

Article

Exploratory Research to Improve Energy-Efficiency of a Ground-Coupled Heat Pump Utilizing an Automatic Control Device of Circulation Pump Speed

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Abstract: Ground-coupled heat pumps (GCHPs) are an efficient thermal energy production system that can satisfy the gap between heating and air-conditioning. Be that as it may, exploratory research on GCHPs is still lacking. The first objective of this article is to describe a utilitarian energy-efficiency improvement device for a vertical GCHP system that includes a buffer tank (BT) between the heat pump unit and the fan coil units and user supply, utilizing the quantitative regulation of water flow rate with a variable-speed circulation pump. At that point, the investigative estimations are utilized to test the performances of the GCHP system in various operating modes. Fundamental efficiency parameters (coefficient of performance (COP) and CO_2 emission) are achieved for one month of running utilizing two control strategies of the GCHP-standard and optimized regulation of the water pump speed—and a benchmarking of these parameters is achieved. Exploratory research has indicated higher efficiency of the system for the flow regulation solution utilizing a BT and programmed control equipment for the circulation pump speed compared with the standard regulation solution (COP_{sus} with 7–8% higher and CO_2 emission level 7.5–8% lower). The second objective is to elaborate a simulation model of the necessary heat/cold in heating and air-conditioning periods, utilizing the Transient Systems Simulation (TRNSYS) program. Finally, the simulation, acquired utilizing the TRNSYS program, is analyzed and compared with experimental information, leading to a good agreement and, along these lines, the simulation model is approved.

Keywords: renewable energy; vertical GCHP; optimized regulation; exploratory investigations; efficiency evaluation; simulation model

1. Introduction

The European Green Deal is an ambitious growth strategy aiming to transform Europe, setting out to achieve net zero emissions by 2050 [1,2]. A priority area for change is the building sector that accounts for up to 40% of energy use [3]. A large amount of energy used for space heating and cooling in buildings comes from fossil fuels [4], which, with their high carbon dioxide (CO_2) emissions, considerably contribute to greenhouse emissions. To limit the impact of fossil fuels, the renewable energy technologies are increasingly used. Among the various technologies to heat and cool buildings, the heat pumps (HPs) are certainly one of the most efficient [5]. The HP moves heat from a thermal source to buildings and rejects the thermal load from the building to the heat sink in the heating and cooling modes, respectively. The use of ground-coupled heat pump (GCHP) systems for heating, ventilating and air conditioning (HVAC) in buildings has been pushed in the last few years as a promising good solution for reducing CO_2 emissions and operating costs [6]. Various GCHPs have been utilized in residential and commercial structures globally in light of their recognizable high



proficiency and environmental friendliness. Several analyses of GCHP systems are reported in the literature by a few researchers [6–14].

For GCHP, the coupling between the HP unit and the ground is obtained by means of heat exchangers [15] where the heat-carrier fluid flows, thus connecting the ground with the evaporator or the condenser of the HP unit. Ground heat exchangers (GHEs) can be vertically or horizontally oriented, but vertical configurations (borehole heat exchangers, BHEs) are more frequently used in order to limit the use of land area [16], especially in commercial buildings. When outdoor air is used as a thermal source/sink, the energy efficiency of the HP is affected by the variation in its temperature throughout the year. Ground temperature is more stable during the year and also closer to the indoor air temperature of the building than the outdoor air temperature; consequently the energy efficiency of the GCHP is higher than that of air-source HPs (ASHPs) [17].

Few researches were centered around various processes of the GCHP system, such as design [18–20], performance [7,21,22], and economic analysis [23–25]. Notwithstanding, most existing analyses of GCHPs focus on theoretical and simulation model analysis [26–29] or in situ checking of the heat transmission in GHE [30–34]. Just a couple of researchers examined the functioning efficiency of exploratory GCHPs. Hwang et al. [32] showed the genuine air-conditioning efficiency of a GCHP mounted in Korea for 1 day of running. Pulat et al. [35] assessed the effectiveness of a horizontal GCHP mounted in Turkey in cold climatic conditions. The COP_{sys} and the COP_{hp} were obtained as 2.45–2.57 and 4.02–4.17, respectively.

Esen et al. [36] compared a GSHP with an ASHP in heating mode from techno-economic point of view, concluding that GSHP was economically convenient due to lower operating costs involved by the higher energy efficiency. Additionally, Lu et al. [37] found similar results in the Melbourne climate. Man et al. [38] made an in situ running efficiency trial of a GCHP for heating and air-conditioning solutions in a temperate zone. The trial demonstrated that the GCHP efficiency was influenced by its irregular or persistent operating modes.

