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Heating system with vapour compressor heat pump and vertical U-tube ground heat exchanger

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Abstract In the paper a heating system with a vapour compressor heat pump and vertical U-tube ground heat exchanger for small residential house is considered. A mathematical model of the system: heated object – vapour compressor heat pump – ground heat exchanger is presented shortly. The system investigated is equipped, apart from the heat pump, with the additional conventional source of heat. The processes taking place in the analyzed system are of unsteady character. The model consists of three elements; the first containing the calculation model of the space to be heated, the second – the vertical U-tube ground heat exchanger with the adjoining area of the ground. The equations for the elements of vapour compressor heat pump form the third element of the general model. The period of one heating season is taken into consideration. The results of calculations for two variants of the ground heat exchanger are presented and compared. These results concern variable in time parameters at particular points of the system and energy consumption during the heating season. This paper presents the mutual influence of the ground heat exchanger subsystem, elements of vapour compressor heat pump and heated space.

Keywords: Heat pump; Ground heat exchanger; Heating system, Numerical modelling

Nomenclature

A	–	area of the heat transfer in the exchanger, m^2
c	–	specific heat capacity, J/(kg K)
h	–	specific enthalpy, J/kg

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k	–	heat transfer coefficient for the heat exchanger, W/(m^2 K)
L	–	length of pipes, m
$\dot{m}$	–	mass flow rate, kg/s
n	–	number of pipes,
N_{el}	–	electric driving power of the compressor, W
$\dot{Q}$	–	heat capacity, W
$\dot{q}_l$	–	heat flux, W/m
T	–	temperature, K
$\dot{V}$	–	volumetric flow rate, m^3/s

Greek symbols

ρ	–	density, kg/m^3
η_{is}	–	internal efficiency of the compressor

Subscripts

a	–	air in a heated space
c	–	condenser
e	–	evaporator
el	–	electrical
f	–	refrigerant
gl	–	intermediate fluid in a ground heat exchanger
ghe	–	ground heat exchanger
hs	–	heated space
s	–	additional source of heat
w	–	water

1 Introduction

Heat pump systems utilizing the low-temperature energy sources have been applied more and more frequently [1–4]. Nowadays their number exceeds 70 million [5]. Two types of heat pumps may be applied in heating systems i.e. absorption or vapour compressor heat pumps. Usually they are used in small objects: individual housing buildings, swimming pools, schools etc. Vapour compressor heat pumps are more popular than absorption heat pumps. These heat pumps may use sources with relatively low temperature e.g. atmospheric air, ground, water flow, natural water container or sewage waste water. In a situation when a heat pump system uses geothermal source, the heat is collected from the ground via an intermediate medium flowing inside the horizontal or vertical heat exchanger pipes. Investigations into ground heat exchangers and heat pump systems have been carried out for years in many countries [3,6–14]. They have an experimental and computational form concerning vertical and horizontal ground heat exchangers.

This paper presents shortly the numerical model of the system: heated space – vapour compressor heat pump – vertical U-tube ground heat exchanger. The model allows thermodynamic analysis of this system and investigations of the mutual influence of the U-tube ground heat exchanger, elements of the vapour compressor heat pump and the heated space. The paper presents and compares calculation results for two cases of geometrical parameters of vertical U-tube ground heat exchanger.

2 Mathematical model of the system: heated object – vapour compressor heat pump – vertical ground heat exchanger

The heating system considered (Fig. 1) consists of three circuits: an intermediate fluid, a working fluid of the vapour compressor heat pump and heating water. The intermediate fluid is heated in vertical U-tube ground heat exchanger and then transfers the collected heat to the refrigerant in the evaporator of heat pump. The vapour compressor heat pump working according to the Linde cycle with irreversible adiabatic compression consists of four elements: evaporator, scroll compressor, throttle valve and a condenser. The refrigerant R134a is a working fluid in the heat pump. The heat is transferred from the condenser to the heat exchanger in the heated space by water.

As mentioned, the computational model of the system consists of three separate subsystems and bases on the models presented in [15–21]. The models of the heated space and the vapour compressor heat pump described shortly in this paper were used in the numerical model and analysis of heating system with horizontal ground heat exchanger [21]. The model of U-tube vertical ground heat exchanger presented in the paper allows to take into consideration variation of ground thermal parameters along the depth. It is a new element of the modified model, which bases on the previous models of the U-tube ground heat exchanger [19,20]. Additionally it permits to consider changes of volumetric flow rate of the intermediate fluid, which are connected with the hydraulic flow resistances of this fluid [20]. The new, recently developed computer program may be used for thermal calculations of the system: heated object – vapour compressor heat pump – vertical U-tube ground heat exchanger. As mentioned, this program allows to perform calculations for a long operational time of the system.

