pump, in W.

In the cooling mode, a HP operates exactly like a central air conditioner. The energy efficiency ratio (EER) is analogous to the COP but tells the cooling performance. The EER_{hp} in Btu/(Wh), is defined as:

$$EER_{hp} = \frac{Q_0}{P_e} \quad (3)$$

where Q_0 is the cooling power of heat pump, in British Thermal Unit per hour (Btu/h); P_e is the compressor power, in W.

The coefficient of performance of heat pump in cooling mode is obtained by the following equation:

$$COP_{hp} = \frac{EER_{hp}}{3.412} \quad (4)$$

where value 3.412 is the transformation factor from Watt in Btu/h.

Figure 2 illustrates the COP variation of heat pumps in the heating operation mode, according to the source temperature t_s and the temperature at the consumer t_u [5].

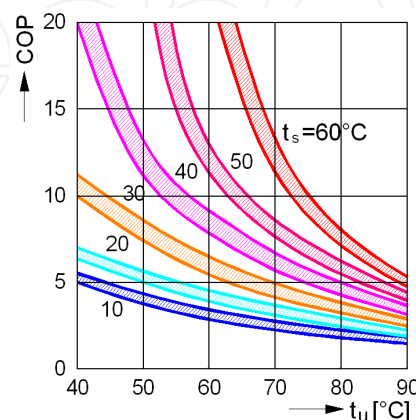


Figure 2. Efficiency variation of heat pumps

The GSHP systems intended for ground-water or oven-system applications have heating COP ratings ranging from 3.0 to 4.0 and cooling EER ratings between 11.0 and 17.0. Those systems intended for closed-loop applications have COP ratings between 2.5 and 4.0 and EER ratings ranging from 10.5 to 20.0 [8]. The characteristic values of the SPF of modern GSHPs are commonly assumed to be approximately 4, meaning that four units of heat are gained per unit of consumed electricity.

The sizing factor (SF) of the HP is defined as the ratio of the heat pump capacity Q_{HP} to the maximum heating demand Q_{max} :

$$SF = \frac{Q_{HP}}{Q_{max}} \quad (5)$$

The SF can be optimized in terms of energy and economics, depending on the source temperature and the used adjustment schedule.

3.1.2. Profitability and capabilities of heat pump

The factors that can affect the life-cycle efficiency of a HP are (1) the local method of electricity generation; (2) the local climate; (3) the type of heat pump (ground or air source); (4) the refrigerant used; (5) the size of the heat pump; (6) the thermostat controls; and (7) the quality of work during installation.

Considering that the HP has over-unit efficiency, to evaluate the consumed primary energy uses a synthetic indicator [5]:

$$\eta_s = \eta_g \text{COP}_{hp} \quad (6)$$

in which:

$$\eta_g = \eta_p \eta_t \eta_{em} \quad (7)$$

where η_g is the global efficiency and η_p , η_t and η_{em} are the electricity production, the transportation and the electromotor efficiency, respectively.

For justifying the use of a heat pump, the synthetic indicator has to satisfy the condition $\eta_s > 1$. Additionally, the use of a heat pump can only be considered if the $\text{COP}_{hp} > 2.78$.

The COP of a heat pump is restricted by the second law of thermodynamics:

- in heating mode:

$$\text{COP} \leq \frac{t_u}{t_u - t_s} = \varepsilon_c \quad (8)$$

- in the cooling mode:

$$\text{COP} \leq \frac{t_s}{t_u - t_s} \quad (9)$$

where t_u and t_s are the absolute temperatures of the hot environment (condensation temperature) and the cold source (evaporation temperature), respectively, in K.

The maximum value ε_c of the efficiency can be obtained in the reverse Carnot cycle.

3.2. Economic indicators

In the economic analysis of a system, different methods could be used to evaluate the systems. Some of them are: the present value (PV) method, the net present cost (NPC), the future value (FV) method, the total annual cost (TAC) method, the total updated cost (TUC) method, the annual life cycle cost (ALCC), and other methods.

- The PV of a future payment can be calculated using the equation [9]:

$$PV = \frac{C}{(1+i)^\tau} \quad (10)$$

where C is the payment/cost on a given future date; τ is the number of periods to that future date; i is the discount (interest) rate. Therefore, PV is the present value of a future payment that occurs at the end of the τ -th period.

Similarly, the PV of a stream of costs with a specified number of fixed periodic payments can be expressed as:

$$PV = u_r C = \frac{C}{CRF} \quad (11)$$

where the update rate u_r is defined as:

$$u_r = \sum_{n=1}^{\tau} \frac{1}{(1+i)^n} = \frac{(1+i)^\tau - 1}{i(1+i)^\tau} = \frac{1}{CRF} \quad (12)$$

where C is the periodic payment that occurs at the end of each period; n is the number of periods (years); CRF is the capital recovery factor.

- Another economic indicator is total updated cost:

$$TUC = I_0 + \sum_{n=1}^{\tau} \frac{C}{(1+i)^n} \quad (13)$$

where I_0 is the initial investment cost, in the operation beginning date of the system; C is annual operation and maintenance cost of the system; i is the discount (inflation) rate; τ is the number of years for which is made update (20 years).

Taking into account Eq. (12), Eq. (13) gets the form:

$$TUC = I_0 + u_r C \quad (14)$$

- Usually, the HP achieves a fuel economy ΔC (operating costs) comparatively of the classical system with thermal station (TS), which is dependent on the type of HP. On the other hand, the HP involve an additional investment I_{HP} from the classical system I_{TS} , which produces the same amount of heat [2].

Thus, it can be determined the recovery time RT , in years, to increase investment, $\Delta I = I_{HP} - I_{TS}$, taking into account the operation saving achieved through low fuel consumption $\Delta C = C_{TS} - C_{HP}$:

$$RT = \frac{\Delta I}{\Delta C} \leq RT_n \quad (15)$$

where RT_n is normal recovery time.

It is estimated that for RT_n a number 8–10 years is acceptable, but this limit varies depending on the country's energy policy and environmental requirements.

3.3. Calculation of greenhouse gas emissions

Due to the diversity in each country with respect to heating practices, direct geothermal energy use by GSHPs, and primary energy sources for electricity, country-specific calculations are provided.

The annual heating energy provided by GSHPs is defined as E_t . The annual primary energy consumption from heat pump electricity use is then:

$$E_{el} = \frac{E_t}{SPF} \quad (16)$$

Because heat pump electricity consumption is considered the most important source for greenhouse gas (GHGs) emissions [10], other potential contributors (e.g., heat pump life cycle, heat pump refrigerant, and borehole construction) are neglected. Applying an emission factor g_p , in kg CO₂/kWh, the annual GHG emissions C_{GSHP} , in kg CO₂, from GSHP operation can be obtained:

$$C_{GSHP} = g_p E_{el} \quad (17)$$

The emission factor typically varies among different countries and characterises the GHG intensity of electricity production. Note that although carbon dioxide (CO₂) represents the most important greenhouse gas, there exist several other compounds that contribute similarly to climate change. Their combined impact is commonly normalised to the specific effect of CO₂, and all emissions are expressed in CO₂ equivalents. For the sake of readability, however, the emissions are expressed only in kg CO₂.

Thus, the CO₂ emissions C_{CO_2} of the GSHP during its operation can be evaluated with the following equation:

$$C_{CO_2} = g_{el} E_{el} \quad (18)$$

where g_{el} is the specific CO₂ emission factor for electricity. The average European CO₂ emission factor for electricity production is 0.486 kg CO₂/kWh and for Romania is 0.547 kg CO₂/kWh [11].

4. Ground-Source heat pump systems

4.1. Generalities

Heat pumps are classified by (1) the heat source and sink; (2) the heating and cooling distribution fluids; and (3) the thermodynamic cycle. The following classifications can be made according to:

- function: heating, cooling, domestic hot-water (DHW) heating, ventilation, drying, heat recovery, etc.
- heat source: ground, ground-water, water, air, exhaust air etc.
- heat source (intermediate fluid)-heat distribution: air-to-air, air-to-water, water-to-water, antifreeze (brine)-to-water, direct expansion-to-water, etc.

Recently, the GSHP system has attracted more and more attention due to its superiority of high energy efficiency and environmental friendliness [4,5,12]. Renewable forms of energy such as solar, wind, biomass, hydro, and earth energy produce low or no GHG emissions. The temperature of the ground is fairly constant below the frost line. The ground is warmer in the middle of winter and cooler in the middle of summer than the outdoor air. Thus, the ground is an efficient heat source. A GSHP system includes three principle components: (1) a ground connection subsystem, (2) heat pump subsystem, and (3) heat distribution subsystem.

The GSHPs comprise a wide variety of systems that may use ground-water, ground, or surface water as heat sources or sinks. These systems have been basically grouped into three categories by ASHRAE [13]: (1) ground-water heat pump (GWHP) systems, (2) surface water heat pump (SWHP) systems, and (3) ground-coupled heat pump (GCHP) systems. The schematics of these different systems are shown in Figure 3. Many parallel terms exist: geothermal heat pump (GHP), earth energy system (EES), and ground-source system (GSS).

