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THE NUMERICAL SIMULATION OF UTILIZER OF SOIL HEAT AND HEAT RECEIVER IN JOINT OPERATION

The numerical model of the joint work of borehole heat exchanger and evaporator of the heat pump is developed. The design of the evaporator is shelltube heat exchanger with segmental baffles with a boiling in the U-shaped tubes. The design of ground heat exchanger is heat exchanger with two U-shaped tubes, which are arranged in a vertical borehole. The effect on the system temperature of soil, the multiplicity of circulation in the evaporator heat exchanger, the filler of borehole is modeled by the developed model.

Introduction. The development of renewable energy sources uses as a promising direction in energy saving technology includes low-potential heat utilization of the soil upper layers.

Despite long-term experience in operating these plants in developed countries, there is not quite a reliable method for their design, which prevents from using heat pumps widely in the Republic of Belarus.

The main disadvantage of the mathematical models upon which modern methods of heat pump design for low-grade heat utilization of soil are based is a simplified analysis of one of the elements of the system: ground heat exchanger (GHE) or heat pump (HP) which gives rise to doubt in data verification.

The solution to this problem is the development of a complex mathematical model of the systems, which includes joint operation of ground heat exchanger, heat pump unit and heat consumers, through which it will be possible to design, refine and evaluate the efficiency of heating systems using low-grade heat of the soil.

Description of the object of the study. The object of mathematical modeling is a system consisting of the heat pump evaporator and several GHE. The evaporator is a shell-and-tube heat exchanger with segmental baffles with boiling in horizontal U-shaped tubes. The heat rejected to the GHE from the ground, passed the intermediate refrigerant in the evaporator to the boiling refrigerant R134a. As the coolant an aqueous solution of ethylene glycol is used as the coolant. Fig. 1 shows the system.

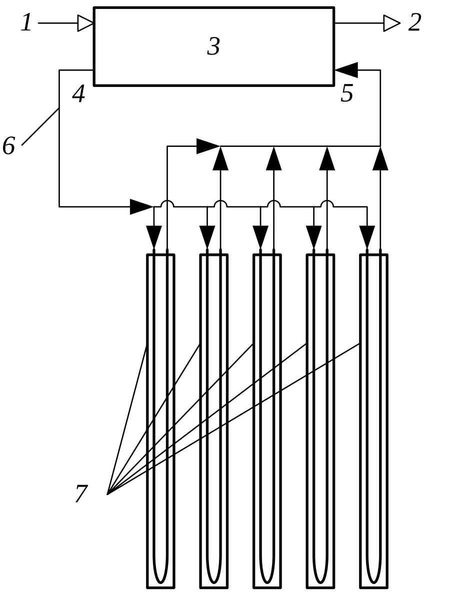

Fig. 1. Diagram of the system:
1 – refrigerant input; *2* – refrigerant output; *3* – heat pump evaporator; *4* – coolant from the evaporator output; *5* – coolant input in the evaporator; *6* – coolant circuit; *7* – ground heat exchangers

Mathematical description of the GHE. The article deals with GHE consisting of two U-shaped plastic tubes 32×3 mm in diameter, located in the borehole. There are 5 heat exchangers; diameter of borehole is 120 mm. To improve the thermal contact with the soil the space between the borehole and the heat exchanger tubes pipe was filled with heat-conductive suspension based on bentonite and cement [1]. Fig. 2 shows the cross section of the heat exchanger.

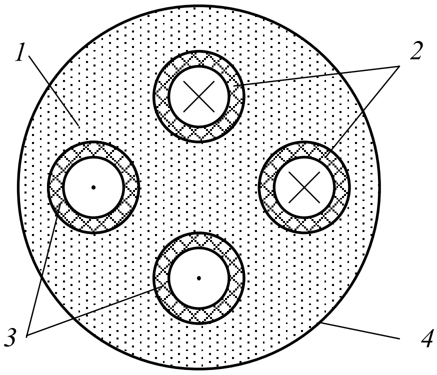

Fig. 2. The cross-section of the ground heat exchanger:
1 – well filler; *2* – descending pipes; *3* – ascending pipes; *4* – borehole wall