Petit and Meyer [24] analyzed the thermal performances of a GCHP and an ASHP system, concluding that the GCHP, with horizontal or vertical GHEs, was more favorable regarding financial feasibility. Esen and Inalli [39] utilized an in situ thermal response test for a ground-source HP system in Turkey, finding that the thermal resistance of the ground fluctuated marginally with depth. Del Col et al. [40] effectuated a test of the ideal operation mode of a ground-source HP system with variable-speed compressor, water pumps and fans in the loops. They presented the modeling of this HP system type and the model obtained was compared with trial information taken during a heating season. Zeng et al. [41] additionally performed an exploratory performance analysis of GCHP for heating and air-conditioning operating modes in the karsts area. Following the results of the study, they concluded that the GCHP efficiency was affected by its operating modes.

This article focuses on the energy and environmental analysis of an existing vertical GCHP system mounted in an experimental laboratory located in Timisoara, Romania, in a continental temperate climate. The system consisted of a simple GCHP with a single U-tube borehole heat exchanger that had been operational since 2012. Exploratory measurements was utilized to determine the GCHP performances at various operating modes. Generally, the circulation pump of GCHP system operates at fixed speed and supplies a constant flow rate. That is why a variable flow rate control method of the circulation pump is proposed to improve the efficiency of the system and save energy during the heating/cooling (air-conditioning) operation. Thus, the development and implementation of a system energy optimization equipment utilizing quantitative regulation with a buffer tank (BT) and a variable-speed circulation pump is one of the principal creative contributions of this survey. A comparative analysis of primary efficiency parameters (e.g., coefficient of performance (COP), CO_2 emission) achieved for one month of working, utilizing the two control strategies—the standard and optimized regulation of the water pump speed in various operating modes is accomplished. In addition, a model of numerical simulation of heat/cold consumption in heating/air conditioning operating modes utilizing Transient Systems Simulation (TRNSYS) is built. At last, the simulation acquired utilizing TRNSYS program is investigated and compared with test estimations.

2. Testing Laboratory Description

Exploratory analysis of GCHP efficiency was led in a laboratory (Figure 1) situated on the ground floor of the Civil Engineering Faculty building of Timisoara, Romania. The City of Timisoara is located in the West of Romania, and its latitude and longitude are 45°47′ N and 21°17′ E, respectively. The climate is continental temperate with four unique seasons. The heating season in Timisoara runs from 1 October to 30 April, and the air-conditioning season runs from 1 May to 30 September.



Figure 1. Exploratory laboratory.

The lab room has an area of 47 m², and its height is 3.70 m. The exterior walls are made of a 200 mm permeable block with a 100-mm thermal protecting layer and a 20 mm lime mortar. The U-values are as per the following: walls, $0.345 \text{ W/m}^2\text{K}$, and twofold windows, $2.22 \text{ W/m}^2\text{K}$. The window area is 16 m², and the interior door area is 2.1 m². The indoor air set temperature is 20 °C for the cold season (winter) and 26 °C for the hot season (summer). The outside air set temperature is -15 °C for the cold season and 32.6 °C for the hot season.

The GCHP was mounted in this exploratory lab room heated and cooled through fan coil units. With the input information referred to above, a heating demand of 3.11 kW and an air-conditioning demand of 2.15 kW were determined [42]. The lab room area was assimilated with a three-man apartment area in Timisoara. Taking into account the average daily consumption of domestic hot water (DHW) of 50 l/person, a tank hot-water temperature of 45 °C and a cold water temperature of 20 °C, a DHW load of 4.36 kW was calculated. Figure 2 delineates the month to month energy requirement for heating (positive values) and air-conditioning (negative values) of the research space.



Figure 2. Monthly energy demand for laboratory heating/air-conditioning.

3. Description of Experimental Set-Up

The GCHP exploratory system comprised of a BHE, HP unit, BT, water circulation pumps, fan coil units, sink, information acquisition tools, and auxiliary elements, as illustrated in Figure 3.



Figure 3. Scheme of the investigational GCHP with optimized flow rate regulation: 1—GCHP; 2—Fan coil unit; 3—Wash stand; 4—DHW expansion tank; 5—BHE expansion tank; 6—Heating/cooling loop expansion tank; 7—Heat meter; 8—Ultrasonic flow meter for heating/cooling loop; 9—Temperature sensors; 10—BHE loop ultrasonicloop meter; 11—Automatic airvent; 12—BHE; 13—Electric panel; 14—Three phase electronic meter; 15—Protection system; 16—DHW meter; 17—Outdoor temperature sensor; 18—Buffer tank; 19—Circulation pump; 20—Flow meter.