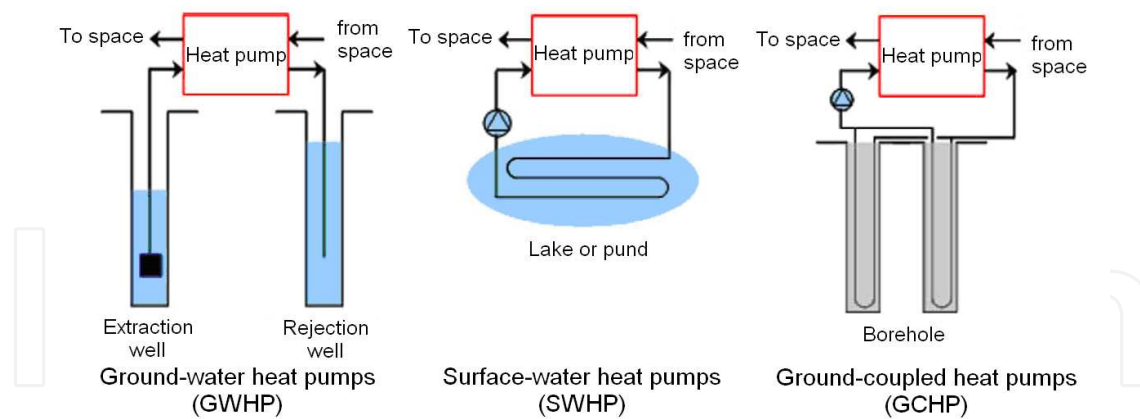


Figure 3. Schematic of different ground-source heat pumps

Among the various GSHP systems, the vertical GCHP system has attracted the greatest interest in research field and practical engineering. Several literature reviews on the GCHP technology have been reported [14].

In a GCHP system, heat is extracted from or rejected to the ground via a closed-loop, i.e., ground heat exchanger (GHE), through which pure water or antifreeze fluid circulates. The GHEs commonly used in the GCHP systems typically consist of HDPE pipes which are installed in either vertical boreholes (called vertical GHE) or horizontal trenches (horizontal GHE) [3]. In direct expansion systems, the heat stored in the ground is absorbed directly by the working fluid (refrigerant). This results in an increased coefficient of performance. Horizontal GHEs are mainly used with this system.

4.2. Description of SWHP systems

Surface water bodies can be very good heat source and sinks, if properly used. The maximum density of water occurs at 4.0°C , not at the freezing point of 0°C . This phenomenon, in combination with the normal modes of heat transfer to and from lakes, produces temperature profile advantageous to efficient heat pump operation. In some cases, lakes can be the very best water supply for cooling. Various water circulation systems are possible and several of the more common are presented [13].

The closed-loop systems consist of water-to-air or water-to-water heat pumps connected to a piping network placed in a lake, river, or other open body of water. A pump circulates water or a water/antifreeze solution through the heat pump water-to-refrigerant heat exchanger and the submerged piping loop, which transfers heat to or from the body of water.

Open-loop systems can use surface water bodies the way cooling towers are used, but without the need for fan energy or frequent maintenance. In warm climates, lakes can also serve as heat sources during winter heating mode, but in colder climates where water temperatures drop below 7°C , closed-loop systems are the only viable option for heating.

Lake water can be pumped directly to water-to-air or water-to-water heat pumps or through an intermediate heat exchanger that is connected to the units with a closed piping loop. Direct

systems tend to be smaller, having only a few heat pumps. In deep lakes (12 m or more), there is often enough thermal stratification throughout the year that direct cooling or precooling is possible. Water can be pumped from the bottom of deep lakes through a coil in the return air duct. Total cooling is possible if water is 10°C or below. Precooling is possible with warmer water, which can then be circulated through the heat pump units.

Advantages of closed-loop SWHPs are (1) relatively low cost because of reduced excavation costs, (2) low pumping energy requirements, and (3) low operating cost. Disadvantages are (1) the possibility of coil damage in public lakes and (2) wide variation in water temperature with outdoor conditions.

4.3. Description of GWHP systems

A GWHP system removes ground-water from a well and delivers it to a heat pump (or an intermediate heat exchanger) to serve as a heat source or sink [13]. One widely used design places a central water-to-water heat exchanger between the ground-water and a closed water loop, which is connected to water-to-air heat pumps in the building. A second possibility is to circulate ground-water through a heat recovery chiller, and to heat and cool the building with a distributed hydronic loop.

Direct systems (in which ground-water is pumped directly to the heat pump without an intermediate heat exchanger) are not recommended except on the very smallest installations. Although some installations of this system have been successful, others have had serious difficulty even with ground-water of apparently benign chemistry. The specific components for handling ground-water are similar. The primary items include (1) wells (supply and, if required, injection), (2) a well pump (usually submerged), and (3) a ground-water heat exchanger. The use of a submerged pump avoids the possibility of introducing air or oxygen into the system. A back-washable filter should also be installed. The injection well should be located from 10 to 15 m in the downstream direction of the ground-water flow.

In an open-loop system, the intermediate heat exchanger between the refrigerant and the ground-water is subject to fouling, corrosion, and blockage. The required flow rate through the intermediate heat exchanger is typically between 0.027 and 0.054 l/s. The ground-water must either be reinjected into the ground by separate wells or discharged to a surface system such as a river or lake. The drill diameter should be at least 220 mm (larger for sandy conditions to prevent sand entry).

The ground-water flow rate G must be capable of delivering the full capacity required from the heat source. This depends on the evaporator cooling power Q_0 and the water cooling degree and is given by the following equation:

$$G = \frac{Q_0}{\rho_w c_w (t_{wi} - t_{we})} \quad (19)$$

where ρ_w is the water density; c_w is the specific heat of water; and t_{wi} and t_{we} are the water temperatures at the heat pump inlet and the heat pump outlet, respectively.

Table 1 summarises the calculated COP values of GWHP and SWHP systems, operating as water-to-water heat pumps.

Water temperature at evaporator inlet t_s [°C]	Water temp. at condenser outlet, t_u [°C]				
30	35	40	45	50	
5	4.55	4.10	3.70	3.40	3.15
10	5.30	4.65	4.15	3.75	3.45
15	6.25	5.35	4.70	4.20	3.85
20	7.70	6.35	5.45	4.80	4.30
25	9.95	7.80	6.45	5.55	4.85
30	14.10	10.10	7.95	6.55	5.60

Table 1. The COP of water-to-water GWHP and SWHP systems

The "Geotherm" system [5] uses a specially built heat exchanger (Figure 4) placed in an extraction well with a 1.0 m diameter and a depth of 2.0 m. The heat exchanger is mounted between a GCHP and a ground-water source with a reduced flow rate and of any water quality. This heat exchanger consists of a set of four coaxial coils made of HDPE tubes with a diameter of 25 mm, immersed in a cylindrical reservoir made of glass fibre reinforced resins (0.8 m diameter and 1.2 m height) supplied with ground-water at the bottom side.

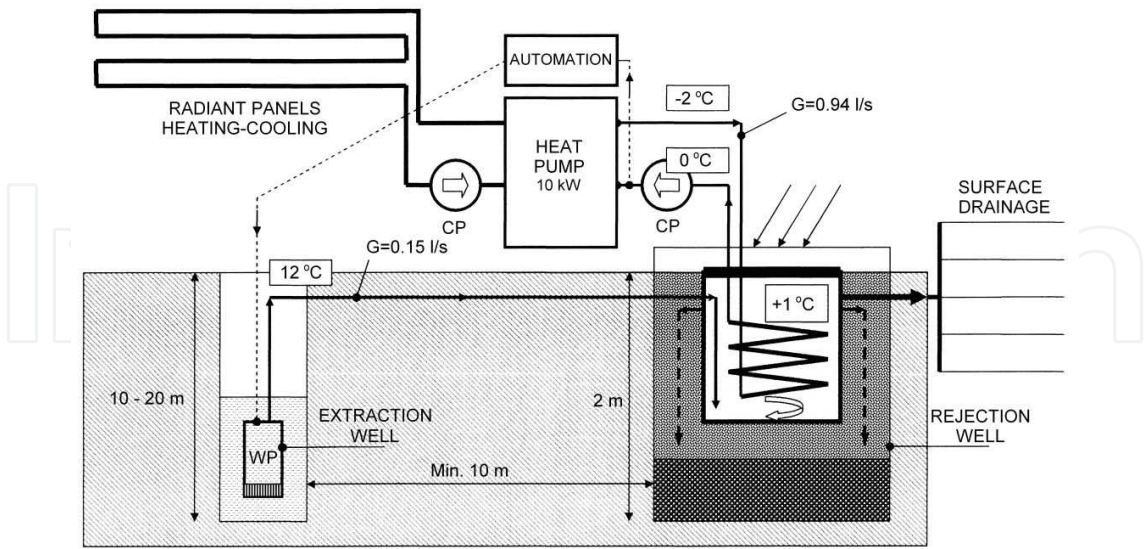


Figure 4. Schematic of "Geotherm" GWHP system

The heat pump used in conjunction with the intermediate heat exchanger is a GCHP system of 10 kW with a COP of 4.

The secondary circuit of the heat exchanger (towards the heat pump) circulates an antifreeze solution (glycol 20%), which enters the heat pump at 0°C and leaves at -2°C, transported by a circulation pump with a flow rate of 0.94 l/s. The glycol flow in the tubes is ensured by the circulation pump within the heat pump. Outside of the coils, the ground-water from the cylindrical reservoir is involved in a flow among the spires of coils by a submersible pump. The relatively small pressure loss on the secondary circuit of the heat exchanger allows the use of a reduced power circulation pump for the glycol [2].

In the primary circuit of the heat exchanger (outside the tubes), ground-water enters with a temperature of 12°C and is evacuated to approximately 1°C (in heating mode). Because the temperature drop is 11°C, compared to 4°C in the usual systems, it is possible to obtain the same thermal power with a ground-water flow rate nearly three times lower. The pressure loss on the primary circuit of the heat exchanger is 26 kPa for the mentioned flow rate. The heat exchange is realized mainly by the ground-water supply, and the heat exchanged directly with the ground around the extraction well is also important. The heat transfer surface is 20 m², and the heat transfer coefficient is 154 W/m²K.