Based on the fact that most of the ground heat exchanger is located below the seasonal fluctuations of soil temperature, that is, in areas where the soil temperature varies with depth slightly, it is assumed that the wall temperature is constant over the depth of the well. In this case, for the mathematical description of a quasi-stationary process of heat transfer in the GHE was used the general solution of Eskilson and Claesson [2], which was modified for the boundary condition of the 1-st kind on the wall of the borehole according to which the temperature of the coolant at the outlet of the GHE is:

$$t' = \frac{\cosh(\gamma L_{Bor}) - \sinh(\gamma L_{Bor})\left(\delta - \frac{\beta_{12}}{\gamma}\right)}{\cosh(\gamma L_{Bor}) + \sinh(\gamma L_{Bor})\left(\delta - \frac{\beta_{12}}{\gamma}\right)} t'' + \frac{(\beta_2 + \beta_1)\sinh(\gamma L_{Bor})}{\gamma \cosh(\gamma L_{Bor}) + (\gamma\delta - \beta_{12})\sinh(\gamma L_{Bor})} t_{soil}, \quad (1)$$

where L_{Bor} – GHE well depth, m; t'' – the temperature of the coolant at the entrance to the GHE, ºC; t_{soil} – the average temperature of the borehole wall (temperature of the adjoining soil), ºC.

The equation (1) includes the following factors:

$$\beta_1 = \beta_2 = \frac{1}{R_1^{\Delta} G_1 c}, \quad (2)$$

$$\beta_{12} = \frac{1}{R_{12}^{\Delta} G_1 c}, \quad (3)$$

$$\gamma = \sqrt{\beta_1^2 + 2\beta_{12}\beta_1}, \quad (4)$$

$$\delta = \frac{1}{\gamma}(\beta_{12} + \beta_1), \quad (5)$$

where R_1^{Δ}, R_{12}^{Δ} – resistance to heat transfer between the coolant flow and the borehole wall, between the ascending and descending coolant flow respectively, m · K/W; G_1 – mass flow rate of the coolant, kg/s; c – heat capacity of the coolant, J/(kg · K).

Calculation of heat transfer coefficients. When boiling heat transfer coefficient of moving refrigerant within the evaporator tubes for vapor mass fraction area from 0 to 1 was determined according to the principle of superposition of macro- and micro-convective heat transfer coefficient method described in [3].

In this case, the heat transfer coefficient is:

$$\alpha_2(x) = \sqrt[3]{\alpha(x)_k^3 + \alpha(x)_B^3}, \quad (6)$$

where $\alpha(x)_k$ – macro-convection heat transfer coefficient, W/(m^2 · K); $\alpha(x)_B$ – micro-convection heat transfer coefficient, W/(m^2 · K); x – vapor mass fraction.

The correlations for the components of equation (6) have the form

$$\alpha(x)_k = \\ = \alpha_{LO}\left\{(1-x)^{0.01} A^{-2.2} + x^{0.01} B^{-2}\right\}^{-0.5}, \quad (7)$$

$$A = (1-x) + 1.2x^{0.4}\left(\frac{\rho'}{\rho''}\right)^{0.37}, \quad (8)$$

$$B = \frac{\alpha_{GO}}{\alpha_{LO}}\left(1 + 8\cdot(1-x)^{0.7}\left(\frac{\rho'}{\rho''}\right)^{0.67}\right), \quad (9)$$

$$\frac{\alpha(x)_B}{\alpha_0} = C_F\left(\frac{q}{q_0}\right)^{n(p^*)} F_{p*} F_d F_W F_{m,x}, \quad (10)$$

where α_{LO} – heat transfer coefficient with vapor mass fraction is 1, W/(m^2 · K); ρ', ρ'' – the density of the liquid and vapor flux, respectively, kg/m^3; α_{GO} – heat transfer coefficient with vapor mass fraction equal to 0, W/(m^2 · K); α_0 – normalized heat transfer coefficient (for R134a – 3,500 W/(m$_2$ · K)); C_F – coefficient taking into account the properties of the refrigerant; q – specific heat flux, W/m^2; q_0 – normalized specific heat flux (for R134a – 20,000 W/m^2); $n(p^*)$ – reduced pressure and flow regime of the refrigerant correction; F_{p*} – reduced pressure correction; F_d – the diameter of the pipe correction; F_W – pipe roughness correction; $F_{m,x}$ – the mass velocity m and vapor mass fraction correction.

