Open Access Journal

Journal of Power Technologies 94 (2) (2014) 1–9

journal homepage:papers.itc.pw.edu.pl

Two-seasonal thermal analyzes of systems with a vapor compressor heat pump and horizontal ground heat exchangers

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Abstract

Selected results of thermal analyzes of heating systems with vapor compressor heat pumps and horizontal ground heat exchangers are presented in this paper. Operation of this system for two lengths of heat exchanger pipes was considered and compared. The thermal analyzes concern two consecutive heating seasons. During heating seasons the heat pump works periodically, supplying heat to a small residential house. Heat is also periodically transferred from the ground to the heat pump unit. The demand for heat for this house, variable in time, was calculated. The heat pump is supported by an additional, conventional heat source when heat pump productivity is insufficient. The characteristic operating parameters of this system (among others: characteristic temperatures, heat fluxes, electric driving power of the compressor) are variable in time. The thermal state of the ground is also variable in time. Natural, thermal regeneration of the ground during spring and summer, when the heat pump system does not operate, was taken into account. The thermal analyzes of the system enable one to compare the electric energy consumption in the system: heat pump— ground heat exchanger and energy transferred from the additional heat source to the heated space during the first and the second heating seasons. The results presented in this paper also make it possible to determine the influence of the length of the horizontal ground heat exchanger pipes on the functioning parameters of this system during two-seasonal operation.

Keywords: Heat pump, Horizontal ground heat exchanger, Heating system, Numerical modeling

1. Introduction

Heating systems with heat pumps are very popular—numbering over 70 million worldwide [1]. In many countries experimental and computational investigations of ground heat exchangers and systems with heat pumps have been carried out [2–7]. In Poland these systems have achieved popularity as well, because they are touted as renewable sources of heat. Heat pump systems can use relatively low temperature heat sources to produce heat.

Atmospheric air, ground, water flow, natural water container or sewage waste water may be low heat sources for a vapor compressor heat pump, which is one of the heat pump types. These heat pumps are more popular than the others, for example absorption heat pumps. In heating systems with vapor compressor heat pumps only electricity is needed. Utilization of low-temperature, natural, renewable heat sources is an advantage of these systems but a disadvantage is

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the high investment cost, especially when the ground is used as a low heat source of a heat pump. In this situation the heat exchanger installed in the ground consists of horizontal or vertical pipes. During the heating season heat is collected from the ground via an intermediate agent flowing inside the pipes of the ground heat exchanger and thermal state of the ground is variable in time. The choice of length of pipes is very important on account of the heat pump productivity and operating parameters of the heating system, for example energy consumption. Since the heat demand for the heated building, which is dependent on environment parameters, varies during the heating season, the heat pump may be supported by an additional, conventional heat source. This is desirable so as to cover times when the demand for heat is higher than the supply from the heat pump. The conventional heat source should have the function of a peaking source.

As mentioned, the thermal analyses presented in this paper concern heating systems with a vapor compressor heat pump and additional heat source. A numerical model of the system: heated object—vapor compressor heat pump—horizontal ground heat exchanger [8–10] allows one to investigate the influence of the length of the ground heat exchanger pipes on the operating parameters of this heating system. Using this model it is possible, among others, to determine the frequency of functioning of the additional heat source, total energy consumption during the heating season and total operation time of heat pump for various lengths of pipes.