The ground-water is then evacuated through the top of the heat exchanger by gravity in the rejection well. If the rejection well cannot retrieve all of the ground-water flow rate, surface drainage through a network of perforated pipes buried at 50–80 cm or another evacuation solution (lake, river, or sewer) is recommended.

Regardless of the outdoor air and ground temperature, the heat pump will always operate at the same optimum temperatures because of the automation. The automation starts the ground-water inlet (electro-valve or submersible pump) only when the return water–glycol temperature goes below 1°C. The ground-water flow rate is limited to 4–12 l/min depending on the thermal power of the heat pump (4–12 kW).

During the summer, the intermediate heat exchanger can operate in a passive cooling mode in which the heat pump only produces domestic hot-water using heat recovered from the air-conditioned space. In this case, the heat carrier from the heaters is transported with the circulation pump directly to the intermediate heat exchanger.

4.4. Description of GCHP systems

The GCHP is a subset of the GSHP and is often called a closed-loop heat pump. A GCHP system consists of a reversible vapour compression cycle that is linked to a GHE buried in the soil (Figure 4). The GCHP is further subdivided according to GHE type: horizontal GHE and vertical GHE.

4.4.1. Types of horizontal GHEs

Horizontal GHEs (Figure 5) can be divided into at least three subgroups: single-pipe, multiple-pipe, and spiral. Single-pipe horizontal GHEs consist of a series of parallel pipe arrangements laid out in trenches. Typical installation depths in Europe vary from 0.8 to 1.5 m. Antifreeze fluid runs through the pipes in a closed system. The values of the specific extraction/rejection

power q_E for ground [15] are given in Table 2. For a specific power of extraction/rejection, q_E can be obtained from required ground area [16]:

$$A = \frac{Q_0}{q_E} \tag{20}$$

where $Q_0 = Q_{HP} - P_e$ is the cooling power of heat pump.

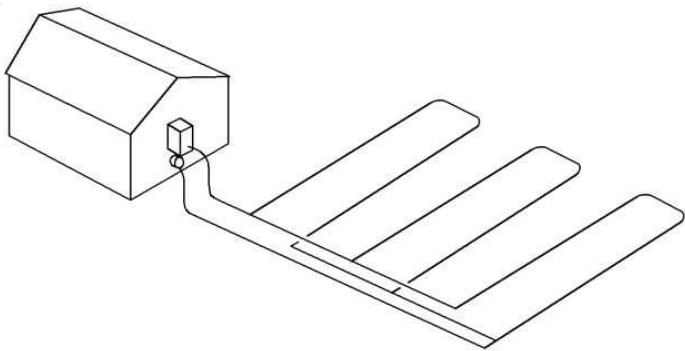


Figure 5. Horizontal ground heat exchanger

No.	Type of ground	q_E [W/m ²]
1	Dry sandy	10...15
2	Moist sandy	15...20
3	Dry clay	20...25
4	Moist clay	25...30
5	Ground with ground-water	30...35

Table 2. Specific extraction/rejection power for ground

To save required ground area, some special GHEs have been developed [7]. Multiple pipes (two, four, or six), placed in a single trench, can reduce the amount of required ground area. The spiral loop (Figure 6) is reported to further reduce the required ground area. This consists of pipe unrolled in circular loops in trenches with a horizontal configuration. For the horizontal spiral loop layout, the trenches are generally a depth of 0.9 to 1.8 m. The distance between coil tubes is of 0.6–1.2 m. The length of collector pipe is of 125 m per loop (up to 200 m). The ends of parallel coils 1 are arranged by a manifold-collector 2 in a heart 3, and then the antifreeze fluid is transported by main pipes 4 at heat pump. Disadvantages of the horizontal systems are: (1) these systems are more affected by ambient air temperature fluctuations because of their proximity to the ground surface, and (2) the installation of the horizontal systems needs much more ground area than vertical system.

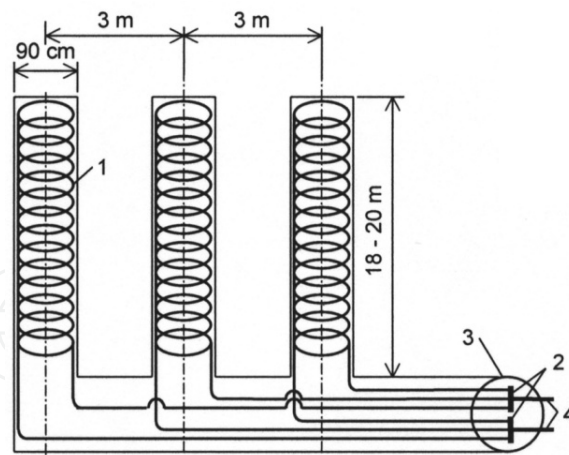


Figure 6. Spiral ground coil

4.4.2. Types of vertical GHEs

There are two basic types of vertical GHEs or borehole heat exchangers (BHE): U-tube and concentric- (coaxial-) tube system configurations (Figure 7). BHEs are widely used when there is a need to install sufficient heat exchanger capacity under a confined surface area, such as when the earth is rocky close to the surface, or where minimum disruption of the landscape is desired. The U-tube vertical GHE may include one, tens, or even hundreds of boreholes, each containing single or double U-tubes through which heat exchange fluid are circulated. Typical U-tubes have a nominal diameter in the range of 20–40 mm and each borehole is normally 20–200 m deep with a diameter ranging from 100 to 200 mm [17]. Concentric pipes, either in a very simple method with two straight pipes of different diameters or in complex configurations, are commonly used in Europe. The borehole annulus is generally backfilled with some special material (grout) that can prevent contamination of ground-water.

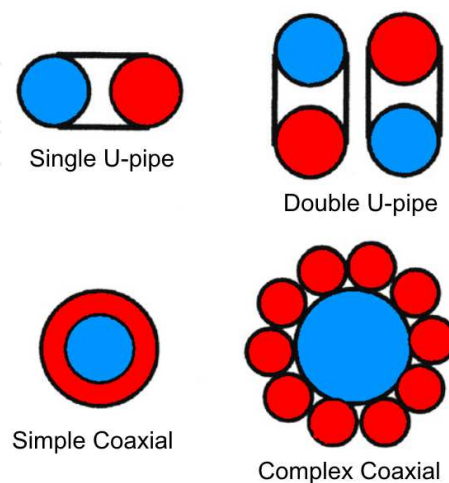


Figure 7. Common vertical GHE designs

A typical borehole with a single U-tube is illustrated in Figure 8. The required borehole length L can be calculated by steady-state heat transfer equation as follows [13]:

$$L = \frac{qR_g}{t_g - t_f} \quad (21)$$

where q is the heat transfer rate, in kW; t_g is the ground temperature, in K; t_f is the heat carrier fluid (antifreeze, refrigerant) temperature, in K; R_g is the effective thermal resistance of ground per unit length, in (mK)/kW.

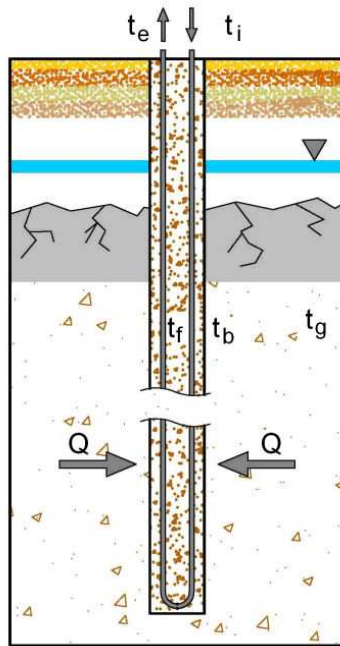


Figure 8. Schematic of a vertical grouted borehole

The GHE usually are designed for the worst conditions by considering that these needs to handle three consecutive thermal pulses of various magnitude and duration: yearly average ground load q_a for 20 years, the highest monthly ground load q_m for 1 month, and the peak hourly load q_h for 6 h. The required borehole length to exchange heat at these conditions is given by [18]:

$$L = \frac{q_h R_b + q_a R_{20a} + q_m R_{1m} + q_h R_{6h}}{t_g - (t_f + \Delta t_g)} \quad (22)$$

where R_b is the effective borehole thermal resistance; R_{20a} , R_{1m} , R_{6h} are the effective ground thermal resistances for 20 years, 1 month, and 6 h thermal pulses; Δt_g is the increase of

temperature because of the long-term interference effect between the borehole and the adjacent boreholes. Alternative methods of computing the thermal borehole resistance are presented by Bernier [18] and Hellström [19].

Advantages of the vertical GCHP are that it (1) requires relatively small ground area, (2) is in contact with soil that varies very little in temperature and thermal properties, (3) requires the smallest amount of pipe and pumping energy, and (4) can yield the most efficient GCHP system performance. Disadvantage is the higher cost because of the expensive equipment needed to drill the borehole.

4.4.3. Simulation models of GHEs

The main objective of the GHE thermal analysis is to determine the temperature of the heat carried fluid, which is circulated in the U-tube and the heat pump, under certain operating conditions. Actually, the heat transfer process in a GHE involves a number of uncertain factors, such as the ground thermal properties, the ground-water flow rate and building loads over a long lifespan of several or even tens of years. In this case, the heat transfer process is rather complicated and must be treated, on the whole, as a transient one. In view of the complication of this problem and its long time scale, the heat transfer process may usually be analysed in two separated regions [3]. The heat transfer models for the two separate regions are as follows.

