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METHOD OF CALCULATION AND ANALYSIS OF JOINT OPERATION OF CIRCUIT LOOP OF GROUND HEAT EXCHANGERS AND HEAT PUMP

The method of heat calculation of the joint work of borehole heat exchangers and heat pump circuits in quasi-stationary regime is developed. The effect of the heat pump efficiency of heat exchangers design values, refrigerant superheating in evaporator, temperatures of heat source and heat consumer were analyzed on the base of the developed method.

Introduction. One of the problems of modern society is the depletion of fossil energy resources and the necessity to find new ways to produce energy. One of the solutions of this problem is to use of renewable energy sources. This problem is relevant for the Republic of Belarus.

National Program of development of local and renewable energy sources in 2011-2015 [1] involves putting into operation heat pumps (HP) of the total thermal capacity of about 8.9 MW, including compression heat pump for recycling low-potential ground heat by 2015.

Economic feasibility of using such facilities in heat supply systems is determined by their energy efficiency. The indication of energy efficiency of HP is coefficient of performance φ. To achieve high coefficient of performance of HP it is necessary at the design stage to take into account a number of factors, such as structural features of heat exchange equipment, refrigerant superheat in the condenser, environment temperature, operating conditions of heat consumer.

This study seeks to develop a method for calculating the joint operation of circuit loop of ground heat exchangers and HP as part of the heating system considering the relevant size of the tubes of the condenser, the refrigerant superheat in the condenser, source and thermal consumer temperature. Research is conducted by numerical simulation.

Description of the investigated object. The investigated system consists of circuit loop HP and several borehole heat exchangers (BHE) (Fig. 1).

The system operates as follows. In the first heat loop circuit the heat rejected to BHE from soil is passed by the intermediate heat carrier in the evaporator to the boiling refrigerant. In the second loop circuit the refrigerant in a superheated state is passed after the evaporator to the compressor where it is compressed to a pressure of the required corresponding saturation temperature.

Refrigerant vapor passes into the condenser, where it is cooled and condensed. After the condenser the liquid refrigerant enters the thermostatic expansion valve, in which the refrigerant is expanded to the evaporator pressure. The permanent refrigerant superheat is maintained in the evaporator output.

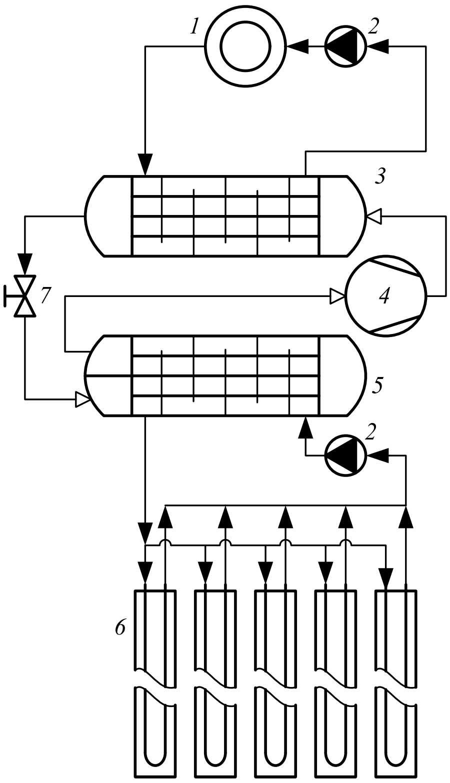

Fig.1 Schematic diagram of the heating system based on utilization low-potential soil heat:
1 – thermal energy consumer; *2* – pump;
3 – condenser, *4* – compressor; *5* – evaporator;
6 – BHE; *7* – thermostatic expansion valve

Mathematical description of the joint operations of BHE and evaporator HP. To determine the parameters of the joint work of the evaporator HP and several BHE, the method proposed in [2] and [3] is used. According to this method the common solution of the problem of heat transfer in BHE for variable temperature conditions of the surface of the well [4] leads to specific solution for constant temperature. In this case, heat carrier temperature at the output of BHE is determined by the formula:

$$t_1'' = At_1' + Bt_b, \quad (1)$$

where A, B – coefficients; t_1' – temperature of the heat carrier in input of BHE; °C; t_b – the wall temperature of the borehole (the temperature of the adjacent soil), °C.

