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The Analysis of Operating Conditions of Geothermal Heat Pump Units

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ABSTRACT

Thermodynamic performance of successively counterflow, multistage and combined models of heat pump units with regard to their utilization conditions in geothermal heating systems is under study. The numerical simulations of static operating conditions for successively counterflow and multistage schemes of heat pump units for geothermal heating systems are carried out. The results of exergy analysis for different technologic schemes of high-temperature heat pump units are presented.

1. INTRODUCTION

Geothermal waters are one of the promising sources of thermal energy for heating by means of heat pump units (HPUs), e. g. Ogurechnikov (2005).

There are some schemes of geothermal heating system, e. g. Rozenfeld and Serdakov (1967), which provide for HPU work in bivalent operation conditions. In this case HPUs heat some of «return» water from a heating system up to the required temperature, the rest part serves as a source of low-potential heat for HPUs. The lower the specified temperature in a heating system is, the more the share of heat utilization of geothermal waters in that scheme is. As a rule a medium-temperature refrigerant is used as working fluid in HPUs.

With utilization of high-temperature single-component refrigerants or nonazeotropic mixture on their basis as working fluids, it is possible to realize monovalent operation conditions of HPU work with the direct conversion of low- potential heat from geothermal water. The peculiar feature of utilization of such a scheme lies in the possibility to reduce the consumption of geothermal water through the selection of optimal cooling temperature, e .g. Kaminsky (2007).

It is possible to expand the work range of geothermal heating systems with HPUs by means of connection of several units in a successively counterflow scheme. In such a scheme water heating in a condenser and cooling of geothermal water in an evaporator are realized in stages, besides every successive cycle happens in a higher interval space of the refrigerant evaporation and condensation temperatures. Thus, in specified temperature limits of the basic working cycle of HPUs the approximation to the Lorenz cycle, with typical for this cycle anisothermic character of evaporation and condensation processes is reached. The efficiency of approximation to the Lorenz cycle is valued with the ratio of the actual conversion coefficient μ_Δ and the theoretical μ and depends on the number of independent units, e. g. Heinrich, Najork and Nestler (1985) and, see e. g. Ostapenko (2006)

Preliminary calculations carried out for the refrigerants R114, R245fa, R141b showed that with the increase of number of units for more than three for the specified operation conditions we cannot find any noticeable changes of the value μ_Δ/μ_0. The utilization of several working fluids for each HPU in a scheme is possible.

A peculiar feature of successively counterflow schemes as an object of regulation is high sensitivity to the changes of input and output parameters of the heat carrier and geothermal water. Any deviations from specified operating conditions for one HPU lead to the changes of operative parameters of all the other units and consequently of the whole system.

Fig. 1 shows successively counterflow schemes of the geothermal HPU.

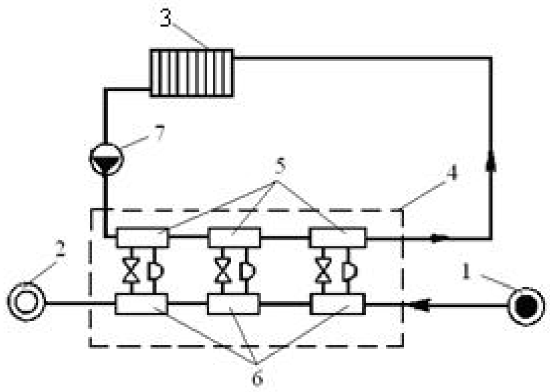

Figure 1: Principal successively counterflow HPU schemes for geothermal heating. 1- production well; 2- injection well; 3- heat consumer; 4- HPU; 5- condensers; 6- evaporators; 7- circulation pump.

Multistage schemes of HPUs, consisting of one common condenser and evaporator and several compressors connected successively, are less sensitive to the changes of input parameters.

Fig. 2 shows multistage of the geothermal HPU.

2. HEAT PUMP PERFORMANCE MODELING

Within the limits of this work it seemed rational to develop a calculation technique and on its basis to fulfill the analysis of operating conditions of geothermal HPUs with different connecting schemes of fundamental equipment elements.

The methods of mathematical modeling of HPU static characteristics are traditionally used to solve the task, e. g. Jin and Spilter (2002).