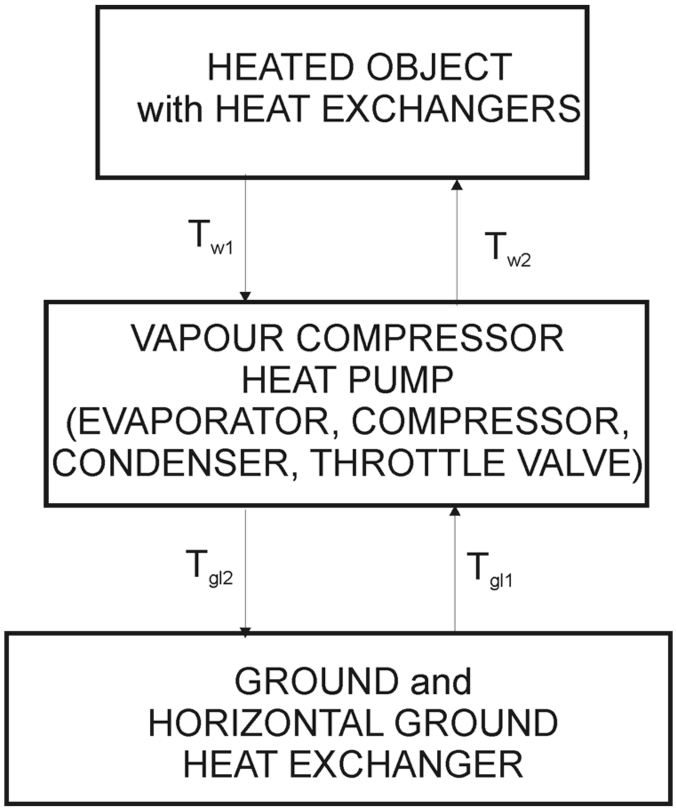

Figure 1: Scheme of the heating system

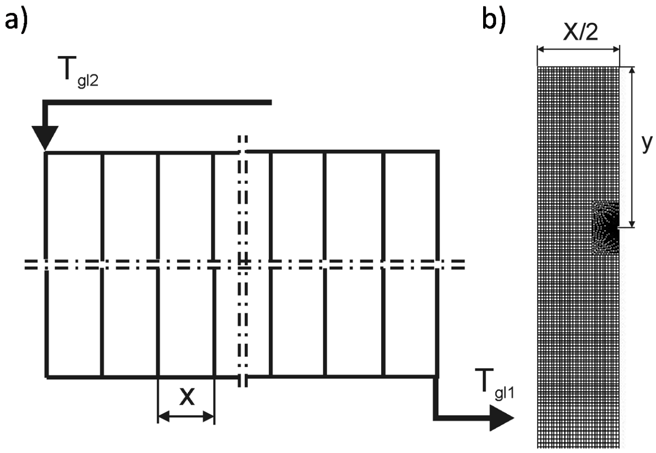

Figure 2: Scheme a) and repetitive element b) of the horizontal GHE

2. Description of the mathematical model of the system: heated object—vapor compressor heat pump—horizontal ground heat exchanger

A schematic diagram of the heating system is presented in Fig. 1. The system consists of three cycles: cycle of heating water, cycle of the refrigerant and cycle of the intermediate agent. The heat exchangers are installed in the house. The water, heated in the condenser of the heat pump, transfers the heat in these exchangers to the air in the heated space. As mentioned earlier, the heating system is also equipped with a supplementary conventional source of heat. It may support the heat pump operation. The working agent in the vapor compressor heat pump—in analyzed cases the refrigerant R134a—performs the Linde cycle, but with irreversible adiabatic compression. The heat pump unit subsystem consists of four elements: the evaporator, the scroll compressor, the throttle valve and the condenser. The heat pump works with constant volumetric capacity of the compressor. The intermediate carrier, e.g., a water solution of glycol, is heated in the pipes of the ground heat exchanger and then transfers the collected heat to the refrigerant in the heat pump evaporator. In analyzed cases the ground heat exchanger consists of a small number of parallel, horizontal pipes (Fig. 2a). The pipes are connected parallel to the evaporator.

In the calculations the heating seasons were divided into three-hour balance periods. The calculations were performed in each time period [9, 10]. Total operation time of the heat pump compressor in the balance period is related to the heat demand of the building. The heat demand, calculated for each period is variable during the heating season and is dependent on environment parameters. In each balance period the compressor works in a continuous way, step by step, until the required demand for heat is fulfilled. During the rest of the period the compressor does not work. If heat pump productivity is insufficient, the heat pump is supported by an additional, conventional heat source. In this case the heat pump provides only part of the heat, working in a constant manner, and the rest of the heat is supplied by the additional source of heat. During compressor operation time the heat is collected from the ground by the intermediate carrier in the ground heat exchanger. It was assumed that during compressor down time the heat is not collected from the ground and the process of partial thermal regeneration in the ground adjacent to the pipes takes place.