The coefficients A and B in the equation (1) depend on the thermo-physical properties of the heat carrier and materials of BHE, geometrical dimensions, flow regime of the heat carrier. The equation (1) is supplemented by the equations for the evaporator:

$$Q = G_1 c_1 \left(t_1' - t_1''\right), \tag{2}$$

$$Q = G_2 \left(h_2'' - h_2'\right), \tag{3}$$

$$Q = F\alpha_1 \left(t_{1m} - t_{w1}\right), \tag{4}$$

$$G_2 r dx = \frac{\left(t_{w1} - t_s\right) dF_b}{\dfrac{d_o}{\alpha_2 d_i} + \dfrac{d_o}{2\lambda_w} \ln \dfrac{d_o}{d_i}}, \tag{5}$$

$$G_2 c_2 dt_2 = \frac{\left(t_{w1} - t_2\right) dF_{sh}}{\dfrac{d_o}{\alpha_2 d_i} + \dfrac{d_o}{2\lambda_w} \ln \dfrac{d_o}{d_i}}, \tag{6}$$

where Q – heat flux, W; G_1 – mass flux in the intermediate heat carrier, kg/s ; c_1 – heating capacity of heat carrier, J/(kg K); G_2 – refrigerant mass flux, kg/s; h_2'', h_2' – enthalpy of the refrigerant at evaporator input and output, J/kg; $F = F_b + F_{sh}$ – the area of the outer surface of the evaporator tubes, m^2; F_b – the surface area of the boiling section of the tube, m^2, F_{sh} – the surface area of the superheat section of the tubes, m^2; α_1 – mean heat transfer coefficient of the heat carrier in the annulus, W/(m^2 K); t_{1m} – mean temperature of the heat carrier, °C; t_{w1} – the mean temperature of the outer surface of the evaporator tube, °C; r –refrigerant specific heat of evaporation, J/kg; x – mass flux of the void fraction; t_s – boiling temperature of the refrigerant, °C; d_o – the outer diameter of tubes, m; α_2 – local coefficient of the refrigerant heat transfer, W/(m^2 K); d_i – internal diameter of tubes, m; λ_w – thermal conductivity of tubes, W/(m·K); c_2 – refrigerant thermal capacity, J/(kg·K); t_2 – refrigerant temperature, ºC.

Mean heat transfer coefficient of the heat carrier in the annulus of the shell tube evaporator with segment partition is calculated by the method [5] taking into account the current scheme of the operating medium and coolant leakages. Equation (5) is true for refrigerant boiling section, and the equation (6) is true for superheat section.

Local heat transfer coefficient of refrigerant inside the evaporator tubes for boiling section is determined by the principle of superposition of macro- and micro-convectional coefficient of heat transfer that presented in the method [5]. In this case, the thermal transfer coefficient is

$$\alpha_2 = \sqrt[3]{\alpha_c^3 + \alpha_b^3}, \tag{7}$$

where α_c – macro-convectional heat transfer coefficient, W/(m^2·K); α_b – micro-convectional heat transfer coefficient, W/(m^2 K).

Flow operation conditions of refrigerant vapor-liquid tubes and equation constituents (7) are determined by the method [5]. Refrigerant local heat transfer coefficient for superheat section is calculated by similarity equations of for the forced convection in channels [5].

Mathematical description heat pump loop circuit. It is assumed that a refrigerant in HP condenses completely. Based on the isenthalpic flash, the refrigerant vapor-liquid content is determined at the output of the expansion valve, respectively, and at the input to the evaporator HP

$$x_2' = \frac{h_3'' - h_2^{liq}}{r_2}, \tag{8}$$

where h''_3 – enthalpy at the condenser output, J/kg; h^{liq}_2 – enthalpy of saturated liquid under pressure in the evaporator, J/kg; r_2 –boiling heat under pressure, J/kg.