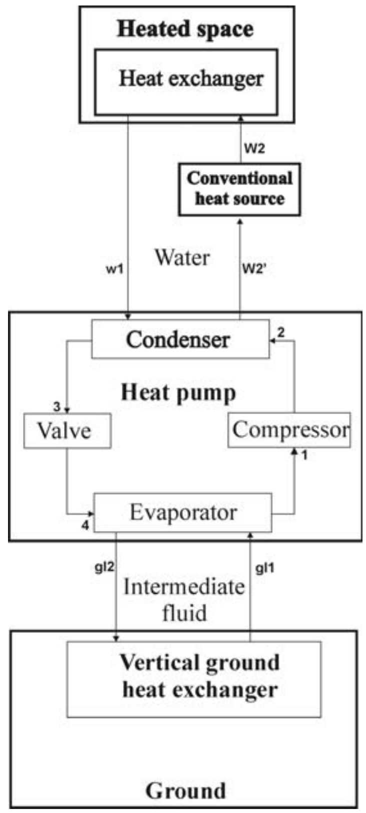


Figure 1. Scheme of the system: heated object – vapour compressor heat pump – vertical ground heat exchanger.

In the calculations the whole heating season under consideration was divided in six-hour periods. The division of each day into four parts allows to consider the influence of solar radiation on the demand for heat and heating system operation. The six-hour demands for heat are known as the calculation results for the heated object. The heat pump works only during a part of each six-hour period until the heat provided by the pump to the heated object reached the value equal to the calculated required heat demand. During the remaining part of the period the heat pump unit does not operate and the heat is not transferred from the ground. If the heat pump heating capacity is not big enough, the heat pump operates in a constant manner throughout all six-hour period and the additional source of heat works as well.

2.1 Model of the heated object

The mathematical model of the heated object is based on energy balance Eq. (1) for elementary time period. This model was presented in [21], as an element of the model of the heating system with horizontal ground heat exchangers. In the model it was assumed that the temperature in analyzed space is constant and the ambient temperature, as well as heat flux of solar radiation, are variable during the heating season. These variations have an hourly character. The structural and material parameters of the building are known. The radiation and convective heat transfer takes place between outer surfaces of the building walls and environment, the heat transfer between heated space and inner wall surfaces has a convective form. The energy balance takes into consideration heat losses and heat gains for the heated space and allows calculation of the heat demand $\dot{Q}_g$ in each elementary time period $\Delta\tau$:

$$\dot{Q}_g\Delta\tau + \sum_j \dot{Q}_{woj}\Delta\tau + \dot{Q}_{wz}\Delta\tau = \\ = \sum_j \dot{Q}_{wsj}\Delta\tau + \dot{Q}_{wd}\Delta\tau + \dot{Q}_{wp}\Delta\tau + \dot{Q}_{zo}\Delta\tau + \dot{Q}_v\Delta\tau + \dot{Q}_m\Delta\tau \ , \quad (1)$$

where:

$\sum_j \dot{Q}_{woj}\Delta\tau$, $\dot{Q}_{wz}\Delta\tau$ – total radiation heat transferred from environment to the heated space through the windows and heat generated inside as a result of the object usage;

$\sum_j \dot{Q}_{wsj}\Delta\tau$, $\dot{Q}_{wd}\Delta\tau$, $\dot{Q}_{wp}\Delta\tau$ – total heat transferred from the heated space to inner surfaces of the walls, roof and floor;

$\dot{Q}_{zo}\Delta\tau$, $\dot{Q}_v\Delta\tau$, $\dot{Q}_m\Delta\tau$ – heat waste through the windows, ventilation system and special construction elements (for example places where the walls and the floor or two walls are jointed;

j – direction of the world.