The computational model of the heating system consists of the models of three subsystems: the heated object, the heat pump unit and the ground heat exchanger [9, 10]. The numerical model of the heated object allows one to calculate demands for heat in the balance periods during the heating season [9, 10]. Radiation and convective heat transfer between the building and environment and heat transfer from the heated space to the ground under the building were taken into account in the calculations. Variable in time temperature distributions in the wall and in the system building floor—ground were determined using the commercial computer program ANSYS FLUENT. This numerical model also takes into consideration heat losses and gains through the windows, inner heat gains and ventilation heat loss. The air temperature in the heated space may be constant during the analyzed period or may be variable in time. In each time step, the operation of the second subsystem—heat pump elements and the operation of the heat exchangers in the object can be described by a set of nonlinear equations [8–12]. It consists of energy balance equations for steady state (for the above-mentioned elements), heat transfer equations for the heat exchangers and thermal and calorific state equations for the refrigerant. The variability of the heat transfer coefficients in the condenser, the evaporator and the heat exchangers in the heated space, as well as the compressor characteristic were also taken into account [9, 10]. In each time step this equation set was solved. The mass flow rate of the intermediate carrier and its temperature at inflow to the evaporator (at outflow from the ground heat exchanger) are common parameters for the heat pump unit and the ground heat exchanger. In the analyses, the model of subsystem ground–ground heat exchanger takes a two-dimensional form [8–11]. It was assumed that during compressor working time the average unit heat flux transferred from the ground is the same as the unit heat flux calculated for the average temperature of the intermediate agent. The part of the vertical cross section as the calculation domain is presented in Fig. 2b and the pipes are located at depth y. On the upper surface of the ground, convective and radiation heat exchange with the environment takes place [9, 10]. The numerical model of the subsystem ground heat exchanger—ground allows one to calculate in each time step the unit heat flux collected in the ground heat exchanger and temperature field in the ground using the computer program ANSYS FLUENT. The temperature of the intermediate carrier at the outflow from the ground heat exchanger is calculated on the basis of the unit heat flux, length of the pipes, temperature at the inflow to the pipes, mass flow rate and thermal capacity of this medium. The volumetric flow rate is variable and is also calculated for the performance characteristic of the intermediate agent pump and the hydraulic characteristic of the flow system of this agent [9, 10]. During operation of the heat pump compressor, when the heat is taken from the ground, the overall heat transfer coefficient between the external surface of the pipe and the intermediate carrier and the average temperature of this agent are the boundary conditions. These values are calculated in each time step. During compressor down time this coefficient is equal to zero. The calculation process for the ground heat exchanger as well as for the heat pump subsystem, in each time step takes an iterative form. The main results of the calculations of the heating system, depending on time, are following: the heat fluxes transferred in the condenser and evaporator, the temperature of the intermediate medium at the outlet from evaporator (at the inlet to the ground heat exchanger pipes) and the evaporation, condensation and heating water temperatures, the driving power of the compressor, the mass flow rate of the intermediate agent and the temperature field in the ground.

3. The results of calculations

The calculations of the system were performed for two variants of the horizontal ground heat exchanger. In both variants the exchanger consists of 10 parallel pipes located at a depth of $y = 1.5$ m and at a distance of $x = 0.75$ m. The outer diameter of the pipes is 0.04 m, the thickness of the walls is 0.0023 m. In variant I the length of a single pipe is 30 m and in variant II 20 m. As mentioned earlier, the calculation domain of the subsystem ground heat exchanger—neighboring ground takes the form presented in Fig. 2b. In calculations the depth of the analyzed part of the ground is 20 m and the ground consists of two parts: upper and lower one. The sample, demonstration calculations were performed for the following data [13]:

- upper part (0...10 m): (unfrozen ground/frozen ground) product of density and specific heat capacity:3737 kJ/(m^{3}K)/2607 kJ/(m^{3}K), thermal conductivity: 1.5 W/(m K)/1.75 W/(m K), latent heat of moist ground 80 kJ/kg

- lower part (10...100 m) (unfrozen ground/frozen ground) product of density and specific heat capacity: 3217 kJ/(m^{3}K)/2327 kJ/(m^{3}K), thermal conductivity: 2.2 W/(m K)/1.2 W/(m K), latent heat of moist ground 60 kJ/kg.