Refrigerant mass flux

$$G_2 = \lambda V_h \rho_2'', \tag{9}$$

where λ – compressor charge efficiency; V_h – volume taken by the compressor piston per unit of time, m^3/s; ρ''_2 – the refrigerant vapor density at the evaporator output, kg/m^3.

The charge efficiency of the compressor:

$$\lambda = a\frac{p_c}{p_e} + b, \tag{10}$$

where a, b – coefficients characteristic of a specific model of the compressor; p_c – the condenser pressure, Pa ; p_e – evaporator pressure, Pa.

Method of common solution. The initial data for the calculation are the thermo-physical properties of materials and media, the temperature of borehole surface, degree of superheat at the evaporator, condensation temperature.

The system of equations (1)–(10) is solved numerically by Newton's iterative method. Equations (5) and (6) were solved by finite difference method.

Further, to determine the power consumption of the compressor and heating effect of HP it is necessary to calculate the refrigerant parameters at input and output of the condenser.

Enthalpy of the refrigerant at the compressor output

$$h_3' = h_2'' + \frac{h_{3i}' - h_2''}{\eta_i}, \tag{11}$$

where h'_{3i} – enthalpy of the refrigerant at the end of adiabatic compression in the compressor, J/kg; η_i – internal indicator compressor efficiency.

Compressor electric power consumption:

$$N = G_2 \frac{h_3' - h_2''}{\eta_{em}}, \qquad (12)$$

where η_{em} – compressor electromechanical efficiency (assumed to be equal to 0.85).

Heating effect of HP

$$Q_c = G_2 \left(h_3' - h_3'' \right). \qquad (13)$$

Coefficient of performance:

$$\varepsilon = \frac{Q_c}{N}. \qquad (14)$$

The results of numerical modelling. On the basis of this method, a numerical study of vapor-compression HP heating supply system based on recycling of low-potential heat of the ground is conducted.

The amount of BHE was 5, its length – 50 m, the diameter of boreholes – 160 mm, the type of BHE – with two U-shaped polyethylene tubes 32×3 mm, the diagonal distance between tubes – 80 mm. Thermal conductivity of tube is assumed to be 0.38 W/(m·K), the thermal conductivity of the borehole grouting – 2.3 W/(m·K). Total mass flux of heat carrier of loop circuit of ground heat exchangers is 0.8 kg/s. The heat carrier of loop circuit of BHE is aqueous solution of ethylene glycol. Thermo-physical properties of heat transfer of soil loop circuit are taken from [6].

The evaporator is tube-in-shell heat exchanger with segmental baffles with boiling in the U-shaped tubes; the tubes are segmented on the tops of the right triangles. The amount of tubes is 40. The tube-in-shell is thermal insulated.

R134a is used as a refrigerant; its thermal properties are in a state of saturation, taken by reference data [5]. Specific volume, enthalpy and entropy in the superheated section are calculated using the state equation of R134a, proposed in [7].

Compressor with 0.00911 m^3/s volume taken by the compressor piston per unit of time is used in the heat pump.

Influence of evaporator design parameters. The influence of the length and diameter of the evaporator tubes on heat supply and HP coefficient of performance. For steel tubes of three sizes 10×1, 12×1 and 16×1.5 mm, active length of evaporator tubes l varies from 0.5 to 2 m. The amount of segmental baffles is chosen so that the distance between them ranges insignificantly (83-95 mm). In this case the heat transfer coefficient in the annulus ranges 3000–3500 W/(m^2·K). The average borehole wall temperature is assumed to be equal 5°C, condensing temperature – 50°C. Superheat of the refrigerant in the evaporator is maintained at 5°C. Fig. 2 shows the change of φ depending on the length of the evaporator tubes.

