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Heat Pumps with High Temperature Heat-Carrier in Decentralized System of Building Heating

Andrey Red′ko, Dyonis Kharlampidi.

Kharkiv State Technical University of Construction and Architecture (Ukraine), 61002, Symska str. 40.

E-mail: andrey.ua-mail@rambler.ru; kharlampidi@ipmach.kahrkov.ua

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ABSTRACT

The possibility of application of high-temperature heat pumps in order to raise utilization efficiency of geothermal sources of low-potential heat in heating systems is analyzed in the work.

1. INTRODUCTION

At present energy conservation technologies based on the application of geothermal sources for providing considerable saving of organic fuel are being adopted abroad.

The conditions, which can be characterized as having prospects for utilization of the Earth abyssal heat, exist on the most part of the Ukrainian territory at moderate depths available for modern equipment. It is known that the total potential of theoretically available geothermal energy in Ukraine is rated as a value equivalent to fuel reserves of 50 10^6 t. e. f., e. g. Razakov (2007). At the same time, the direct heat utilization of geothermal sources in heating systems is of low efficiency as they (geothermal sources) have a relatively low temperature of 40-60ºC.

The economically effective utilization of low-potential heat from geothermal sources is possible with application of heat-pump units (HPUs). Heating systems based on HPUs and using low-potential heat of geothermal sources and also rejected heat of industrial works are in operation in all industrially developed countries, and moreover, their number is increasing with every year, e.g. Lund (2008), Lund, Freeston and Boyd (2005). According to the forecast of the World Energy Committee, by 2020 in economically developed countries the share of HPUs for heating and hot water supply will be up to 75%. In the USA more than 35 % of residential buildings are equipped with HPUs which provide complex heating and cooling all the year round.

The application of HPUs results in fuel saving, in reducing of environmental pollution and in tighter daily schedule of electric load. The economic expediency of their application is determined by the level of capital expenditure, and by the price relations of electric power and fuel. The latter is caused by the fact that HPUs consume electric power but save fuel, owing to the substitution of boiler-houses. Therefore, the cheaper electric power is and the more expensive fuel is, the higher HPUs efficiency will be. As a rule, the period of HPU payback depending on their capacity is from 1,5 to 3 years, and it is much less than the standard payback period defined for heating systems (8,3 years). We must note the preference of geothermal water utilization as a low-potential heat source for HPUs. It is connected with the fact that the temperature of geothermal waters remains constant during the whole heating period. The combination of these factors is the most favorable in the Crimea and Western regions of Ukraine.

Fresh and low-mineralized waters of Pliocene–quaternary hydrothermal section with the temperature of 30-60ºC are the most worthwhile to be used in HPUs. The use of these waters for water supply is not always possible according to sanitary parameters, but it is not a limitation for applying them as a source of thermal energy.

The direct use of heat from underground waters for heating and hot water supply is not always justified. In this sense, the application of combined technological schemes with HPUs will allow to raise the efficiency of a geothermal circulation system.

2. THE ANALYSES OF THERMODYNAMIC EFFECTIVENESS OF DIFFERENT WAYS HEAT GENERATION

Energy effectiveness of electric power use for heat generation is rated by the coefficient of fuel saving Sf, fuel saving works at Sf ≥1. This condition occurs at the HPU conversion coefficient μ > 2,3, e. g. Cocorin, Latyk, Melic-Arkelyan (1987).

To compare thermodynamic effectiveness of different ways of heat generation for heating aims, heating coefficient C_{ht} is used, e. g. Procenko (1994)

$$C_{\mathrm{ht}} = \frac{q_{\mathrm{H}}}{l} \tag{1}$$

where q_{H}, l is the produced heat, the work input.

Let us here consider electric heating for residential buildings (electric boilers, electric heating of floors, electric air heaters etc.). All the electric power input in those units turns into heat output

$$l_{\mathrm{EL}} = q_{\mathrm{H}} \tag{2}$$

Besides, according to (1), C_{ht} = 1. Considering electric power losses in electrical networks (of about 10-15%), the actual heating coefficient will naturally be lower $C_{ht} < 1$.

