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ЕНЕРГЕТИКА ТА ЕНЕРГОЗБЕРЕЖЕННЯ

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Two-Stage Heat Pumps for Energy Saving Technologies

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The problem of energy saving becomes one of the most important in power engineering. It is caused by exhaustion of world reserves in hydrocarbon fuel, such as gas, oil and coal representing sources of traditional heat supply. Conventional sources have essential shortcomings: low power, ecological and economic efficiencies that can be eliminated by using alternative methods of power supply, like the considered one: low-temperature natural heat of ground waters on the basis of heat pump installations application. The considered heat supply system provides an effective use of two stages heat pump installation operating as heat source at ground waters during the lowest ambient temperature period. The calculation method of heat pump installations on the basis of groundwater energy is proposed. The values of electric energy consumption by the compressors' drive, and the heat supply system transformation coefficient µ for a low-potential source of heat from ground waters allowing to estimate high efficiency of two stages heat pump installations are calculated.

Keywords: Heating Loading; Freon; Two Stages Heat Pump Installation; Low-Potential Source of Heat; Energy Efficiency

Двоступеневі теплонасосні установки для енергозберігаючих технологій

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Одним з основних напрямків вдосконалення систем теплопостачання є тенденція переходу на низькотемпературні системи опалення на основі теплонасосних установок. Запропоновано методику розрахунку теплонасосних установок на базі ґрунтових вод. Розроблено алгоритм розрахунку теплонасосної установки, реалізація якої підтверджує ефективність її роботи для теплопостачання споживачів. Встановлено раціональну глибину і кількість свердловин, які забезпечують надійну роботу теплонасосних систем при мінімальній потужності електроприводу насосів, що подають ґрунтову воду до теплонасосної установки. Встановлено, що раціональне розміщення трьох менших за глибиною свердловин в порівнянні з однієї свердловиною дозволяє заощадити витрата електроенергії в 1,7 разів. Шляхом числового моделювання теплових процесів в елементах одноступінчатої і двоступеневої теплонасосних установок встановлено, що більш ефективною є двоступенева теплонасосна установка для системи опалення «тепла підлога». Це досягається за рахунок більш низької температури теплоносія, що подається під внутрібудинкову систему, внаслідок чого для досягнення комфортних температурних режимів потрібно менше електроенергії на привід компресора. Визначено коефіцієнти трансформації теплоти, потужність, споживана електроприводом компресора теплового насоса. Виконано оцінку ефективності використання запропонованих теплонасосних систем для заданих кліматичних умов і встановлені умови ефективного використання теплонасосних систем на базі ґрунтових вод.

Ключові слова: теплонасосна установка; ґрунтова вода; утилізація; низькопотенційна енергія; одноступеневий тепловий насос; двоступеневий тепловий насос; коефіцієнт трансформації теплоти; опалювальна навантаження; ефективність.

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1 Introduction

Unlike fossil fuels, the unconventional geothermal energy resource is not limited. Therefore, one of the main trends as to the replacement of traditional and upgrading of existing heat supply systems consists in the transition to a low-temperature heating system based on the heat pump plants (HPP) using groundwater thermal ener-

gy [1, 2]. To implement the heat transformation processes at HPP various working bodies are applied, but not always their thermodynamic and thermal properties meet the heating systems requirements, especially in climates where the atmospheric temperature is low enough [3]. As the thermal load increases the heat potential augments and its maximum value implies also the maximum heat carrier temperature in the heating system. Therefore, during periods of highest temperature gap between the low potential heat source and the heating systems' heat carrier (especially in the sever climatic regions) to cover the full heating load, the two-stage HPP are used. Analysis of recent research and publications. In recent years, there arises the need to improve the HPP efficiency [4].

2 General material

The article deals with a promising concept of the two-stage heat pump system (Figure 1), useful for heating systems with low potential heat source [5, 6].

