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Comparative analyses of systems with vapour compressor heat pumps and ground heat exchangers

This paper presents selected results of thermal analyses of heating system with vapour compressor heat pumps. The operation of this system was considered and compared for various numbers of horizontal ground heat exchanger pipes during the heating season. The main results of the variant thermal calculations are, among others, variability of operation times, heat fluxes and daily amounts of heat taken from the ground and transferred to the heated space, temperatures of working mediums and mass flow rate of the intermediate agent in the ground heat exchangers, variability of daily amounts of electric driving energy of the compressor as well as total energy consumption during the heating season. The results of these analyses were used in economic calculations connected with the ground heat exchanger. The examples of calculations concerning the optimization number of pipes and an influence of unit installation cost of the horizontal ground heat exchanger on the discounted cost are presented.

1 Introduction

Heating systems with vapour compressor heat pump are more and more popular in the world and in Poland [6,7,10]. The number of heat pumps is bigger than 70 million [7]. In many cases ground is a lower heat source for heat pumps. Heat is collected from the ground by an intermediate agent in horizontal or vertical ground heat exchangers. In practice, in heating systems, additional heat sources are installed. When the heat pump productivity is not big enough, an additional source of heat supports the heat pump. Thermal state of the ground is variable during the heating system operation. Preliminary, variant simulations only for ground heat exchangers proved that geometrical parameters of these exchangers and temperature of an intermediate agent influence the thermal processes in the ground adjoining to the tubes [1,9]. The results of analyses of the heating season

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with the horizontal or vertical ground heat exchangers presented in [2–5] show mutual connections between the heat pumps operation and the thermal states of the ground. Characteristic parameters of the system are variable during the heating season and the thermal processes in the ground have an impact on processes inside the heat pump unit. Configurations of the ground heat exchangers have an influence on these parameters, on number of the heat pump and additional heat source functioning, on total energy consumption in the heating system and on costs. In this paper, the results of thermal and economic analyses for various number of horizontal ground heat exchanger pipes are compared.

2 Operation of the system: heated object – vapour compressor heat pump – horizontal ground heat exchanger

The considered heating system is presented in Fig. 1. It was assumed that this system consists of three cycles: the cycle of the intermediate agent, cycle of the heat pump with a refrigerant and the cycle of water in heated object. As mentioned, ground is a lower heat source for the heat pump. Heat is transferred from the ground to the heat pump evaporator by the intermediate medium (water solution of glycol), which flows inside the horizontal pipes of the ground heat exchanger (Fig. 1). This exchanger is located in upper layer of the ground and its pipes are connected parallel with the evaporator. In the heat pump cycle the refrigerant R134a is the working medium. It was assumed that the heat pump unit has a typical form – it consists of a compressor, the condenser, the throttle valve and the evaporator. The third cycle comprises the condenser, heat exchangers in the heated object, the water pump and the additional conventional heat source. The heating water is heated in the condenser and then it transfers the heat collected in the heat exchangers to air in the heated space.

As the earlier analyses proved, the processes taking place in this system are of an unsteady character [2–5]. Such parameters as heat flux collected from the ground, heat fluxes transferred from the heat pump and the additional heat source to the heated space, characteristic medium temperatures in the cycles and mass flow rate of the intermediate agent, as well as temperature field in the ground are variable during the heating season. The operation of this system during one heating season for several variants of ground heat exchanger configuration (number of pipes) has been performed and the results have been compared. The mathematical model of the heating system presented in [2,3] is based on the assumptions concerning the functioning of system elements.

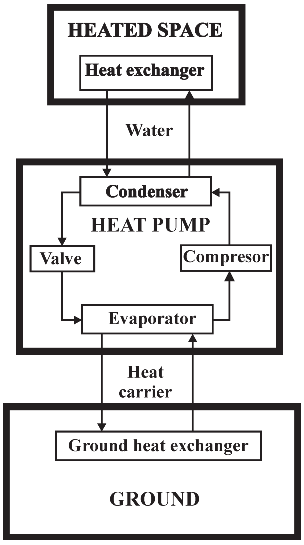

Figure 1. Scheme of the heating system with vapour compressor heat pump and ground heat exchanger.