Below we will analyze a heating boiler-house where, as is known, heat is received by means of organic fuel burning and there is no work input here. But at the electric power station this fuel can be used to produce work. Thus, the heating coefficient of a boiler-house is defined as the ratio of generated heat, considering the boiler efficiency, to the possible lost work of the condensing electric power station, considering its efficiency:

$$C_{ht}^{b} = \frac{\eta_{bh}}{\eta_{ces}} \tag{3}$$

where η_{bh}, η_{ces} is the boiler-house heating efficiency, the electric power station efficiency

With the boiler-house heating efficiency η_{bh}= 0,9 ÷ 0,92 and the electric power station efficiency η_{ces} = 0,4 ÷ 0,42, we get the heating coefficient. $C_{ht}^{b} = 2$

The heating coefficient of HPU is determined by temperature limits of the thermodynamic cycle

$$C_{ht}^{HPU} = \frac{\eta_{em} T_c}{(T_c - T_e)} \tag{4}$$

where T_c, T_e, η_{em} is the condensation temperature of the working fluid, the evaporation temperature, electromechanical efficiency of the compressor.

The results of the comparison of different types of heating are summarized in Table 1. We see that the HPUs with geothermal source have the highest value of C_{ht}.

Table 1. Results of the comparison of different ways of heat supply.

Way of heating	Values of heating coefficient
Electric heating	0,9 ÷ 1
Heating boiler-house	2 ÷ 2,2
HPUs with air source	2,5 ÷ 3
Thermal electric power station	4,5 ÷ 5
HPUs with geothermal source	6,5 ÷ 7,5

It should be noted that the principal cause of this essential difference in values of the heating coefficient for different types of HPUs lies in the maximum attainable values of the HPU coefficient of performance μ which, likewise C_{ht}^{HPU}, depends on the cycle temperatures of evaporation and condensation.

3. THE INFLUENCE OF A TECHNOLOGICAL CONNECTION SCHEME OF THE HPU ELEMENTS ON THE ENERGY EFFICIENCY

The utilization of geothermal waters at the temperature of 30-60°C as a low-potential heat source for HPUs will allow to raise the condensation temperature up to 90-95°C which, in its turn, will provide a high value of coefficient of performance μ corresponding to the energy efficiency of electric power utilization for heating and hot water supply Sf > 1.

Meanwhile, to obtain high condensation temperatures in HPUs it is not sufficient to have the temperature of a low-potential source of more than 30°C. As is known, e. g. Sokolov and Brodyansky (1967), at the rise of condensation temperature in the cycle the specific heat of phase transfer during condensation process lowers, and the inner irreversible losses during throttling process increase, which causes the reduction of μ. It is possible to provide high temperature of a heat-carrier after a single-stage HPU (85-90°C) by choosing high-temperature working fluids as refrigerants. However, at the same time, the specific total heat output, defining the geometric dimensions of heat-exchange and compression equipment of HPUs, may decrease. In any case, HPU working fluid selection for predetermined temperature limits of a thermodynamic cycle is a settlement by compromise.

In this connection, in the paper on the basis of numerical simulation of power characteristics of a geothermal HPU with a given technological scheme we consider it reasonable to find a working fluid which will provide the effective exploitation of HPUs under high-temperature conditions.

The influence of a technological connection scheme of the HPU elements on the heat transfer efficiency will be in our consideration.

The thermodynamic cycle of HPUs and the principal technological schemes are shown in Fig. 1, Fig. 2. and Fig. 3.

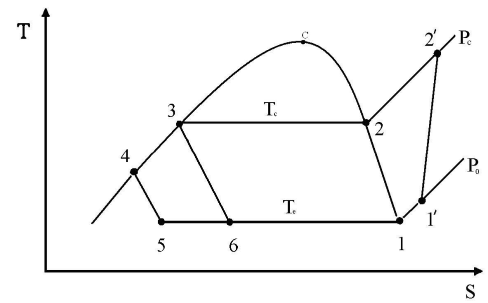

Figure 1: Thermodynamic cycle of a heating pump in T - S diagram. 1–1′- isobar steam superheating; 1′–2′ - isentropic compression, 2′–2 - removal of steam superheating, 2–3 - condensation, 3–4 - condensate supercooling, 4–5 - throttling of supercooled fluid, 3–6 - throttling

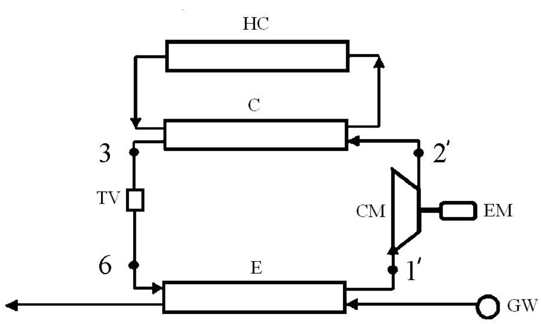

Figure 2: Principal technological scheme of simple single-stage HPU. HC- heat consumer, C-condenser, E- evaporator, CM - compressor, EM- electric motor, TV- throttle valve, GW-geothermal well.