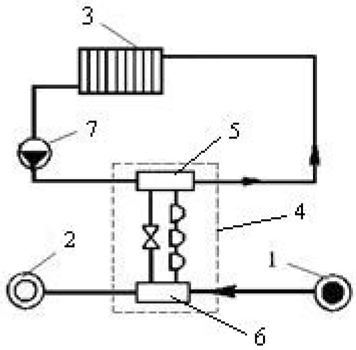

Figure 2: Principal multistage HPU schemes for geothermal heating. 1- production well; 2- injection well; 3- heat consumer; 4- HPU; 5- condenser; 6- evaporator; 7- circulation pump.

The calculation of HPU static characteristics provides for specification, as initial data, of the geometric parameters of evaporator, condenser, compressor, binding piping of basic equipment and also the initial temperatures of heat-carriers and their flow rates. For the successively counterflow schemes with more than two HPUs the calculation has one peculiar feature.

In the scheme consisting of, for example, three units the temperature of inlet «return» water in the first HPU in water heating and the temperature of inlet geothermal water in the third HPU are known. The outlet temperatures of heat-carrier and geothermal water in the second HPU which actually determine operating conditions of the rest units are not known. For multistage schemes connected successively according to the refrigerant, the parameters of inlet steam in the second and third compressors are unknown.

2.1 The Calculation of the Thermal Water Intake Performance

The well production rate of a hydrodynamically perfect well is found from the equation, e. g. Shterenlicht (1991)

$$G_{gwt} = 1{,}36k\frac{H_0^2 - h^2{}_w}{\lg\dfrac{R_w}{r_w}} \tag{1}$$

where k is the filtration coefficient, m/s; H_0 is the capacity of a water-bearing stratum, m; h_w is the depth of water in the well when there is no pumping out, m; r_w is the radius of a well, m; R_w is the range radius of a well, m.

With the start of pumping out the water level in the well and around it becomes lower. The distance R_w beyond the limits of which there is no change of head pressure can be found from

$$R_w = 3000S_0/k^2 \tag{2}$$

where S_0 is the lowering of water level during pumping out, m.

The equation (1) can also be used for the calculation of injection well.

Motion patterns of geothermal waters in a water-bearing stratum at work of a production well and absorption well are shown in Fig. 3 and Fig. 4.

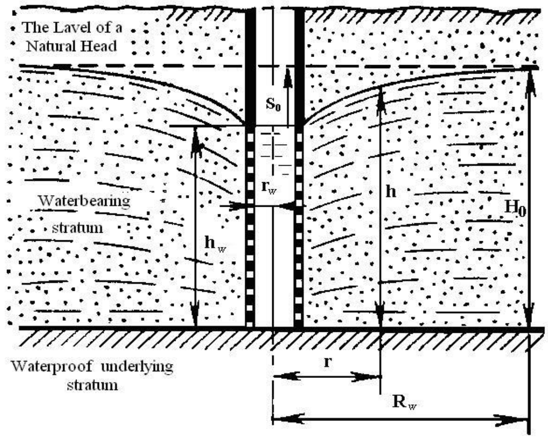

Figure 3: Patterns of geothermal waters inflow to a vertical production well.

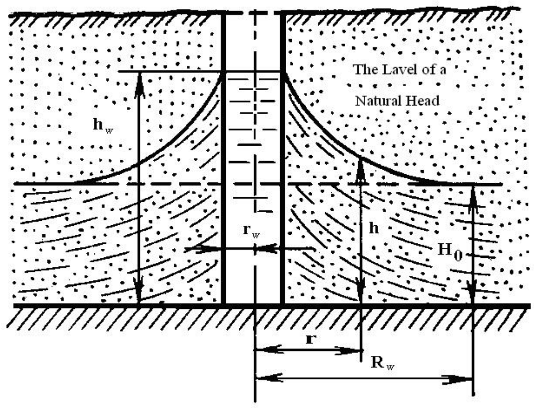

Figure 4: Patterns of geothermal waters outflow from a vertical injection well.

The weighted average temperature of thermal water intake is assumed as specified (rated) temperature of geothermal water t'_{gw} received at thermal water intake which comprises two and more wells

$$t'_{gw} = \frac{t'_{gw1}G_{gw1} + t'_{gw2}G_{gw2} + ... + t'_{gwk}G_{gwk}}{G_{gw1} + G_{gw2} + ... + G_{gwk}} \tag{3}$$

where t'_{gw1}, t'_{gw2}, t'_{gwk} are the temperatures at wellheads, °C; G_{gw1}, G_{gw2}, G_{gwk} are the geothermal well production rates, kg/s.