The condenser has the following geometric parameters: number of pipes: 42, length of pipes: 1 m, outside/inside pipe diameter: 0.012 m/0.010 m. The same evaporator parameters are: number of pipes: 42, length of pipes: 0.7 m, outside/inside pipe diameter: 0.012 m/0.010 m. The value of the product of the heat transfer coefficient and the area of the heat exchangers in the heated space was calculated on the basis of their thermal characteristic as a function of calculated heating water temperatures. It was also assumed that the mass flow rate of the heating water is 0.8 kg/s. The electric driving power of the scroll compressor in each time step is determined on the basis of its performance characteristic; the value of its volumetric capacity is constant at 0.00475 m^3/s. The heating season was divided, as mentioned earlier, into three-hour balance periods. Changes in heat demand for the residential building in these periods were calculated using the mentioned computer code [9, 10] for time step 450 s. In analyzed variants the area of the building is approximately 180 m^2. The changeability of environment parameters was assumed for southern Poland. Calculations were performed assuming constant air temperature in the heated space (22°C), constant convection heat transfer coefficients on the outer and inner surfaces and constant wall emissivity and absorptivity [9, 10]. In Fig. 3 three-hour demands for heat in the heating season are presented.
Calculations for the system: heated object—vapor compressor heat pump—horizontal ground heat exchanger were performed for two values of time steps: 225 s from the start of the heating season to the 30th day and 450 s during the following days to the end of this season.

The presented analyses concern two consecutive heating seasons. The heating seasons were assumed to start in mid-September. Natural, thermal regeneration of the ground adjoining the pipes of the ground heat exchanger was taken into account. This process was assumed to start in mid-May. The initial temperature distribution in the computational domain for the first heating season is the result of heat transfer calculations for the ground during the period of two years prior to the start of the ground heat exchanger operation. It is self-evident that the initial temperatures for the second season are connected with the

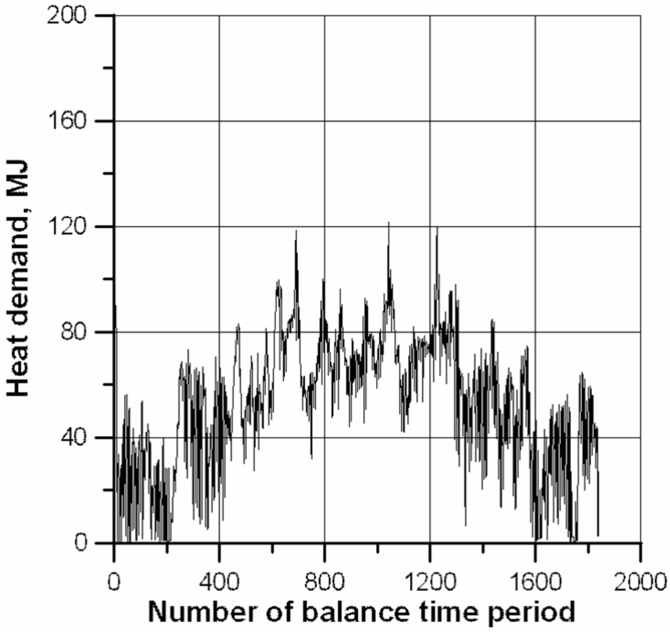

Figure 3: Demand for heat during the heating season

system operation during the first heating season and the thermal regeneration of the ground during spring and summer.

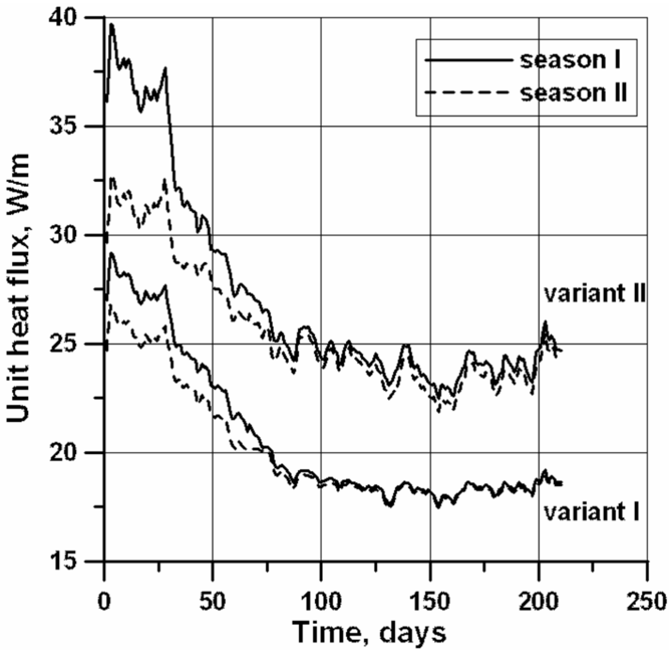

Figure 4: Seasonal changes of daily average unit heat flux transferred from the ground to the GHE

Selected results of the calculations are presented in Figs. 4...13. The presented daily average values of characteristic parameters were calculated as average values during the daily operation time. As the earlier analyzes demonstrated [8–11], unit heat fluxes collected from the ground during first heating season are variable in time and are dependent on the length of the ground heat exchanger pipes.