The heat carrier fluid can be conveyed towards two fan coils units in two discharge regulation types [42]:

- 1. By a distribution pump connected within the HP unit of the GCHP (standard case);
- 2. By a constant-speed flow pump connected with a BT. The GCHP computerization can adjust the circulation pump connected to the BT by ON/OFF exchanging. This equipment improves the functioning of whole framework. The BT permits diminishing the GCHP ON/OFF exchanging due to its thermal inertia, and in this way, the energy performance increments. The solution for heat carrier fluid discharge regulation was advanced, utilizing a programmed control equipment of water pump speed [43]. The principal elements of the programmed equipment for pump speed adjustment are presented in Figure 4 [44].



Figure 4. Scheme of a control device of the variable-speed pump.

For the proposed setting, various estimations of the heat demand were determined, utilizing the Romanian standard SR 1907 and of the cold demand, utilizing the Romanian standard SR 6648, contingent upon the temperature difference Δt among interior and outside, as follows:

- For heating, a range of outdoor air temperature was considered between -20 °C and 22 °C, taking into account that the indoor air temperature is constant (*t_i* = 22 °C), and the outdoor air temperature varies depending on the day–night and seasonal alternation;
- For cooling, the range of the outdoor air temperature was considered to be between 26 °C and 42 °C, taking into account that the indoor air temperature is steady (*t_i* = 26 °C), and the outdoor air temperature fluctuates, relying upon the day–night and seasonal alternation.

Each estimation of the heat and cold demand corresponds to a flow rate of heat carrier fluid that must be pumped in the system by the circulation pump at a certain speed ν of its engine and, in this way, the speed control of the circulation pump joined to the BT is controlled.

Following the calculations performed, the variation curve—by points of the frequency v, in Hz—of the frequency converter was obtained depending on the temperature difference Δt in °C. Applying the geometric interpolation by the numerical method of the smallest squares [45], the analytical expression of this curve was determined, under the forms:

For heating:

$$\nu = 7.652\Delta t^{0.284} \tag{1}$$

• For cooling:

$$\nu = 14.845\Delta t^{0.175} \tag{2}$$

Figures 5 and 6 graphically illustrate both the curve of the function given by points for heating and cooling, respectively, and the corresponding geometric interpolation curve analytically expressed by Equation (1) or Equation (2), which shows a good concordance between them.



Figure 5. Graph of the function $v(\Delta t)$ for heating operation.



Figure 6. Graph of the function $v(\Delta t)$ for cooling operation.

Figure 7 illustrates a scheme of the programmed control equipment of the circulation pump speed as per the heating/air-conditioning request of the lab room. In contrast to the traditional solution, in which the circulation pump ON/OFF exchanging is adjusted by the GCHP computerization, the optimized solution guarantees both the ON/OFF exchanging and the speed adjustment of the circulation pump.



Figure 7. Scheme of the programmed control equipment for circulation pump: 1—Feeder 220 Vca/24 Vcc; 2—Frequency converter; 3—PLC; 4—Circulation pump; 5—Heating/cooling switcher; 6—Indoor temperature sensor; 7—Outdoor temperature sensor.

The temperature difference between within and outside of the heated/air-conditioned room is estimated by temperature sensors TS1 and TS2, coupled with a programmable logic controller (PLC).

In the interior memory of the PLC, a calculation algorithm, dependent on Equations (1) and (2), is actualized for ascertaining the frequency of the frequency converter, which relies upon the measured temperature difference. The PLC sends to the frequency converter the comparing frquency incentive to guarantee the fluid discharge, as indicated by the heating/air-conditioning requirement at that time. Additionally, the PLC permits the ON/OFF exchanging control of the circulation pump. For the concurrent ON/OFF exchanging of the circulation pump and GCHP, common temperature values of this procedure were determined.

3.1. Borehole Heat Exchanger

In the GCHP application, the BHE ensure that the heat extraction or rejection from the ground. The running the BHE instigates concurrent heat and moisture flows in the soil in which the loop is installed. The heat exchange among BHE and soil is fundamentally practiced by thermal conduction and, in a specific way, by moisture movement. In this manner, soil types and temperature, similar to moisture gradients, have critical impacts on the heat exchange process [41]. Thus, the properties of the soil at the trial site are fundamental.

The straightforward vertical borehole of this exploratory GCHP has a depth of 80 m. Antifreeze solution (glycol 30%) flows in a sole polyethylene U-tube of 32 mm interior diameter, with a 60-mm partition between the return and supply tubes, placed in the borehole. The borehole with a total diameter of 110 mm was loaded up with sand and finished with a bentonite layer at the top to stay away from the interruption of contaminations in the aquifers. The mean temperature over the full borehole depth was 15.2 °C.