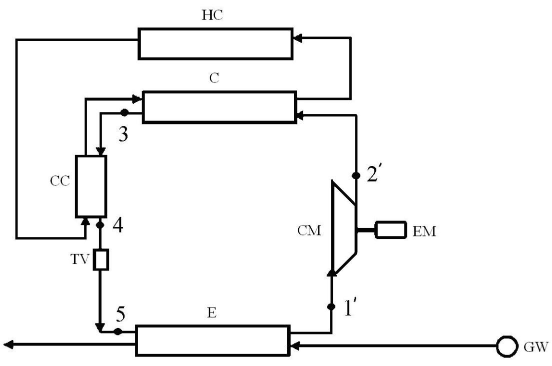

Figure 3: Principal technological scheme of simple single-stage HPU with a condensate cooler. HC- heat consumer, C- condenser, CC– condensate cooler, E - evaporator, CM - compressor, EM- electric motor, TV- throttle valve, GW- geothermal well.

The actual coefficient of performance of an inverse Rankine cycle is found from:

$$\mu = \eta_{\mathrm{i}}\eta_{\mathrm{em}}(i_{2'} - i_3)l_{\mathrm{a}} \tag{5}$$

where η_{i} is the compressor isoentropic efficiency; η_{em} is the compressor electromechanical efficiency; i_2' and i_3' are the enthalpy values of the working substance in typical points of the cycle (Fig. 1); l_{a} is the specific isentropic compression work in the compressor during the process $1'$–$2'$.

Specific isentropic compression work in the compressor l_{a} is defined from the equation:

$$l_{\mathrm{a}} = \frac{k}{k-1}P_0 v_{1'}\left[\left(\frac{P_{\mathrm{c}}}{P_0}\right)^{\frac{k-1}{k}} - 1\right] \tag{6}$$

where k is the adiabatic parameter; P_0 and P_{c} are the evaporation and condensation pressures, respectively, kPa; v_1' is the specific compressor inlet steam amount, m³/kg.

As stated above, at high temperatures of a working fluid throttling irreversible losses grow and to reduce them the condensate cooler is included in the HPU scheme (Fig. 3).

Working fluid cooling below the condensation temperature allows to increase the specific heat rejection in the evaporator (for a unit of working fluid consumption) and to reduce the specific electric power consumption in the compressor for a unit of the converted heat. What is more, the higher the condensation temperature is, the deeper the cooling of liquid condensate should be.

The throttling process 3–4 (Fig. 2) or 4–5 (Fig. 3) is isoenthalpic, that is $i_3 = i_6$ and $i_4 = i_5$.

The temperature reduction of liquid working substance in the condensate cooler (3–4 process) from T_{c} to T_4 results in the increase of the specific heat output for the value $i_3 - i_4$. The coefficient of performance μ for HPU with a cooler is given by:

$$\mu = \eta_{\mathrm{i}}\eta_{\mathrm{em}}\left[(i_{2'} - i_3) + (i_3 - i_4)\right]/l_{\mathrm{a}} \tag{7}$$

Working fluid consumption in the HPU scheme is calculated by:

- in the HPU scheme without a condensate cooler

$$m_1 = \frac{M_{\mathrm{w}}c_{\mathrm{pw}}(T_1 - T_2)}{i_{1'} - i_3} \tag{8}$$

- in the HPU scheme with a condensate cooler

$$m_2 = \frac{M_{\mathrm{w}}c_{\mathrm{pw}}(T_1 - T_2)}{(i_{1'} - i_3) + (i_3 - i_4)} \tag{9}$$

where M_{w} is the mass water flow rate, kg/s; c_{pw} is the water heat capacity, kJ/(kg K); T_1 and T_2 are the HPU evaporator inlet and outlet water temperature, °C.