- *Heat conduction outside borehole.* A number of simulation models for the heat transfer outside the borehole have been recently reported, most of which were based on either analytical methodologies or numerical methods [3].
- *Kelvin's line-source.* The earliest approach to calculating the thermal transport around a heat exchange pipe in the ground is the Kelvin's line-source theory, i.e. the infinite line source [20]. According to the Kelvin's line-source theory, the temperature response in the ground due to a constant heat rate is given by:

$$t(r, \tau) - t_0 = \frac{q}{4\pi\lambda} \int_{r^2/4a\tau}^{\infty} \frac{e^{-u}}{u} du \quad (23)$$

where r is the distance from the line-source and τ the time since start of the operation; t is the temperature of the ground at distance r and time τ ; t_0 is the initial temperature of the ground; q is the heating rate per length of the line source; λ and a are the thermal conductivity and diffusivity of the ground. The solution to the integral term in Eq. (23) can be found from the related references [21].

- *Cylindrical source model.* The cylindrical source solution for a constant heat transfer rate was developed by Carslaw and Jaeger [22], then refined by Ingersoll et al. [21], and later employed in a number of research studies [23]. Based on the governing equation of the transient heat conduction along with the given boundary and initial conditions, the temperature distribution of the ground can be given in the cylindrical coordinate:

$$\begin{aligned}
\frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} &= \frac{1}{a} \frac{\partial t}{\partial \tau} & r_b < r < \infty \\
-2\pi r_b \lambda \frac{\partial t}{\partial \tau} &= q & r = r_b, \tau > 0 \\
t - t_0 &= 0 & \tau = 0, r > r_b
\end{aligned} \tag{24}$$

where r_b is the borehole radius.

The cylindrical source solution is given as follows:

$$t - t_0 = \frac{q}{\lambda} G(z, p) \tag{25}$$

where $z = a\tau/r_b$, $p = r/r_b$.

As defined by Carslaw and Jaeger [22], the expression $G(z, p)$ is only a function of time and distance from the borehole centre. An approximate method for G was proposed by Hellström [19].

- Eskilson's model. Both the one-dimensional model of the Kelvin's theory and the cylindrical source model neglect the axial heat flow along the borehole depth. A major progress was made by Eskilson [24] to account for the finite length of the borehole. The basic formulation of the ground temperature is governed by the heat conduction equation in cylindrical coordinates:

$$\begin{aligned}
\frac{\partial^2 t}{\partial r^2} + \frac{1}{r} \frac{\partial t}{\partial r} + \frac{\partial^2 t}{\partial z^2} &= \frac{1}{a} \frac{\partial t}{\partial \tau} \\
t(r, 0, \tau) &= t_0 \\
t(r, z, 0) &= t_0 \\
q(\tau) &= \frac{1}{L} \int_D^{D+L} 2\pi r \lambda \left. \frac{\partial t}{\partial r} \right|_{r=r_b} dz
\end{aligned} \tag{26}$$

where L is the borehole length; D means the uppermost part of the borehole, which can be thermally neglected in engineering practice.

In Eskilson's model, the numerical finite-difference method is used on a radial-axial coordinate system to obtain the temperature distribution of a single borehole with finite length. The final expression of the temperature response at the borehole wall to a unit step heat pulse is a function of τ/τ_s and r_b/L only:

$$t_b - t_0 = -\frac{q}{2\pi\lambda} f(\tau/\tau_s, r_b/L) \tag{27}$$

where $\tau_s = L^2/9a$ means the steady-state time. The f -function is essentially the dimensionless temperature response at the borehole wall, which was computed numerically.

- Finite line-source solution. Based on the Eskilson's model, an analytical solution to the finite line source has been developed by a research group which considers the influences of the finite length of the borehole and the ground surface as a boundary [3]. The solution of the temperature excess was given by Zeng et al. [25]:

$$t(r, z, \tau) - t_0 = \frac{q}{4\pi\lambda} \int_0^L \left[\frac{\operatorname{erfc}\left(\frac{\sqrt{r^2 + (z-l)^2}}{2\sqrt{a\tau}}\right)}{\sqrt{r^2 + (z-l)^2}} - \frac{\operatorname{erfc}\left(\frac{\sqrt{r^2 + (z+l)^2}}{2\sqrt{a\tau}}\right)}{\sqrt{r^2 + (z+l)^2}} \right] dl \quad (28)$$

It can be seen from Eq. (28) that the temperature on the borehole wall, where $r = r_b$, varies with time and borehole length. The temperature at the middle of the borehole length ($z = L/2$) is usually chosen as its representative temperature. An alternative is the integral mean temperature along the borehole length, which may be determined by numerical integration of Eq. (28).

- *Heat transfer inside borehole.* The thermal resistance inside the borehole, which is primarily determined by thermal properties of the grouting materials and the arrangement of flow channels of the borehole, has a significant impact on the GHE performance. The main objective of this analysis is to determine the entering and leaving temperatures of the circulating fluid in the borehole according to the borehole wall temperature, its heat flow, and the thermal resistance [3].
- One-dimensional model. A simplified one-dimensional model has been recommended for GHE design, which considers the U-tube as a single "equivalent" pipe [26].
- Two-dimensional model. Hellström [19] derived the analytical two-dimensional solutions of the thermal resistances among pipes in the cross-section perpendicular to the borehole axis, which is superior to empirical expressions and one-dimensional model.
- Quasi-three-dimensional model. On the basis of the two-dimensional model mentioned above, a quasi-three-dimensional model was proposed by Zeng et al. [27], which takes account of the fluid temperature variation along the borehole depth.

4.4.4. Ground thermal response test

In the case of vertical closed-loop GCHP systems, the determination of the parameters to calculate the vaporization thermal power that must be provided from the ground is laborious. To know how many loops must be set, which is a function of the energy that must be given to the heat pump, evaluating the thermal conductivity of the ground and the effective thermal resistance of the borehole are very important. In this respect, taking a thermal response test (TRT) of the ground is necessary, using a borehole in which a simple ground loop is placed.

During an in-situ test, a ground electric heater usually provides heat to the circulating fluid (water or glycol) through the ground loop while the inlet (t_i) and outlet (t_e) fluid temperatures

are measured (Figure 9). The average of these two instantaneous temperature reading is usually taken to represent the average temperature in the vertical ground loop at a given time. In an ideal test, the measured circulating flow rate and the heat input rate remain constant throughout the test [28]. The first TRT in Romania was performed in 2009 by the GEOTHERM PDC company of Bucharest [29].

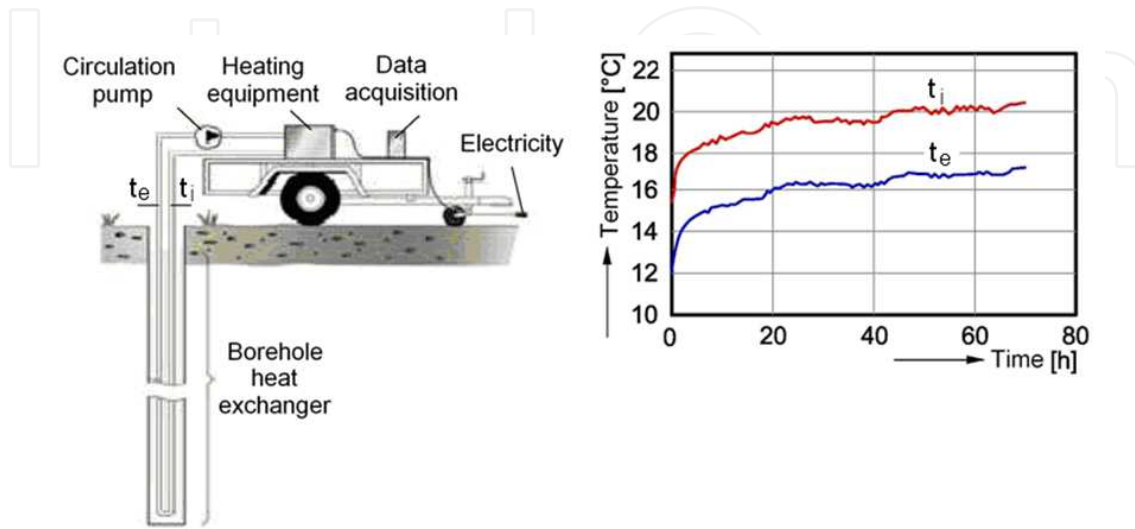


Figure 9. Schematic of equipment for thermal response test

To estimate the minimum duration τ_{\min} of the test, the following equation can be used [24]:

$$\tau_{\min} = \frac{5r_b^2}{a} \quad (29)$$

where r_b is the borehole radius and a is the ground thermal diffusivity.

For data analysis and final evaluation of ground thermal conductivity λ and borehole thermal resistance R_b , some methods were developed [15,30] that use one of the simulation models of GHEs previously presented. Through the ground thermal response test, the length of the borehole is properly determined, the operating performance of the system is provided, and supplementary costs (extra loops, boreholes, glycol, etc.) are avoided. This operation is performed using specialized software.

5. Heat pump heating and cooling systems

5.1. Radiator heating system

A hot-water radiator heating system is a type of central heating. In the system, heat is generated in the boiler. For the generation of the heat, a natural gas boiler is used where the chemical

energy of natural gas is transferred into the heat. Then, the heat is distributed by hot-water (heat carrier) to the radiators. The radiators heat the rooms. The hot-water is circulated by a water circulation pump, which operates continuously. The radiators, as rule of thumb, are located next to the cold surfaces of the envelope. They significantly influence the thermal comfort. The radiators release the highest amount of heat to the heated room by convection and one part by heat radiation [31]. The convective heat transfer will lead to a lower relative humidity of the air, and, at high radiator surface temperature, dust particles can be burned, leading to lower indoor air quality. Thus, emitters should be implemented with a radiation factor as high as possible in the case of high-temperature water supplies. The highlights of the convective thermal field achieved with radiators were illustrated in [32].