3 Description of the mathematical model and the method of calculation

The heating season is divided into several hours of balance periods. The heat demand of the building in each period is known and it depends on variable environment parameters. It is the main result of calculations for the heated building. The heat pump system works in three ways. If the heat pump productivity for the balance period is greater than the periodical heat demand, the heat pump functions in continuous way until the required heat demand is fulfilled [2–4]. During the remaining part of this balance period the heat pump does not operate, the heat is not collected from the ground and partial thermal regeneration of the ground adjoining the pipes of the heat exchanger takes place. If the heat capacity of the heat pump is not big enough, the heat pump compressor operates in a constant way during all balance period [2–4]. Part of heat is provided by the heat pump and the remaining part of required heat is transferred to the heated space from the additional heat source. The range of work parameters of the compressor is determined. In case when condensation or evaporation temperatures are outside of this range, the compressor does not work and only the additional heat source delivers heat to the heated space.

Mathematical model of the heating system contains models of three subsystems: heated object, vapour compressor heat pump and horizontal ground heat exchanger [2,3]. Variability of heat demand during the heating season is calculated on the basis of the heated object model [2,3]. The momentary heat demand, calculated on the basis of the energy balance for the building in each elementary time step is dependent on variable during the heating season parameters (temperatures and solar radiation heat fluxes). The model takes into consideration heat losses and gains through the windows, inner heat gains and ventilation heat waste, radiation and convective heat transfer between outer wall surfaces and environment, as well as heat transfer from the heated space to the ground under the building. In the heated space the convective heat transfer between air and the inner wall surfaces was assumed. In the numerical procedure the commercial computer program ANSYS Fluent is used for calculations of the heat fluxes transferred from air in the heated space to the inner surfaces of the building as well as the temperature fields in the walls, roof and in the system: floor – ground. The values of heat demand for several hours periods during the heating season are the final results of the calculations.

As the changes of characteristic parameters in the system are very slow, it was assumed that in the elementary time step the operation of the heat pump elements are described by energy balance equations for the steady state, and the evaporator and condenser additionally by Peclet number equations [2–5]. Functioning of the heat exchangers in the heating system is described by analogous equations. The value of the product of heat transfer area and heat transfer coefficient is determined on the basis of the technical data for these heat exchangers as a function of heating water temperatures. In the mathematical model volumetric capacity of the scroll compressor is constant, and its electric driving power and internal efficiency are calculated for condensation and evaporation temperatures [2–4]. Equations for calculations of heat transfer coefficients in the evaporator and the condenser, and state equations for the refrigerant (R134a) constitute the supplement of this model [3]. The main results of iterative calculations, in each time step, are following: heat flux transferred in the condenser and evaporator, temperature of the intermediate medium at the outlet of the evaporator (at the inlet to the ground heat exchanger pipes) and evaporation, condensation and heating water temperatures.

The numerical model of the ground heat exchanger [2,3] uses the computer software ANSYS Fluent to determine the unsteady temperature field in the ground and the unit heat flux transferred from the ground to the evaporator. The calculations have been performed in a two-dimensional system. A part of the calculation domain is presented in Fig. 2B. The convective and radiation heat exchange with environment takes place on the upper surface of the ground. As mentioned, the environment parameters are changeable during the heating season. The assumption of the two-dimensional model entails on additional assumption: in each time step, when the heat pump operates, the average, unit heat flux transferred from the ground is the same as the unit heat flux calculated for the average temperature of this agent. This temperature is calculated in an iterative way on the basis of the unit heat flux, pipes length, temperature at the inflow to the pipes, mass flow rate and thermal capacity of this medium. Thermal parameters of the intermediate agent are dependent on its average temperature. Boundary conditions on the outer surface of the pipe are the average temperature of intermediate agent and the substitute heat transfer coefficient. It was assumed that in compressor operation time, the value of this coefficient is calculated. When the compressor does not work, this coefficient is equal to zero. The values of volumetric flow rate of the intermediate agent are also variable and they are connected with average temperatures of this medium. In each time step a new value of volumetric flow rate is calculated on the basis of the intermediate medium pump performance and the hydraulic characteristic of the flow system of this agent. In the model the change phase of moisture in the ground is taken into account. The calculation process for the ground heat exchanger in each time step has an iterative form because of the assumptions and connections between average temperature of the intermediate medium, its mass flow rate, thermal state of the ground and heat flux collected from the ground [2,3].

Temperatures at the outflow of the ground heat exchanger (evaporator inflow) and at inflow to the ground heat exchanger (evaporator outflow) influence cooperation of the heat pump unit with the exchangers in the heated space and functioning of the subsystem: ground-ground heat exchanger. It is the reason for the iterative calculation procedure [2,3].