The capacity expended on the HPU compressor drive is calculated from:

- for the scheme without a condensate cooler

$$N_1 = \frac{m_1(i_{2'} - i_{1'})}{\eta_{\mathrm{em}}} \tag{10}$$

- for the scheme with a condensate cooler

$$N_2 = \frac{m_2(i_{2'} - i_{1'})}{\eta_{\mathrm{em}}} \tag{11}$$

It is possible to value the advantage of any HPU scheme for given conditions quantitatively adopting the relations N_1/N_2 and Q_1/Q_2.

$$\frac{N_1}{N_2} = \frac{m_1}{m_2} = 1 + \frac{i_3 - i_4}{i_1 - i_3} \tag{12}$$

The HPU heat capacity for the scheme without a condensate cooler

$$Q_1 = m_1(i_{2'} - i_3) \tag{13}$$

The HPU heat capacity for the scheme with a condensate cooler

$$Q_2 = m_2(i_{2'} - i_4) \tag{14}$$

Thus,

$$\frac{Q_1}{Q_2} = \left(1 + \frac{i_3 - i_4}{i_{1'} - i_3}\right)\left(\frac{i_{2'} - i_3}{i_{2'} - i_4}\right) \tag{15}$$

With addition of a condensate cooler to the HPU scheme under other equal conditions, the compressor drive capacity and heat capacity reduce. Besides, the reduction degree of N is higher than that of Q, which indicates the higher coefficient of performance μ for the HPUs with a condensate cooler.

A set of calculation of thermodynamic cycle parameters has been carried out for the schemes noted above. To calculate thermal physical properties of refrigerants, data base of REFPROP software product was used (NIST thermodynamic properties of refrigerants and refrigerant mixtures database).

The calculations have been carried out for Freon R142b, isobutane and nonazeotropic mixture of iso-butane and isopentane (C_4H_{10}/C_5H_{12}) at the concentration of 90/10%.

The thermodynamic properties of some HPU working fluids are represented in Table 2.

Table 2. Thermodynamic properties of HPU working fluids

Working fluid	Critical temperature T_{cr}, K	Critical Pressure P_{cr}, MPa
Freon R142b	409,65	4,14
Isobutane C_4H_{10}	407,85	3,63
Mixture C_4H_{10}/C_5H_{12}	434,33	3,52

The utilization of high-temperature working fluids is limited by pressure value in a condenser. Thus, at condensation temperature T_c = 90ºC the pressure value for isobutane C_4H_{10} is P_c = 1,63 MPa, and for Freon R142b it is 1,74 MPa. For other working fluids the value P_c is far higher, for example, for ammonia P_c is 5,12 MPa what, according to technical and economical calculation, makes their utilization not reasonable as it requires the reinforcement of durability of HPU basic elements.

The thermodynamic parameters of HPU cycles with different working fluids are presented in Table 3. The evaporation temperature in the calculations was assumed as T_e = 30ºC, the condensation temperature was T_c = 90°C.

Table 3. Thermodynamic parameters of HPU cycles

Working fluid	Specific cold output, q_0, kJ/kg	Coefficient of performance μ
Freon R142b	116,4	3,14
Isobutane C_4H_{10}	191,2	2,86
Mixture C_4H_{10}/C_5H_{12}	215,0*	2,43*
	290,0**	3,20**

* the average temperature of the mixture in the evaporator was 24,5°C, in the condenser it was 95°C.

** the average temperature of the mixture in the evaporator was 37,5°C, in the condenser it was 95°C.

As we may see from Table 3, all the selected working fluids let accomplish the process of thermal conversion when the conversion coefficient μ is higher than economically justified μ > 2,3.

Therefore, we consider nonazeotropic mixture C_4H_{10}/C_5H_{12} (90/10)% to be worthwhile high-temperature working fluid for HPUs. Let us make a detailed analysis of the thermodynamic properties of the cycle for this mixture.

Fig. 5 and Fig. 6 show thermodynamic cycles in *P–i* -diagram for the nonazeotropic mixture C_4H_{10}/C_5H_{12} at different temperatures of evaporation.

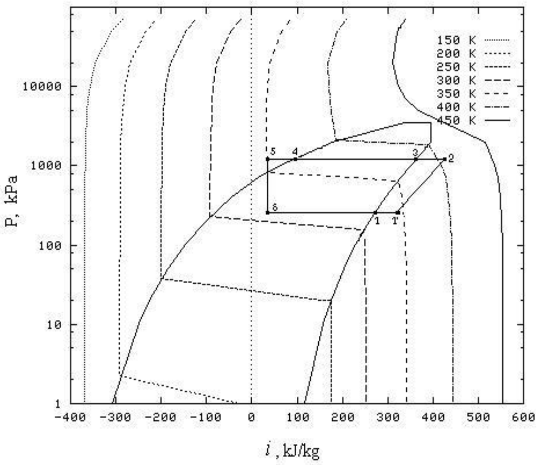

Figure 5: Calculations of HPU cycle parameters for nonazeotropic mixture C_4H_{10}/C_5H_{12} at the average temperature of 37,5°C during the process of evaporation