Closing relations of system of equations are valid for horizontal pipes and according to [3] have a maximum deviation from the experimental data by 30%.

Average heat transfer coefficient in the shell and tube heat exchanger shell side with segmental baffles is calculated according to the method [3], with the flow diagram of media and coolant leakages.

Solution method. The system of equations (1)–(10) is complemented by the relations that connect the evaporator and the GHE:

$$Q = F\alpha_1(t_{1av} - t_{w1}), \quad (11)$$

$$Q = G_1 c(t' - t''), \quad (12)$$

$$G_2 r dx = \frac{(t_{w1} - t_s)dF}{\frac{d_{ext}}{\alpha_2(x) d_{int}} + \frac{d_{ext}}{2\lambda_w}\ln\frac{d_{ext}}{d_{int}}}, \quad (13)$$

$$Q = G_2 r(x' - x''), \quad (14)$$

where Q – heat flux, W; F – exterior surface of the evaporator tubes, m^2; α_1 – mean heat transfer coefficient of coolant in the evaporator shell side, W/(m^2 · K); t_{1av} – average coolant temperature, ºC; t_{w1} – mean temperature of the evaporator tubes exterior surface, ºC; G_2 – mass flow of the refrigerant, kg/s; r – enthalpy of evaporation, J/kg; t_s – the boiling temperature of the refrigerant, ºC; d_{ext} – external diameter of the evaporator pipes, m; $\alpha_2(x)$ – a local boiling heat transfer coefficient, W/(m^2 · K); d_{int} – internal diameter of the evaporator pipes, m; λ_w – thermal conductivity of the evaporator pipes material, W/(m · K); x' – vapor mass fraction of the refrigerant at the evaporator inlet; x'' – vapor mass fraction of the refrigerant at the evaporator outlet.

The mean temperature of the external wall of the evaporator pipes is taken as a boundary condition. The refrigerant vapor mass fraction and its mass flow are specified at the pipes inlet. Vapor mass fraction of the refrigerant at the evaporator outlet is calculated by solving of the differential equation (13) by the finite difference method with the explicit scheme. The notation of the equation for the refrigerant vapor mass fraction change in finite differences as follows:

$$G_2 r\left(x_i - x_{i-1}\right) = \frac{\left(t_{w1} - t_s\right) n\pi d_{int}\Delta z}{\dfrac{d_{ext}}{\alpha_2\left(x_{i-1}\right) d_{int}} + \dfrac{d_{ext}}{2\lambda_w}\ln\dfrac{d_{ext}}{d_{int}}}, \quad (15)$$

where i – node number of finite difference mesh; n – number of the evaporator pipes on one refrigerant stroke; Δz – size of finite-difference mesh, m.

The solution of the system of the equations (1)–(14) can not be obtained analytically. In this study, the system is solved numerically by the Gauss-Newton method [4].

The initial data for solving the problem were refrigerant boiling temperature, the temperature of the borehole wall, corresponding to the adjacent soil temperature, mass flows of refrigerant and coolant, vapor mass fraction of the refrigerant at the evaporator inlet.

Results of the numerical experiment. When used a mathematical developed model for the prototype system, a series of numerical experiments were carried out.

We take into account the boiling temperature of the refrigerant subject to the temperature difference in the evaporator 3–6ºC, mass flow 0.08 kg/s. Vapor mass fraction at the evaporator inlet 0.15. Thermal conductivity of the borehole filler was assumed to be 2.3 W/(m · K), which was a part of its range of possible values for the data of the work [1].

A characteristic feature of the heat pump for the utilization of low-potential heat of soil previously established in works [5–7] is a gradual temperature decrease of the GHE borehole wall due to the heat rejection from the adjacent soil mass, which affects the parameters of the system, in the first place, this leads to heat flow reduction.