In the paper the selected results for various numbers of the horizontal ground heat exchanger pipes are presented. In each case the manner of system operation and technical parameters of particular elements are analogous. The variant analysis presented have a comparative character.

The results of thermal calculations for the heating system are the total energy consumption by the compressor and the pump of the intermediate agent and the total energy transferred from the additional heat source to the heated space during the heating season. These values are used in economic analysis performed to determine the number of the ground heat exchanger pipes. This analysis allows to determine the influence of the cost of ground heat exchanger installation per

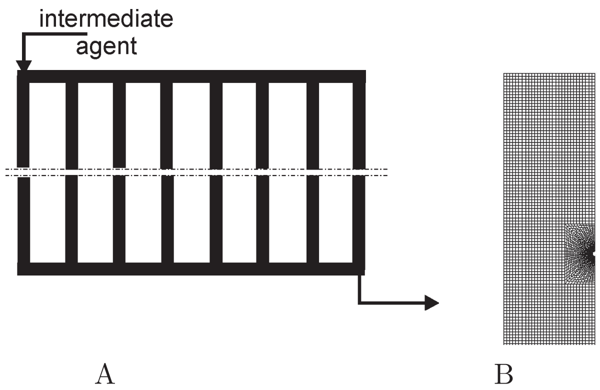

Figure 2. The horizontal ground heat exchanger A) scheme, and the computational domain B).

unit and number of pipes on the cost [3]:

$$K = K_{ghe} + \sum_{t=1}^{n_t} \frac{E_{el\,t}}{(1+r)^{n_t}} \ , \tag{1}$$

where K_{ghe} is the investment cost of ground heat exchanger, $E_{el\,t}$ is the cost of electrical energy, and r is the rate of discount, n_t – the number of years (time of the system operation). Cost of electrical energy is calculated on the basis of the total seasonal energy consumption and its unit cost [3]

$$E_{el\,t} = (E_{el\,hp} + E_{el\,ghe} + E_{el\,wp} + E_{el\,hs})\, e_{el} \ , \tag{2}$$

where: $E_{el\,hp}$, $E_{el\,ghe}, E_{el\,wp}$, are respectively the driving electrical energy of the compressor, of the pump, of the intermediate agent and of the heating water pump, $E_{el\,hs}$ is electrical energy used by additional electric heat source [kWh]; and e_{el} is the unit cost of electrical energy [PLN/kWh]. Installation cost of a ground heat exchanger is described by the following equation:

$$K_{ghe} = k_{jF} F \ , \tag{3}$$

where k_{jF} is the unit investment cost of a horizontal ground heat exchanger [PLN/m^2], and F is the ground area occupied by the heat exchanger [m^2],

$$F = L_{ghe} n_{ghe} x \ , \tag{4}$$

where L_{ghe} – is the length of a single pipe, n_{ghe} is the number of the pipes, and x is the distance between the pipes, [m].

Minimal cost calculated on the basis of Eqs. (1)–(4) is a factor that decides on the number of pipes assuming the same operation of the heating system and technical parameters of system elements. If the rate of discount and the period of system operation are assumed, unit installation cost of a ground heat exchanger and unit cost of electric energy have influence on the discounted cost, Eq. (1).

4 Calculation results

The calculations have been performed for four variants of a number of pipes: 8, 10, 15, 20. Other geometrical parameters of the ground heat exchanger are the following: length of a single pipe – 30 m, depth of location – 1.5 m, distance between the pipes – 0.75 m. The outer and inner diameter of the pipes are equal to 0.04 m and 0.0354 m. Convection heat transfer coefficient on the outer ground surface is equal to 16 W/(m^{2}K) and environment parameters (temperatures and solar, radiation heat fluxes) are characteristic for Silesia [3]. The initial temperature distribution in the ground is the result of heat transfer calculations for the ground without the heat exchanger during the period of two years prior to the start of the heating system operation. In the analysed variants the constant temperature equal to 9 °C was assumed on the bottom surface of the computational domain (20 m in depth). Thermal parameters of the ground are [7]:

- upper part (0–10 m): unfrozen ground – product of density and specific heat capacity 3740 kJ/(m^{3}K), thermal conductivity 1.5 W/(mK), frozen ground – product of density and specific heat capacity 2600 kJ/(m^{3}K), thermal conductivity 1.75 W/(mK), latent heat 80 kJ/kg;
- lower part (10–100 m): unfrozen ground – product of density and specific heat capacity 3220 kJ/(m^{3}K), thermal conductivity 1.2 W/(mK), frozen ground – product of density and specific heat capacity 2330 kJ/(m^{3}K), thermal conductivity 2.2 W/(mK), latent heat 55 kJ/kg.

