УДК 621.577

S. A. Filatov, PhD student (BSTU);
V. I. Volodin, D.Sc. (Engineering), professor (BSTU)

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 m^3/s is used in HP.

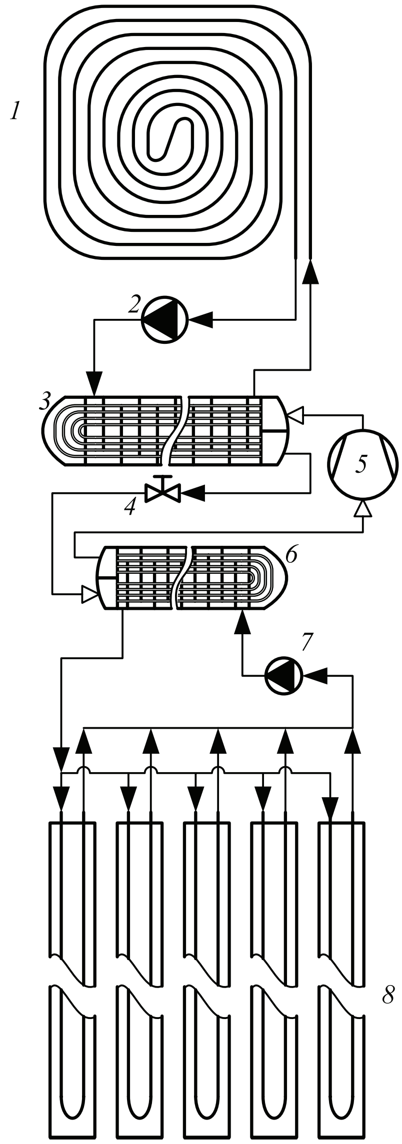

Fig. 1. Schematic diagram of the heating system based on utilization low-grade heat of soil:
1 – the consumer of thermal energy (panel heating);
2, *7* – circulation pumps; *3* – the condenser;
4 – expansion valve; *5* – compressor;
6 – evaporator; *8* – BHE

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}):

$$Q = q_0 F \left(t_{hs} - t_{oa} \right), \quad (1)$$

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:

$$Q = \alpha_{rc} F_{hp} \left(t_{hp} - t_{hs} \right), \quad (2)$$

where $\alpha_{rc} = \alpha_c + \alpha_r$ – radiantly-convective heat transfer coefficient, W/(m^2 · K); α_c – free convective component, W/(m^2 · K); α_r – radiant component , Вт/(м^2 · К); F_{hp} – the surface area of heating panels ($F_{hp} \le F$), m^2; t_{hp} – temperature of heating panels surface, °C.

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

$$\alpha_c = 2.2 \left(t_{hp} - t_{hs} \right)^{0.25}. \quad (3)$$

Component α_r is determined by the formula

$$\alpha_r = \varepsilon \sigma_0 \frac{T_{hp}^4 - T_{hs}^4}{T_{hp} - T_{hs}}, \quad (4)$$

where ε – the degree of blackness of the surface of heating panels; σ_0 – Stefan-Boltzmann constant, W/(m^2 · 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°C (the average temperature of the coldest five days for Minsk [8]);
- indoor air temperature t_{hs} = 18°C (according to [9] for the residential and commercial buildings);
- the specific characteristics of the building heating q_0 = 0,654 W/(m^2 · K) [9];
- heated area F = 450 m^2;
- the average ground temperature t_g = 5°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 ε 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 $t_{hp} \le 24$°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_h. 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°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 φ of heat pump, which in this case is 4.2. The high value of φ 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 ε, 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_c, coefficient of performance φ, heat flow Q_e 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_g 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 undesired 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_c and Q_e depending on the t_g. It can be seen that increasing the t_g from 5 to 8°C, the growth of Q_c by 10.4% (an increase by 1.3 kW), and Q_e 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_c by 16.3% (a decrease by 2.0 kW) and Q_e by 18.6% (a decrease by 1.8 kW).

Fig. 3 shows the change in φ 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 φ 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 φ by 9.7%.