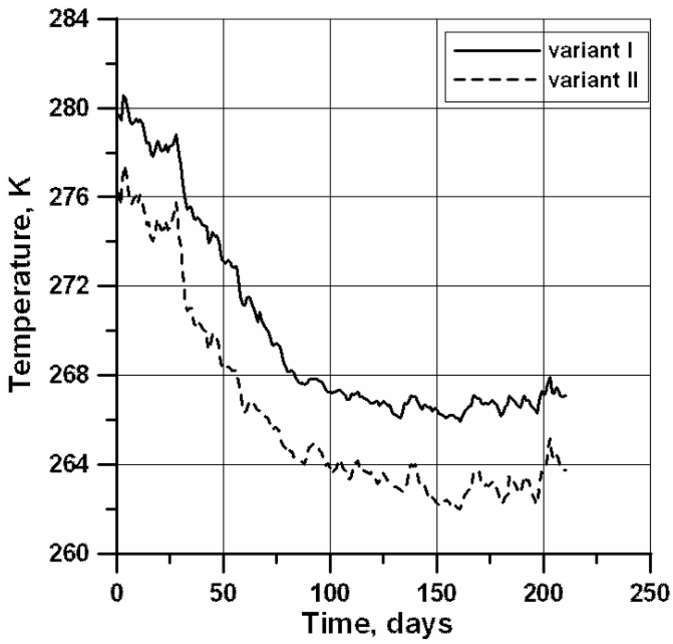

Figure 5: Seasonal changes in daily average temperature of the intermediate carrier in the GHE—I heating season

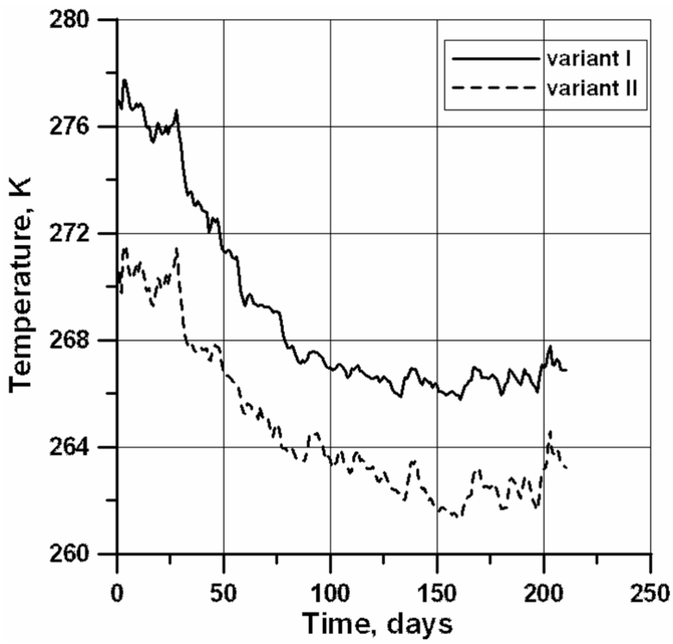

Figure 6: Seasonal changes in daily average temperature of the intermediate carrier in the GHE—II heating season

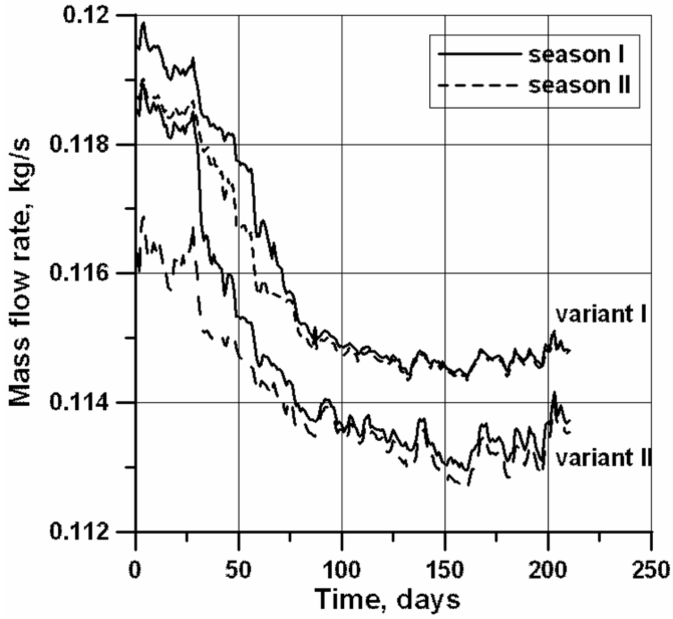

Figure 7: Seasonal changes in daily average mass flow rate of the intermediate medium in the GHE