As mentioned, the model of heated building takes into account accumulation of energy in the walls, the roof as well as the heat transfer to the ground under the building. The unsteady processes of heat conduction in theses elements are analyzed to determine heat capacity values $\dot{Q}_{wsj}$, $\dot{Q}_{wd}$, $\dot{Q}_{wp}$ at each time step of considered heating season. In Fig. 2 the repetitive elements of the walls and the floor situated on the ground and boundary conditions are presented. The form of the roof repetitive element is analogous in the walls. As mentioned, environment temperatures T_0 and solar radiation heat fluxes $\dot{q}_{rj}$ for various direction of the world j, as well as temperatures of horizon are depended on time The heat transfer coefficients on the inner α_a and outer α_0 surfaces of the building, the temperature T_a inside the heated space and values of the emissivity and absorptivity for outer surfaces of the walls and the roof are constant.

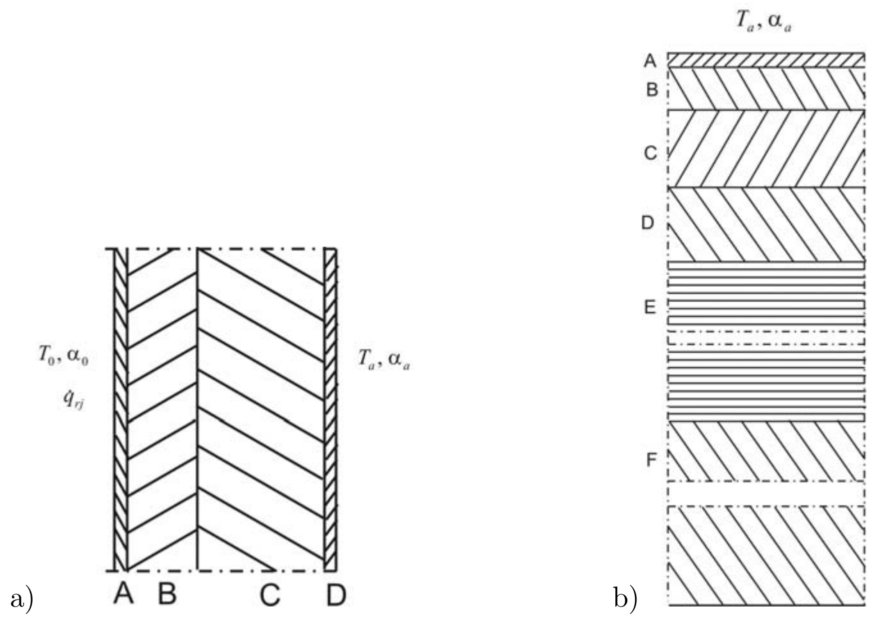


Figure 2. Repetitive element a) of the building walls: A, D – plasters, B – thermal insulation, C – Porotherm wall; b) of the subsystem floor-ground: A, – terracotta, B – jointless floor, C – thermal insulation, D – lean concrete, E – sand, F – ground.

The professional numerical code ANSYS FLUENT was used to determine the unsteady temperature fields in these elements and unitary heat fluxes transferred from the heated space to its inner surfaces. In the calculations, the described model was used to determine the demand for heat in each six-hour periods during all heating season, for exemplary building.

The beginning of the heating season has been assumed from the mid-September and encompassing 230 days during the year.

2.2 Mathematical model of the vapour compressor heat pump

The model of the vapour compressor heat pump [21] is based on the previous models presented in [15–17,19]. In calculations it was assumed that in each time step operation of the main elements of the vapour compressor heat pump can be described by energy equations for steady state. It is possible because changes of the characteristic working parameters of the system are very slow. The heat pump operation is described by energy balance equations for particular elements as well as Peclet number equations for heat exchangers. Analogous equations for the additional heat source and the heat exchanger in the heated space describe their operation in each time step. These equations are supplemented by the use of thermal and calorimetric state equations for refrigerant R134a and equations for calculated heat transfer coefficients in the evaporator and the condenser [22–24]. The exemplary equations describing operation of the condenser, heat exchanger in the heated object and additional heat source are Eqs. (2)–(7) [16]:

$$\dot{Q}_c = \dot{m}_w c_w (T'_{w2} - T_{w1}) \ , \tag{2}$$

$$\dot{Q}_c = \dot{V}_f \rho_1 (h_2 - h_3) \ , \tag{3}$$

$$\dot{Q}_c = k_c A_c \frac{T'_{w2} - T_{w1}}{\ln \frac{T_c - T_{w1}}{T_c - T'_{w2}}} \ , \tag{4}$$

$$\dot{Q}_s = \dot{m}_w c_w (T_{w2} - T'_{w2}) \ , \tag{5}$$

$$\dot{Q}_{hs} = \dot{m}_w c_w (T_{w2} - T_{w1}) \ , \tag{6}$$

$$\dot{Q}_{hs} = (kA)_{hs} \frac{T_{w2} - T_{w1}}{\ln \frac{T_{w2} - T_a}{T_{w1} - T_a}} \ , \tag{7}$$