This scheme operating principle is following: water from the heat generator 9 reaches the pump 8, delivering it for heating to the capacitors 6 and 7, serially connected

to the water feeding system. The cooling zone condensers separating superheated steam from the condensation zone thus provide the countercurrent working fluid - and - heated water circulation therefore enabling the temperature increase of the heated water that leaves the condenser concurrently reducing the energy loss from irreversible heat exchange. The primary stage capacitor 7 is heating the water from the temperature t_{s1} to a certain intermediate temperature t_i. then the water enters the second-stage capacitor 6 and is heated therein to t_{s2} temperature. The low potential groundwater heat entering the evaporator 3 is passed to boiling working agent, which vapor at a pressure P_0 reaches the lower-stage compressor 1, where it is compressed to a pressure P_i, and divided into two streams. One stream flows into the capacitor 7, wherein during the heat transfer to the heated water it is condensed, and the other reaches the upper-stage capacitor 2 where it is compressed to a pressure P_c, and passing into the capacitor 6 it heats the carrier fluid from the intermediate temperature to a temperature ts2. Then, the working medium condensation through an expansion valve 5 passes into the condenser 7, and the total flow of condensate from the capacitor 7 is supplied via the throttle valve 4 to the evaporator [7].
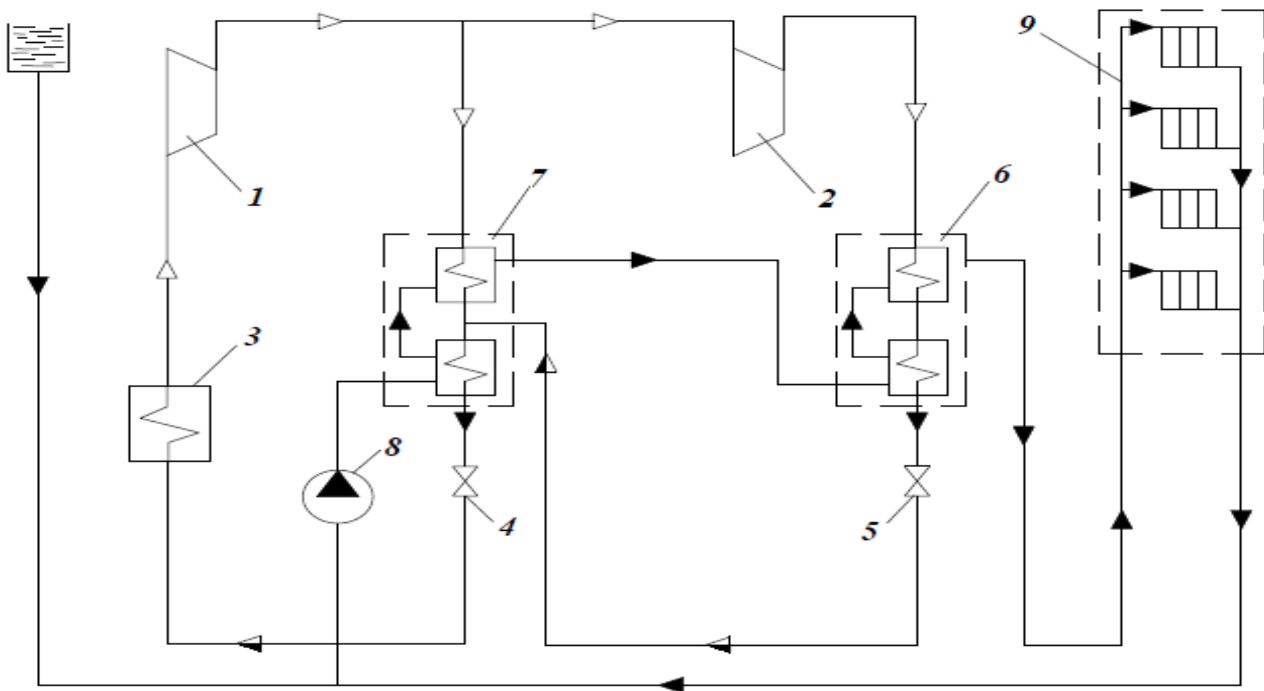
Figure 1 – Two-stage HPP principal diagram