To ensure ever-changing heat demand in a room, qualitative, quantitative or mixed control systems are used.

5.2. Radiant heating and cooling systems

In low-energy buildings, the low-temperature heating system usually works with a supply water temperature below 45°C [33]. Embedded radiant systems are used in all types of buildings. Radiant heating application is classified as panel heating if the panel surface temperature is below 150°C [34]. In thermal radiation, heat is transferred by electromagnetic waves that travel in straight lines and can be reflected. The water temperatures are operated at very close to room temperature and, depending on the position of the piping, the system can take advantage of the thermal storage capacity of the building structure.

Figure 10 shows the available types of embedded hydronic radiant systems [35]. Panel heating provides a comfortable environment by controlling surface temperatures and minimising air motion within a space. A radiant system is a sensible heating system that provides more than 50% of the total heat flux by thermal radiation. The controlled temperature surfaces may be in the floor, walls, or ceiling, with the temperature maintained by circulation of water or air. The radiant heat transfer is, in all cases, 5.5 W/(m²K). The convective heat transfer then varies between 0.5 and 5.5 W/(m²K), depending on the surface type and on heating or cooling mode. This shows that the radiant heat transfer varies between 50 and 90% of the total heat transfer [36].

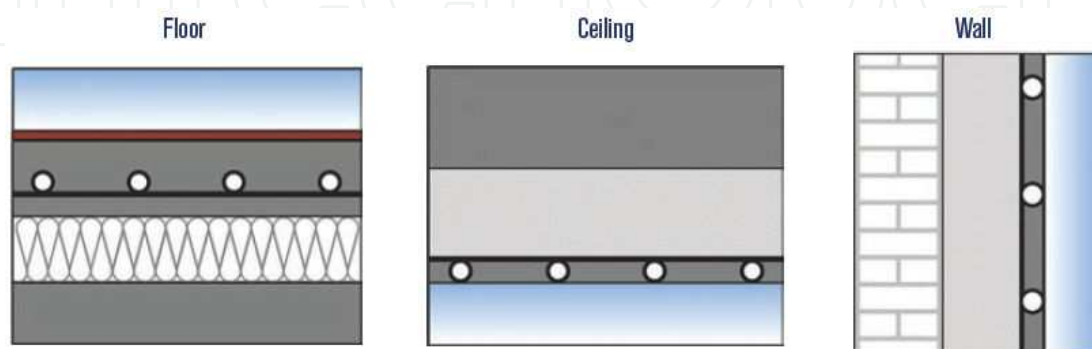


Figure 10. Examples of water-based radiant systems

Radiant panel heating is characterised by the fact that heating is associated with a yielding of heat with low temperature because of physiological reasons. Thus, at the radiant floor panels, the temperature must not exceed $+29^{\circ}\text{C}$, and at the radiant ceiling panels the temperature will not exceed $35\text{--}40^{\circ}\text{C}$, depending on the position of the occupier (in feet) and the occupier distance to the panels, in accordance with thermal comfort criteria established by ISO Standard 7730 [37]. For cooling, the minimum floor temperature is 19°C . A vertical air temperature difference between head and feet of less than 3°C is recommended.

5.3. Performance assessment of radiator and radiant floor heating systems connected to a GCHP

5.3.1. Description of office room

Experimental investigations of GCHP performance were conducted in an office room (Figure 11) at the Polytechnic University of Timisoara, Romania, located at the ground floor of the Civil Engineering Faculty building. The Timisoara city has a continental temperature climate with four different seasons. The heating season runs in Timisoara from 1 October to 30 April. The following data are known: heat transfer resistance ($1/U$ -value) of building components: walls ($2.10\text{ m}^2\text{K/W}$), ceiling ($0.34\text{ m}^2\text{K/W}$), windows and doors ($0.65\text{ m}^2\text{K/W}$); glass walls surface, 8.2 m^2 ; total internal heat gain (e.g. from computers, human and lights), 25 W/m^2 ; and heat demand, 1.35 kW . The indoor and outdoor air design temperatures are 22 and -15°C , respectively.

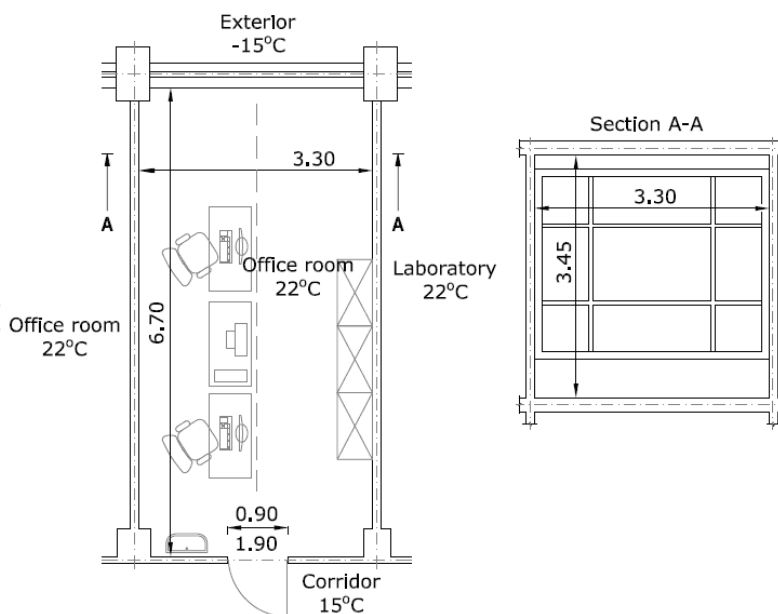


Figure 11. Heated office room

This space is equipped both with a floor heating system and steel panel radiators to analyse the energy and environmental performances of these systems. These two heating systems are

connected to a mechanical compression GCHP, type WPC 5 COOL. In the GCHP system, heat is extracted from the ground by a closed-loop vertical GHE with a length of 80 m.

5.3.2. Experimental facilities

The GCHP experimental system consisted of a BHE, heat pump unit, circulating water pumps, floor/radiator heating circuit, data acquisition instruments and auxiliary parts as shown in Figure 12.

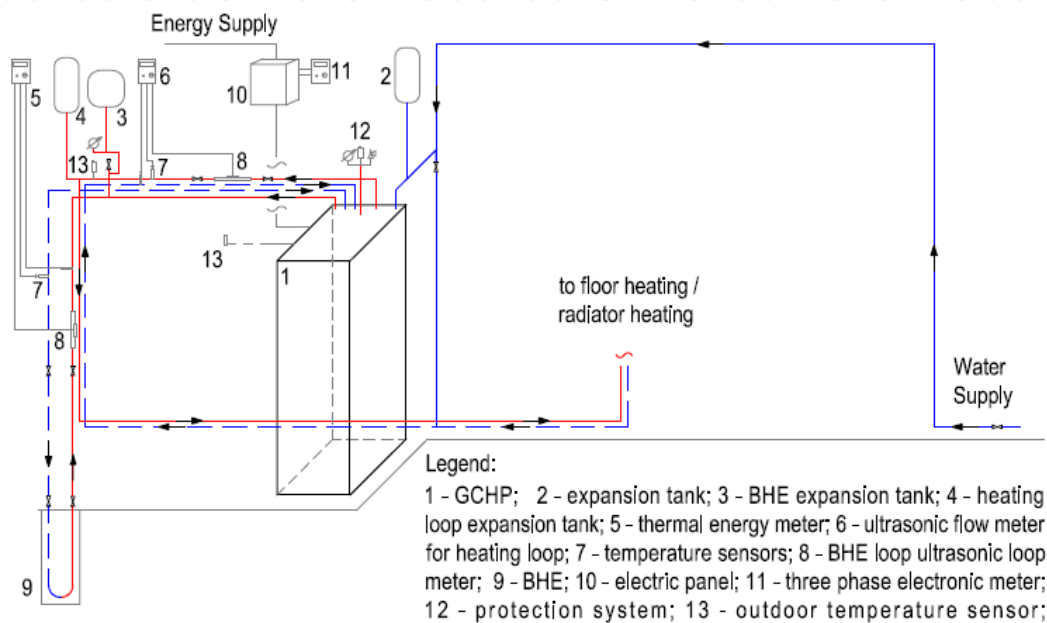


Figure 12. Experimental GCHP system

- **Borehole heat exchanger.** The GHE of this experimental GCHP consisted of a simple vertical borehole that had a depth of 80 m. Antifreeze fluid (30% ethylene glycol aqueous solution) circulates in a single polyethylene U-tube of 32 mm internal diameter, with a 60 mm separation between the return and the supply tubes, buried in borehole. The borehole overall diameter was 110 mm. The borehole was filled with sand and finished with a bentonite layer at the top to avoid intrusion of pollutants in the aquifers. The average temperature across the full borehole depth tested was 15.1°C. The ground characteristics are based on measurements obtained from the Banat Water Resources Management Agency [38]. The average thermal conductivity and thermal diffusivity of the ground from the surface to 80 m deep tested were 1.90 W/(m K) and 0.79×10^{-6} m²/s, respectively [39]. The boreholes were completely backfilled with grout mixed with drilling mud, cement and sand in specific proportions. The thermal conductivity and thermal diffusivity of the grout tested by manufacturer were 2.32 W/(m K) and 0.93×10^{-6} m²/s, respectively.
- **Heat pump unit.** The heat pump unit is a reversible ground-to-water scroll hermetic compressor unit with R410A as a refrigerant and the nominal heating capacity of 6.5 kW.