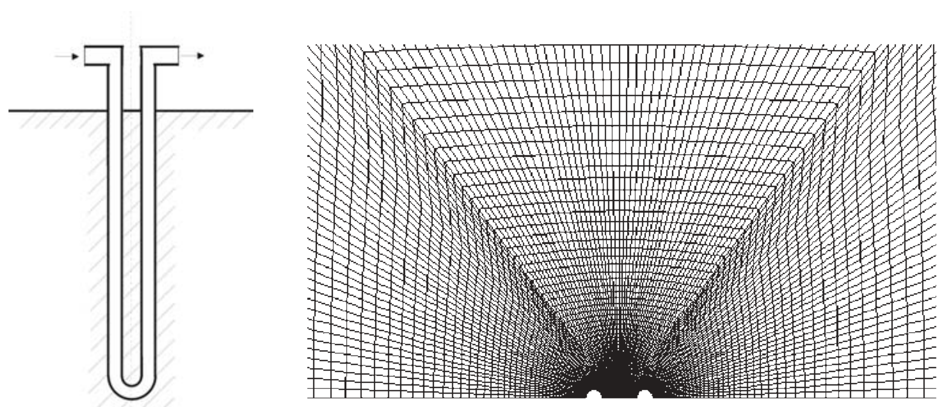
$$N_{el} = N(T_c,\ T_e) \ , \tag{8}$$

$$\eta_{is} = \eta\,(T_e,\ T_c) \ . \tag{9}$$

The electric driving power of the scroll compressor in each time step is determined on the basis of its performance (8) [21]. The compressor works only in definite ranges of the evaporation and condensation temperatures [16,21]. In case when these temperatures are not within the work ranges the heat pump does not operate. It means that the heat is supplied to the heated object only by the additional heat source. In the heat pump model the internal efficiency of compressor is dependent on characteristic temperatures in the condenser and the evaporator (9) [21].

2.3 Mathematical model of the vertical ground heat exchanger

The ground heat exchanger consists of two elongated U-type shape pipes, which are connected parallel to the evaporator of the heat pump (Fig. 3a). Since the thermal parameters of the ground depend on the depth, the vertical ground heat exchanger was divided in sections. In a case when the ground consists of j different horizontal parts, the U-tube vertical heat exchanger is divided into j vertical sections.



(A) (B)

Figure 3. Scheme (A) and fragment of computational domain (B) of vertical ground heat exchanger.

Vertical ground heat exchanger model is two-dimensional and it is based on the model proposed in [18–20]. The assumption of the 2D model is related to additional assumption: 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 intermediate fluid. This is substantiated because of the small value of the temperature increase in ground heat exchangers (from 2 to 4 K) [5].

For $L_{ghe\,i}$ length pipe in section j average heat flux q_{li}, total heat flux collected from the ground $\dot{Q}_{ei}$, average temperature of intermediate fluid in pipe $T_{av\,gli}$, and the outlet temperature of this fluid T_{gl1i} were calculated on the basis of the relations (10)–(13):

$$q_{li} = q_{li}\left(T_{av\,gli}\right) \ , \tag{10}$$

$$\dot{Q}_{ei} = \dot{q}_{li} L_{ghei} \ , \tag{11}$$

$$T_{av\,gli} = \frac{T_{gl1i} + T_{gl2i}}{2} \ , \tag{12}$$

$$T_{gl1i} = T_{gl2i} + \frac{\dot{Q}_{ei}}{\dot{V}_{gli} c_{gl} \rho_{gl}} \ . \tag{13}$$