The ground proprieties are based on estimations acquired from the Banat Water Resources Management Agency [46]. The mean thermal conductivity and thermal diffusivity of the ground over the full borehole depth were 1.91 W/(mK) and 0.80×10^{-6} m²/s, respectively. The boreholes were backfilled with grout, blended in with penetrating mud, cement, and sand in explicit extents. The thermal conductivity and diffusivity of the grout tried by the maker were 2.31 W/(mK) and 0.92×10^{-6} m²/s, respectively.

3.2. Heat Pump Unit Equipment

The HP unit utilized in this study is a ground-to-water HP with a scroll air-tight compressor that works with the refrigerant R410A. The nominal heating and air-conditioning limits are 6.5 kW and 3.8 kW, respectively. The HP unit is an electrically-powered HP, having an interior refrigerating system and a DHW tank with a 175-L capacity. Running the HP was administered by an electronic regulator, which, contingent upon the water return temperature, turned the HP compressor on or off. The circulation pump of heat source was adjusted by the HP regulator that starts the heat source pump 30 s before compressor actuation.

The energy efficiency of a reversible GCHP is expressed by COP_{hp} , which can be calculated utilizing the following formulas:

For heating operating mode:

$$COP_{hp} = \frac{E_t}{E_{el}}$$
(3)

in which E_t is the useful heat and E_{el} is the electricity consumption of HP.

• For cooling operating mode:

$$COP_{hp} = \frac{EER_{hp}}{3.412} \tag{4}$$

in which the energy efficiency ratio EER_{hp} , in Btu/(Wh), describes the cooling performance, and the value 3.412 is the transformation factor from Watt to Btu/h (British Thermal Units per hour).

Another way of expressing COP_{hp} is dependent on the temperature of the hot environment (T_h) and that of the cold environment (T_c) [10]:

$$COP_{hp} = \frac{1}{\frac{T_h}{T_c} - 1}$$
(5)

The COP of the overall system (COP_{*sys*}) is calculated with Equation (3), in which E_{el} is the electricity consumption of the GCHP, including the electricity consumption of the HP compressor, circulation pumps, fan coil units, frequency converter, and PLC.

The carbon dioxide (CO₂) emissions C_{CO_2} during the heating system operation are evaluated as follows:

$$C_{\rm CO_2} = g_{el} E_{el} \tag{6}$$

where $g_{el} = 0.547$ kg CO₂/kWh is the specific CO₂ emission factor for electrical energy [47].

The thermal power E_t for heating/cooling operating modes and electrical consumption E_{el} , utilized by the HP unit or the GCHP system, must be measured to obtain the COP_{hp} or COP_{sys} and CO₂ emissions.

3.3. Water Circulation Pumps

The water circulation circuits of the GCHP comprised of a GCHP-BT circuit and BT-fan coil unit circuit. Two centrifugal pumps with the rated flow rate of 2.8 and, respectively, 5.5 m³/h were used for the two water circuits. The primary circulation pump (constant-speed pump coupled to HP unit) was adjusted by the GCHP computerization, and the subsequent pump (variable-speed pump coupled to BT) was adjusted by a programmed control equipment.

3.4. Fan Coil Units

As terminal units of the GCHP, two parallel coupled fan coil units, whose global thermal power was 3.2 kW, were utilized.

3.5. Information Acquisition System

To more readily comprehend the behavior of the GCHP, an information acquisition system was created to screen the most relevant parameters of the system. The GCHP information procurement system comprises the inside and outside temperatures, supply/return temperatures, heat source temperature (outlet BHE temperature), DHW temperature, relative air humidity, and primary working parameters of the system elements.

3.6. Measurement Instrumentations

A network of sensors was created to permit the registration of the most important parameters of the system, which are described in detail in [42], together with the other measuring instruments, including their accuracy.

4. Methodology

Various experiments were performed on the GCHP system under steady-state operation regimes to assess its COP and CO₂ emission [42]. The tests were partitioned into heating and cooling tests with or without the production of DHW. The system was examined for 2 years. The exploratory estimations were accomplished for two strategies of the flow rate regulation of the heat carrier fluid (water) in the system: solution (1)—standard regulation, by constant-speed circulation pump coupled with HP unit, and solution (2)—optimized regulation, by variable-speed circulation pump coupled with BT. In the heating operating mode, the tests were directed for a 1-month time frame for every one of the two analyzed situations, from 25 January 2013 to 23 February 2013 and 18 January 2014 to 16 February 2014.