The heat pump unit is a compact type model having an inside refrigeration system. The operation of the heat pump is governed by an electronic controller, which, depending on the system water return temperature, switches the heat pump compressor on or off. The heat source circulation pump was controlled by the heat pump controller, which activates the source pump 30 s before compressor activation [40].

- GCHP data acquisition system. The GCHP data acquisition system consists of the indoor and outdoor air temperature, dew point temperature, supply/return temperature, heat source temperature (outlet BHE temperature), relative air humidity, and main operating parameters of the system components.
- Heating systems. The heating systems are supplied via a five-circuit flow/return manifold as follows. The first two circuits supply the floor heating system. The third and fourth circuits are coupled to a radiator heating system, and the fifth circuit is for backup. The flow/return manifold is equipped with a circulation pump to ensure the chosen temperature of the heat carrier (hot-water). A three-way valve and a thermostatic valve are provided to adjust the maximum hot-water temperature of the floor's heating system. Thus, for higher temperatures, the hot water is adjusted to achieve a circulation loop in the heating system.

To achieve higher performances of the heating systems, a thermostat is provided for controlling the start/stop command of the circulation pump when the room reaches the set point temperature. At the same height as this thermostat, there is also an ambient thermostat that controls the starting and stopping of the heat pump to ensure optimum operation of the entire heating system. The start–stop command of the flow/return manifold circulation pump is controlled by an interior thermostat relay, situated at a height of approximately 1.00 m above the floor surface. This height has been determined to provide adequate comfort for the office occupants.

1. *Radiant floor heating system* consists of two circuits connected to a flow/return manifold (Figure 13), designed to satisfy the office heating demand of 1.35 kW. The first circuit has a length of 54 m and is installed in a spiral coil, with the closest step distance to the exterior wall of the building to compensate for the effect of the heat bridge, and the second circuit, with a length of 61 m, is mounted in the coil simple. The mounting step of the coils is between 10 and 30 cm. The floor heating pipes are made of cross-linked polyethylene with an external diameter of 17 mm and a wall thickness of 2 mm. The mass flow rate for each circuit is controlled by the flow/return manifold circuit valves. They are adjusted to satisfy the heat demand according to Timisoara's climate ($t_e = -15^\circ\text{C}$).
2. *Radiator heating system*. The low-temperature radiator heating system (45/35°C) has two steel panel radiators, each one with two water columns and a length of 1000 mm, height of 600 mm and thermal power of 680 W (Figure 14), connected to a flow/return manifold and dimensioned to satisfy the office heating demand of 1.35 kW. They are installed on a stand at 15 cm above the floor surface to ensure optimal indoor air circulation. The heating radiator system pipes are made of cross-linked polyethylene with an external diameter of 17 mm and a wall thickness of 2 mm. The mass flow rate for each radiator is controlled by the flow/return manifold circuit valves, adjusted to satisfy the heat demand of office room.

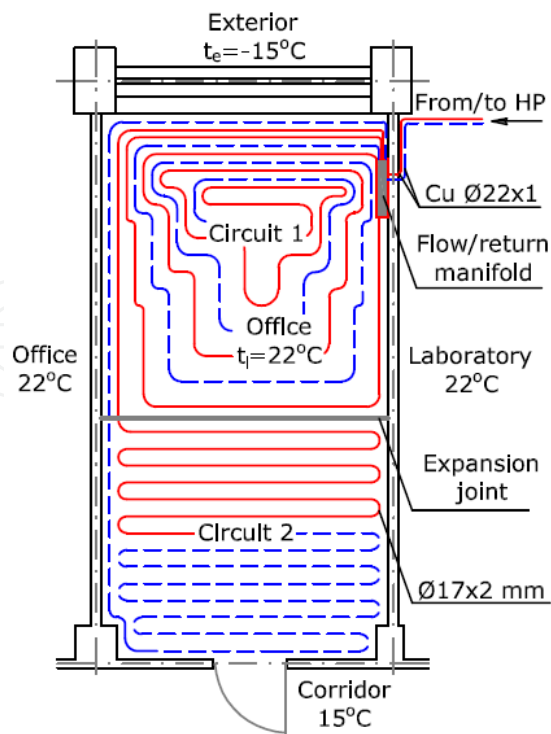


Figure 13. Schematics of floor heating circuit

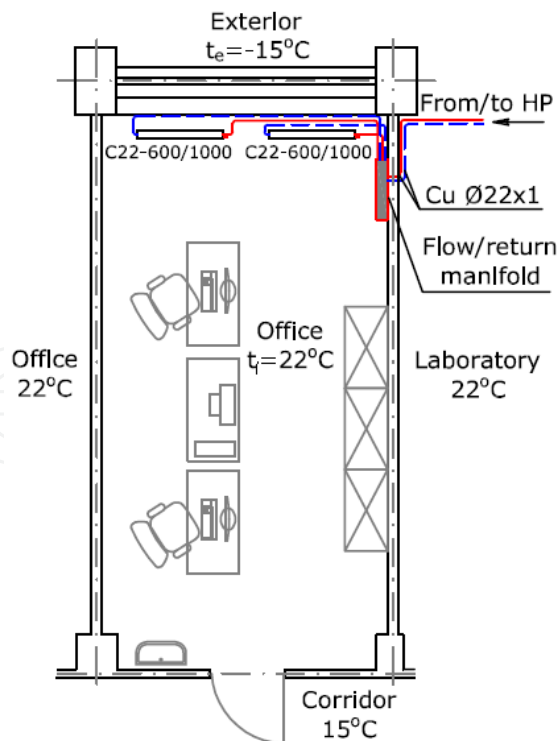


Figure 14. Schematics of radiator heating system

5.3.3. Auxiliary equipment

A network of sensors was setup to allow monitoring of the most relevant parameters of the system [40]. Two thermal energy meters were used to measure the thermal energy produced by the GCHP and the extracted/injected thermal energy to the ground. A thermal energy meter was built with a heat computer, two PT500 temperature sensors and an ultrasonic mass flow meter. The two PT500 wires temperature sensors with an accuracy of $\pm 0.15^\circ\text{C}$ were used to measure the supply and return temperature for a hydraulic circuit (the water–antifreeze solution circuit or the manifold circuit). Also, an ultrasonic mass flow meter measured the mass flow rate for a hydraulic circuit. The thermal energy meters were AEM meters, model LUXTERM, with a signal converter IP 67 and accuracy $< 0.2\%$. A three-phase electronic electricity meter measured the electrical energy consumed by system (the heat pump unit, the circulating pumps, a feeder 220 Vca/ 24 Vcc, a frequency converter, and a programmable logic controller) and another three-phase electronic electricity meter measured the electrical energy consumed by the heat pump compressor. The two three-phase electronic electricity meters were multifunctional type from AEM, model ENERLUX-T, with an accuracy grade in $\pm 0.4\%$ of the nominal value. The monitoring and recording of the experiments were performed using a personal computer (PC). The indoor and outdoor air temperature was measured by AFS sensors and supply/return and heat source temperature were recorded by PTC immersion sensors, all connected to the GCHP data acquisition system and having an accuracy of $\pm 0.2^\circ\text{C}$ [40].

5.3.4. Experimental Results

- Comparison between energy performances of systems. The two heating systems were monitored for two months. The experiments were conducted for a one-week heating period for each of the two analysed heating systems, from the 7th of December 2013 to the 6th of January 2014 and from the 15th of January 2014 to the 14th of February 2014. The outdoor temperature varied in the range of -5.6 to 9.7°C . The weekly mean values of the outdoor temperature during the two periods were almost equal.

The energy performance of heating system is determined based on the coefficient of performance (COP_{sys}), which can be calculated using Eq. (1). The carbon dioxide emission (C_{CO_2}) of the heating system during its operation is calculated with Eq. (18). To obtain the COP and CO_2 emissions, it is necessary to measure the heating energy and electricity used in the system.

During the cold season, measurements were performed at the appreciatively same average outdoor air temperature and the heat source temperature for both the radiant floor heating system and the radiator heating system. The following average values were recorded: outdoor air temperature (t_e), indoor air temperature (t_i), heat source temperature (t_{hs}), supply hot-water temperature (t_d), electricity consumption (E_{el}) and useful thermal energy for heating (E_t). In addition, the CO_2 emission and the ON/OFF switching of the heat pump were determined in both heating systems.

Figure 15 shows a comparison between the indoor air temperatures $t_{i,\text{RAD}}$ and $t_{i,\text{RF}}$ obtained by radiator heating and radiant floor heating. It is observed that due to the small thermal inertia

of the radiators, a high level of ON/OFF switch is needed for the heat pump of the radiator heating system, leading to large fluctuations of indoor air temperature compared with the floor heating system, along with reduced thermal comfort. Table 3 presents a summary of the experimental results.

