INFLUENCE OF GROUND HEAT SOURCE PARAMETERS AND CONSUMER ON CHARACTERISTICS OF GEOTHERMAL HEAT PUMP SYSTEM

Features of heat supply system based on the use of ground are investigated with the help of the method of numerical simulation. The variables are the parameters characterizing the heat source and the consumer: the average temperature of the soil, the area and the radiative properties of the surface heating panels, air temperature of heated space. It was found that the maximum energy efficiency of the system is achieved by using a low-temperature floor heating system.

Introduction. Low-grade heat of ground using heat pumps (HP) is considered to be one of the modern trends of renewable energy utilization. The main elements of such systems are interconnected heating contours of ground heat exchangers, a heat pump and a heat consumer which determine the operation of the whole system.

The mutual dependence of the parameters of the elements of such a system requires the design and evaluation of the use of complex numerical models that have been used successfully in the analysis of similar systems: refrigerators [1], HP for heat recovery ventilation emissions [2], HP with horizontal ground heat exchangers [3].

In this paper, based on the extended numerical model studies [4–6], the analysis of the influence of the features of the system heating using heat of the ground is made, which is complemented by the equations for the heat consumer. The heating system is regarded as the object of heat consumption by the building.

Object of study. We are studying a system consisting of contour HP, several borehole heat exchangers (BHE) and panel floor heating system (Fig. 1). The number of BHE is 5, their length – 60 m, the diameter of boreholes – 200 mm, BHE with two U-shaped polyethylene pipes 32×3 mm, the diagonal distance between the tubes is 150 mm. Thermal conductivity of BHE pipes is assumed to be equal to 0.38 W/(m·K); the thermal conductivity of grouting is 2.3 W/(m·K). Coolant flow rate of the contour of ground heat exchangers is 0.8 kg/s. Coolant of the BHE circuit is 12.2% aqueous solution of ethylene glycol.

The evaporator and condenser of HP are tube heat exchangers with segmental baffles with boiling in the U-shaped steel pipes. The number of tubes is 40, the length of the condenser tubes is 4 m, of the evaporator – 2 m. Pipes sized 9×1 mm are used in the condenser, in the evaporator – 1×10 mm. R134a is used as a refrigerant.

Compressor Bitzer 4FES with a volumetric displacement 22.72 m³/s is used in HP.

Heat requirement for heating residential buildings is determined by its specific heating characteristic $q_0$ and the temperature difference
between the outside air \((t_{oa})\) and the air of heated space \((t_{hs})\):

\[
Q = q_0 F (t_{hs} - t_{oa}),
\]

where \(F\) – heated area, \(m^2\).

**Method of analysis.** For the numerical analysis of the parameters of the heat supply system based on the use of low-grade heat of soil we applied the advanced numerical model of a ground heat pump, presented earlier in [4–6] and which differs from the analogues in the possibility of a joint analysis of heat and mass transfer processes in its basic contours: ground heat exchangers, HP and the consumer.

It is assumed that the heat transfer in the floor panels is limited by radiantly-convective heat exchange on the surface when taking into account the parameters of the heat consumer. In this case, the heat flow transmitted to the room air is equal to:

\[
Q = \alpha_c F_{hp} (t_{hp} - t_{hs}),
\]

where \(\alpha_c = \alpha_c + \alpha_r\) – radiantly-convective heat transfer coefficient, \(W/(m^2 \cdot K)\); \(\alpha_r\) – free convective component, \(W/(m^2 \cdot K)\); \(\alpha_c\) – radiant component, \(W/(m^2 \cdot K)\); \(F_{hp}\) – the surface area of heating panels \((F_{hp} \leq F)\), \(m^2\); \(t_{hp}\) – temperature of heating panels surface, \(^\circ C\).

Here component \(\alpha_c\) is determined by the formula of V. S. Zhukovsky for free convection on a horizontal surface [7]:

\[
\alpha_c = 2.2 (t_{hp} - t_{hs})^{0.25}.
\]

Component \(\alpha_r\) is determined by the formula

\[
\alpha_r = \varepsilon \sigma_0 \frac{T_{hp}^4 - T_{hs}^4}{T_{hp} - T_{hs}},
\]

where \(\varepsilon\) – the degree of blackness of the surface of heating panels; \(\sigma_0\) – Stefan-Boltzmann constant, \(W/(m^2 \cdot K^4)\); \(T_{hp}\) – surface temperature of the heating panels, \(K\); \(T_{hs}\) – air temperature of heated space, \(K\).