2.2 The Calculation of the Heat Pump Performance

2.2.1 The technique of calculation of a successively counterflow geothermal HPUs connecting scheme

The heating capacity, withdrawn from geothermal water in HPU evaporator, and corresponding to it HPU cooling capacity are found from the equations

$$Q_0 = G_{gw}c_{gw}(t_{gw1} - t_0)\eta_0 \tag{4}$$

$$Q_0 = G_{rf} q_0 \quad (5)$$

where G_{rf} is the mass flow of a refrigerant in the cycle, kg/s; q_0 is the specific cooling capacity in the cycle, kJ/kg; t_0 is the evaporation temperature of a working fluid, °C; t_{gw1} is the inlet temperature of geothermal water into the evaporator; η_0 is the evaporator cooling coefficient.

At the calculation of operation parameters of the first HPU, according to the motion of a low-potential heat-carrier, the temperature $t_{gw1} = t'_{gw}$.

The evaporator cooling coefficient η_0 is

$$\eta_0 = 1 - e^{-\frac{k_e F_e}{G_{gw} c_{gw}}} \quad (6)$$

where $k_e F_e$ is the intensity of heat transfer in the evaporator, kW/k.

The intensity of heat transfer in the evaporator is found from the equation, e. g. Wang and Barnet (2000)

$$k_e F_e = \frac{1}{c_1 G_{gwt}^{-0.8} + c_2 Q_0^{-0.745} + c_3}. \quad (7)$$

The heating capacity withdrawn from the HPU condenser into the heating system is found from

$$Q_c = G_{rf} q_c \quad (8)$$

$$Q_c = G_{wt} c_{wt} (t_k - t_{wt1}) \eta_c \quad (9)$$

where q_c is the specific heating efficiency in the cycle, kJ/kg; t_c is the condensation temperature of a refrigerant, °C; t_{wt1} is the inlet water temperature in the condenser; G_{wt} is the water flow through HPU condensers, kg/s.

The cooling coefficient for the HPU condenser η_c is defined from

$$\eta_c = 1 - e^{-\frac{k_c F_c}{G_{wt} c_{wt}}} \quad (10)$$

where $k_c F_c$ is the intensity of heat transfer in the condenser, kW/k.

The intensity of heat transfer in the condenser is found from the equation, e. g. Kempiak and Crawford (1992)

$$k_c F_c = \frac{1}{c_4 G_{wt}^{-0.5} + c_5 G_{rf} + c_6} \quad (11)$$

The evaporation and condensation temperatures in the second and following approximations are calculated from the equations:

$$t_0 = t_{gw1} - \frac{Q_0}{G_{gw} c_{gw} \eta_0} \quad (12)$$

$$t_c = t_{wt1} + \frac{Q_c}{G_{wt} c_{wt} \eta_c} \quad (13)$$

As is known, e. g. Koshkin (1989), while changing operating conditions some part of HPU cooling capacity is spent on cooling of the evaporator (if t_0 is dropping). In the case, when HPU cooling capacity is increasing, t_0 is also increasing.

Changes of cooling capacity are analyzed as follows:

$$\Delta Q_0 = M_e c_e \Delta T_0 \quad (14)$$

where M_e and c_e are the mass and the heat capacity of the evaporator; ΔT_0 is the change of evaporation temperature.

The same equation can be used for the condenser

$$\Delta Q_c = M_c c_c \Delta T_c \quad (15)$$

where M_c and c_c are the mass and the heat capacity of the condenser; ΔT_c is the change of condensation temperature.

At the preselected theoretical displacement of the compressor V_{th} the mass flow of a refrigerant G_{rf} is equal to

$$G_{rf} = \frac{V_{th} \lambda}{\nu_1} \quad (16)$$

where ν_1 is the specific volume of refrigerant vapor at suction in the compressor, m^3/kg.

Under the condition of isoenthalpic nature of the process in the HPU expansion valve, the flow of a refrigerant through the throttle is defined from, e. g. Chernyvsky (2006)

$$G_{rf} = \varpi f \sqrt{2 p_{in} (P_c - P_0)} \quad (17)$$

where f is the flow area of the throttle, m^2; ω is the efflux coefficient; P_{in} is the fluid density before the HPU expansion valve, kg/m³; P_c and P_0 are the condensation and evaporation pressures, kPa.