The values of volumetric flow rate of the intermediate agent are variable and are connected with average temperatures of this fluid in the U-tube ground heat exchanger. In each time step a new value of whole volumetric flow rate is calculated on the base of the intermediate fluid pump performance (14) and the hydraulic characteristic of flow system of this fluid (15) [23]:

$$H_P = f\left(\dot{V}_{gl}\right) \ , \tag{14}$$

$$H_R = f\left(\dot{V}_{gl},\, T_{gl\,av}\right) . \tag{15}$$

The results of calculations in each time step include the temperature field in the analyzed parts of the ground, the volumetric flow rate and the outlet temperature of the intermediate fluid. The temperature of this fluid at the inlet to the pipe of first section of the ground heat exchanger is given as a result of calculations of heat pump unit. The temperature is equal to the intermediate fluid temperature at the outlet from the evaporator. The professional computer code ANSYS FLUENT was adopted to determine the unsteady temperature field in the ground and the unit heat flux transferred from the ground. The computational domain for each section of vertical exchanger contains some part of the horizontal plane limited by the adiabatic line crossing the axis of pipes, the adiabatic line in the middle of the space between neighboring U-type elements and two perpendicular lines, which have a constant temperature, far enough from the heat exchanger tubes. The considered parts of the ground covers the areas located approximately at the mid-point of the pipe sections length. The portion of computational domain for the section of the vertical ground heat exchanger is presented in

Fig. 3B. In calculations the geometrical parameters of the whole computational domain are 8.5 m×6 m. The boundary conditions on the surface of the pipe, calculated at each time step, are the average temperature of the intermediate fluid and overall heat transfer coefficient between external surfaces of the pipes and the fluid [16]. It was assumed that if the compressor does not work, then the outer surfaces of exchanger pipes are regarded as insulated $k_{gl} = 0$. The initial temperature for each computational domain is constant.

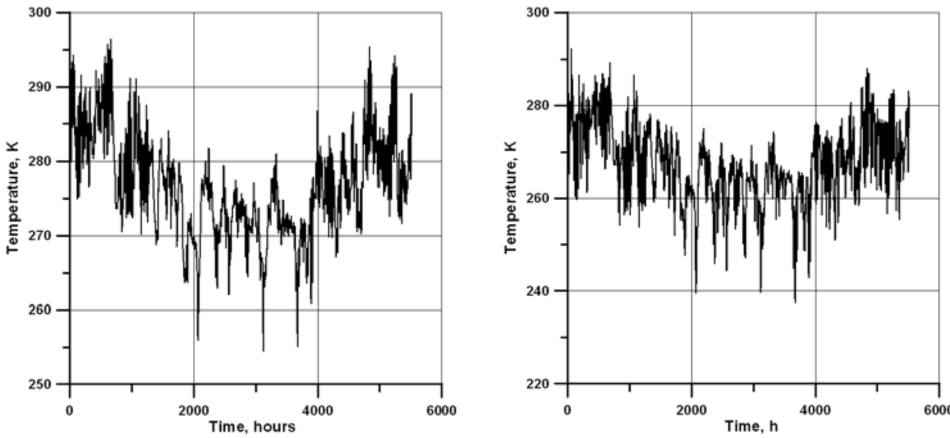
$$T_{gl\,avi} = f_T(\tau), \quad k_{gl} = k(\dot{V}_{gl}, T_{gl\,avi}) \ . \tag{16}$$

3 The data for calculations

Calculations have been performed for the system shown in Fig. 1. Two versions of the configuration of vertical ground heat exchanger were assumed. The vertical U-shaped pipes are separated by 3 m and reach a depth of up to 80 m in the first case and 100 in second one . The outer diameter of the pipes is equal to 0.04 m and the thickness of the walls is equal to 0.0023 m. In analyzed cases it was assumed, that the ground consists of two different parts. Thermal parameters of first, upper parts – from outer surface of the ground to 10 m depth – are different than parameters of parts located below the upper one. For this reason in the calculations the vertical ground heat exchanger was divided into two sections. The calculations have been performed for the following thermal characteristic parameters of the ground:

- upper part (0–10 m) density 1700 kg/m^3, specific heat capacity 2000 J/(kg K), thermal conductivity 1.5 W/(m K), specific latent heat of the moist ground 60 kJ/kg;
- lower part (10–100 m) density 1700 kg/m^3, specific heat capacity 2000 J/(kg K), thermal conductivity 1.2 W/(m K), specific latent heat of the moist ground 40 kJ/kg.