The main geometric parameters of the condenser are: number of the pipes – 42, length of the pipe – 1 m, outside/inside pipes diameter – 0.012 m/0.010 m. The same parameters for the evaporator are: number of the pipes – 42, length of the pipe – 0.7 m, outside/inside pipes diameter – 0.012 m/0.010 m. It also has segment walls. Water mass flow rate is equals 0.8 kg/s, and the value of the compressor volumetric capacity is equal to 0.00475 m^3/s. The calculations have been made assuming six-hour balance periods and the elementary time step of 450 s. The beginning of the heating season was assumed for 210 days from mid-September. The presented daily average values of characteristic parameters were calculated as average values during the daily operation time. The heat demands for the balance periods were calculated for modern residential building, assuming constant air temperature in the heated space (22 °C), convection heat transfer coefficients on the outer and inner surfaces, wall emissivity and absorptivity, and changeability of environment parameters [3]. In analyzed variants the area of the building is equals approximately 180 m^2. Six-hour demands for heat in the heating season are presented in Fig. 3. The selected results of thermal

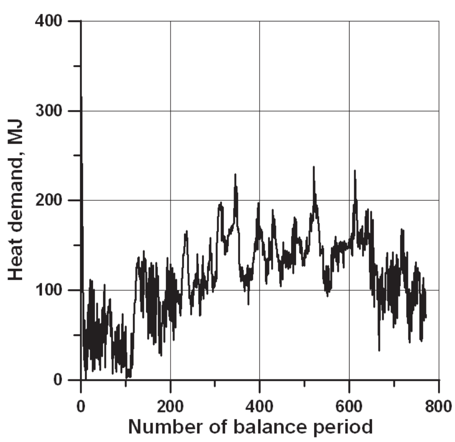

Figure 3. Demand for heat in six-hour periods during the heating season.

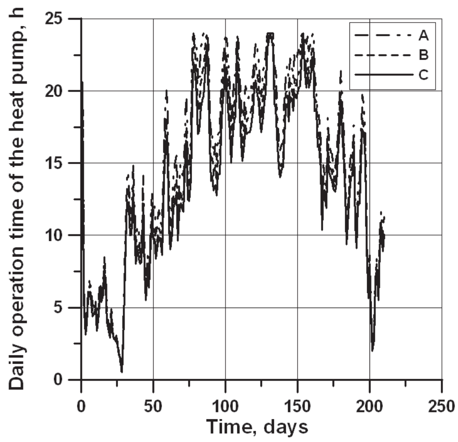

Figure 4. Daily operation time of the heat pump during the heating season: A) 10, B) 15, C) 20 pipes.

analyses are presented in Figs. 4–10. As mentioned before, heat is provided to the heated space by the heat pump or by the heat pump supported by the additional heat source. As the earlier analyses also proved [2–4], operation time of the heat pump (Fig. 4) and frequency of the additional heat source functioning are dependent on the heat demand and on variable heating capacity of the heat pump (Fig. 5). These analyses also showed that the mutual connections take place between characteristic parameters of the system: heated object – heat pump unit – ground heat exchanger [2–4]. In the considered variants thermal state of the ground, as well as values of unit and total heat fluxes collected by the intermediate agent from the ground (Figs. 6 and 5), and hence also the heat fluxes transferred from the heat pump condenser to the heated space are strictly connected with the number of pipes. In the analysed variants, mass flow rates of the intermediate agent in the single pipe (Fig. 7), assuming the same pump of this agent in each case, are different. They and the average intermediate agent temperatures (Fig. 8) have influence on thermal states of the ground and

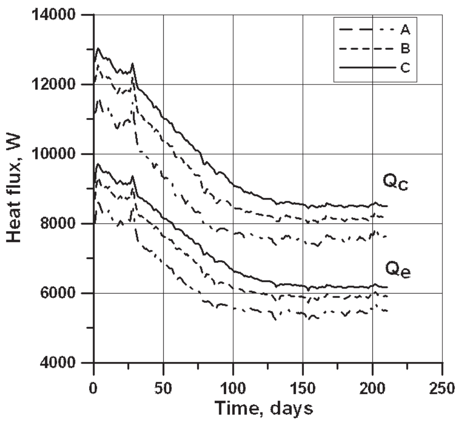

Figure 5. Daily average heat flux transferred from the heat pump to the heated space Q_c and collected from the ground Q_e: A) 10, B) 15, C) 20 pipes.