The volumetric efficiency of the compressor can be presented as, e. g. Morozuk (2006)

$$\lambda = \lambda_c \lambda'_{wt} \quad (18)$$

where λ_c is the volumetric efficiency coefficient taking into account the effect of «dead space»

$$\lambda_c = 1 - c\left[\left(\frac{P_c}{P_0}\right)^{\frac{1}{m}} - 1\right] \quad (19)$$

where c is the relative value of «dead space» c = 0,015…0,05; m is the polytropic exponent of the reverse expansion from «dead space», m = 1,0…1,5.

The coefficient λ'_{wt} in the formula (18), taking into account the volumetric losses which were caused by vapor throttling in valves, vapor heating from the cylinder walls during the suction process, flow from the compression chamber to the suction chamber, is defined from the expression:

$$\lambda'_{wt} = \frac{T_0 + \Delta T_{\text{sup}}^{suc}}{\alpha T_c + \beta \Delta T_{\text{sup}}^{suc}} \quad (20)$$

where ΔT_{sup}^{suc} is the total superheating of a refrigerant at the suction in the compressor; T_0 and T_c is the evaporation temperature and condensation temperature of a refrigerant, K; α = 1,12; β = 0,5 are the coefficients taking into account the working fluid influence.

The compressor adiabatic capacity is

$$N_a = G_{rf} l_{cm} \tag{21}$$

where l_{cm} is the compression work in the compressor

$$l_{cm} = \frac{k}{k-1} P_0 \nu_1 \left[\left(\frac{P_{dis}}{P_{suc}} \right)^{\frac{k-1}{k}} - 1 \right] \tag{22}$$

where k is the adiabatic exponent; P_{suc} and P_{dis} are the suction and discharge pressures respectively, kPa.

Considering pressure losses in the suction ΔP_{suc} and discharge ΔP_{dis} pipelines and the compressor valves, the values P_{suc} and P_{dis} are found from the following relations:

$$P_{suc} = P_0 - \Delta P_{suc} \tag{23}$$

$$P_{dis} = P_{cm} + \Delta P_{dis} \tag{24}$$

The values ΔP_{suc} and ΔP_{dis} are defined considering recommendations of the work, e.g. Bratuta, Sherstyk and Kharlampidi (2007).

The isoentropic capacity of the compressor is

$$N_i = \frac{N_a}{\eta_i} \tag{25}$$

where η_i is the isoentropic efficiency factor of the compressor.

In the general case, the value η_i can be determined using the empirical dependence, e. g. Morozuk (2006)

$$\eta_i = \lambda'_{wt} + 0{,}0025 t_0 \tag{26}$$

The power capacity consumed by the compressor electric motor from the network N_{el} is as follows:

$$N_{el} = \frac{V_{th} p^i_{fr} + N_i}{\eta_{el.m}} \tag{27}$$

where $\eta_{el.m}$ is the efficiency factor of the electric motor, $\eta_{el.m}$ = 0,85…0,9; p^i_{fr} is the average indicator pressure of friction, p^i_{fr} = 30…50 kPa.

The actual coefficient of performance of HPU is

$$\mu = \frac{Q_c}{N_{el}} \tag{28}$$

The coefficient of performance for the whole successively counterflow connection scheme of three HPUs is equal to

$$\mu = \frac{\sum_{n=3} Q_c}{\sum_{n=3} N_{el} + N_{aux}} \tag{29}$$

where N_{aux} is the power spent on the drive of auxiliary equipment.

Fig. 5 shows the calculation block diagram of HPU static characteristics.

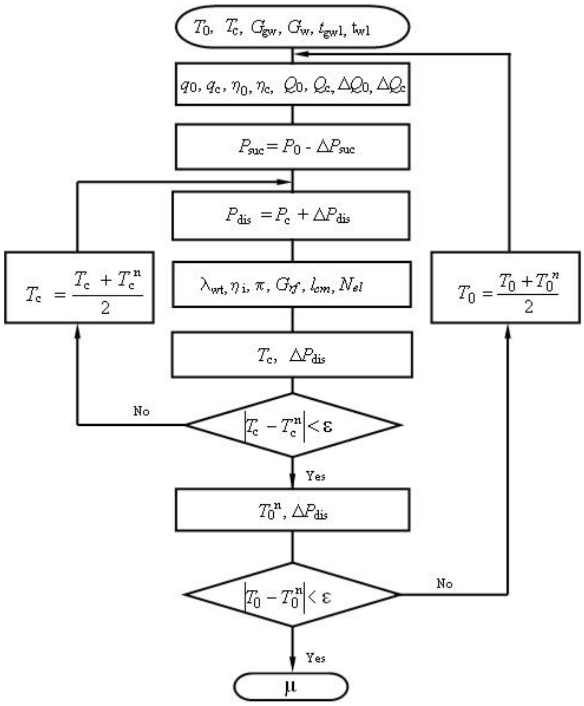

Figure 5: Calculation block diagram of HPU static characteristics.