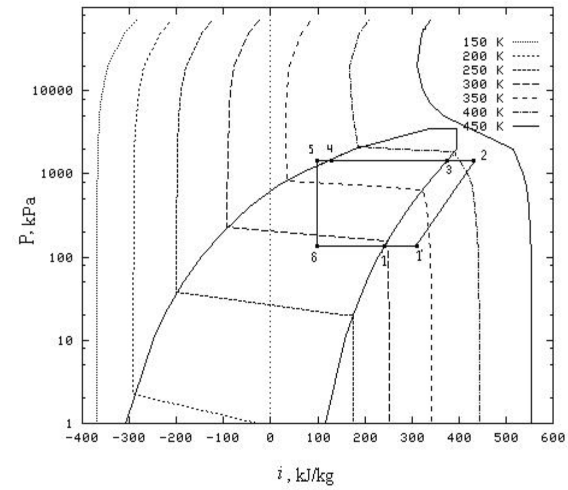

Figure 6: Calculations of HPU cycle parameters for mixture C_4H_{10}/C_5H_{12} at the average temperature of 24,5°C during the process of evaporation.

The analysis of calculation results for the cycles showed that because of rather rapid slope of the right boundary curve for the given mixture, considerable steam superheating (1–1′ process) after evaporation is needed for point 2 not to appear in the wet steam area in the end of compression process. Besides, the lower the mixture temperature during the evaporation process is, the higher the value of steam superheating should be.

The results of calculation for the HPU cycle fulfilled for two variants are represented in Table 4.

As Table 4 shows, the coefficient of performance μ grows at the rise of average evaporation temperature and at the increase of fluid supercooling in the condenser. It should be noted that in contrast to the cycles with single-component working fluids, fluid supercooling after condenser in the cycle with nonazeotropic mixture at the fixed average temperature in the process of evaporation leads to the rise of evaporation pressure and, consequently, reduces compression work in the condenser. It is connected with the temperature glade of evaporation and condensation processes for the mixture of a certain concentration which depends on the angle of isobar and isoterm intersection on the thermodynamic diagram. So, in order to change pressure in the evaporator in the cycle with nonazeotropic mixture, it is necessary to affect the value of fluid supercooling in the condensate cooler.

Table 4. Thermodynamic properties of HPU cycle for mixture C_4H_{10}/C_5H_{12}

Properties	Var.1	Var.2
Average temperature of the mixture during the process of evaporation T_0, °C	37,5	24,5
Inlet temperature of evaporator T_6, °C	33	20,5
Outlet temperature of evaporation area T_1 , °C	42	28,5
Temperature glade in evaporation process ΔT_e, °C	8	8
Inlet steam temperature in the compressor, $T_1^{\prime}$°C	65	40
Outlet steam temperature during compression T_2, °C	135	141
Inlet temperature in condensation area T_3, °C	97	97
Average temperature during condensation T_c, °C	95	95
Outlet temperature in the condenser T_4, °C	92	92
Temperature of supercooling T_5, °C	77	82
Temperature glade in condensation process ΔT_c, °C	5	5
Compressor indicator efficiency η_i	0,6	0,6
Specific cold productivity in the cycle q_0, kJ/kg	290	215
Specific heat productivity in the cycle q_c, kJ/kg	420	365
Specific compression work in the compressor l_a, kJ/kg	130	150
Coefficient of performance μ	3,2	2,43

The overall performance of geothermal HPU substantially depends of the thermodynamic efficiency of evaporator. In Table 5 the results of comparison the coefficients of performance of HPU for various type evaporators are considered. Type A is a flooded type evaporator. Type B is a horizontal evaporator with intratubal boiling of the refrigerant. Calculations are executed for two variants (see Table 4).

Table 5. Coefficients of performance for different types of evaporators

Concentration of Mixture (C_4H_{10}/C_5H_{12}) %	Coefficient of performance μ			
	Var. 1		Var. 2	
	Tape A	Tape B	Tape A	Tape B
90/10	3,2	3,16	2,43	2,37
50/50	2,98	3,46	2,04	2,89

CONCLUSION

Thus, the results of numerical simulation of HPU power properties let us conclude about the expediency of utilization of working fluids R142b, R600a (C_4H_{10}) and nonazeotropic mixture C_4H_{10}/C_5H_{12} for obtaining high temperatures of heat-carrier for heating system

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