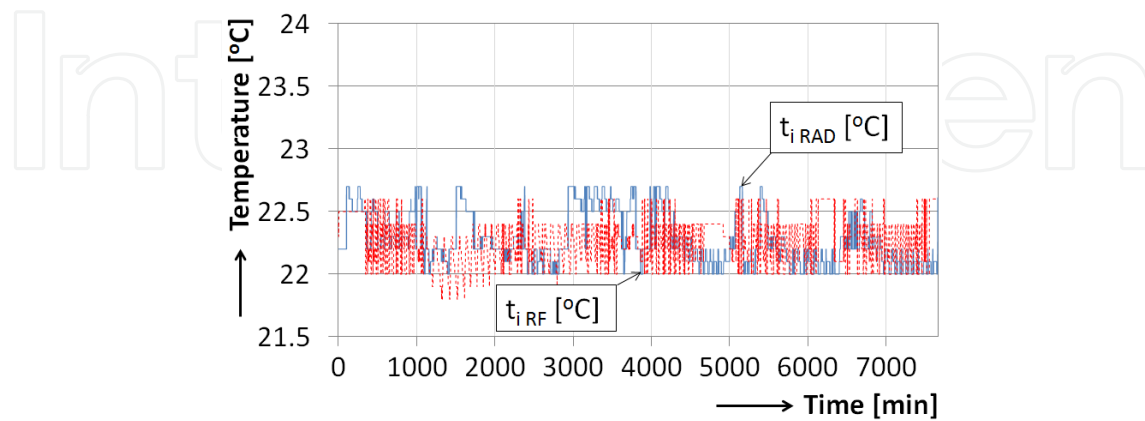


Figure 15. Variation of indoor air temperature

Heating system	t_e [°C]	t_i [°C]	t_{hs} [°C]	t_d [°C]	E_{el} [kWh]	E_t [kWh]	C_{CO_2} [kg]	On/Off switch	COP_{sys}
Radiant floor	9.39	22.28	18.77	28.12	5.77	32.78	3.16	48	5.68
Radiator	9.00	22.30	17.62	30.62	6.35	34.42	3.47	140	5.42

Table 3. Experimental results

The two heating systems have small differences (4.5%) in their energy performance coefficient (COP_{sys}) value, but the ON/OFF switching in the case of radiator heating system is almost three times higher than that for radiant floor heating system, leading to higher wear on the heat pump equipment. In addition, there was 10% higher energy consumption and CO_2 emission for the radiator heating system compared with the floor heating system under the same operating conditions. Energy consumption can be influenced by the building occupants' activity and the floor surface material. If the floor surface material exhibits good heat transfer, such as with stone or tile, the floor feels cold even at a temperature of approximately 24 to 25°C.

- Uncertainty analysis (the analysis of uncertainties in experimental measurement and results) is necessary to evaluate the experimental data. An uncertainty analysis was performed using the method described by Holman [41]. A result Z is a given function of the independent variables $x_1, x_2, x_3, \dots, x_n$. If the uncertainties in the independent variables $w_1, w_2, w_3, \dots, w_n$ are all given with same odds, then uncertainty in the result w_z having these odds is calculated by the following equation [40]:

$$w_z = \sqrt{\left(\frac{\partial Z}{\partial x_1} w_1\right)^2 + \left(\frac{\partial Z}{\partial x_2} w_2\right)^2 + \dots + \left(\frac{\partial Z}{\partial x_n} w_n\right)^2} \tag{30}$$

In the present study, the temperatures, thermal energy and electrical energy were measured with appropriate instruments explained previously. Error analysis for estimating the maximum uncertainty in the experimental results was performed using Eq. (30). It was found that the maximum uncertainty in the results is in the COP_{sys} , with an acceptable uncertainty of 3.9 and 3.1% for radiant floor heating system and radiator heating system, respectively.

5.3.5. Thermal comfort assessment

The office room with geometrical dimensions from Figure 11 is considered. The following data are known: indoor air temperature, 22°C; relative humidity of air, 55%; thermal power of heater, 1360 W; floor temperature, 20°C for radiator heating and 29°C for radiant floor heating.

Assessment of thermal comfort in the office room is performed using the PMV (predicted mean vote)–PPD (predicted percent dissatisfied) model [42]. A comparative study of PMV and PPD indices is performed using the computer program Thermal Comfort [43] in several points situated on a straight line (discontinuous), at different distances from the window, function of metabolic rate (i_M), and clothing thermal resistance (R_{cl}). The results of the numerical solution obtained for the pairs of values 3.4 met-0.67 clo (intense activity, normal clothes), 1 met-0.90 clo (reading seated, winter clothes), and 1.1 met-0.29 clo (writing, light clothes) are reported in Table 4.

Heating type	Distance from the window [m]	3.4 met – 0.67 clo			1 met – 0.90 clo			1.1 met – 0.29 clo		
		t_r [°C]	PMV [–]	PPD [%]	t_r [°C]	PMV [–]	PPD [%]	t_r [°C]	PMV [–]	PPD [%]
Radiant floor	1.0	23.00	2.17	84	23.00	–0.35	8	23.00	–1.63	58
	1.5	23.70	2.22	86	23.70	–0.26	6	23.70	–1.51	52
	2.0	24.30	2.26	87	24.30	–0.18	6	24.30	–1.41	46
	2.5	24.70	2.28	88	24.70	–0.12	5	24.70	–1.34	42
	3.0	25.00	2.31	88	25.00	–0.08	5	25.00	–1.28	39
	3.5	25.20	2.32	89	25.20	–0.06	5	25.20	–1.25	38
	4.0	25.30	2.32	89	25.30	–0.04	5	25.30	–1.23	37
	4.5	25.50	2.34	89	25.50	–0.02	5	25.50	–1.19	35
	5.0	25.50	2.34	89	25.50	–0.02	5	25.50	–1.19	35
Radiator	1.0	20.60	2.01	77	20.60	–0.67	14	20.60	–2.05	79
	1.5	21.20	2.05	79	21.20	–0.59	12	21.20	–1.94	74

Heating type	Distance from the window [m]	3.4 met – 0.67 clo			1 met – 0.90 clo			1.1 met – 0.29 clo		
		$t_r [^{\circ}\text{C}]$	PMV [-]	PPD [%]	$t_r [^{\circ}\text{C}]$	PMV [-]	PPD [%]	$t_r [^{\circ}\text{C}]$	PMV [-]	PPD [%]
	2.0	21.70	2.08	80	21.70	-0.53	11	21.70	-1.86	70
	2.5	22.10	2.11	82	22.10	-0.48	10	22.10	-1.79	67
	3.0	22.40	2.13	82	22.40	-0.43	9	22.40	-1.74	64
	3.5	22.60	2.14	83	22.60	-0.41	8	22.60	-1.70	62
	4.0	22.70	2.15	83	22.70	-0.39	8	22.70	-1.69	61
	4.5	22.80	2.16	83	22.80	-0.38	8	22.80	-1.67	60
	5.0	22.80	2.16	83	22.80	-0.38	8	22.80	-1.67	60

Table 4. Numerical results of THERMAL COMFORT computer program

According to the performed study, it was established that the PMV index has values close to zero only for the pair of values 1 met-0.9 clo. For any other pair of values i_M - R_{cl} , the percent of people dissatisfied with their thermal comfort would be greater than 35%. In addition, the PMV index values for the pair 1 met-0.9 clo are lower with 47–94% in the case of the radiant floor heating system than in the case of the radiator heating system. Therefore, the first system leads to increased thermal comfort.

5.3.6. Numerical simulation of useful thermal energy and system COP using TRNSYS software

One of the main advantages of TRNSYS software [44] for the modelling and design of ground-source heat pumps is that it includes components for the calculation of building thermal loads, specific components for HVAC, heat pumps and circulating pumps, modules for borehole heat exchangers and thermal storage, as well as climatic data files, which make it a very suitable tool to model a complete air-conditioning/heat pump installation to provide heating and cooling to a building.

Some statistical methods, such as the root-mean square (RMS), the coefficient of variation (c_v), the coefficient of multiple determinations (R^2), and percentage difference (relative error) (e_r) may be used to compare simulated (computed) and actual values for model validation [40]:

$$\text{RMS} = \sqrt{\frac{\sum_{i=1}^n (y_{\text{sim},i} - y_{\text{mea},i})^2}{n}} \quad (31)$$

$$c_v = \frac{\text{RMS}}{|\bar{y}_{\text{mea},i}|} 100 \quad (32)$$

$$R^2 = 1 - \frac{\sum_{i=1}^n (y_{\text{sim},i} - y_{\text{mea},i})^2}{\sum_{i=1}^n y_{\text{mea},i}^2} \quad (33)$$

$$e_r = \frac{|y_{\text{mea},i} - y_{\text{sim},i}|}{y_{\text{mea},i}} 100\% \quad (34)$$

where n is the number of measured data in the independent data set; $y_{\text{mea},i}$ is the measured value of one data point i ; $y_{\text{sim},i}$ is the simulated value; $\bar{y}_{\text{mea},i}$ is the mean value of all measured data points.

- Simulation of thermal energy used for office room heating. To simulate the thermal energy used to cover the heating load of the office room, the operational connections were established between the building and all internal and external factors. Figure 16 presents the operational scheme built in TRNSYS, where the building thermal behavior was modelled using a “Type 56” subroutine. This subroutine was processed with the TRNBuild interface by introducing the main construction elements, their orientation and surface, shadow factors, and indoor activity type. Weather data for the Timisoara were obtained from the Meteonorm data base [45] and the weather data reader “Type 109” and “Type 89d” were used to convert the data in a form readable from TRNSYS. The simulation model took into account the outdoor air infiltrations, heat source type, and interior gains. To extract the results, an online plotter (“Type 65”) is used.