As mentioned, the temperatures of atmospheric air, temperatures of horizon as well as solar radiation heat fluxes are variable during the considered period (Fig. 4 and 5). The variation of these quantities is characteristic for southern Poland [25]. Six-hour demands for heat in January and March calculated for the sample building are presented in Fig. 6A, and demands for heat in all considered heating season is presented in Fig. 6B.



(A) (B)

Figure 4. Changes of the ambient temperature (A) and changes of the horizon temperature (B) during the heating season.

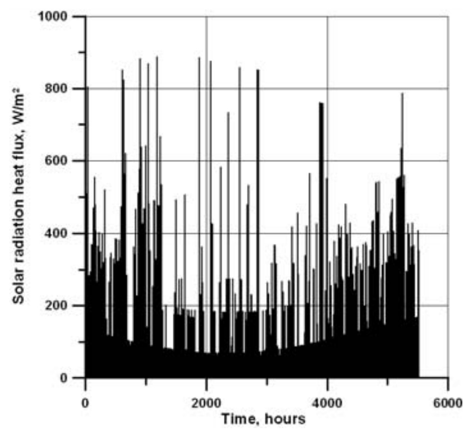
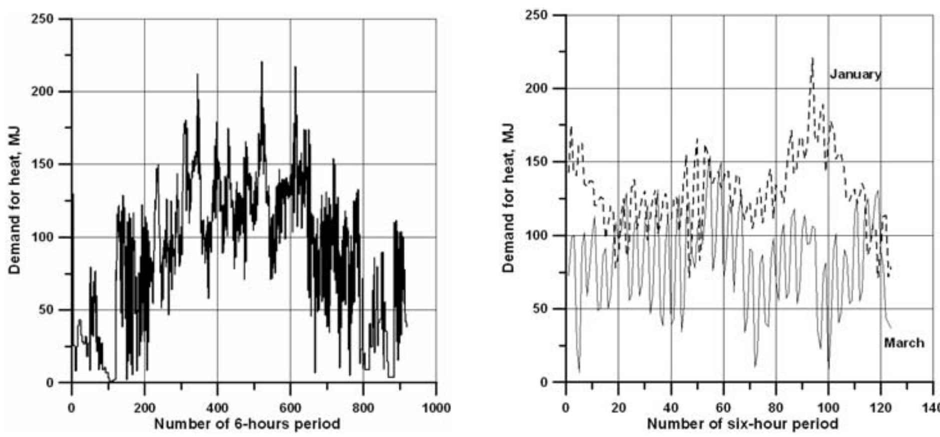


Figure 5. Changes of the solar radiation heat flux for the horizontal surface during the heating season.

The initial temperatures in computational domains are the results of heat transfer calculations for the part of the ground without the heat exchanger, during the period of two years prior to the start of the heating system. This region is limited by the two vertical adiabatic surfaces, the outer ground surface from the top and from the bottom – the plane located about 100 m deep. On the upper boundary of the ground, convection



(A) (B)

Figure 6. Demand for heat in six-hour periods: during the heating season (A), in January and March (B).

and radiation heat transfer with atmospheric air was assumed. The starting value of temperature in calculation domain for first section situated in upper parts of the ground (to 10 m) equals 281.4 K and for second, bottom section 282.15 K. It was also assumed that the value $(kA)_{hs}$ for the heat exchanger in the heating system equals 550 W/K, water mass flow rate equals 0.8 kg/s, and the value of the compressor volumetric capacity equals to 0.00475 m^3/s. The condenser has the following geometric parameters: number of pipes – 42, length of pipes – 1 m, outside/inside diameter – 0.012 m/0.010 m. The same evaporator parameters equal to: number of pipes – 42, length of pipes – 0.7 m, outside/inside diameter – 0.012 m/0.010 m.

4 Results of the calculations, conclusions

The heating system with the heat pump and the additional source of heat must provide required heat quantity for the building in each six-hour period. As mentioned, thermodynamic analysis has been conducted for two cases, i.e. for different length of vertical pipes of a U-pipe heat exchanger. Figures 7–13 present results of calculations for the period of one heating season. Average values of parameters such as heat fluxes (Fig. 7), intermediate medium temperatures (Fig. 9), mass flow rate of the intermediate fluid (Fig. 10) as well as evaporation and condensation temperatures (Fig. 12) and coefficient of performance (Fig. 13) were calculated for a time period equal to the working time of the compressor over the course of each day.