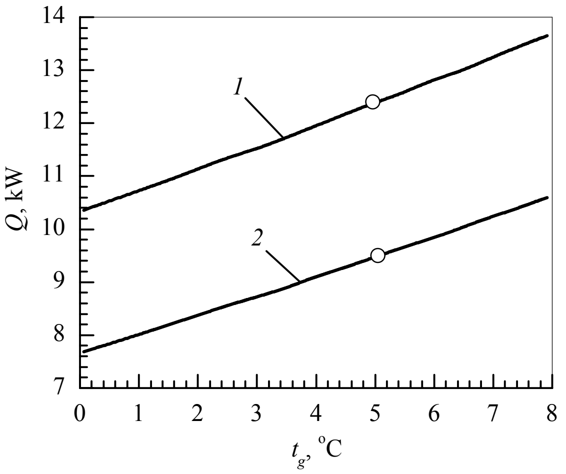

Fig. 2. Change of Q_c (curve *1*) and Q_e (curve *2*) depending on the t_g: ○ – nominal mode

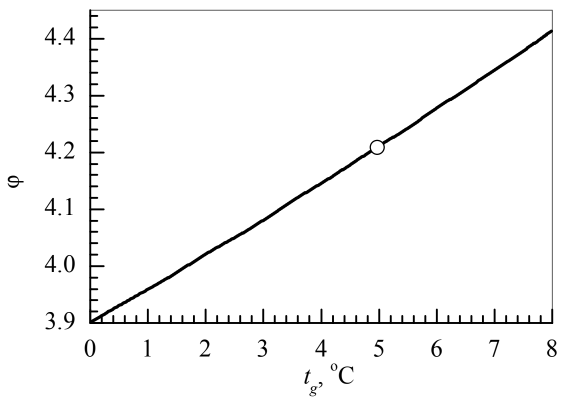

Fig. 3. Change of φ depending on the t_g: ○ – nominal mode

Fig. 4 shows the change of the t_{hp} depending on the t_g. It is seen that in the mainly range of change t_g = 0–6.8°C the floor temperature of the t_{hp} does not exceed 24°C. In the range of the t_g = 6.8–8.0°C the temperature of the t_{hp} exceeds the allowable due to the growth of heat output.

The influence of the surface area of the heating panels. The advantage of the studied floor panel heating system used in a basic variant is the low temperature of the coolant, which is achieved through the advanced heat exchange surface. Low temperature promotes increasing the Q_c and φ due to the setting of low pressure in the condenser and consequently, lower compression refrigerant cycle of the heat pump.

Nevertheless, the shortcomings of the floor heating can be the following: higher installation costs, material consumption, the need for thermal insulation, the presence of the free surface of the floor, not occupied by furniture or equipment. 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.

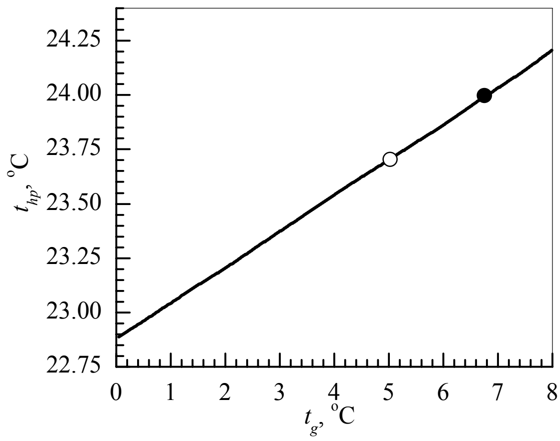

Fig. 4. Change of the t_{hp} depending on the t_g: ○ – nominal mode; ● – maximum allowable t_{hp}

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).

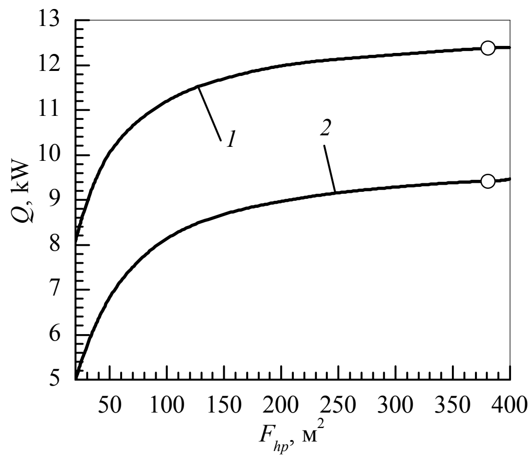

Fig. 5. Change of Q_c (curve *1*) and Q_e (curve *2*) depending on F_{hp}: ○ – nominal mode

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 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.