For further comparison of two-stage HPP energy efficiency a heating diagram based on single stage HPP operating with the groundwater energy source is represented (Figure 2), where low-grade heat source (ground water), transmits heat to freon in the evaporator 2, the freon is boiling, and that steam absorbed with compressor 1, it is compressed. Then the freon is supplied to the water-

cooled condenser 3, the water representing a high-potential heat source, which is fed into the heating system 6, equipped with a pump 5. Upon heat exchanging the freon steam is condensed and in the state of liquid refrigerant through the expansion valve 4, where its temperature and pressure drop, flows back to the evaporator [1].
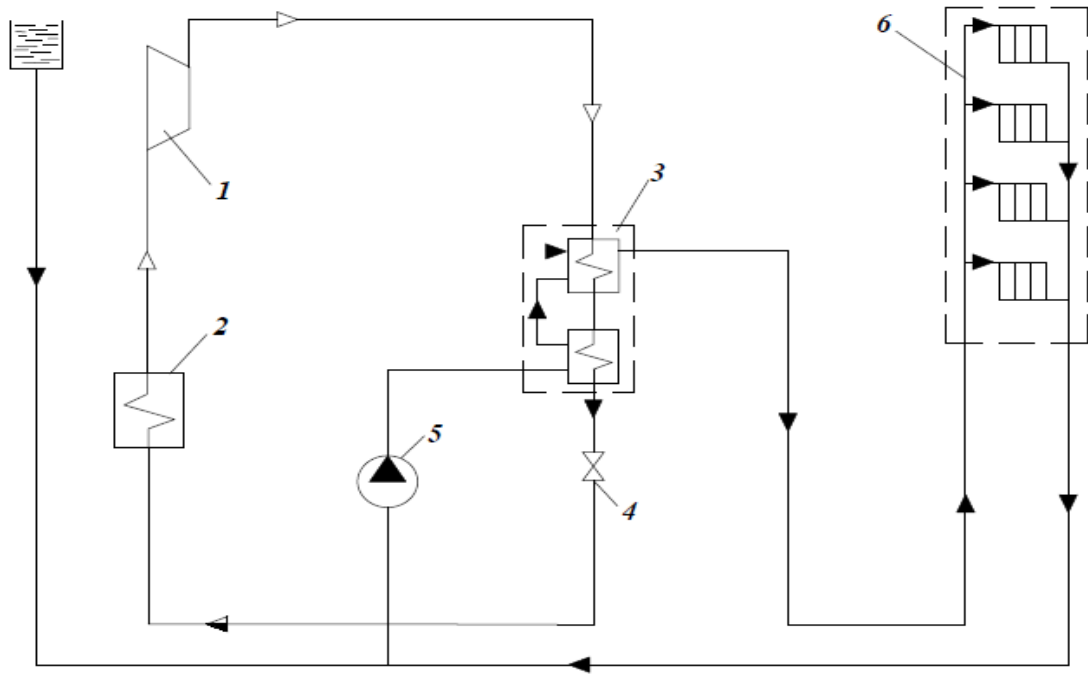
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Figure 2 – Single-stage HPP principal diagram

3 Method of analysis

The aim of research includes a comparative analysis of the considered two-stages HPP energy efficiency schemes using environmentally friendly refrigerant R152a for heating systems, with low-potential heat source represented by ground water in the temperature range t_{st1} = 4…12°C, [8], for a full coverage of the heat load on the heating system when the outdoor temperature t_0 = −18°C.

In order to analyze the considered two-stages HPP energy efficiency schemes we shall use the method [1, 9], implemented with the auxiliary of the Cool Pack program p, h-diagram (Figure 3, 4) [10].

The freon evaporation temperature t_{fe}, allowing to estimate the enthalpy h_1 and pressure P_0 of freon at the evaporator's exit:

$$t_{fe}=t_{1p2} - \Delta t_e,$$

where:

t_{1p2} – low-potential heat source temperature, °C;

Δt_e – temperature drop at the evaporator's exit, К.

The freon condensation temperature t_{fc}, allowing to estimate the enthalpy h_3 and pressure P_c of freon at the evaporator's exit:

$$t_{fc} = t_{hp2} + \Delta t_c,$$

where:

t_{hp2} – high-potential heat source temperature at the condenser's exit, °C;

Δt_c — temperature drop at the condenser's exit, К.