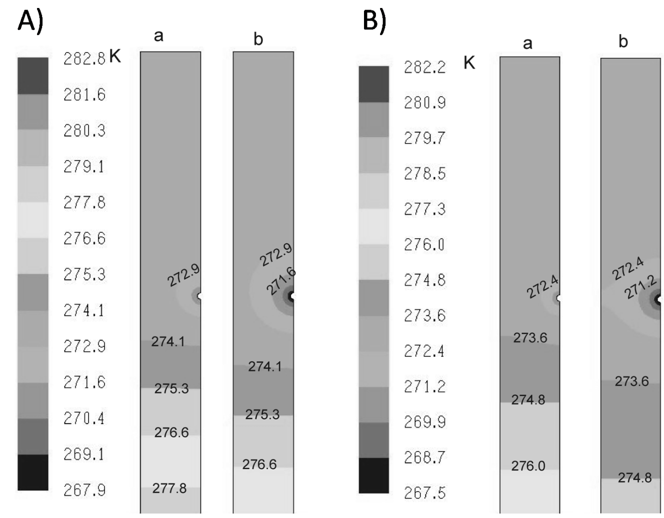

Figure 8: Temperature distributions in the ground—120 days of the heating season: A) first season, B) second season, a) variant I, b) variant II

Visible differences between these unit heat fluxes calculated for assumed lengths of the pipes (variant I and II), for seasons I and II (Fig. 4) relate to different temperatures and mass flow rates of the intermediate agent (Fig. 5, 6, 7) as well as different temperature distributions in the ground (Fig. 8) in the analyzed cases. In the initial operation time periods heat fluxes taken from the ground in season II are smaller than in season I for both variants, as well as the temperature of the intermediate medium (Fig. 5, 6). These fluxes are connected with only partial thermal regeneration of the ground during the spring-summer period, when the heat pump unit does not operate. The differences between the mentioned heat fluxes are more visible for shorter pipes. Higher unit heat fluxes taken from the ground for variant II compared to variant I result from lower temperatures of the intermediate medium in the ground heat exchanger in variant II. Changes in the intermediate medium temperature mean that the mass flow rates of this medium also vary during the heating season (Fig. 7), which in turn affects other operating parameters. The presented results of calculations show the influence of length of ground heat exchanger pipes on changes in the temperature distributions in the ground during the two heating seasons and the same on characteristic parameters of system functioning.

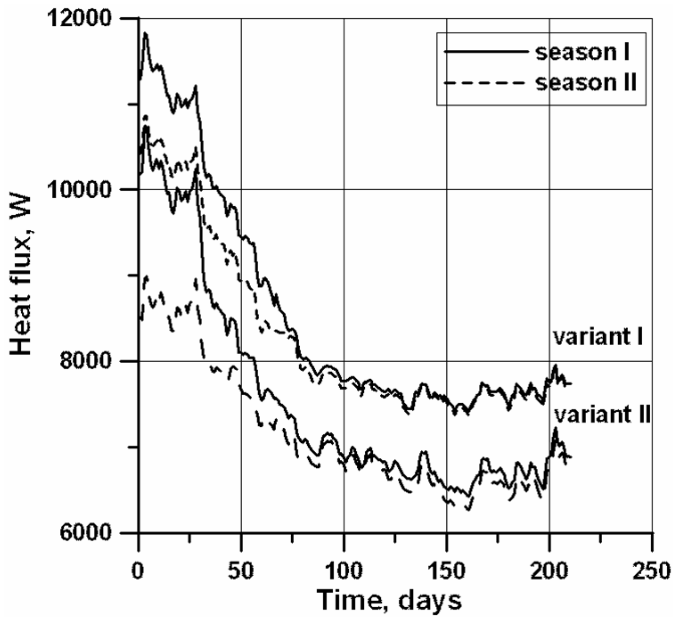

Figure 9: Seasonal changes in the daily average heat flux transferred from the heat pump to the heated space