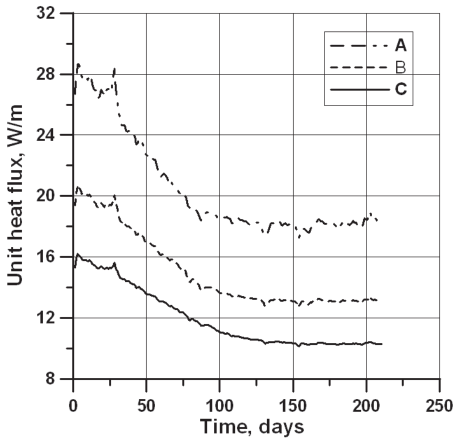

Figure 6. Daily average unit heat flux collected from the ground: A) 10, B) 15, C) 20 pipes.

the unit heat fluxes collected in the ground heat exchangers. Although during the heating season these unit heat fluxes are lower for 20 pipes (C) than for 10 pipes (A), it is obvious that the total heat flux collected in the ground heat exchanger is higher for variant C (Fig. 5). The characteristic operation parameters are connected with cooperation of the heat pump and the subsystem: ground heat exchanger – ground and depend on the demand for heat. For the 10 pipes decrease of such parameters like the heat fluxes and the temperature of the intermediate medium during the heating season is more visible comparing with other variants. The evaporation temperatures change according to the changes of the intermediate agent temperatures (Fig. 9). The character of changes of the condensation temperature is analogous and its momentary increases are connected with the additional heat source. Examples of daily amounts of heat transferred from the additional heat source to the heated space and daily driving electric energy of the compressor in the variant B (15 pipes) are presented in Fig. 10. In the considered cases, assuming analogous manner of the heating system operation, the additional heat source provided approximately 620 MJ – variant C, 980 MJ – variant B, 2050 MJ – variant A. Total values of electric energy consumed by the compressor and the pump of intermediate agent and total time of the heat pump operation were equal to respectively 30170 MJ, 30790 MJ and 31400 MJ and 2815 h, 2992 h, 3225 h. Analogous values for 8 pipes were equal to 3180 MJ,

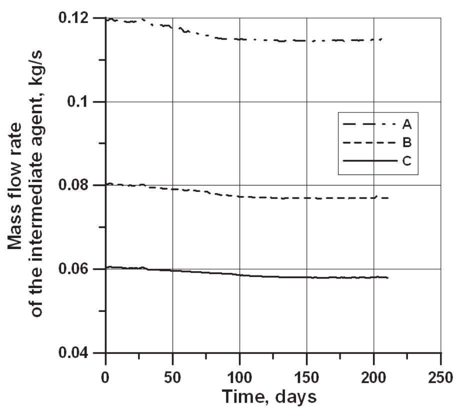

Figure 7. Daily average mass flow rate of the intermediate agent (single pipe): A) 10, B) 15, C) 20 pipes.

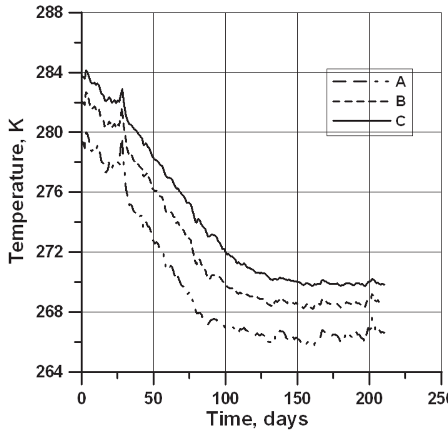

Figure 8. Daily average temperature of the intermediate medium (average value in the pipe): A) 10, B) 15, C) 20 pipes.

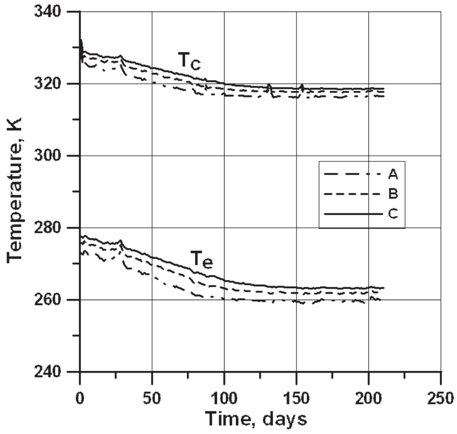

Figure 9. Daily average condensation and evaporation temperatures: A) 10, B) 15, C) 20 pipes.