According to work [2], the temperature of the soil below the seasonal temperature fluctuations corresponds to an average annual temperature of the soil surface, which is for the Republic of Belarus is 6–8ºC subject to the locality according to [8]. In the calculations, this temperature corresponds to the temperature of the borehole wall at the beginning of the heating season.

Fig. 3 shows the change of vapor mass fraction at the evaporator outlet and the evaporator heat flux depending on the temperature of the borehole wall. Heat flux is reduced by about 63% by the temperature decrease from 8 to 2 ºC.

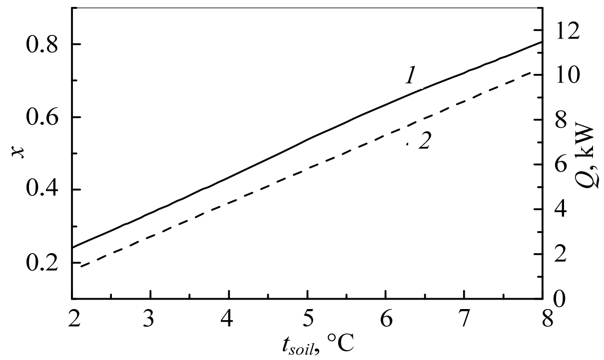

Fig. 3. Changing of vapor mass fraction and heat flux depending on the temperature of the soil at t_s = 0ºC; m = 79.6 kg/(m^2 · s):
1 – vapor mass fraction at the evaporator outlet;
2 – heat flux

As can be seen from Fig. 3, decrease of soil temperature results in decreasing of heat flux and refrigerant vapor mass fraction at the evaporator outlet, which can be compensated by decrease of refrigerant boiling temperature.

Fig. 4 shows the change of the refrigerant boiling temperature and the corresponding saturation pressure from the soil temperature. Such a distribution of parameters allows maintaining a constant heat flux of the evaporator.

It turned out that the local boiling heat transfer coefficient and vapor mass fraction change irregularly along the length of pipes (Fig. 5). Microconvection is the main here.

It is related to the dependence of the heat transfer coefficient of the vapor mass fraction, and on the flow regime varied along the length of the pipe. Thus, when steam quality is 0.15–0.6 and 0.8–1.0 one can observe wave flow regime and when steam quality is 0.6–0.8 – annular flow.

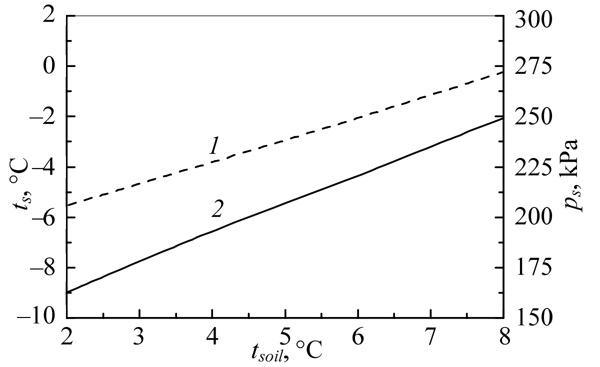

Fig. 4. Changing of the boiling temperature and the saturation pressure depending on the soil temperature at Q = 13.8 kW; x'' = 1; m = 79.6 kg/(m^2 · s):
1 – saturation pressure; *2* – soil temperature

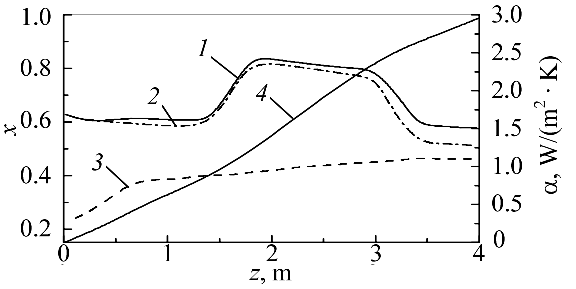

Fig. 5. Changing of the heat transfer coefficient and vapor mass fraction along the length of the pipes:
1 – the local heat transfer coefficient;
2 – the micro-component of the heat transfer coefficient;
3 – the macro-component of the heat transfer oefficient; *4* – the local vapor mass fraction