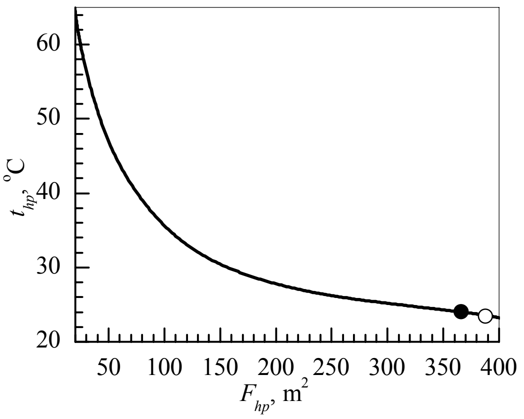

Fig. 6. Change of the t_{hp} depending on the F_{hp}: ○ – nominal mode; ● – maximum allowable t_{hp}

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 φ from the nominal value of 4.3 at F_{hp} = 380 m^2 to that of 2.7 at F_{hp} = 20 m^2 (drop by 37%).

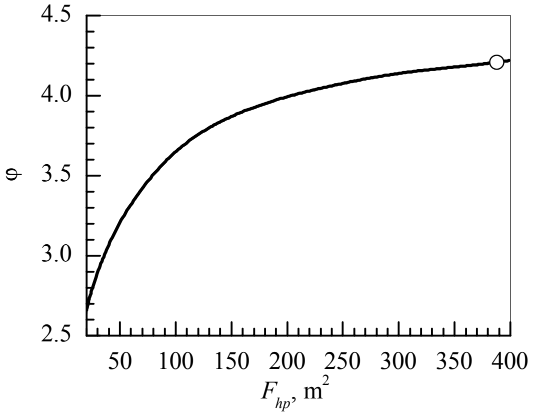

Fig. 7. Change of φ depending on the F_{hp}: ○ – nominal mode

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 α_{rc} is determined by the radiant component α_r. Consequently, the system parameters will depend on the emissivity factor of the surface of heating panels ε.

Fig. 8 shows the change in Q_c and Q_e depending on ε. It is seen that the maximum value of Q_c = 12.57 kW at ε = 0,99 is greater than the nominal by 1.7% (by 0.21 kW), the minimum value of Q_c = 12.09 kW at ε = 0,01 is less than the nominal by 2.2% (by 0.27 kW); the maximum value of Q_e = 9.66 kW at ε = 0,99 is greater than the nominal by 2.3% (by 0.22 kW), the minimum value of Q_e = 9.14 kW at ε = 0,01 is less than the nominal by 3.2% (by 0.30 kW).

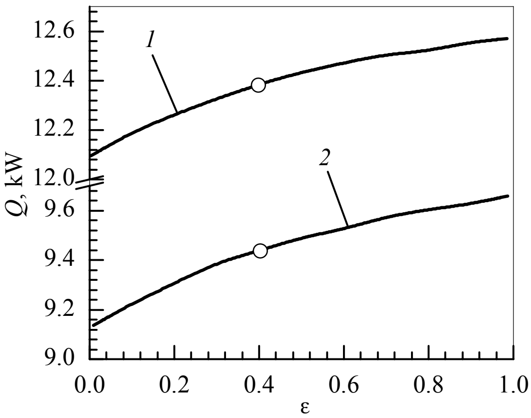

Fig. 8. Change Q_c (curve *1*) and Q_e (curve *2*) depending on the ε: ○ – nominal mode

Fig. 9 shows the change in φ depending on ε. It is seen that the maximum value of φ = 4.32 at ε = 0.99 is greater than the nominal by 2.9% (by 0.12), the minimum value of φ = 12.09 kW at ε = 0,01 is less than the nominal by 3.3% (by 0.14).