Fig. 2 shows that φ increases by 45%, when tube length increases from 0.5 to 2 m. The maximum coefficient of performance corresponds to the tubes 10×1 mm; Q_c changes from 7.1 to 11.2 kW (58% increase), and the character of its change is similar to the character of φ. Decrease in the φ and Q_c for larger diameter tubes is due to the fact that increasing tube diameter decreases the refrigerant flow rate, which leads to a change of vapor-liquid mixture flow regime. Thus, in the tubes 10×1 mm wave flow regime turns into annular one. In the tubes 16×1.5 mm there is a stratified flow regime characterized by lower values of the refrigerant heat transfer coefficient and incomplete wetting of the inner surface of the tube. Decrease in the heat transfer coefficient in the evaporator tubes is accompanied by an increase in temperature difference, required to maintain the refrigerant in a superheated state with a given value of overheating at the output.

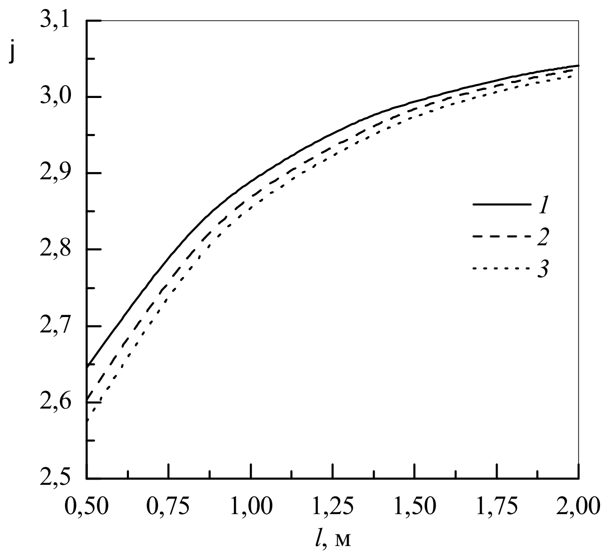

Fig. 2. Coefficient of performance change depending on the length of the evaporator tubes of *1* – 10 mm; *2* – 12×1 mm; *3* – 16×1,5 mm sizes

Influence of refrigerant superheat. To work reliably, the reciprocating compressor refrigerant must be maintained in overheat condition at the output. To estimate the influence of the magnitude of the refrigerant superheat in the evaporator HP on φ and Q_c the numerical experiment was conducted. The average surface temperature of the wells and the condensation temperature were assumed to be the same as in the previous study. Length of the tube of evaporator was taken to be 1.5 m, the amount of segmental baffles – 15, the evaporator tubes – 10×1 mm. The overheating varied from 2 to 15°C. Fig. 3 shows the change in Q_c depending on the magnitude of refrigerant superheat at the evaporator output.

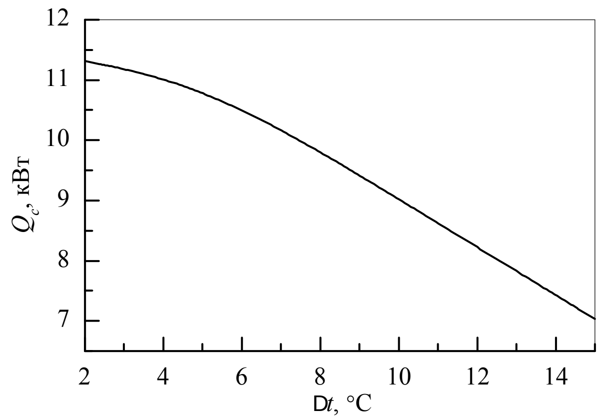

Fig. 3 Change of the heating capacity Q_c depending on refrigerant superheat Δt at evaporator output

Fig. 3, shows Q_c decrease by 39% (from 11.5 to 7 kW) when superheat increases from 2 to 15°C. Q_c decrease corresponds to j decrease by 13% (from 3 to 2.6). The nature of the j change is similar to that of Q_c changes and, it is due to the fact that with increasing refrigerant superheat, the boiling area decreases and the average refrigerant thermal transfer at the output decreases. This is compensated by increasing the temperature difference in the evaporator by reducing the pressure p_e, which increases the degree of compression and corresponding decrease of j and Q_c.