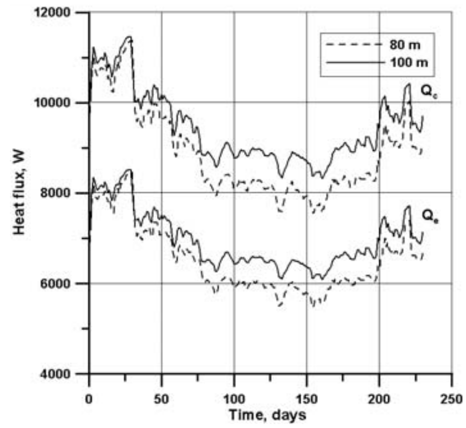


Figure 7. Daily average heat capacity transferred from ground to the evaporator (Q_e) and transferred from the condenser to the heated space (Q_c) during the heating season.

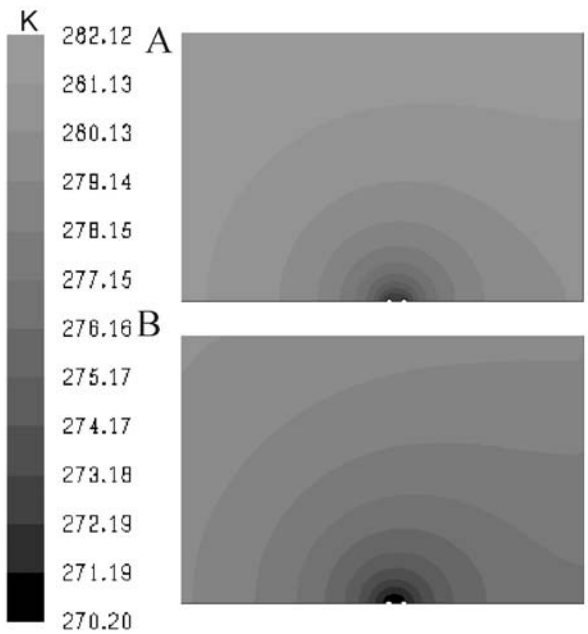


Figure 8. Exemplary temperature distributions in the ground adjacent to the vertical U-tube pipe (bottom section): A) 80-th, B) 160-th day of the heating season (a fragment of the computational domain).

Each analyzed case is characterised by decrease of heating capacity of the heat pump in the initial period of the system operation. It is due to the decrease of heat flux collected from the ground (Fig. 7), which is related to the process of falling the ground temperature in the vicinity of the heat exchanger pipes. Sample temperature distributions in selected days of the heating season are presented in Fig. 8. Increase of heat fluxes values within the end period results from the decrease of heat demand for the object in early spring period (Fig. 7). During the process of collecting heat from the ground, the temperatures of intermediate fluid at the inlet and outlet to the heat exchanger are variable (Fig. 9). The values of temperatures as well as the values of the intermediate medium mass flow rate are related to the length of ground heat exchanger pipes (Fig. 10). As mentioned, the time of operation of the heat pump in each six-hour cycle is depended on required demand for heat. During the compressor downtime, the ground thermal state in the vicinity of the ground heat exchanger pipes recovers. The working time of the compressor on each day of the heating season is presented in Fig. 11. It is obvious that the longest times of system operation are characteristic for the periods of increased demand for heat. The variability of evaporation and condensation temperatures as well as variability of coefficient of performance (the ratio of heat generated by heat pump to internal driving power of compressor) for both variants are presented in Figs. 12 and 13. The courses of the characteristic parameters of the heat pump presented in Figs. 7, 9, 10, 12 and 13 are strictly connected to the ground temperature field in the vicinity of the pipes.

One of the results of the calculations concerns the consumption energy in the heating system subject to consideration. The additional heat source operates only periodically: in the case of 80 m U-pipe it supplied the system in 20 days, in the case of a 100 m U-pipe in 7 days. In the analysed variants the consumption of electricity used to drive the compressor and the pump of the intermediate fluid as well as the energy provided by the additional sources of heat are different. Their values amount to respectively c.a.: for 80 m U-pipe 7390 kWh and 180 kWh, for 100 m U-pipe 7200 kWh and 100 kWh.