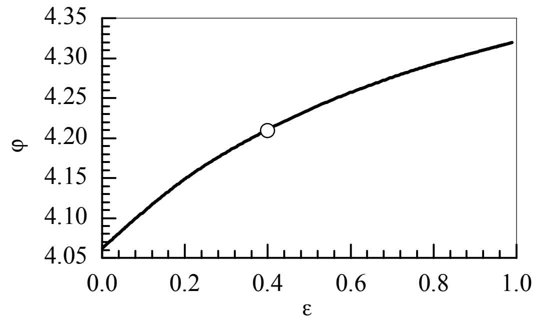

Fig. 9. Change in φ depending on ε: ○ – nominal mode

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

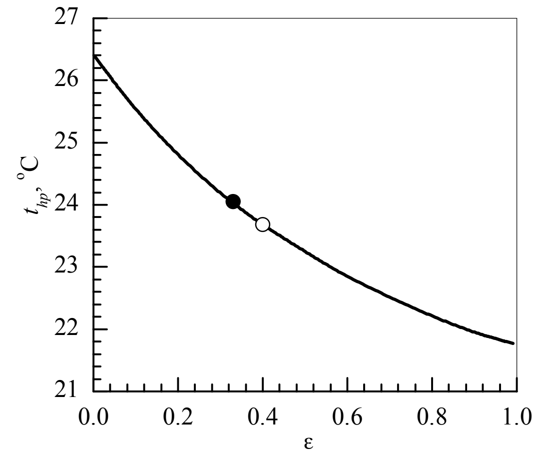

Fig. 10. Change of the t_{hp} depending on the ε: ○ – nominal mode; ● – maximum allowable t_{hp}

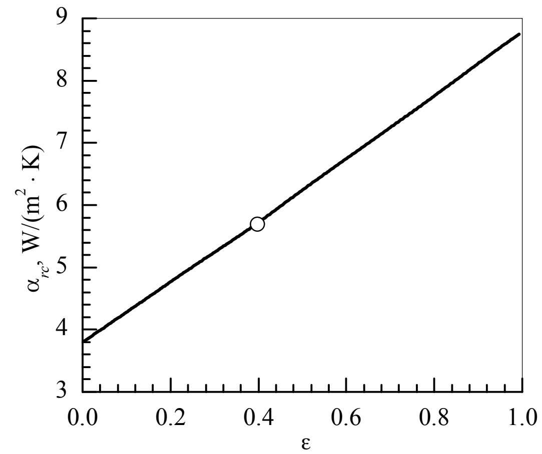

Fig. 11. Change of α_{rc} depending on the ε: ○ – nominal mode

Thus, it is clear that ε has no significant effect on the Q_c, Q_e and φ. It has been found that ε has a significant impact on t_{hp}, the decrease of which with increasing ε is explained by a corresponding increase in α_{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°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.

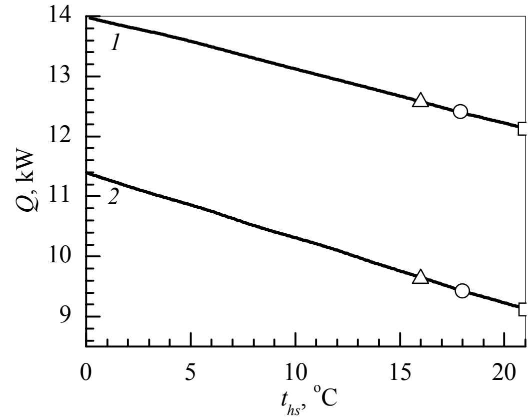

Fig. 12. Change Q_c (curve *1*) and Q_e (curve *2*) depending on the t_{hs}: ○ – nominal mode; «□» – at t_{hs} = 21°C, «Δ» – at t_{hs} = 16 °C

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.

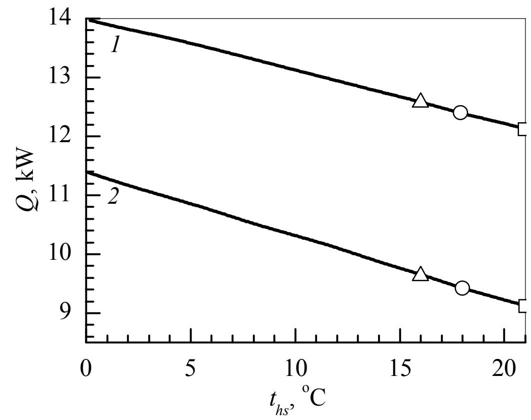

Fig. 13. Change of t_{hp} depending on the t_{hs}: ○ – nominal mode; □ – at t_{hs} = 21°C, Δ – at t_{hs} = 16°C; ● – maximum allowable t_{hp}

At t_{hs} = 21°C heating capacity Q_c is lower than the nominal by 2.1% (a decrease by 0.26 kW), at t_{hs} = 16°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°C from the considered range and was higher than the nominal by 12,8% (an increase by1.6 kW).