In Fig. 8 sample temperature distributions are presented. Visible differences in the temperature field for variant I and II cause the system to operate in a different way during consecutive heating seasons. Although unit heat fluxes collected from the ground are lower for variant I, total heat fluxes are bigger compared to variant II. In this way heat fluxes transferred in the condenser are bigger for longer pipes of the ground heat exchanger. Heat pump productivity

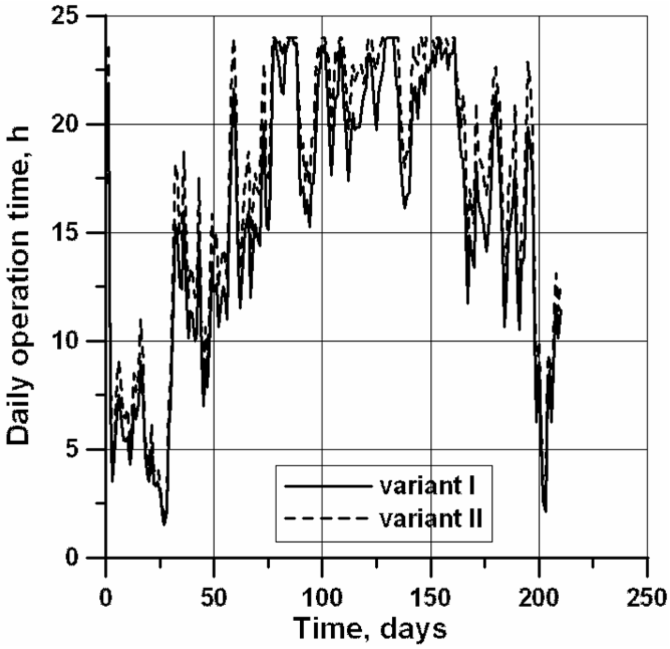

Figure 10: Daily operation time of the VCHP compressor—II heating season

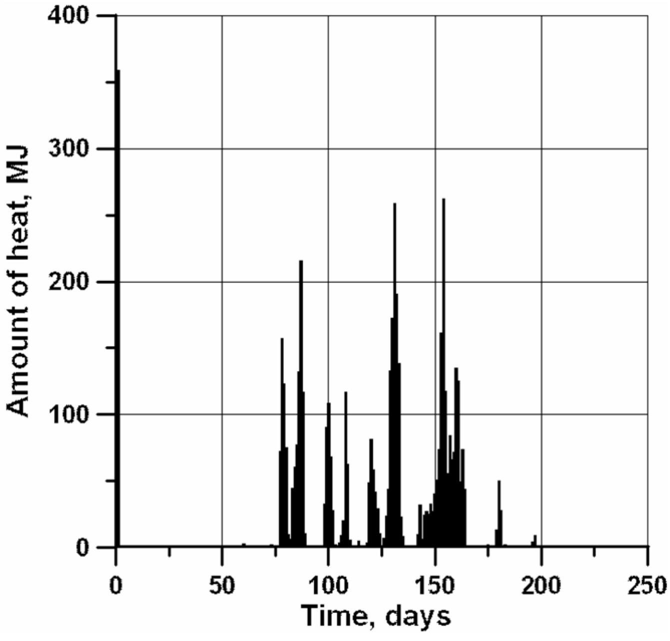

Figure 11: Seasonal changes in the daily amount of the heat transferred from the additional heat source to the heated space (variant II)—I heating season

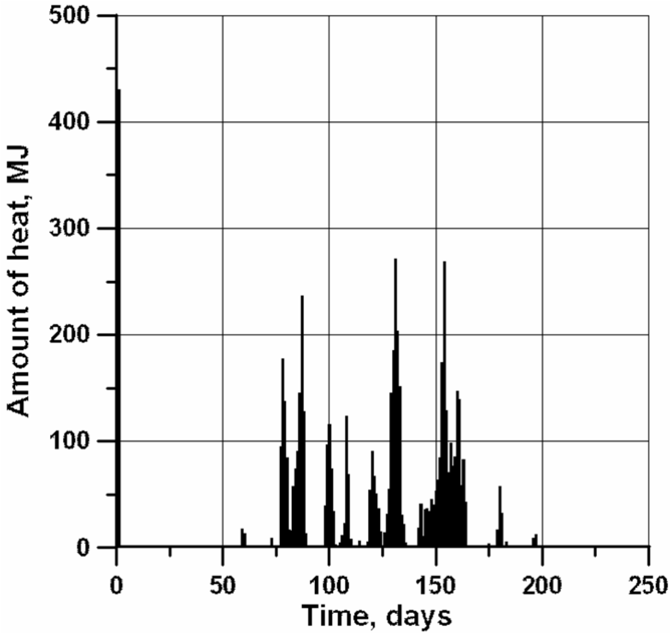

Figure 12: Seasonal changes in the daily amount of the heat transferred from the additional heat source to the heated space (variant II)—II heating season