**System characteristics.** In the presented study the system that meets the following criteria is chosen as a nominal:

- outdoor temperature \(t_{oa}\) = \(-24^\circ C\) (the average temperature of the coldest five days for Minsk [8]);
- indoor air temperature \(t_{hs}\) = \(18^\circ C\) (according to [9] for the residential and commercial buildings);
- the specific characteristics of the building heating \(q_0 = 0.654 W/(m^2 \cdot K)\) [9];
- heated area \(F = 450 m^2\);
- the average ground temperature \(t_g\) = \(5^\circ C\) (takes less ground temperature 6–8°C at the beginning of operation of the system and corresponds to the subsequent quasi-stationary state of the system);
- emissivity factor \(\varepsilon\) of heating panels surface equal to \(0.4\).

In accordance with the sanitary requirements the condition of the surface temperature of heating panels (floor) is defined as follows \(t_{hp} \leq 24^\circ C\) [10].

The heat consumption of the building according to the formula (1) is 12.36 kW and corresponds to the heating capacity of a heat pump \(Q_e\). This heat flow is provided by floor heating panels the area of which is \(F_{hp} = 380 m^2\). In this case, \(t_{hp} = 23.7^\circ C\) and complies with a norm. Evaporator heat flow \(Q_e\), and therefore BHE is 9.44 kW. Energy efficiency is determined by coefficient of performance \(\phi\) of heat pump, which in this case is 4.2. The high value of \(\phi\) indicates the effectiveness of this method of production of thermal energy. Nevertheless, during the operation of a heating system ground temperature \(t_g\) changes that affect the basic parameters of the HP. The impact on system performance parameters of the heat consumer: the surface area of heating panels \(F_{hp}\), its emissivity factor \(\varepsilon\), air temperature of heated space \(t_{hs}\) is also of practical importance.

The following are the results of numerical analysis of the impact of these characteristics on the basic parameters of the system: heating capacity \(Q_e\), coefficient of performance \(\phi\), heat flow \(Q_c\) of BHE, the surface temperature of heating panels \(t_{hp}\).

**Influence of ground temperature.** Ground temperature \(t_g\) at the beginning of HP with BHE is close to the average annual temperature of the Earth's surface for the considered area (6–8°C for the Republic of Belarus [11]), as evidenced by the practice of designing BHE [12] and long-term observations [13]. During the heating period the \(t_{hp}\) will decline through cooling due to the operation of stations. Temperature 0°C is taken as the lower limit of the \(t_g\), at which the freezing of ground moisture starts and the corresponding undesirable deformation and adjacent ground elements of BHE.

Using numerical simulation, it is possible to analyze the changes in the parameters of the heating system when the temperature of soil is in the range of 0–8°C.

Fig. 2 shows the change in \(Q_e\) and \(Q_c\) depending on the \(t_g\). It can be seen that increasing the \(t_g\) from 5 to 8°C, the growth of \(Q_e\) by 10.4% (an increase by 1.3 kW), and \(Q_c\) by 12.3% (an increase by 1.2 kW) is observed compared to the nominal values. The reduction of ground temperature from 5 to 1°C leads to a drop in \(Q_e\) by 16.3% (a decrease by 2.0 kW) and \(Q_c\) by 18.6% (a decrease by 1.8 kW).

Fig. 3 shows the change in \(\phi\) depending on the \(t_g\). It can be seen that with the increasing of the \(t_g\) from 5 to 8°C the growth of \(\phi\) occurs from 4.2 to
4.4, which is 4.8% compared to the nominal value. The reduction of soil temperature from 5 to 1°C leads to a decrease of $\phi$ by 9.7%.

These factors have led to the fact that floor systems in our country did not get a widespread usage unlike the systems with a small area of the heat-transfer surface.

The quantitative estimation of parameters of the heating system based on the use of soil heat at low values of heat-transfer surface area $F_{hp}$. Using numerical simulation, it is possible to analyze the changes in the parameters of the studied heat supply system while substantially reducing $F_{hp}$ to 20 m$^2$.

Fig. 5 shows the change in $Q_c$ and $Q_e$ at different $F_{hp}$. It is seen that with the reduction of $F_{hp}$ drop of $Q_c$ and $Q_e$ occurs. The minimum value of $Q_c$ of the considered range of $F_{hp}$ is below the nominal value by 35% (a decrease by 4.42 kW). Minimum $Q_e$ is below the nominal value by 49% (a decrease by 4.57 kW).