The analyses indicated that parameters in characteristic points of the system: heated object – vapour compressor heat pump – ground heat exchanger, depend on each another. With the same manner of operations of the heating system, a change of one of geometric parameters of the ground heat exchanger has an impact on the characteristic parameters of the remaining elements. The thermal condition of the ground related to the

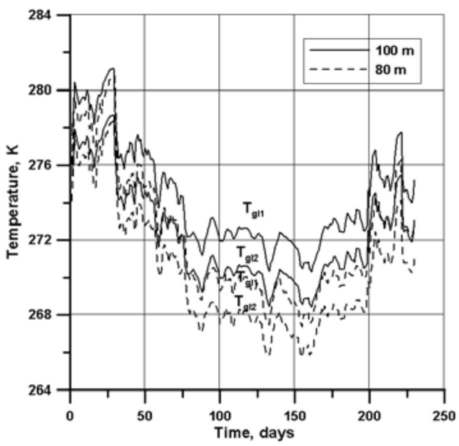


Figure 9. Daily average of inlet (T_{gl2}) and outlet (T_{gl1}) temperatures of the intermediate medium.

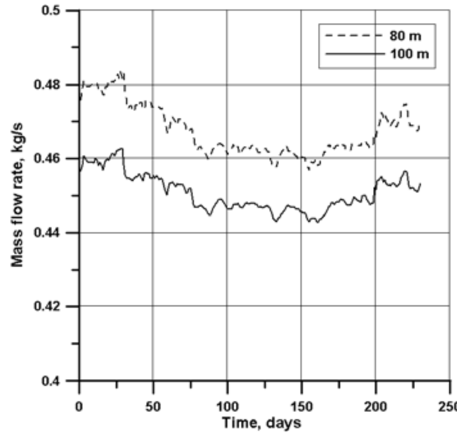


Figure 10. Daily average mass flow rate (in single pipe) during the heating season.

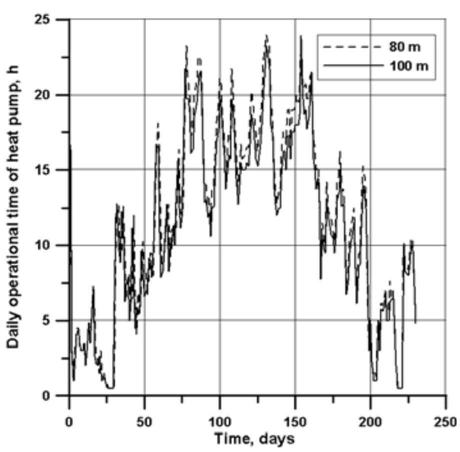


Figure 11. Daily operational time of the vapour compressor heat pump.

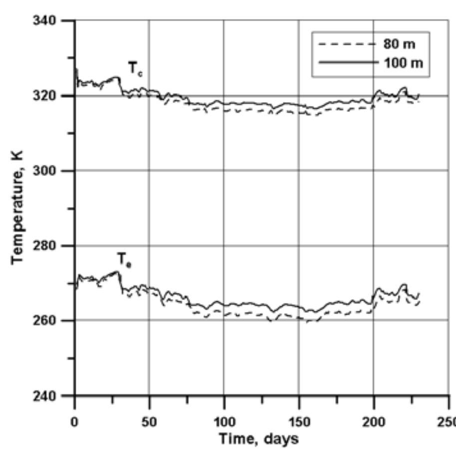


Figure 12. Daily average evaporation (T_e) and condensation (T_c) temperature during the heating season.

length of the ground heat exchangers influences the operation of the remaining sub-systems. The parameters characteristic for the heated object and heat pump have an impact on the thermal condition of the ground.

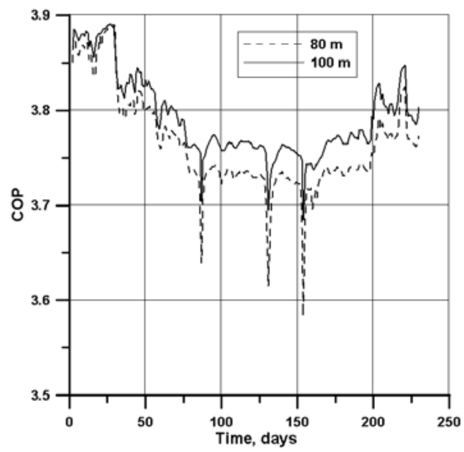


Figure 13. Daily average coefficient of performance of the heat pump cycle during the heating season.

Example of such parameter may include required demand for heat for the heated object; it influences the operational time of compressor and the time of collecting heat from ground as well as its thermal condition.

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