Compressor's adiabatic efficiency [9]:

$$\eta_a = 0{,}98(273 + t_0)/(273 + t_{fc}),$$

where:

t_0 – atmospheric air temperature, °C .

Freon enthalpy at the compressor's exit

$$h_{2f} = h_{f1} + (h_{2a} - h_1)/\eta_a,$$

where

h_{2a} – freon enthalpy upon adiabatic compression completing, kJ/kg. Condenser specific heat load:

$$q_c = h_2 - h_3$$

HPP specific heat load:

$$q_{h1} = q_c.$$

Compression work at the compressor:

$$l_{cw} = h_2 - h_1.$$

Specific energy consumed by the electric motor:

$$W = l_{cw}/(\eta_{e/m} \, \eta_e),$$

where:

$\eta_{e/m}$ – compressor's electro mechanic efficiency;

η_e – electric motor efficiency.

Heat transformation coefficient

$$\mu = q_{h1} / l_{cw}.$$

Freon mass consumption:

$$G_{fmc} = Q_{h1} / q_{h1}.$$

where:

Q_{h1} – HPP heat load, kW.

Electric power consumed by the compressor's drive:

$$N = W G_{fmc}.$$

To analyze the two-stage HPP efficiency (Figure 1) we use the methods [9]. Intermediate compressive pressure
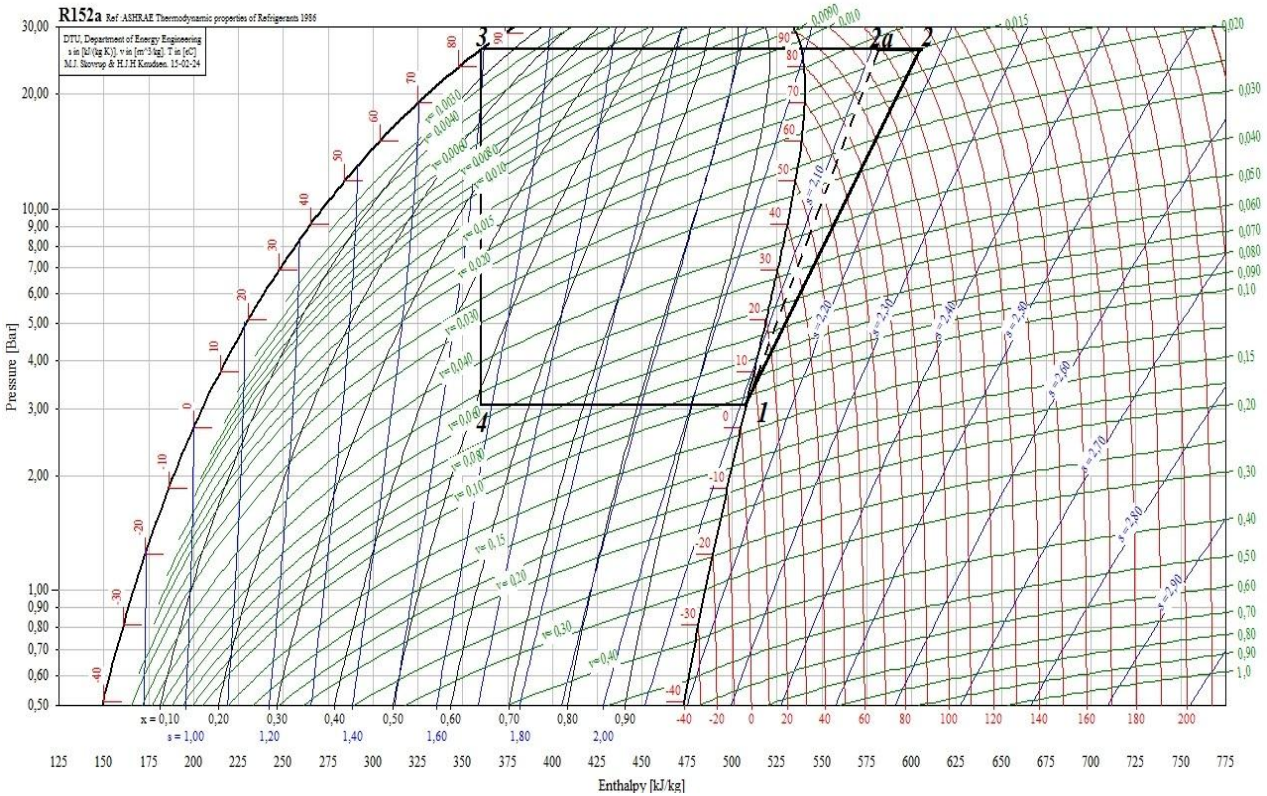
$$P_i =(P_e P_c)^{1/2}.$$

