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A Study of a Heat Pump Ground Collector

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Summary. The paper presents selected issues related to the investigations into a ground collector of a heat pump fitted in a low energy single-family house. The experimental investigations performed in a test station (a single family house fitted with space heating) should allow an estimation of the influence of selected operating parameters of the ground collector on the performance of a heat pump.

Key words: ground source heat pump, vertical collector, spiral compressor, ground heat source, coefficient of performance (COP).

INTRODUCTION

The increasingly expanding range of applications of heat pumps depends not only on their advanced design and functionalities but also on the workmanship and the efficiency of the ground heat source. In many countries, the heat pump market is dominated by the brine/water type [18, 22]. The trend is beginning to change in favor of the air/water heat pumps, despite the fact that ground heat pumps are characterized by high coefficients of performance and reliability. The said potential is conditional upon proper workmanship of the ground heat source. Poorly built installations spoil the opinion of not only the geothermal systems but also the entire heat pump technology.

From the analysis of available data on heat pumps, out of all installed heat pumps the prevailing are ground source heat pumps that, due to a rather cold climate, are the preferred solution in Poland. In Poland, in the ground source heat pump sector, the most frequently applied are pumps fitted with vertical (60%) and horizontal (30%) collectors. As results from the data [8, 18], there are two reasons for the prevailing status of vertical collectors (practically twice the number of horizontal collectors). Firstly, there exists a general conviction that systems fitted with vertical collectors have better performance and reliability and, secondly, systems fitted with horizontal collectors have terrain-related limitations (a plot size of approx. 4 times the area of the heated building is required for proper operation of horizontal collectors).

Energy-related analyses along with the analyses of the efficiency of pumps fitted with ground source collectors (both vertical and horizontal) have already been performed in many scientific centers [17]. Hepbasli et al. [5] have determined the energy effects for a system in which the compressor pump operated with vertical collectors. Following this analysis, thermal efficiency of the ground source collector ducts was determined along with the coefficient of performance. Akpinar and Hepbasli [1] have analyzed energy-related problems and expenses for heat pumps (fitted with vertical collectors) used for heating purposes. The authors have developed a simulation model that can be used for the analysis of expenses when a heat pump is applied for heating purposes. Kurpaska et al. [13] in his experimental research have determined the efficiency of a heat pump fitted with vertical collectors used for the heating of a glasshouse. In the work by Kurpaska et al. [12], a theoretical analysis was performed of the change of the thermal capacity of a ground source collector depending on the ground humidity and temperature. As can be observed, the efficiency of a heat pump depends on a variety of design, configuration and climate related factors. Hence, the purpose of this paper is to analyze the efficiency of a heat pump fitted with vertical collectors.

CHARACTERISTICS OF THE RESEARCH OBJECT

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A heating system utilizing the brine/water heat pump was installed in a two story single-family house of the total usable area of 156 m². Low temperature space heating was applied in the ground and first floors. The water in the heating circuits on the ground and first floors was fed by two separate distributors powered with two separately controlled circulation pumps, which allowed an easy control of the thermal load of the heat pump.

From the analysis of the available literature it results that the demand for heat, for new buildings with good quality thermal insulation, is 40-50 W m^{-2} [8]. Assuming a unit thermal load of 50 W m^{-2} and the usable area of the building of 156 m^2, an approximate thermal load of the building of 7.8 kW was obtained. If the same heat pump is used for the production of domestic hot water then, for a 4 person family, an additional 1 kW needs to be added [30]. Eventually, for a house of the area of 156 m^2 a heat pump of the capacity of 9 kW was applied.

COMPONENTS OF THE HEAT PUMP

The heat pumps used in low energy homes are controlled with thermostatic expansion valves. The realization of the adopted scope of experimental research forced the application of a heat pump controlled with an electronic expansion valve. Purchasing serially manufactured heat pumps was not an option. Instead, components made by reputable manufacturers from the refrigeration industry were used for its construction

R407C was used as refrigerant and plate heat exchangers by WTK were used in the condenser and evaporator circuits

The applied R407C refrigerant is a three-component mixture. Each of these components has a different evaporation temperature. In order to ensure 100% evaporation of each of the components, a super-heater of drawn-in gas was used. This eliminates the fluid uptake into the compressor and improves the coefficient of performance of the thermodynamic cycle.