To check function ability of the technique we carried out the calculation of HPU operating conditions where the following geometric and metering characteristics were represented: V_{th} = 0,02 m^3/s; F_c = 3,0 m^2; F_e = 1,6 m^2; G_{wt} = 0,5 kg/s; G_{gw} = 0,75 kg/s. A new safe for ozone refrigerant R245fa was regarded as working fluid.

With regard to the calculation of a successively counterflow scheme consisting of three HPUs, in the first approximation, besides t_0 and t_c, t_{gw2} and t_{wt2} are also specified, which, subsequently, are defined more precisely according to heat balances of evaporators and condensers.

Fig. 6 and 7 represent the generalized relations for Q_0 and Q_c obtained at different t_0, t_c, t_{wt2}, t_{gw1}. The relation $\mu = f(t_{gw1}, t_{wt2})$ is presented in Fig. 8.

Table 1 represents the calculation results of performance of a successively counterflow scheme consisting of three HPUs.

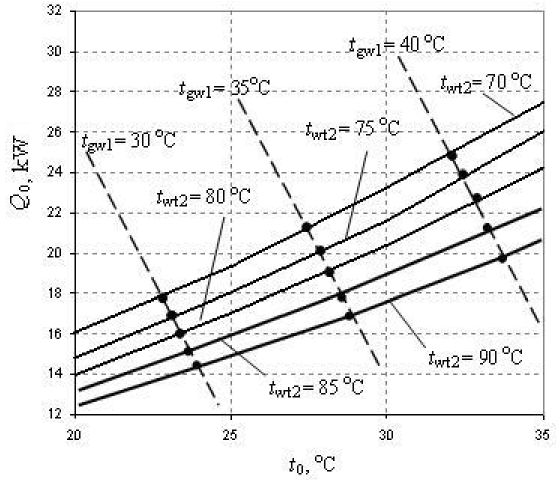

Figure 6: Change of compressor refrigerating capacity and evaporator power capacity subject to t_0 at different t_{gw1} and t_{wt2}. ___ - compressor productivity; _ _ - evaporator refrigerating capacity.

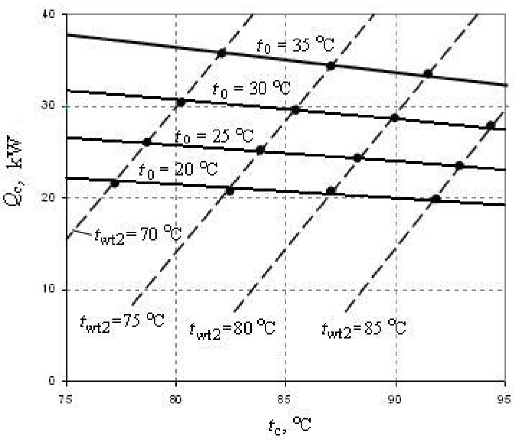

Figure. 7: Change of compressor productivity and condenser power capacity subject to t_c at different t_0 and t_{wt2}. ___ - compressor productivity; _ _ - condenser power capacity.

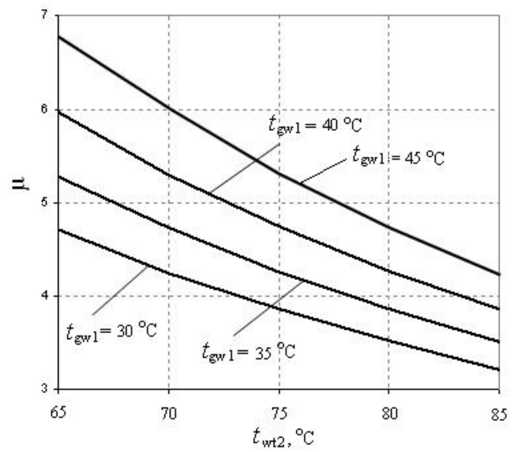

Figure 8: Coefficient of performance μ = $f(t_{gw1}, t_{wt2})$.