As can be seen from Fig. 6, the allowable temperature range of the floor surface has a narrow range. With decreasing of $F_{hp}$ growth of the $t_{hp}$ occurs, with its maximum value reaching 64°C at the
lower boundary $F_{hp} = 20 \text{ m}^2$. These high temperatures are not valid for floor heating, in this case, the system can already be considered as a high-temperature and should include other heating devices, such as radiators or convectors, that is widespread in most of the buildings operated in the Republic of Belarus.

At the same time the using of heat pumps in conjunction with heating devices of small area surfaces is due to the low energy efficiency of such systems, as evidenced by the data in Fig. 7, from which it follows that the reduction of the $F_{hp}$ leads to a drop of $\varphi$ from the nominal value of 4.3 at $F_{hp} = 380 \text{ m}^2$ to that of 2.7 at $F_{hp} = 20 \text{ m}^2$ (drop by 37%).

Therefore, we can conclude that to create an energy-efficient heat pump heating system based on the use of low-grade heat of soil it is necessary to provide low-temperature heating, such as floor panel heating. In the buildings this can be achieved by the modernization of the existing system.

**The influence of emissivity factor of heating surface.** Radiant-convective heat transfer coefficient on the surface of heating panels $\alpha_{rc}$ is determined by the radiant component $\alpha_r$. Consequently, the system parameters will depend on the emissivity factor of the surface of heating panels $\varepsilon$.

Fig. 8 shows the change in $Q_c$ and $Q_e$ depending on $\varepsilon$. It is seen that the maximum value of $Q_c = 12.57 \text{ kW}$ at $\varepsilon = 0.99$ is greater than the nominal by 1.7% (by 0.21 kW), the minimum value of $Q_c = 12.09 \text{ kW}$ at $\varepsilon = 0.01$ is less than the nominal by 2.2% (by 0.27 kW); the maximum value of $Q_e = 9.66 \text{ kW}$ at $\varepsilon = 0.99$ is greater than the nominal by 2.3% (by 0.22 kW), the minimum value of $Q_e = 9.14 \text{ kW}$ at $\varepsilon = 0.01$ is less than the nominal by 3.2% (by 0.30 kW).

Fig. 9 shows the change in $\varphi$ depending on $\varepsilon$. It is seen that the maximum value of $\varphi = 4.32$ at $\varepsilon = 0.99$ is greater than the nominal by 2.9% (by 0.12), the minimum value of $\varphi = 12.09$ kW at $\varepsilon = 0.01$ is less than the nominal by 3.3% (by 0.14).

Fig. 10 shows the change of $t_{hp}$ depending on $\varepsilon$. It can be seen that in this case with the increase of $\varepsilon$ there is a drop of $t_{hp}$, which varies in the range of 21.8–26.4°C. And for the values $\varepsilon = 0.01–0.32$ surface temperature exceeded the allowable taken 24°C. Fig. 11 shows the effect of $\varepsilon$ on $\alpha_{rc}$, which naturally increases with $\varepsilon$. 

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**Fig. 6. Change of the $t_{hp}$ depending on the $F_{hp}$:**
- ○ – nominal mode; ● – maximum allowable $t_{hp}$

**Fig. 7. Change of $\varphi$ depending on the $F_{hp}$:**
- ○ – nominal mode

**Fig. 8. Change $Q_c$ (curve 1) and $Q_e$ (curve 2) depending on the $\varepsilon$:**
- ○ – nominal mode

**Fig. 9. Change in $\varphi$ depending on $\varepsilon$:**
- ○ – nominal mode

**Fig. 10. Change of $t_{hp}$ depending on $\varepsilon$:**
- ○ – nominal mode

**Fig. 11. Effect of $\varepsilon$ on $\alpha_{rc}$,** which naturally increases with $\varepsilon$. 

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Fig. 10. Change of the \( t_{hp} \) depending on the \( \varepsilon \):
○ – nominal mode; ● – maximum allowable \( t_{hp} \)

Thus, it is clear that \( \varepsilon \) has no significant effect on the \( Q_c \), \( Q_e \) and \( \phi \). It has been found that \( \varepsilon \) has a significant impact on \( t_{hp} \), the decrease of which with increasing \( \varepsilon \) is explained by a corresponding increase in \( \alpha_{rc} \), which varies in the range of 3.8–8.8 W/(m\(^2\) · K).