The plate heat exchangers were selected through Avogadro2 rev. 2.0. software. The standard working parameters for a brine heat pump are B0/W35, which denotes the refrigerant evaporation temperature (0 °C) and the condensation temperature (35 °C). Assuming an expected difference in the glycol circuit amounting to $\Delta Tgs=3$ °C and 5 °C for superheating and subcooling of the refrigerant, a P7 series 30-plate heat exchanger was selected [27]. Fig. 1 presents a diagram of the applied heating system.

SELECTION OF THE GROUND SOURCE

The efficiency of a heat pump increases as the difference between the ground heat source and air heat source decreases. The most effective ground heat source is ground water, whose temperature, irrespective of the season, remains on a constant level of approx. 10 °C. This is also a cost effective solution if the depth to ground water does not exceed 7 m and the wells (in a dual well system) are drilled no deeper than 15 m.

The simplest and the cheapest ground heat source solution is the air drawn to the heat pump, however the application of air as an external heat source in our climate is ineffective. In the winter, when the temperatures outside are below zero, the performance of such a system decreases to the value of COP=2-3 along with its heating capacity, which may render the space insufficiently heated [10, 14].

There are also vertical or horizontal collectors (flat or spiral). The latter are cheaper but cover a large area of the plot – two to four times greater than the usable area of the house.

Due to limited plot development possibilities and bad soil conditions (sand was found at the depth of 1-2 m) the construction of a ground heat source in the form of a horizontal collector was given up. Due to high additional costs of the determination of the ground thermal capacity, the examination of the heat transfer capacity by geothermal probes was also given up.