The outside temperatures varied from -5.8 °C to 10.1 °C. The month to month average estimations of the outside temperature during the two-time intervals were practically equivalent. In the cooling operating mode, the tests were directed for 1-month for every one of the two investigated situations, from 25 May 2013 to 23 June 2013 and from 30 May 2014 to 28 June 2014. The outside temperature varied in the interval of 15.3–34.8 °C. The month to month average estimations of the outside temperature during the two-time intervals were practically equivalent.

5. Experiment Results and Discussion

5.1. GCHP Efficiency in Various Operating Modes

5.1.1. Heating Operation

Variation in the indoor air temperature (t_i) and outside temperature (t_e), registered for a 1-month time frame, is illustrated in Figure 8. It ought to be noticed that, in the solution (2), a diminished indoor temperature was acquired, around the design temperature of 22 °C, which leads to all the more suitable comfort in the heated space. Furthermore, the diminished wavering of the antifreeze solution temperature prompts a lower heat source requirement. The heat source temperature in the cold season is up to 12–13 °C higher than the outside temperature, which raises the capacity and improves the performance of a GCHP. Table 1 presents the summarization of the average estimation of the temperatures (t_i , t_e , t_s), power consumption (E_{el}), helpful heat for heating (E_t), COP of the system (COP_{sys}) and the HP unit (COP_{hp}), and CO₂ emission (C_{CO2}).



Figure 8. Registered indoor and outside temperature during heating operating mode.

Table 1. GCHP system performance in heating operating mode.

Flow	t _i	t _e	ts	E _{el}	E _t	COP _{sys}	COP _{hp}	C _{CO2}
Regulation	(°C)	(°C)	(°C)	(kWh)	(kWh)	(–)	(-)	(kg)
(1) Standard	22.64	3.24	16.23	125.17	510.61	4.06	4.81	50.46
(2) Optimized	21.84	3.76	17.08	116.47	512.54	4.40	5.07	46.94

The COP_{hp} estimations of the HP unit for standard and improved solution are 4.81 and 5.07, respectively. For the optimized solution, the $\text{COP}_{sys} = 4.40$ is 7.5% higher and the CO_2 emissions level is 7% lower than on the classic solution. Because of the characteristics of the climate, ground, and the site where the estimations were performed and a higher underground water flow rate, the heat source temperature is increased, and the COP_{hp} and COP_{sys} are remarkable values for the two cases.

Consequently, the comparison of these results with others obtained in similar analyses [6,48], but in different local conditions, showed that the efficiency obtained here are significantly improved.

5.1.2. Cooling Operation

Figures 9 and 10 show the variation in the time of the characteristic temperatures (t_e , t_i , and t_s) recorded during the experiments in the cooling operating mode. It should be noted that, in case (2), a more decreased indoor air temperature was acquired, around the design temperature of 26 °C, prompting higher comfort in the heated space. Table 2 summarizes the average estimations of the temperatures (t_i , t_e , t_s), power consumption (E_{el}), useful heat for air-conditioning (E_t), system EER (EER_{sys}) and CO₂ emissions (C_{CO₂}). The energy efficiency ratio EER_{sys} of the GCHP system was 8% lower in the optimized solution than in the classic solution.



Figure 9. Variation in outside temperature during the air-conditioning operating mode.



Figure 10. Registered indoor and heat source temperature during air-conditioning operating mode.

Flow	t _i	t _e	ts	E _{el}	E _t	<i>EER_{sys}</i>	COP _{sys}	C _{CO2}
Regulation	(°C)	(°C)	(°C)	(kWh)	(kWh)	(Btu/Wh)	(-)	(kg)
(1) Standard	26.21	24.53	20.49	71.00	287.22	13.78	4.03	28.61
(2) Optimized	25.72	24.89	20.66	65.15	288.44	15.10	4.43	26.24

Table 2. GCHP system performance in air-conditioning operating mode.

The correlation of the exploratory GCHP efficiency in the heating and air-conditioning period (Tables 1 and 2) demonstrate that the efficiency in heating and air-conditioning mode was practically equivalent. This is because the heating burden was higher than the air-conditioning demand, and additionally, the power consumption in the heating mode was higher than the power consumption in air-conditioning operation.

The COP estimations of the system were compared with the current COP values recorded in GCHPs investigation. The exploration of Man et al. [38] uncovered COPs of GCHPs of 4.18–4.57 in the winter and 3.9–4.54 in the summer. Likewise, the investigation in the hot season of Michopoulos et al. [31], which utilized a vertical GHE at a depth of 80 m, obtained a COP_{sys} of 4.4–4.5. It is seen that the efficiency esteems achieved here are comparative.