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Figure 3 – *Cycle of single stage HPP operating with ground water energy for heating system at* t_0 = -18°C

At two-stages HPP calculation, the freon consumption at the low G^l_{fmc} and high G^h_{fmc} pressure circuits are found from the power balance condition:

$$G^l_{fmc}(h_4 - h_9) = G^h_{fmc}(h_4 - h_8),$$

where:

h_4, h_8, h_9 – enthalpy at the process working points 4, 8, 9, kJ/kg (Fig. 4).

In such a way we obtain the ratio between freon consumption levels at the high G^h_{fmc} pressure and low G^l_{fmc} pressure circuits

$$G^h_{fmc} / G^l_{mc} = (h_4 - h_9) / (h_4 - h_8) = (\delta + 1)/1,$$

where:

δ – the proportion of freon vapor produced by 1st stage condenser and this one produced by 1st stage compressor.

Enthalpy of freon vapor entering the 2nd stage high pressure compressor:

$$h_3 = (h_2 + \delta h_4)/(1 + \delta),$$

where:

h_2 – freon enthalpy at the 1st stage compressor exit, kJ/kg.

Freon consumption at the high pressure circuit:

$$G^h_{fmc} = Q_{h1}/(h_5 - h_7),$$

where:

h_5 – freon enthalpy at the 2nd stage compressor exit, kJ/kg.

Freon consumption at the low pressure circuit:

$$G^l_{fmc} = G^h_{fmc}/(1 + \delta).$$

Mechanical energy consumed by the 1st and 2nd stage compressor drives:

$$N_{m\Sigma} = G^h_{fmc}(h_5 - h_3) + G^l_{mc}(h_2 - h_1).$$

Electrical energy consumed by the compressor drive:

$$N = N_{m\Sigma}/\eta_{e/m}.$$

Heat transformation coefficient:

$$\mu = Q_{h1}/N.$$

4 The results of numerical simulation single-stage and two-stages hpp operating with ground water energy

To calculate the power efficiency of the considered HPP circuits for heat supply system, adopted are the following initial data of that system thermal scheduling of 95…50°C with cutoff at 80°C:

– ground water temperature at the evaporator's entrance HPP t_{st1} = 4…12°C;

– ground water temperature at the evaporator's exit HPP t_{lp2} = 1…9°C;

– temperature drop at the heat exchangers' exit Δ_{te}=5 К, Δt_c=5 К;

– compressor's electromechanical efficiency $\eta_{e/m}$= 0,95;

– HPP heat load Q_{hl} = 502 kW (5-fllors building for 60 apartments).

– atmospheric air temperature t_0= −18°C;