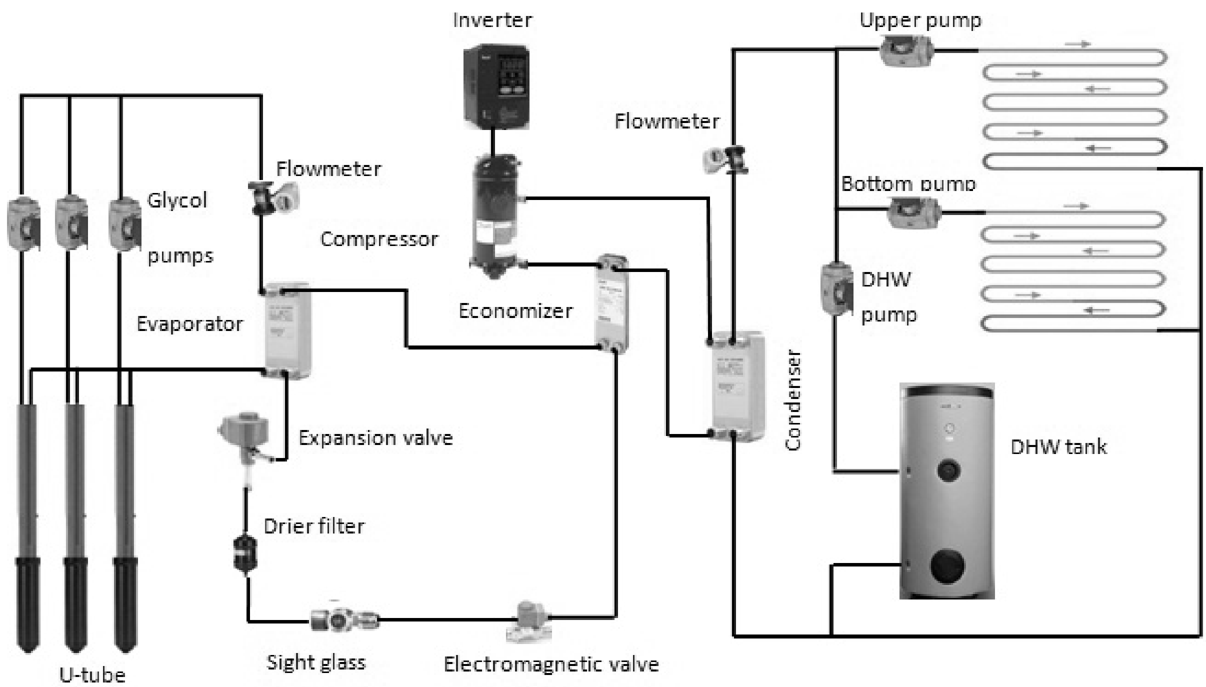

Fig. 1. Diagram of the heating system utilizing a heat pump

Dedicated Energeo software (dedicated by Aspol, a supplier of the vertical collector probes) was used for the selection of the vertical collector. Upon entering of the parameters of the heat pump to the Energeo software for the assumed coefficient of ground heat capacity of 38 W m^{-1}, a total required length of the vertical collector of 186 m was obtained using U probes filled with 20% solution of ethylene glycol. Three probes were used, 62 m each connected to a wall distributor located in the boiler room, hydraulically balanced with ball valves.

THE TESTING PROCEDURE

The heat pump was used in the space heating in constant cycles:

– Cycle 1 working-time 60 minutes, dwell time 60 minutes,
– Cycle 2 working-time 120 minutes, dwell time 60 minutes

During the tests, basic operating parameters, needed for the determination of the energy balance were recorded.

The application of two-speed circulation pumps in individual circuits of the ground source and space heating allowed an easy adjustment of the mass flow of water and volumetric flow of brine. The application for an electronic expansion valve in the freon circuit allowed investigating the heat pump for the refrigerant evaporation temperatures in the range from -5 °C to 2.5 °C.

The tests of the vertical collectors were performed for the space heating mode on and production of domestic hot water. For the assumed constant refrigerant evaporation temperatures a series of tests were carried out for varied demand for heat.

In the space heating circuit the tests were performed for the following cases:

– Space heating on the ground floor only,
– Space heating on the ground and first floors simultaneously.

For the determination of the heating capacity of the condenser a standard relation was applied:

$$Q_{con} = C_{p,w}\, M_w \left(T_{w,out} - T_{w,in}\right), \tag{1}$$

where:

Q_{con} denotes the heat at the outlet from the condenser, M_w denotes the mass flow of water. The cooling capacity of the evaporator was calculated analogically.

The coefficient of performance of the heat pump at a moment in time (t) was determined from a standard relation:

$$COP_{HP} = \frac{Q_{con}(t)}{W_{com}(t)}. \tag{2}$$

A change in the thermal load (change in the mass flow of water, change in the capacity of the receiver) forces a change in the refrigerant and water condensation temperatures in the heating circuit downstream of the condenser, which is why for individual measurement cycles (τ) the average value of the coefficient of performance (COP) was determined according to relation (3)

$$COP = \frac{\int_0^{\tau} Q_{con}(t)dt}{\int_0^{\tau} W_{com}(t)dt}. \tag{3}$$

RESULTS OF EXPERIMENTS AND DISCUSSION

The preliminary research was conducted in the 2012/2013 heating season. During the research, basic operating parameters, needed for the determination of the heat pump energy balance were recorded.

Designations on the figures:

– Tgs, out– temperature of glycol at the inlet to the evaporator,
– Tgs, int – temperature of glycol at the outlet from the evaporator,
– Tsh,out – temperature of return water from the heating circuit,
– Tsh,in – temperature of water fed to the heating circuit,
– Tfl – floor temperature on the ground floor,
– Tev – refrigerant evaporation temperature.

The volumetric flow of glycol in the ground circuit obtained during the experimental research was 1.92 m3 h^{-1} (speed 1) and 2.30 m3 h^{-1} (speed 2) for a simultaneous operation of all three loops of the ground source.

In the heating circuit, the mass flow of water was 0.09 kg s^{-1} (speed I) and 0.13 kg s^{-1} (speed 2) for the circuit pump activated on the ground floor and 0.16 and 0.22 kg s^{-1} for a simultaneous operation of the pumps on the ground and first floors.

COP and the heating capacity were determined for each measurement cycle. To asses the measurement uncertainty for all experiments, a method recommended by Moffat was applied [20]. The calculated measurement error was ±2.45% and ±3.15% respectively.

Fig. 2 presents the trend in the changes of the temperature of brine for the space heating operating on the ground and first floors. During the operation of a heat pump the temperature of the brine, as recorded at the inlet to and outlet from the evaporator, drops gradually. When heat is collected from the ground by the vertical collectors the ground temperature around these collectors decreases.