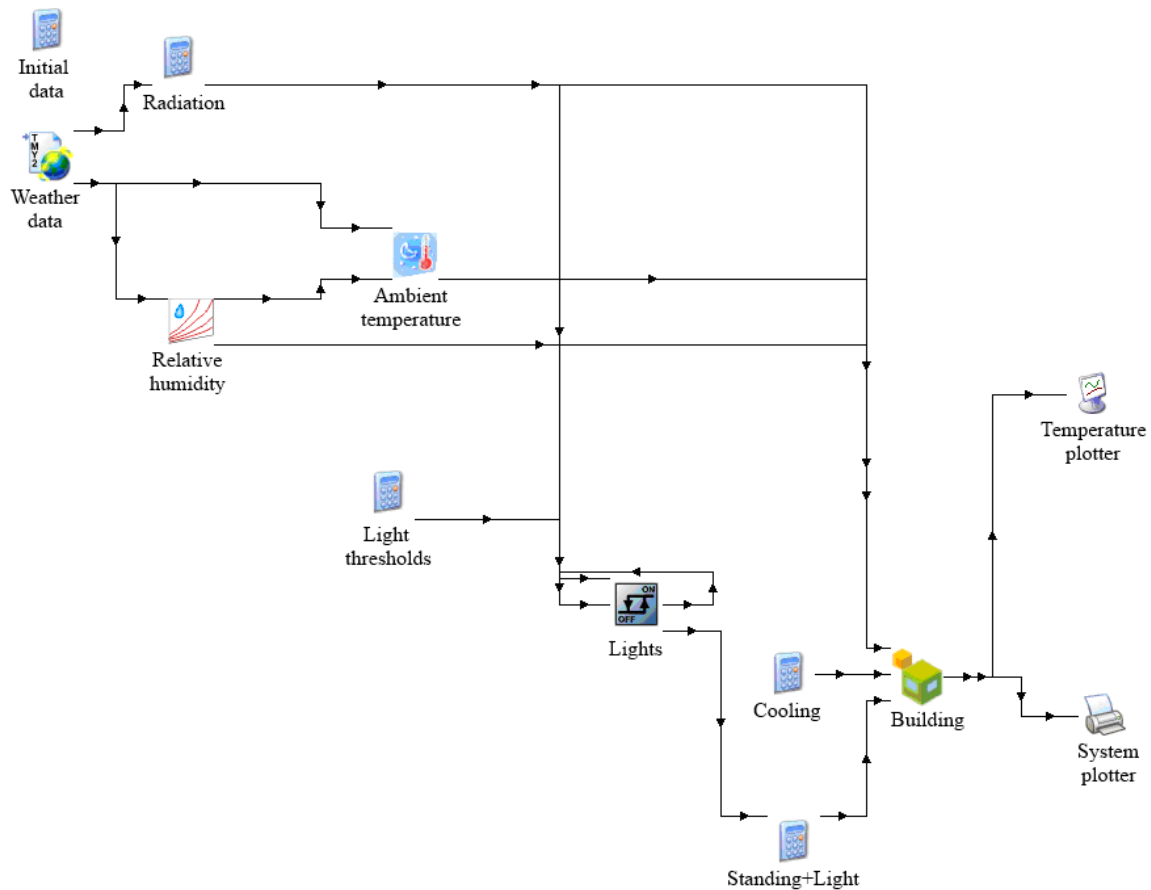


Figure 16. Scheme of the system model built in TRNSYS to simulate the useful thermal energy

Performing simulations for a one-year period (8760 h), the values of thermal energy used for heating were obtained and are presented beside the measured values in Table 5. Statistical values such as RMS, c_v and R^2 are also given in the same table.

Month	Heating energy [kWh]		Percentage difference e_r [%]	RMS	c_v [%]	R^2
	Simulated	Measured				
January	252.50	256.24	1.57	2.72187	1.409	0.99990075
February	195.70	195.06	0.32			
March	151.61	150.44	0.77			
April	49.73	48.95	1.59			
May	0.00	0.00	0.00			
June	0.00	0.00	0.00			
July	0.00	0.00	0.00			
August	0.00	0.00	0.00			
September	0.00	0.00	0.00			
October	94.85	95.66	0.84			
November	174.45	172.62	1.06			
December	238.75	240.11	0.57			

Table 5. Thermal energy used for office room heating

There was a maximum difference between the measured and TRNSYS simulated values for the heating period of approximately 1.59%, which is very acceptable. The RMS and c_v values in heating mode are 2.722 and 1.41%, respectively. The R^2 -values are about 0.9999, which can be considered as very satisfactory. Thus, the simulation model was validated by the experimental data.

- COP simulation of GCHP system. For COP simulation of the GCHP system, the operational scheme built in TRNSYS from Figure 17 was utilised. The assembly of GCHP system consists of the standard TRNSYS weather data readers “Type 15-6”, a GCHP model “Type 919”, a BHE “Type 557a”. Also, in the simulation model were defined single-speed circulating pumps “Type 114” for the antifreeze fluid in the BHE and “Type 3d” for heat carrier fluid of the manifold. A “Type 14” for the load profile and a daily load subroutine were created, this approach improving significantly the numerical convergence of the model. Finally, two model integrators (“Type 25” and “Type 24”) were used to calculate daily and total results for thermal energy produced.

COP simulation of the GCHP integrated both with radiator and radiant floor heating system was performed for 1 month period. The results of the simulation program are presented beside

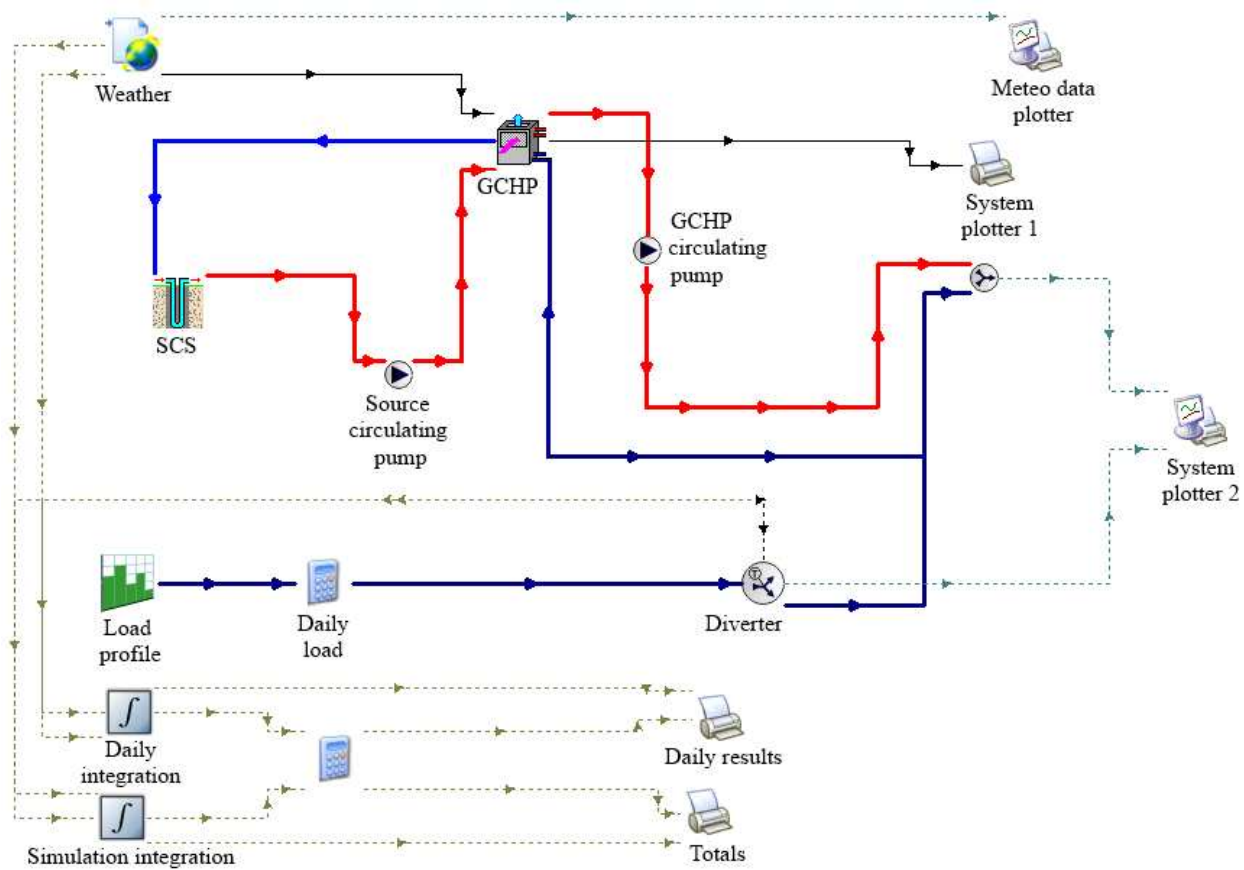


Figure 17. Scheme of the system model built in TRNSYS for COP simulation

the experimental data in Table 6. A comparative analysis of these results indicates that the COP_{sys} values simulated with TRNSYS program were only 3.52% lower than the measured values for radiant floor heating system and only 4.98% lower than the measured values for radiator heating system. Thus, the simulation model is validated experimentally.

Heating system	COP_{sys}		Percentage difference e_r [%]
	Simulated	Measured	
Radiant floor	5.48	5.68	3.52
Radiator	5.15	5.42	4.98

Table 6. The COP values for GCHP system

6. Conclusions

The GSHPs are suitable for heating and cooling of buildings and so could play a significant role in reducing CO₂ emissions. The GWHPs have the low costs, but with some limitations on

the big water flow rate and the clogging of extraction well with appreciable sediment quantities. The new GWHP system “Geotherm”, having $COP = 4$, removes these disadvantages by using a special heat exchanger.

Through the ground thermal response test, the length of the vertical GHE is properly determined and supplementary costs (extra loops, boreholes, glycol, etc.) are avoided.

This study showed that radiator heating and radiant floor heating systems have small differences (4.5%) in their energy performance coefficient (COP_{sys}) value, but the ON/OFF switch in the case of a radiator heating system is almost three times higher than that for a radiant floor heating system, leading to higher wear on the heat pump equipment. In addition, the radiator heating system showed 10% higher energy consumption and CO_2 emissions compared to the floor heating system under the same operating conditions.

The developed TRNSYS simulation models can be used as a tool to determine the GCHP performance connected with different heating systems to optimise their energy efficiency and ensure the user’s comfort throughout the year.

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