5.1.3. Heating and DHW Provision

For the comparative analysis of the two flow control strategies, an equivalent DHW volume was utilized, $V_{dhw} = 1.21 \text{ m}^3$. The variation in the indoor air (t_i) , outside air (t_e) , DHW (t_{dhw}) and heat source (t_s) temperatures—registered for seven days' time span for every one of the two investigated solutions, from 15 January 2013 to 21 January 2013 and 10 January 2014 to 16 January 2014—is illustrated in Figures 11 and 12. The heat source temperature in the cold season is roughly 15 °C higher than the outside temperature.



Figure 11. Registered indoor and outside temperatures during heating and DHW provision.



Figure 12. Variation in DHW and heat source temperature during heating and DHW provision.

Table 3 summarizes the average estimations of the characteristic temperatures (t_i , t_e , t_{dhw} , t_s), DHW volume (V_{dhw}), electrical energy consumption (E_{el}), helpful heat (E_t), COP of the system (COP_{sys}) and of the HP unit (COP_{hp}), and CO₂ emission (C_{CO₂}).

Flow	<i>t</i> _i	<i>t</i> _e	ts	t _{dhw}	<i>V_{dhw}</i>	E _{el}	E _t	COP _{sys}	COP _{hp}	C _{CO2}
Regulation	(°C)	(°C)	(°C)	(°C)	(m ³)	(kWh)	(kWh)	(–)	(-)	(kg)
(1) Standard	21.27	1.03	16.79	42.63	1.21	82.66	266.99	3.22	3.80	33.31
(2) Optimized	21.61	0.93	16.27	42.64		80.12	269.13	3.36	3.96	32.28

Table 3. GCHP efficiency for heating and DHW operation.

Although the COP_{sys} recorded equivalent values for the two solutions, the results of the experiments show that, when utilizing programmed control equipment for the circulation pump speed, an electrical energy saving of 3% and a CO₂ emissions reduction of 3% were obtained.

Examining the test information (Tables 1 and 3) results demonstrate that the COP_{sys} of the system working in heating and DHW provision, comparative with the heating operating mode, diminished fundamentally in the domain 20.6–23.9%, in contrast with the two solutions, from 4.06–4.40 to 3.22–3.36, respectively. The COP_{hp} estimations of the HP unit for solutions (1) and (2) are 3.80 and 3.96, respectively.

5.1.4. Cooling and DHW Provision

To establish the GCHP efficiency in summer, test estimations over 1 week for every one of the two analyzed control strategies was accomplished from 26 June 2013 to 2 July 2013 and 29 June 2014 to 5 July 2014. During the experiments, both the cooling demand and the DHW demand for a family that utilizes a DHW volume $V_{dhw} = 1.35 \text{ m}^3$ were provided. Figure 13 outlines the evolution of indoor air (t_i) and outside air (t_e) temperatures, and Figure 14 shows the variation in the DHW (t_{dhw}) and heat source (t_s) temperatures.



Figure 13. Registered indoor and outside temperatures during air-conditioning and DHW provision.

Table 4 includes a summarization of the average estimations of the characteristic temperatures (t_i, t_e, t_{dhw}, t_s) , DHW volume (V_{dhw}) , power consumption (E_{el}) , helpful heat (E_t) , COP of the system (COP_{sys}) and CO₂ emissions (C_{CO2}).

COP_{sys} Flow t_i t_e V_{dhw} E_{el} E_t C_{CO_2} t_s t_{dhw} Regulation (°C) (°C) (°C) (°C) (m³) (kWh) (kWh) (--) (kg) 25.40 24.96 20.62 38.92 50.80 198.62 3.90 20.47 (1) Classic 1.36 25.28 39.12 (2) Optimized 25.4520.6048.22 195.06 4.05 19.43

Table 4. GCHP efficiency for air-conditioning and DHW operation.



Figure 14. Variation in DHW and heat source temperatures during air-conditioning and DHW operation.

Though the COP_{sys} recorded values about equal for the two solutions, the trial results show that, when utilizing the programmed control equipment for the circulation pump speed, an electrical energy saving of 5% and a CO₂ emissions reduction of 5% were obtained.

The test information (Tables 2 and 4) on the COP_{sys} of the GCHP in air-conditioning and DHW operation, comparative with air-conditioning operating, show that there is an abatement in the domain of 3.2–8.6% compared with the two solutions, from 4.03–4.43 to 3.90–4.05, respectively.

Figures 15 and 16 sums up the efficiencies of the GCHP system in the diverse operating modes to reveal the exploratory estimations of the COP_{sys} and CO_2 emissions (C_{CO_2}).



Figure 15. Evolution of COP_{*sys*} in various operating modes.



Figure 16. Evolution of CO₂ emissions in various operating modes.