For the refrigerant evaporation temperature 0 °C an average ΔTgs=2.6 °C was observed along with a drop in the glycol temperature at the inlet to the evaporator (42%) and the outlet from the evaporator (56%) in a 60 minute measurement cycle. For the refrigerant evaporation temperature –2.5 °C a ΔTgs=2.9 °C was observed along with a drop in the temperature of 45% at the inlet and 60% at the outlet.

During the first measurement cycle, for the assumed constant refrigerant evaporation temperature of 0 °C the efficiency in the heat transfer in the evaporator drops, which is a result of a reduction of the temperature difference between the glycol and the freon circuits. As a result, a higher brine temperature can be observed at the outlet from the evaporator. The temperature of the brine at the inlet to the evaporator remains constant.

Fig. 3 presents the influence of the change in the volumetric flow of glycol in the ground source circuit on the

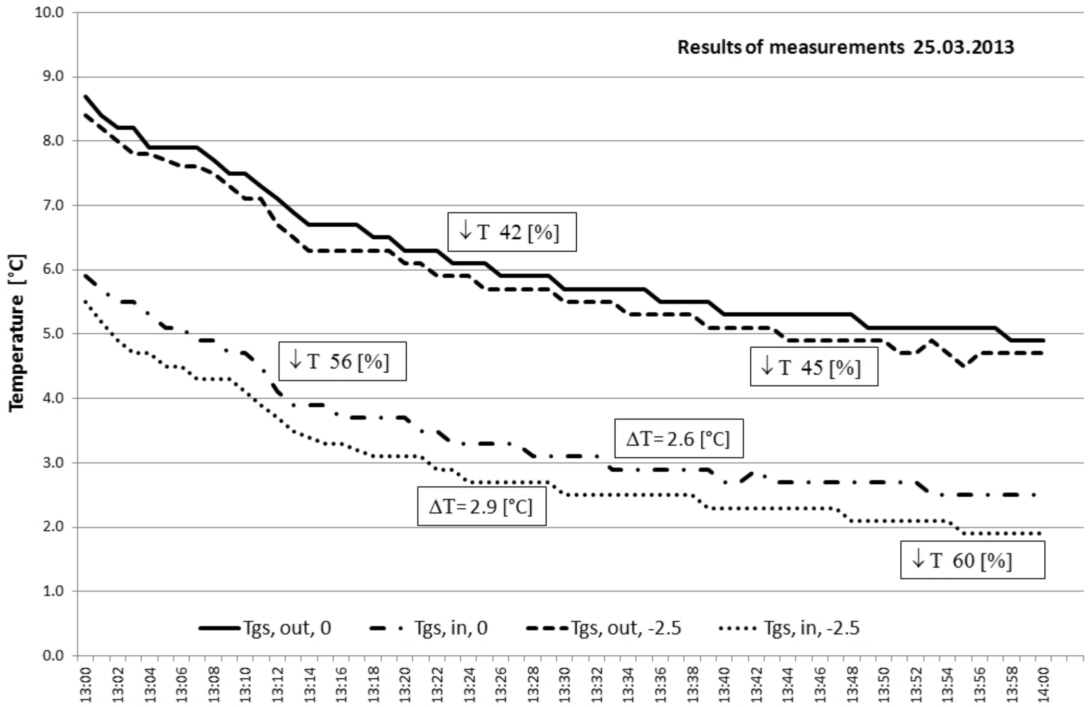

Fig. 2. Changes in temperature in the ground source circuit for the refrigerant evaporation temperature of -2.5°C and 0°C