At t_{hs} = 21°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°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°C from the considered range and was higher than the nominal by 20.8% (an increase by1.96 kW).

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

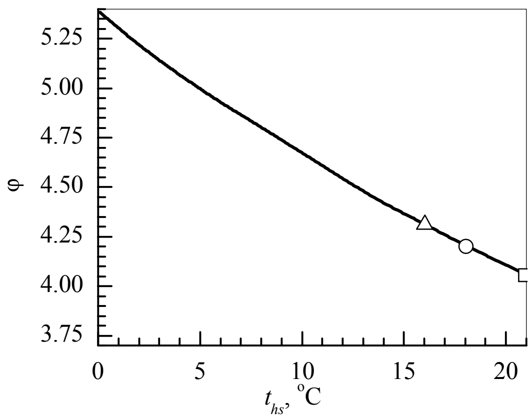

Fig. 14. Change of the φ depending on the t_{hs}:
○ – nominal mode □ – at t_{hs} = 21°C;
Δ – at t_{hs} = 16°C

At t_{hs} = 21°C the transformation coefficient φ is lower than the nominal by 3.3% (drop by 0.14), at t_{hs} = 16°C – 2.6% higher (an increase by 0.11). The maximum value of φ corresponds to the minimum value of t_{hs} = 0°C from the considered range 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

1. Володин В. И. Комплексный подход к расчету параметров компрессионной холодильной машины // Холодильная техника. 1998. № 2. С. 8–10.

2. Здитовецкая С. В., Володин В. И. Утилизация теплоты в системе приточно-вытяжной вентиляции с использованием теплового насоса // Труды БГТУ. Сер. III. Химия и технология неорган. в-в. 2009. Вып. XVII. С. 171–173.

3. Тарасова В. А., Харлампиди Д. Х., Шерстюк А. В. Моделирование тепловых режимов совместной работы грунтового теплообменника и теплонасосной установки // Восточно-европейский журнал передовых технологий. 2011. Т. 53, № 5/8. С. 34–40.

4. Филатов С. О. Численное моделирование и анализ энергетических параметров теплового насоса с многотрубными вертикальными грунтовыми теплообменниками // Экология и промышленность. 2013. № 3. С. 61–66.

5. Филатов С. О. Влияние параметров энергетических свай на работу теплового насоса системы теплоснабжения здания // Будівельні конструкції: Міжвідомчий науково-технічний збірник наукових праць. 2013. Вип. 77. С. 131–135.

6. Филатов С. О., Володин В. И. Метод расчета и анализ совместной работы контура циркуляции грунтовых теплообменников и теплового насоса // Труды БГТУ. 2013. № 3: Химия и технология неорган. в-в. С. 161–165.

7. Кирпичев М. В., Михеев М. А., Эйгенсон Л. С. Теплопередача. М.: Государственное энергетическое издательство, 1940. 292 с.

8. Строительная климатология: СНБ 2.04.02-2000. Введ. 01.07.01. Минск: Стройтехнорм, 2001. 37 с.

9. Строительная теплотехника. Строительные нормы проектирования: ТКП 45-2.04-43-2006. Введ. 01.07.07. Минск: Стройтехнорм, 2007. 40 с.

10. Андреевский А. К. Отопление / под ред. М. И. Курпана. Минск: Выш. школа, 1982. 364 с.

11. Справочник по климату СССР / Глав. упр. гидрометеорол. службы при Совете Министров СССР, Упр. гидрометеорол. службы БССР, Минская гидрометеорол. обсерватория. 1965. Вып. 7: Белорусская СССР. Ч. 2: Температура воздуха и почвы. 246 с.

12. Huber A., Schuler O. Berechnungsmodul für Erdwärmesonden. Zürich: Bundesamt für Energiewirtschaft, 1997. 74 p.

13. Kasuda T., Archenbach R. Earth temperature and thermal diffusivity at selected stations in the United States // ASHRAE Transactions. 1965. Vol. 71, No. 1. P. 61–75.

Received 21.02.2014