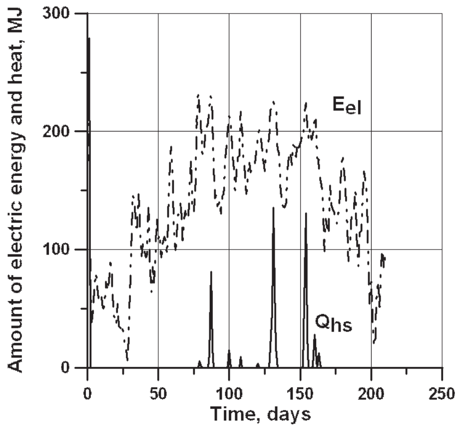

Figure 10. Daily driving electric energy of compressor E_{el} and daily amount of heat transferred from the additional heat source to the heated space Q_{hs} (15 pipes).

31900 MJ and 3360 h. Longer operation time has influence on thermal state of the ground and heat capacity of the heat pump and causes increase of energy consumption by the additional heat source.

As mentioned before, the results of comparative thermal analyses were used in economic calculations. In Fig. 11 discount costs are presented as a function

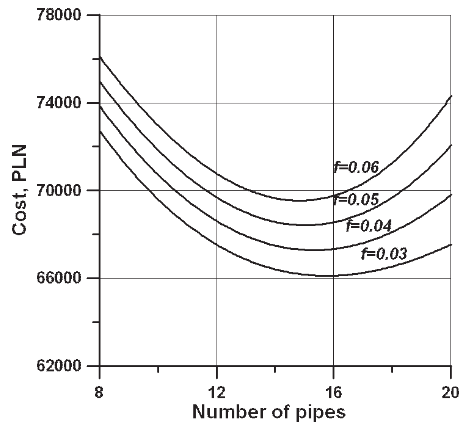

Figure 11. Discount costs.

of the number of ground heat exchanger pipes and the ratio f of the unit investment cost of the horizontal ground heat exchanger, k_{jF}, to the unit cost of electric energy, e_{el}, ($f = k_{jF}/e_{el}$). Variant calculations were performed assuming that $e_{el} = 500$ PLN/MWh and $f = 0.03$, 0.04, 0.05, 0.06, with rate of discount $r = 0.06$, and with $n_t = 30$ years and that efficiency of the electric additional heat source is equal to 1. Electric energy consumption by the heating water pump was determined on the basis of calculated operation times of the heating system with the assumption that average electric driving power of this pump is equal to 100 W. The results of variant analyses presented in Fig. 11 show that total discount cost, concerning investment cost of the ground heat exchanger and the cost of electric energy, strongly depend on the number of pipes and relationship between the unit cost of electric energy and the unit cost of ground heat exchanger installation. Minimal discount cost in each case is characteristic for different number of pipes of the horizontal ground heat exchanger with the same other geometric parameters (length of single pipes, distance between pipes and location depth). The presented scheme of the calculation (Eqs. 1–4) allows to compare the influence of the geometric parameters and unit installation costs of the ground heat exchanger, the unit cost of electric energy and the rate of discount on the discount costs [3].

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Analizy porównawcze układów z parowymi sprężarkowymi pompami grzejnymi i z gruntowymi wymiennikami ciepła

S t r e s z c z e n i e

W pracy zaprezentowano wybrane rezultaty analiz termodynamicznych układu grzewczego z parową sprężarkową pompą ciepła. Źródłem ciepła dla pompy grzejnej jest grunt. Rozpatrywano i porównano funkcjonowanie układu dla różnych konfiguracji gruntowego poziomego wymiennika ciepła. Głównymi wynikami obliczeń termodynamicznych są: zmienność strumienia ciepła pobieranego z gruntu, strumienia ciepła przekazywanego do ogrzewanej przestrzeni, zmienność temperatur czynników roboczych w układzie oraz strumienia czynnika pośredniczącego w gruntowym wymienniku ciepła, jak również całkowite zużycie energii podczas sezonu grzewczego. Rezultaty tych analiz zostały wykorzystane w techniczno-ekonomicznych obliczeniach związanych z gruntowym wymiennikiem ciepła. Przedstawiono przykłady obliczeń dotyczące optymalizacji liczby rur oraz wpływu jednostkowego kosztu instalacji gruntowego poziomego wymiennika ciepła na koszt zdyskontowany.