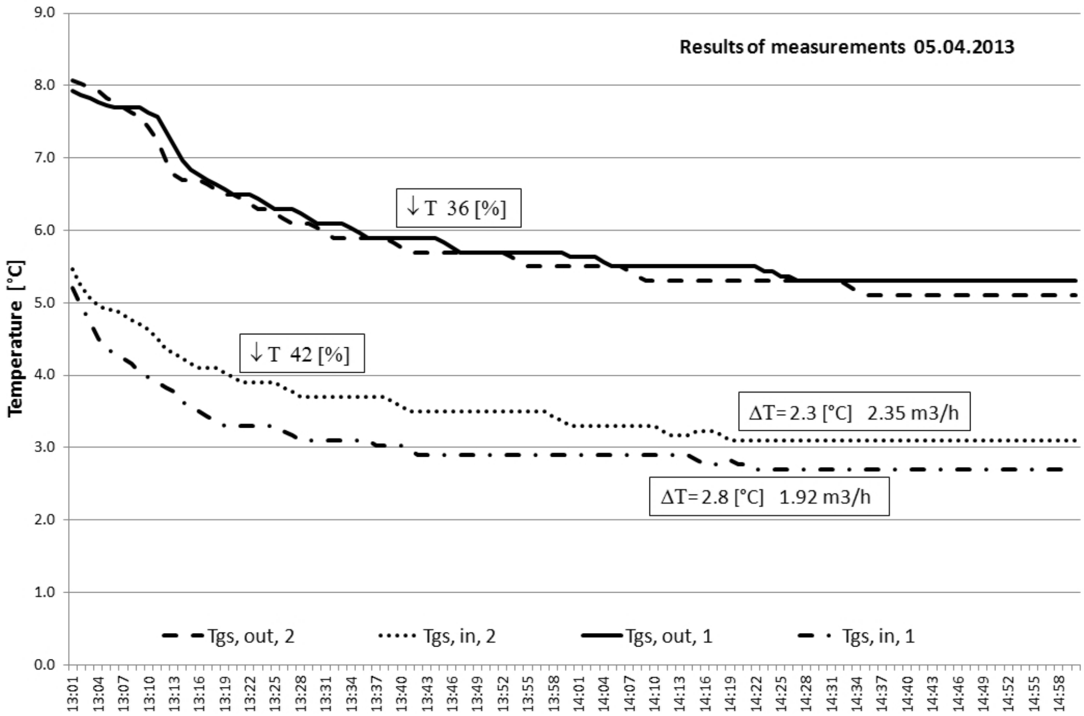

Fig. 3. Changes in the temperature in the ground source circuit for two values of the volumetric flow of glycol – 1.92 and 2.35 m3 h-1

brine temperature upstream and downstream of the evaporator. A drop in the temperature of the brine at the inlet to the evaporator remains constant. The recorded temperature of the brine at the inlet to the evaporator reaches the maximum possible temperature resulting from the temperatures of the ground around the vertical collectors. Identical demand for cooling power in the evaporator for two measurement cycles forced a greater temperature difference of ΔTgs=2.8 °C for the volumetric flow of glycol of 1.92 m3 h^{-1}.

For all three vertical collectors activated the average temperature difference between the inlet to and outlet from the evaporator on the level of ΔTgs=2.1 °C was recorded along with a drop in the glycol temperature at the inlet to the evaporator on the level of 25%. For two collectors activated simultaneously, the values were ΔTgs=2.9 °C and 38% respectively. Such an examination allows evaluating the dynamics of changes of the operating parameters of the ground source if one of the vertical collectors fails.