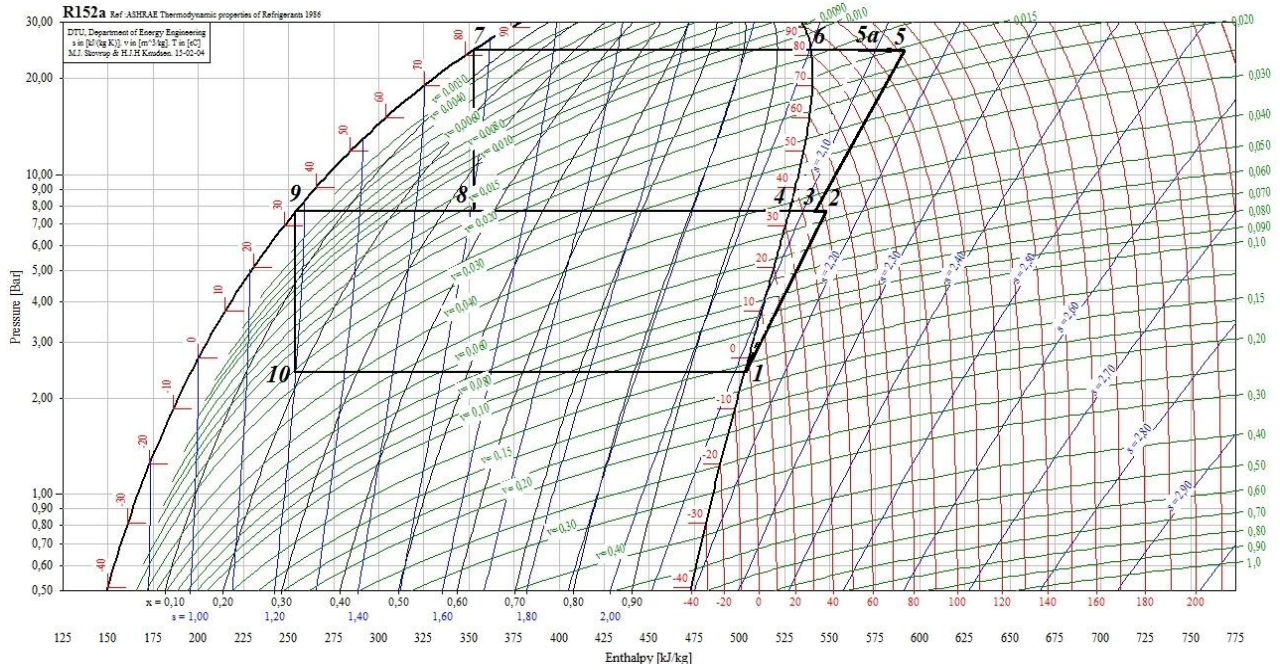– heating intrabuilding devices – sectional radiators.

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Figure 4 – *Cycle of two-stage HPP operating with ground water energy to supply for heating system at $t_0 = -18°C$*

In order to analyze the energy efficiency of the considered HPP circuits application for heat supply system using ground water as low-potential heat source, ceteris paribus the following values have been obtained (Table 1, Figure 5,

6). As the refrigerant selected is the environmentally friendly agent R152a, convenient for enhancing the HPP efficiency due to its thermal properties [3].

Table 1 – *HPP power efficiency parameters*

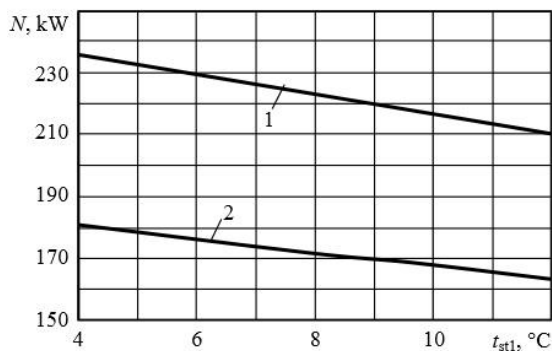| T_{st1},°C	N , kW		μ	
	Single-stage HPP	Two-stage HPP	Single-stage HPP	Two-stage HPP
10	221,2	174,7	2,387	2,871
14	208,7	168,9	2,530	2,971
18	195,7	155,4	2,698	3,228
22	183,2	149,4	2,883	3,359
26	170,7	137,1	3,093	3,659

Figure 5 – *Dependence between the electric power consumption N and the ground water temperature t_{st1} at the evaporator entrance for single-stage (1) and two-stage (2) HPP*
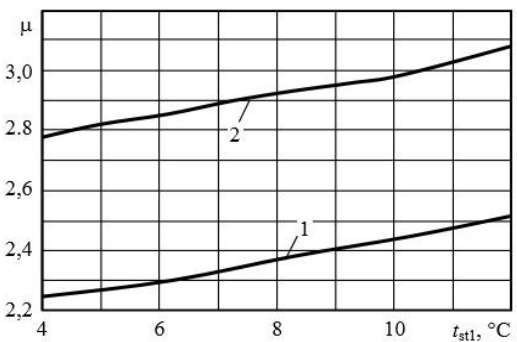