Effect of temperature receiver and the source of heat. One of the factors that significantly affect the operation of HP is the temperature of sources and receivers of heat. Decrease in the temperature difference of the consumer (heat supply system) and heat source (soil mass) is accompanied by increase of j and Q_c. The developed numerical model allows one to quantitatively evaluate the dependence of j and Q_c on this factor. In this problem, the temperature of the heat consumer determines the temperature of the refrigerant condensing in the condenser, which is given above. The temperature heat source corresponds to the average surface temperature of BHE wells.

The following numerical experiment was conducted. Initial data: tube length – 2 m, the number of segmental blocks – 20; tube evaporator – 10×1 mm; superheat at the evaporator – 5°C. Surface temperature of BHE well (soil) ranged from –5 to 11°C. This range of soil temperatures is characteristic for the considered systems and is determined by the gradual decrease in temperature due to cooling [8, 9]. Condensing temperature changed from 30 to 50°C. The results of calculation are shown in Fig. 4 and 5.

Fig. 4 and 5 show a substantial dependence of the HP temperature parameters of the TH on the temperature of well wall (soil). When soil temperature decreases from 11 to –5°C, Qc decreases by 45-50%, j – for 21–27%, depending on the condensation temperature. When the condensation temperature decreases from 50 to 30°C, Qc increases by 14-26%, and j – for 42–55%.

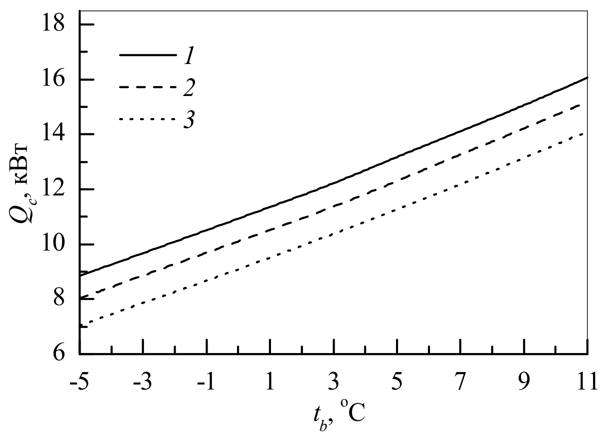

Fig. 4. Change of Q_c depending on t_b at the condensation temperatures:
1 – 30°C, *2* – 40°C, *3* – 50°C

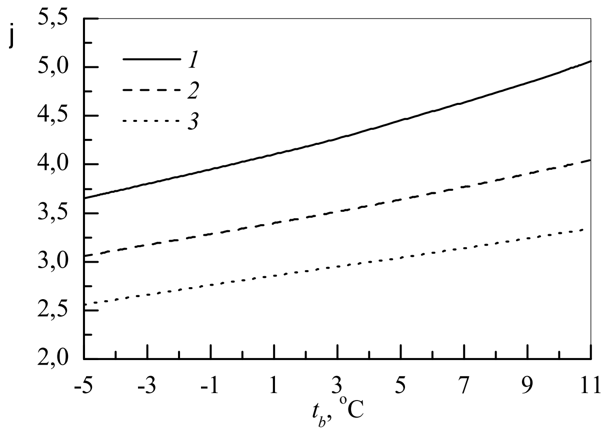

Fig. 5. Change of j depending on t_b at the condensation temperatures:
1 – 30°C, *2* – 40°C, *3* – 50°C

Conclusion. The developed method for calculating the joint work of BHE circuits and HP compression, allows carrying out numerical studies evaluating the impact of design and regime parameters on the coefficient of performance and heat supply.

In the given conditions, when the length of the evaporator tubes changes from 0.5 to 2 m, the coefficient of performance and heat supply increase for 45% and 58%, respectively. Increase of the tube diameter from 10 to 16 mm resulted in a slight decrease in the HP energy values. When the refrigerant superheat increases from 2 to 15°C, coefficient of performance and heat supply decrease by 13 and 39%, respectively. The soil temperature decrease from 11 to –5°C results in a decrease of heating capacity by 45–50% and coefficient of performance decrease by 21–27%. When the condensing temperature decreases from 50 to

30°C, it causes an increase in heat supply to 14–26%, and coefficient of performance to 42–55%. The data should be considered when designing and evaluating the energy efficiency of heating systems with the soil heat pumps.

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