**The influence of the indoor air.** Depending on the purpose and mode of operation of the building the indoor air temperature \( t_{hs} \) may also vary. According to [9] for the residential, office and residential buildings the estimated temperature of indoor air \( t_{hs} \) is 18°C, for preschool and children's hospitals \( t_{hs} = 21^\circ\text{C} \). In case of using heating standby mode \( t_{hs} \) can be reduced to 16°C. It is also known that the \( t_{hs} \) in premises for agricultural purposes varies in a very wide range from 0 to 22°C [10].

In this paper we establish the dependence of the main parameters of the system on the \( t_{hs} \), varying in the range from 0 to 21°C.

Fig. 12 shows the change in \( Q_c \) and \( Q_e \) depending on the \( t_{hs} \). The increase of the \( t_{hs} \) from 0 to 21°C leads to a decrease in \( Q_c \) and \( Q_e \). At the considered temperatures \( Q_c \) varies from 13.97 to 12.13 kW, and \( Q_e \) from 11.39 to 9.12 kW.

![Fig. 12. Change \( Q_c \) (curve 1) and \( Q_e \) (curve 2) depending on the \( t_{hs} \): ○ – nominal mode; □ – at \( t_{hs} = 21^\circ\text{C} \), Δ – at \( t_{hs} = 16^\circ\text{C} \); ● – maximum allowable \( t_{hp} \)](image)

The increase of the \( t_{hs} \) leads to the increase in the temperature of the heating panels (Fig. 13), which in turn affects the condensation temperature of the working medium of HP, the cause of which is the reduction of its heating capacity \( Q_c \).

![Fig. 13. Change of \( t_{hp} \) depending on the \( t_{hs} \): ○ – nominal mode; □ – at \( t_{hs} = 21^\circ\text{C} \), Δ – at \( t_{hs} = 16^\circ\text{C} \); ● – maximum allowable \( t_{hp} \)](image)

At \( t_{hs} = 21^\circ\text{C} \) heating capacity \( Q_c \) is lower than the nominal by 2.1% (a decrease by 0.26 kW), at \( t_{hs} = 16^\circ\text{C} \) it is higher by 1.5% (an increase by 0.19 kW). The maximum value of \( Q_c \) met the minimum value of the \( t_{hs} = 0^\circ\text{C} \) from the considered range and was higher than the nominal by 12.8% (an increase by 1.6 kW).

At \( t_{hs} = 21^\circ\text{C} \) the heat flow of BHE \( Q_e \) is lower than the nominal by 3.3% (a decrease by 0.31 kW), at \( t_{hs} = 16^\circ\text{C} \) – by 2.7% higher (an increase by 0.25 kW). The maximum value of \( Q_e \) met the minimum value of \( t_{hs} = 0^\circ\text{C} \) from the considered range and was higher than the nominal by 20.8% (an increase by 1.96 kW).

Fig. 14 shows the change in the \( \phi \) depending on the \( t_{hs} \). The increase of the \( t_{hs} \) leads to the decrease of the \( \phi \). At the considered temperatures \( \phi \) ranges from 4.06 to 5.38.

![Fig. 14. Change of \( \phi \) depending on the \( t_{hs} \): ○ – nominal mode; □ – at \( t_{hs} = 21^\circ\text{C} \), Δ – at \( t_{hs} = 16^\circ\text{C} \); ● – maximum allowable \( t_{hp} \)](image)
The maximum value of heating panels on the efficiency of the heat pump is reduced from 4.2 to 2.7 changing the transformation coefficient of the heat pump system with a standard 64°C temperature through the use of low-grade heat of the ground.

The transition from the low-temperature heat pump system will be characterized by the high coefficients of performance of HP in the case of use for space heating with reduced temperature of heated space (temperature of heated space) have a significant impact on the performance of the heating system and higher than the nominal by 28.0% (an increase by 1.18).

The obtained calculated data show that for the heating systems based on the use of low-grade heat of ground will be characterized by the high coefficients of performance of HP in the case of use for space heating with reduced temperature of indoor air.

Conclusion. The data obtained show that the considered parameters of the source (ground temperature) and heat consumers (the area and emissivity factor of the surface of heating panels, air temperature of heated space) have a significant impact on the performance of the heating system through the use of low-grade heat of the ground.

The transition from the low-temperature heat pump system with a standard 64°C temperature leads to a significant reduction in its energy efficiency, when the transform coefficient of the heat pump is reduced from 4.2 to 2.7 changing the defining parameters.

The obtained data of numerical analysis also shows the need to consider the effect of soil temperature and room air, the radiative properties of heating panels on the efficiency of the heat pump system using the heat of the ground.

References


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