Figure 6 – *Dependence between the heat supply system transformation coefficient μ and the ground water temperature t_{st1} at the evaporator entrance for single-stage (1) and two-stage (2) HPP*

The analysis of heat supply systems' parameters (Table 1) using HPP when temperatures schedule 95...50 °C with cutoff at 80°C operating the low-potential heat source of ground water t_{st1}= 4...12°C does evidence that the smaller is the temperature drop between the low-potential heat source and the municipal system heat carrier, the higher is the HPP efficiency.

Therefore, from power viewpoint, more efficient is a two stages HPP based heat supply system, that is illustrated with the heat transformation coefficient μ, which average increase is 19% when compared to a system based onto a single-stage HPP at the expense of lower electric power quantity consumed by compressors' drive N. In other words, two-stages HPP allow producing more heat energy for the heat supply system under the same electric power consumption level. Therefore, to satisfy completely the heat loads of heat supply system, especially at the environmental temperatures below t_0 = = −18 °C, essential is to apply alternative heat supply systems using two-stage HPP.

5 The results of numerical simulation of hpp at ground water inflow from the borehole for various heating intrabuilding devices – sectional radiators and underfloor heating

Fig. 7 shows a diagram of inflow of groundwater the vertical borehole. The borehole must be provided with special suction pump destined to supply ground water to the evaporator HPP. Groundwater get into the vertical borehole through the water permeable in the radial direction along the entire height of the borehole. The height of the aquifer is equal H_0. After starting pumping the groundwater level begins to drop and the free surface and the free surface take the form of so-called depression funnel [3].

Volumetric flow of groundwater, m³/s [9]:

$$V_w = 1{,}36 \, k \, (H_0^2 - h_{bor}^2) \, / \, \lg (R_{bor}/r_{bor}) \, ,$$

where:

k – filtration coefficient, m/s;

H_0 – depth of the borehole relative to the natural groundwater level, m;

r_{bor} – borehole radius, m;

R_{bor} – range of radius, where the influence of borehole is considerable, m.

Heat load from the groundwater flow in the evaporator of the heat pump

$$Q_0 = G_w c_p \Delta t \, ,$$

where:

c_p – heat capacity of the ground water, kJ/(kg·K);

Δt – difference between the temperature of the ground water at the inlet and outlet of the evaporator HPP, К

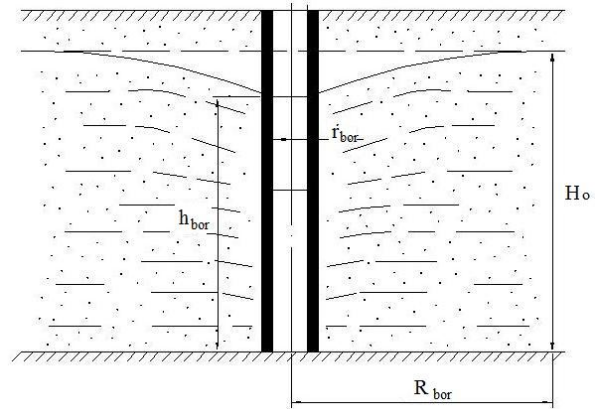G_w – mass flow rate of groundwater, kg/s

$$G_w = V_w \rho_w$$

Figure 7 – *Scheme of inflow of ground water to the vertical borehole*

Heat power of the evaporator of single-stage HPP [10]

$$Q_0 = G_{fmc} \, (h_1 - h_4), \text{ kW} \, .$$

Heat power of the evaporator of two-stages HPP [11]

$$Q_0 = G'_{fmc} \, (h_1 - h_{10}), \text{ kW}.$$

Electrical energy consumed by the pump drive pumping out the ground water from a borehole and providing a required pressure to supply water to the evaporator:

$$N_{bor} = V_w \Delta P / (1000 \, \eta_k), \text{ kW},$$

where:

ΔP – pressure drop depending on depth of the borehole H_0, Pa;

η_k – electric motor efficiency of pump.