Process-related parameters of the system strongly depend on the properties of the used materials. Thermal conductivity of the borehole filler determines the thermal resistance of the well, which affects the operation of the system as a whole. Thermal conductivity of bentonite-cement grouting used as borehole fillers according to the test data [1] depending on the composition can range from 0.7 to 2.4 W/(m · K). Fig. 6 shows the change of the boiling temperature of refrigerant required to maintain it in a dry saturated state at the evaporator outlet (x'' = 1), with the thermal conductivity of the borehole filler material. As the thermal conductivity of the filler is increased. The behavior of the soil and boiling refrigerant temperature difference $\Delta t_{soil/s}$ has the opposite tendency – it is reduced. This will affect the operation of the system in whole and will lead to a decrease in the coefficient of performance of the heat pump.

The effect of the multiplicity of circulation in the evaporator system was studied. Fig. 7 shows the behavior of the heat flux and vapor mass fraction of refrigerant at the evaporator outlet when the multiplicity of circulation changes. While increasing multiplicity of circulation from 5 to 15 the evaporator heat flux increases by approximately 40%. This shows that the change in the multiplicity of circulation can be used as a way to control the heat flow of the system.

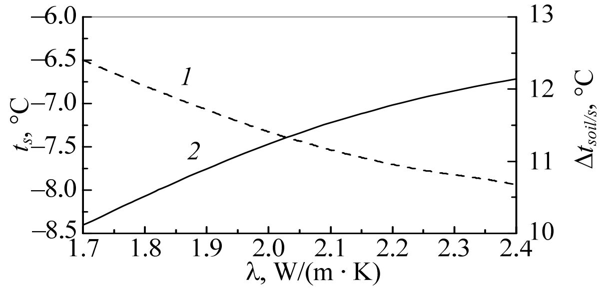

Fig. 6. The effect of the borehole filler thermal conductivity on the boiling temperature of the refrigerant and the temperature difference $\Delta t_{soil/s}$ at t_{soil} = 4ºC; Q = 13.8 kW; x'' = 1; m = 79.6 kg/(m^2 · s):
1 – $\Delta t_{soil/s}$; *2* – the boiling point of the refrigerant

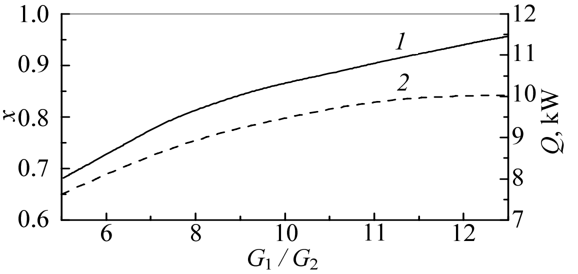

Fig. 7. The effect of multiplicity of circulation on the heat flux and refrigerant vapor mass fraction at the evaporator outlet at t_{soil} = 8ºC; t_s = 0ºC; m = 79.6 kg/(m^2 · s):
1 – heat flux; *2* – vapor mass fraction of the refrigerant at the evaporator outlet

Conclusion. A mathematical model was developed for the study of joint operation of the utilizer of the soil heat and the heat pump evaporator. The numerical simulations revealed that under these conditions the heat flux considerably depends on the temperature of the soil. Temperature reduction from 8 to 2ºC causes the decrease of the heat flux approximately by 63%. When boiling along the pipe length the degree of irregularity was revealed through the change of vapor mass fraction and refrigerant flow regime. The effect of borehole filler thermal conductivity of GHE on the system operation is shown. For the conditions above in the range of its values 1.7–2.4 W/(m · K) the boiling temperature required to maintain the refrigerant at the evaporator outlet in a dry saturated state varied within –8.4 to –6.8ºC. It was established that with increasing of circulation multiplicity factor in the evaporator from 5 to 15 heat flow increases up to about 40%.

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