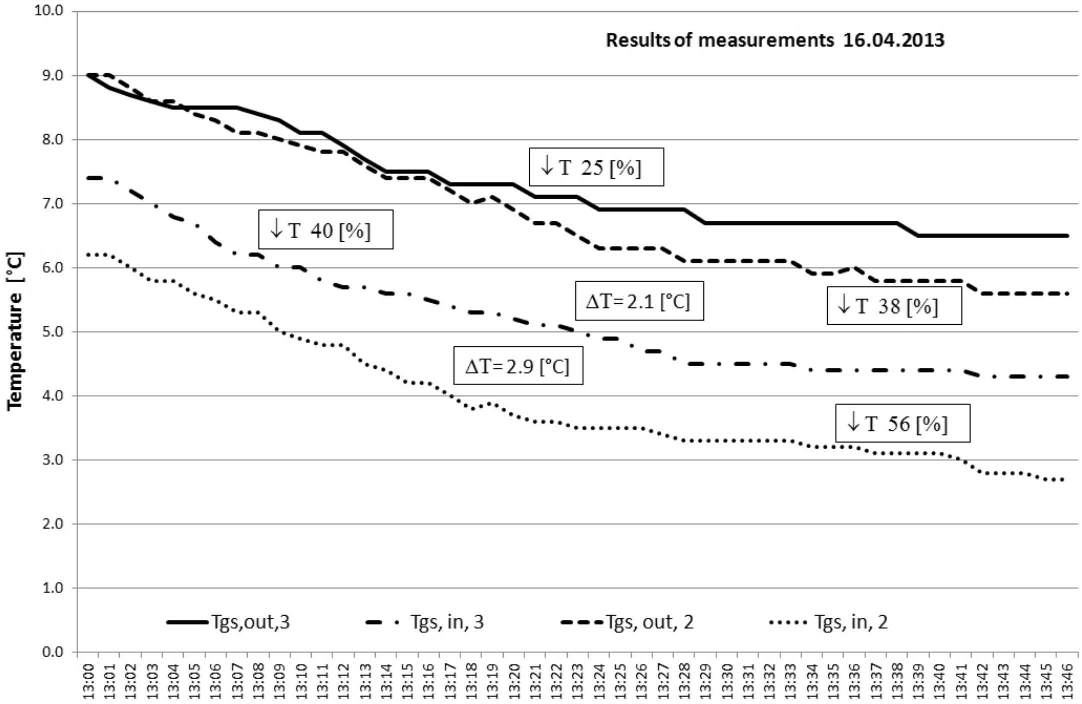

Fig. 4. Changes in the temperature in the ground source circuit if one of the collectors is disabled

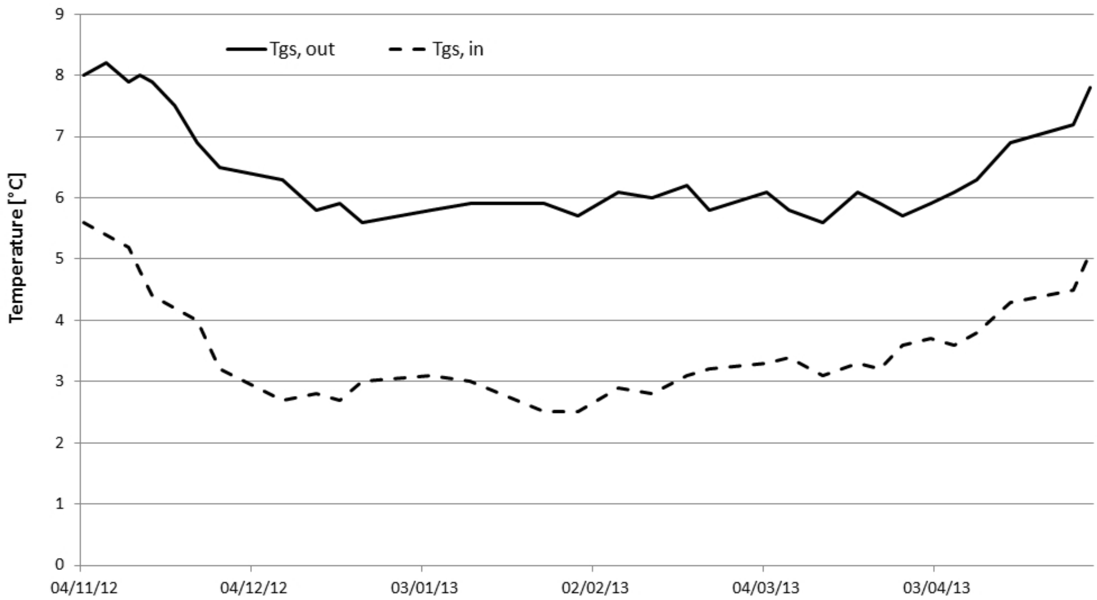

Fig. 5. Changes in the temperature in the glycol circuit in the heating season

In the heating season, an increased demand for heat results in a chilling of the ground around the vertical collectors, which manifests itself by a reduction of the glycol temperature at the inlet to the evaporator. As shown in the trend in Fig. 5, the lowest temperature values were observed from December through March (the greatest demand for heat). Usually, in March, the demand for heat drops drastically, the effect of which is a growth in the average temperatures of the glycol circuit.

Fig. 6 presents the trend in the changes of temperature in the glycol circuit for cycle 2 (operation time 2 h, dwell time 1 h). From the analysis of the presented data, we may conclude that the adopted heat pump dwell time is sufficient for the ground temperature around the vertical collectors to stabilize. Additionally, the figure shows the course of changes of the floor temperature when the heating circuit on the ground floor is on.

CONCLUSIONS

The circuit pumps applied in the experiment consumed on average 35 W (speed 1) and 50 W (speed 2). The total power consumption of the pumps was 3.5 % of that of the compressor.