5.2. Uncertainty Analysis

Uncertainty analysis (the analysis of vulnerabilities in test estimation and results) is important to assess the exploratory information [49,50]. The methodology utilized to carry out uncertainty analysis was described by Holman [49]. An outcome Z is a function given by the independent variables x_1 , x_2 , x_3 ... x_n . On the off chance that the errors in the independent variables w_1 , w_2 , w_3 ... w_n are all given

with same chance, then uncertainty in the outcome w_Z with this chance is determined utilizing the following formula [49]:

$$w_Z = \sqrt{\left(\frac{\partial Z}{\partial x_1}w_1\right)^2 + \left(\frac{\partial Z}{\partial x_2}w_2\right)^2 + \ldots + \left(\frac{\partial Z}{\partial x_n}w_n\right)^2} \tag{7}$$

In the current survey, the temperatures, heat, and electricity were estimated with suitable instruments, already clarified. Error analysis for assessing the maximum uncertainty in the trial results was conducted utilizing Equation (7). It was discovered that the most extreme uncertainty in the outcomes is in the COP_{sys} , with an acceptable uncertainty interval of 1.31–1.69% in the heating operating mode and of 2.29–3.38% in the air-conditioning operating mode. The accuracy of the COP_{hp} was evaluated in the range of 1.82–2.33% in the heating mode.

6. Numerical Simulation of Useful Thermal Energy Using TRNSYS Software

The essential rule of the TRNSYS software [51] is the implementation of algebraic and first-order conventional differential equations depicting physical components into programming subroutines (types) with a standard interface. The STEC library is based on steady-state energy preservation formulated in thermodynamic amounts (temperature, pressure, and enthalpy). One of the principal advantages of TRNSYS program for the modeling and design of HP systems is that it incorporates parts for the computation of building thermal loads, specific components for HVAC systems, HP systems, circulation pumps, modules for vertical GHEs and storing heat, similar to climatic information files, which make it an entirely appropriate instrument to model a GCHP system to provide heating and air-conditioning to a building. The METEONORM program [52] has a database of over 7000 worldwide stations that can generate data on monthly, daily, or hourly time scales.

For the validation of the numerical simulation model, four statistical assessment criteria (indices) were utilized—the coefficient of multiple determinations (R^2), the root mean square error (RMSE), the coefficient of variation (c_v), and relative error (e_r)—as defined below [53,54].

The coefficient R^2 presents the general concordance among measured (real) and simulated (predicted) time series, and ranges from 0, for a poor model, to 1 for a perfect model, and is expressed as:

$$R^{2} = 1 - \frac{\sum_{j=1}^{n} (y_{\sin,j} - y_{\max,j})^{2}}{\sum_{j=1}^{n} y_{\max,j}^{2}}$$
(8)

in which *n* is the number of data points; $y_{\text{mea},j}$ is the measured value of data point *j*; $y_{\text{sim},j}$ is the predicted value.

The RMSE is a measure of overall performance over the entire range of the data set expressed by the formula:

$$\text{RMSE} = \sqrt{\frac{\sum_{j=1}^{n} (y_{\sin,j} - y_{\text{mea},j})^2}{n}}$$
(9)

Greater or equal to 0, the RMSE indicates an ideal model fit when it equals 0.

The c_v (in %) can be interpreted as an order of magnitude of the repeatability relative standard uncertainty and it is defined by:

$$c_{v} = \frac{\text{RMSE}}{\left|\overline{y}_{\text{mea},j}\right|} 100 \tag{10}$$

where $\overline{y}_{\text{mea},i}$ is the average value of all measured data points.

The relative error e_r , in %, can be calculate as follows:

$$e_r = \frac{y_{\min,j} - y_{\max,j}}{y_{\max,j}} 100 \tag{11}$$

6.1. Definition of the Operational Plan

To simulate the heat energy utilized for the heating/air-conditioning of the exploratory laboratory space, the operational connections were set up between the construction and all inner and outer factors.

Figure 17 shows the operation plan made in TRNSYS, where the construction thermal behavior was modeled utilizing a "Type 56" subroutine elaborated with the TRNBuild interface. Climate information for the Timisoara was obtained from the METEONORM information base [52], and the climate information readers, "Type 109" and "Type 89d", were utilized to change over the information in a form intelligible from TRNSYS. The simulation model considered the outside air penetration, heat/cold source type, and internal gains. Likewise, light thresholds, air-conditioning, and shading were characterized for a decent way to deal with the model. The results were extracted with an online plotter ("Type 65").



Figure 17. Simulation scheme of the thermal energy consumption in TRNSYS software.

6.2. Comparison of Simulated Results with Exploratory Information

The simulated values of heat energy utilized for space heating and conditioning were obtained from the simulations performed for a period of 1 year (8760 h) and included the values measured in Table 5. The statistical criteria (RMSE, c_v , and R^2) for the assessment of simulation model accuracy are presented in Table 6, considering the operation of GCHP system in various modes, and the relative errors e_r , which are included in Table 5.