Table 1 Performance of a Counterflow Scheme Consisting of Three HPUs

Parameters	HPU 1	HPU 2	HPU 3
Heating capacity, *kW*	30	21	20
Refrigerating capacity, *kW*	18	16	14
Inlet temperature of geothermal water into the evaporator, °C	37	30,6	25
Evaporation temperature of refrigerant, °C	33	23	20
Outlet temperature of geothermal water from the evaporator, °C	30,6	25	21,5
Condensation temperature of refrigerant, °C	100	87	83
Inlet water temperature in the condenser, °C	85	75	65
Outlet water temperature from the condenser, °C	95	85	75

2.2.2 The Technique of Calculation of a Multistage Geothermal HPU Scheme

The calculation of characteristics of a multistage HPU is a coordination of the characteristics of its separate elements. On the one hand, the volumetric capacity of all the stages must be equal to the refrigerant flow rate appropriate to heat load on the evaporator. On the other hand, the compressor productivity must correspond to the condensation pressure which goes with the condenser heating efficiency and the heat carrier parameters of a heating system.

Despite the equality of flow rates through all the compressors in a multistage scheme, the values of specific vapor volumes at suction in each compressor are different. Therefore, it is necessary to coordinate the work of each stage considering intermediate pressure.

For a three-stage HPU there are two levels of intermediate pressures which divide the range of working pressures from P_0 to P_c into three stages of compression. The first stage is from P_0 to P_{int1}, the second stage is from P_{int1} to P_{int2} and the third stage is from P_{int2} to P_c.

Thus, taking into account recommendations of the work Morozuk (2006), we can put down

$$\frac{P_c}{P_{\text{int}}} = \frac{P_{\text{int 2}}}{P_{\text{int 1}}} = \frac{P_{\text{int 1}}}{P_0} = \sqrt[3]{\frac{P_c}{P_0}} \tag{29}$$

then

$$P_{\text{int 1}} = \sqrt[3]{P_c^2 \cdot P_0} \tag{30}$$

$$P_{\text{int 1}} = \sqrt[3]{P_c \cdot P_0^2} \tag{31}$$

The coefficient of performabce for a three-stage HPU is

$$\mu = \frac{Q_c}{\sum_{n=3} N_{el} + N_{aux}} \tag{32}$$

Table 2 represents the calculation results of power capacity of the compressors of a three-stage HPU for the following geometric and metering characteristics: $\Sigma V_{th} = 0{,}06$ m^3/s; $F_c = 9{,}0$ m^2; $F_e = 5{,}0$ m^2; $G_{wt} = 0{,}5$ kg/s; $G_{gw} = 0{,}75$ kg/s.

Table 2. Calculation of Power Capacity of the Compressor Drive of a Three-Stage HPU.

Compressor stage	Power capacity of compressor drive N_{el}, кW			
	t_{wt2}, ºC			
	73	77	83	87
t_{gw1} = 26°C				
stages 1	3,964	4,330	4,703	5,082
stages 2	4,179	4,589	5,013	5,453
stages 3	4,179	4,672	5,087	5,659
t_{gw1} =32°C				
stages 1	3,559	3,916	4,278	4,648
stages 2	3,744	4,139	4,548	4,972
stages 3	3,726	4,193	4,589	5,129
t_{gw1} = 37°C				
stages 1	3,169	3,515	3,868	4,228
stages 2	3,325	3,707	4,101	4,510
stages 3	3,296	3,738	4,118	4,626

The analysis of calculation results showed that under other equal conditions the sensitivity of μ to the changes of input parameters of geothermal water and heat carrier of a heating system for a three-stage HPU scheme is less than for a successively counterflow scheme.

The average value of μ for a successively counterflow scheme and also the cooling value of geothermal water in HPU evaporators are higher. These factors allow to recommend a successively counterflow HPU scheme for the projects with a low well production rate.

3. COMBINED SCHEME OF GEOTHERMAL HEAT SUPPLY WITH HPU

Natural conditions of the most Ukrainian geothermal sources practically eliminate application of the flowing techniques since known sites are characterized by limited supply of natural thermal fluids and by small water production in the section of water-bearing reservoirs, e. g. Razakov (2007). Thereby, at utilization of low-potential heat it is necessary to organize operation of two wells: a production well and an injection well. The methods of water production from the well and water injection into the well, including its design and processing equipment, can be selected with the employment of existing projects, e.g. Murawyev (1978).