As the R407C refrigerant evaporation temperature grew, the COP increased but in the case of the tested heat pump,

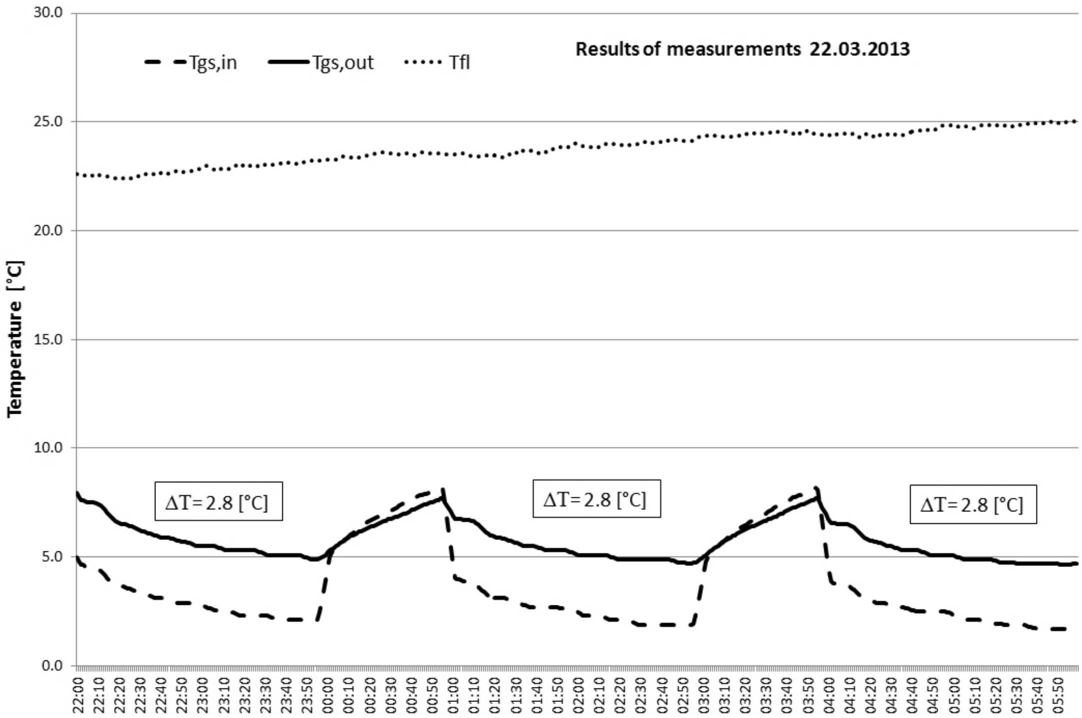

Fig. 6. Changes in the temperature in the glycol circuit for cycle 2

the maximum values of COP were obtained for the refrigerant evaporation temperature in the range from –2.5 °C to 0 °C. The obtained results were heavily influenced by the temperature of glycol in the ground source circuit.

The capacity of the heat pump in the tested period changed in the range from 8.4 kW to 9.2 kW. Higher values were obtained for the case of a simultaneous heating on the first and second floors. Setting higher values of the water mass flow in the space heating circuit forces a reduction of the condensation temperature, which leads to an increase in the COP and heating capacity.

From the presented research results, it follows that the total length of the vertical collectors was sufficient to match the cooling capacity of the heat pump. The application of three vertical collectors allows further operation of the heat pump even if one of the collectors fails.

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BADANIA WYMIENNIKA GRUNTOWEGO POMPY CIEPŁA

Streszczenie. W pracy zostały zaprezentowane wybrane zagadnienia dotyczące badania wymiennika gruntowego pompy ciepła zainstalowanej w domu jednorodzinnym wykonanym w technologii niskoenergetycznej. Przeprowadzone badania eksperymentalne na stanowisku badawczym (domek jednorodzinny z ogrzewaniem podłogowym) pozwolą oszacować wpływ wybranych parametrów eksploatacyjnych wymiennika gruntowego na sprawność energetyczną pompy ciepła.

Słowa kluczowe: gruntowa pompa ciepła, kolektor pionowy, sprężarka spiralna, źródło dolne, współczynnik efektywności energetycznej.