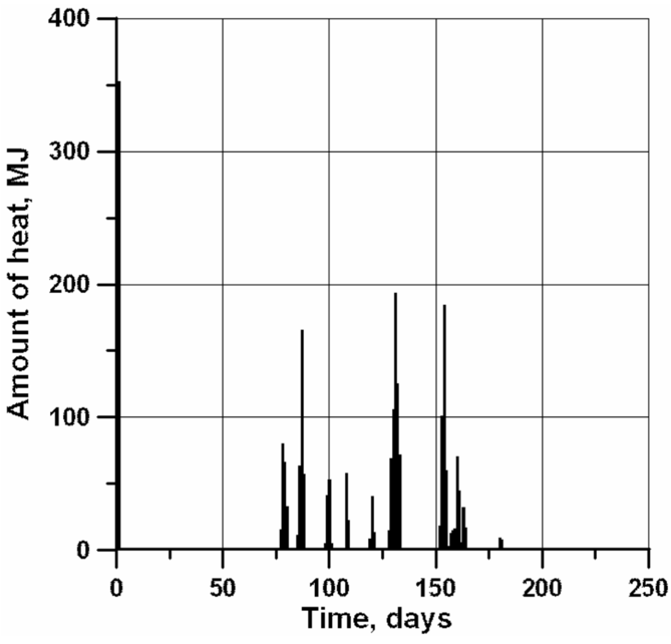

Figure 13: Seasonal changes in the daily amount of the heat transferred from the additional heat source to the heated space (variant I)—II heating season

decreases during the heating season and its values are lower in the second season (II) than the first season (I), especially in variant II (Fig. 9). Due to this fact the total compressor working time is the longest for shorter pipes, during the second season (Table 1). It is visible that in the second heating season for this case the compressor was working 24 hours a day more frequently than for variant I (Fig. 10). Accordingly, the additional heat source was used in the second season more often for variant II (Fig. 12) and

Table 1: Energy consumption and operation time of the heating system during heating seasons

	Variant II, season I	Variant II, season II	Variant I, season I	Variant I, season II
Electric energy, MJ	34658	34968	33708	33957
Energy from additional heat source, MJ	5011	5776	2094	2370
Compressor working time, h	3753	3849	3435	3487

it provided the biggest volume of consumed energy (Table 1). The amount of energy transferred from the conventional heat source to the heated space and the operational frequency of this heat source are lower in the first heating season (Fig. 11) and in the case of longer pipes (Fig. 13).

4. Discussion, conclusions

The presented results of thermal analyzes as well as analyzes presented earlier, e.g., in [8–12] clearly show that there are mutual connections between operating parameters of the heating system with the heat pump unit, demand for heat and the thermal state of the ground. The numerical results presented in this paper demonstrate the influence of the lengths of ground exchanger pipes on the operating parameters, the thermal state of the ground and electric energy consumption during the long term operation of the system.

The more general aim of this paper is to present computation possibilities of a complex model consisting of three described subsystems: the heated object, the heat pump unit and the ground with the pipes of the ground heat exchanger. It should be stressed that the presented model can be used for multi-variant comparative calculations. This model allows one to investigate the influence of, among others, parameters of the ground heat exchanger, heat exchangers in a heated space, compressor, evaporator and condenser, as well as various kinds of intermediate agents on operating parameters of the system during long-term functioning. The presented results take the form of examples. The final result of complex analyzes ought to take the form of thermal-economic optimization. Such optimization definitely requires a much larger number of calculations than shown in this paper and its aim is to select the most suitable values of characteristic parameters, first of all the summary length of ground exchanger pipes [10]. Choosing a greater length increases the investment costs, while a shorter length increases the cost of electricity for the additional heat source. Evidently, some other optimal characteristic parameters, e.g., the productivity of the compressor, are connected with length of ground heat exchanger pipes.

Acknowledgments

This work was sponsored by the Ministry of Science and Higher Education from science financial resources 2010...2012 under research project NN 512 317238

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