The particular feature of operation for all geothermal designs comprising doublet well systems (one production well and one injection well) is strongly marked unsteady character of hydraulic processes in the initial period of operation of thermal water production system.

The analysis carried out in the work, e. g. Razakov (2007), showed that before entering quasisteady operation of the system at first fast pressure drop in the outlet of production well and pressure rise in the injection well occur, besides, the intensity of water flow considerably changes. Concerning HPU, changes of thermal water flow will lead to changes of its most operation parameters, especially in a successively counterflow operation design. In this connection it is reasonable to offer the configuration of HPU technological scheme which would allow to compensate this deficiency.

Fig. 9 shows a combined design of geothermal heat supply with HPU, consisting of two units. The first, according to thermal water flow direction, HPU operates in a simple single-stage cycle, where the refrigerant R245fa serves as working fluid; the second HPU operates in a two-stage cycle, the scheme of which includes an intermediate heat exchanger and isobutane R600a serves as refrigerant. Evaporators in the combined scheme are connected successively, condensers are connected parallel.

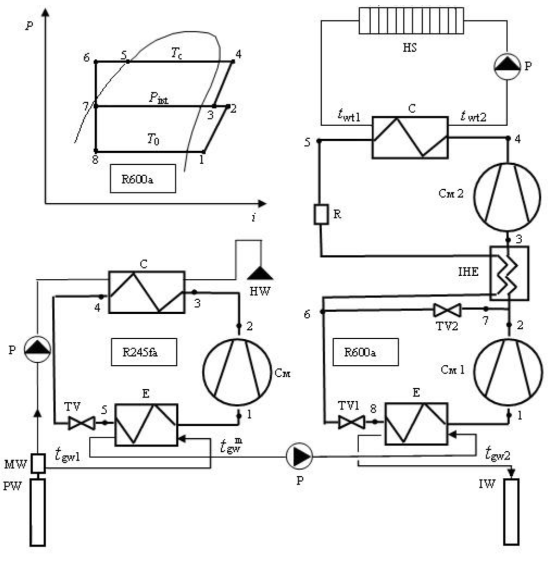

Figure 9: Combined scheme of geothermal heat supply with HPU and thermodynamic cycle of two-stage HPU with refrigerant R600a in *P* - *i* diagram. Cm1- compressor at low pressure side; Cm2- compressor at high pressure side; C- condenser; E- evaporator; R- receiver; IHE- intermediate heat exchanger; TV1, TV2- throttling valves; MW- modular unit of water cleaning; P- pump; HS- heating system; HW- system of hot water supply; PW- production well; IW- injection well

The principle of combined scheme operation is the following. Some part of thermal water at temperature of 42°C enters the evaporator of the first, according to water flow direction, HPU where it is cooled to the temperature of

36°C, the other part of it after passing the modular unit of water cleaning comes to the condenser inlet of HPU1 where it is heated up to 70°C, after that water is delivered to the system of hot water supply. In the evaporator of the second, according to water flow direction, HPU2 deeper water cooling happens, after that thermal water enters the injection well. In the condenser of HPU2 heating of thermal water up to the temperature of 85-87°C is accomplished. HPU2 realizes the two-stage cycle where after the stage of low pressure (Cm1) compressed vapor R600a enters the intermediate heat exchanger (IHE) and there it is cooled. In the same heat exchanger supercooling of refrigerant condensate coming from the condenser (C) is carried out. For the purpose of more complete cooling of superheated steam before suction to the second stage compressor (Cm2), after the throttling process in the control valve (TV2) some part of liquid refrigerant flow is injected. At the same time not only the vapor from Cm1 undergoes the process of compression in Cm2 but also the vapor formed during the process of boiling in IHE.

At the thermodynamic parameters calculation of the combined scheme the following data were taken. Temperatures of the cycle for HPU1: T_0 = 35°C; T_c = 81°C; steam superheating ΔT_{sh} = 7°C; supercooling ΔT_{sc} = 3°C; Q_0 = 27 kW; Q_c = 33,58 kW. For HPU2: T_0 = 30 °C; T_c = 95°C; supercooling in the condenser ΔT_{sc}= 5°C; intermediate pressure P_{int} = 0,871 MPa; saturation temperature appropriate to intermediate pressure T_{int} = 60°C; cooling temperature in intermediate heat exchanger T_{IHE} = 70°C; steam superheating before suction into Cm1 ΔT_{sh1} = 25°C; steam superheating before suction into Cm2 ΔT_{sh2} = 15°C; heat capacity of a low-potential source Q_0 = 100 кW.