The maximum relative error e_r of the simulated values in TRNSYS, compared to those measured, was about 1.60% during heating and around 1.65% during air-conditioning, which is very satisfactory. The values of the RMSE and c_v performance indices in the heating period are approximately 2.7222 and 0.0141, respectively, and in the air-conditioning period they are approximately 3.080 and 0.0238, respectively. The R^2 -values in the two operating modes are approximately 0.9998, which is very acceptable. Accordingly, the simulation model was validated by the experimental measurements.

Month	Heating En	ergy (kWh)	Relative	Cooling En	Relative		
Wonth	Predicted	Measured	Error e_r (%)	Predicted	Measured	Error e_r (%)	
Jan.	505.00	512.48	-1.45	0.00	0.00	0.00	
Feb.	391.40	390.12	+0.32	0.00	0.00	0.00	
Mar.	303.21	300.87	+0.77	0.00	0.00	0.00	
Apr.	99.45	97.89	+1.59	0.00	0.00	0.00	
May	0.00	0.00	0.00	208.00	211.23	-1.52	
Jun.	0.00	0.00	0.00	242.00	242.82	-0.33	
Jul.	0.00	0.00	0.00	337.90	333.12	+1.43	
Month	Heating En	Heating Energy (kWh)		Cooling Energy (kWh)		Relative	
month	Predicted	Measured	Error e_r (%)	Predicted	Measured	Error e_r (%)	
Aug.	0.00	0.00	0.00	437.80	445.14	-1.64	
Sep.	0.00	0.00	0.00	324.30	319.20	+1.59	
Oct.	189.70	191.31	-0.84	0.00	0.00	0.00	
Nov.	348.90	345.23	+1.06	0.00	0.00	0.00	
Dec.	477.50	480.21	-0.56	0.00	0.00	0.00	

Table 5. Heat energy utilized for laboratory heating and air-conditioning.

Table 6. Performance criteria for simulation model.

Operating Mode	RMSE	c_v	<i>R</i> ²
Heating	2.72187	0.01409	0.99990075
Air-conditioning	3.08003	0.02382	0.99977802

7. Conclusions

The utilization of HPs in present-day buildings with improved thermal protection and the diminished heat demands is a good option, in contrast to the old-style heating/air-conditioning and DHW heating solutions. This article focused on the assessment of the GCHP system efficiency, giving heating/air-conditioning and DHW a laboratory trial. The performance of the GCHP system, in both optimized and standard mode, was assessed via both experiments and computer simulations with TRNSYS. The principal conclusions of this experimental survey are as follows:

- 1. The accomplished trial investigations showed higher efficiency of the GCHP for the flow rate regulation solution utilizing a BT and programmed control device for circulation pump speed, comparative with an old-style regulation solutions (COP_{sys} with 7–8% increase, and CO₂ emissions with 7.5–8% decrease).
- 2. The overall COP of the GCHP in the two solutions was $COP_{sys} > 4$ for the heating or air-conditioning period, and $3 < COP_{sys} < 4$ for heating or air-conditioning and DHW provision.
- 3. In the standard and optimized flow regulation solutions, the experimental evaluations of the COP for HP unit (COP_{hp}) during the heating and DHW period were 3.80 and 3.96, respectively, and during the heating period they were 4.81 and 5.07, respectively.
- 4. The GCHP system utilized for cooling and DHW provision has higher performance ($COP_{sys} = 3.90$ for the classic case and 4.05 for optimized case) than that for heating and DHW production ($COP_{sys} = 3.22$ for the classic case and 3.36 for optimized case) because the heat extracted from the soil for heating and DHW production is higher than the heat injected into the soil for air-conditioning and DHW heating.
- 5. When utilizing the variable-speed water pump, electrical energy saving and a decrease in the CO₂ emissions of 3% for air heating in the laboratory and 5% for air cooling in the space were obtained simultaneously with DHW generation.

- 6. The proposed numerical simulation model can be utilized as an instrument to establish the GCHP performance in various operating modes in order to optimize the energy efficiency of the system and guarantee the occupant's comfort consistently.
- 7. A limitation of this study is that the optimized solution for regulating the flow rate of heat-carrying fluid using a buffer tank and an automatic control device of the circulation pump speed is applicable only to residential buildings.
- 8. In order to possibly improve the energy efficiency of the GCHP system, in-depth research is needed in the future, aimed at integrating solar photovoltaic collectors into the system, which will produce electricity to drive the circulation pump in the water pumping process.

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