While studying HPU operation conditions it is to estimate reliably their thermodynamic perfection with the help of a detailed analysis of exergy losses in its elements.

Relative value of exergy destruction in the element is

$$\chi = E_{Dk} \Big/ \sum E_{Dk} \tag{33}$$

where E_{Dk} is exergy destruction in the element under study, kW; $\sum E_{Dk}$ is total value of exergy destruction in the unit elements, kW.

The destruction figure of exergy applied to HPU is found from

$$\delta = \frac{E_{Dk}}{N_{el} + Q_0 \tau_0} \tag{34}$$

where τ_0 is the Carnot factor for evaporator.

Exergic weight of the element is

$$\xi = \frac{E_{Dk}}{Q_c \tau_c} \tag{35}$$

where τ_c is the Carnot factor of the condenser.

Exergy efficiency is

$$\eta_{ex} = \frac{Q_c \tau_c}{N_{el} + Q_0 \tau_0} \tag{36}$$

Table 3 represents the calculation results of exergy destruction in the elements of HPU1 (refrigerant R245fa) and HPU2 (refrigerant R600a) which are included in the combined scheme.

Table 3. The calculation of exergy destruction in the HPU elements for different schemes

HPU element	E_{Dk}, кW	χ, %	δ, %	ξ, %
	Combined scheme			
	HPU1 (R245fa)			
Evaporator	0,178	5,29	1,73	2,57
Compressor	1,24	36,9	12,12	17,94
Electric motor	0,34	10,0	3,32	4,92
Condenser	0,88	26,1	8,6	12,7
Throttle	0,723	21,51	7,06	10,4
	HPU2 (R600a)			
Evaporator	2,69	12,0	5,11	8,56
Compressor 1	2,51	11,2	4,77	7,99
Electric motor 1	0,99	4,44	1,88	3,14
Intermediate heat exchanger	3,1	13,9	5,89	9,87
Compressor 2	1,37	6,14	2,62	4,39
Electric motor 2	1,04	4,66	1,97	3,30
Condenser	5,9	26,4	11,2	18,7
Throttle 1	3,87	17,3	7,35	12,3
Throttle 2	0,81	3,63	1,55	2,59
	Exergic efficiency η_{ex} = 0,636			
	Single-stage HPU (R600a)			
Evaporator	6,73	15,3	7,67	16,7
Compressor	9,96	22,6	11,35	24,8
Electric motor	3,52	8,01	4,01	8,76
Regenerative heat exchanger	11,6	26,4	13,2	28,8
Condenser	3,86	8,79	4,4	9,61
Throttle	8,3	18,9	9,46	20,6
	Exergic efficiency η_{ex} = 0,457			

The estimation of thermodynamic perfection of the single-stage HPU with regenerative heat exchanger (refrigerant R600a) is given for the comparison. Calculation of the single-stage HPU was carried out for the same heating efficiency and at the same inlet and outlet parameters of evaporator and condenser as for the combined scheme. Some recommendations of literature, e. g. Yantowsky and Pustovalov (1982), were used to define exergy losses. The refrigerant parameters characterizing system state of equilibrium with environment were taken at T_{env} = 272 K.

As table 3 shows, for a combined scheme exergy efficiency is 28% higher than for a single-stage one. As for the absolute value of exergy destruction in the elements a combined scheme also has some advantage in comparison to a single-stage one. Exergy losses in the condenser are the exception. For a combined scheme they are a little higher than for a single-stage scheme. This can be explained by higher value of the Carnot heat transfer factor for a single-stage scheme (τ_c = 0,244) in comparison to a combined scheme where that value is τ_c = 0,225. It should be noted that exergy losses in the condenser mostly depend on losses in the compressor and are predetermined by the process of thermodynamic cycle. The calculation of relative characteristics χ, δ, ξ allowed to reveal the most imperfect elements in each scheme. For the single-stage HPU such an element is a regenerative heat exchanger and for the combined scheme it is the compressor of HPU1.

CONCLUSION

The introduced relations and calculation technique let analyze different thermodynamic operation conditions of HPU and also to select rational arrangement of its technologic scheme considering operation conditions of thermal water intake system and parameters of heat-transfer agent of heating system.

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