The corresponding results of numerical simulation of energy efficiency of TNI heating systems with two intrahouse heating devices (sectional radiators and floor heating) were obtained (Figure 8, 9).
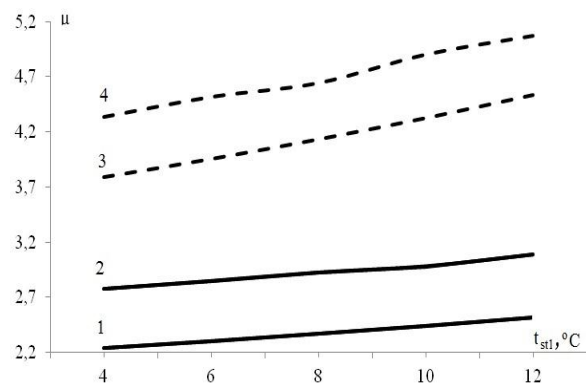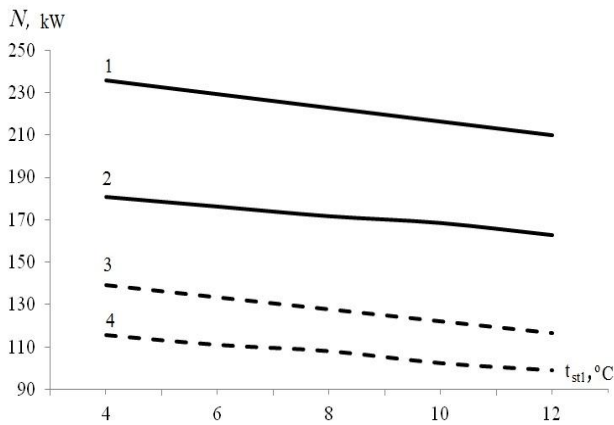

Figure 8 – *Dependence between the transformation coefficient μ and the ground water temperature t_{st1} at the evaporator entrance:*

1 – single-stage HPP with section radiators;
2 – two-stage HPP with section radiators;
3 – single-stage HPP with underfloor heating;
4 – two-stage HPP with underfloor heating

Figure 9 – Dependence between the electrical energy consumed by the pump drive N and the ground water temperature t_{st1} at the evaporator entrance:
  1 – single-stage HPP with section radiators;
  2 – two-stage HPP with section radiators;
  3 – single-stage HPP with underfloor heating;
  4 – two-stage HPP with underfloor heating

In order to maintain the required heat output of the evaporator HPP and saving electrical energy used by the pump drive pumping out the ground water from the bore-hole the rational approach to the placement of equipment must be used.

The values of required depth of the borehole H_0 providing the required volume flow rate of water flowing into the evaporator of HPP and total electric energy consumed by the pump N_{bor} are shown in the Table 2.

The calculations are made for the clay when $k = 10^{6}$ m/s; $r_{bor} = 0,15$ m; $R_{bor} = 100$ m; $h_{bor} = 10$ m.

6 Conclusions

Results of numerical simulation for HPP heating systems in the temperature chart 95...50°C with a temperature of cutting 80°C by using low-grade heat source (ground water) with temperatures $t_{st1} = 4...12$°C shows that the less is temperatures drop between the low-potential heat source and the coolant being supplied to the heating circuit, the higher is HPP efficiency.

Table 2 – Performance indicators depending on the temperature of the ground water

Number of boreholes	HPP	Values	Ground water temperature at the evaporator's entrance of HPP t_{st1}, °C				
			4	6	8	10	12
One borehole	1	Q_0, kW	277,7	283,9	290,0	296,1	302,2
	2	Q_0, kW	330,0	334,3	338,7	341,7	347,2
	3	Q_0, kW	369,4	374,9	380,4	385,8	391,1
	4	Q_0, kW	391,7	396,1	399,0	404,5	407,7
	1	H_0, m	214	216	219	221	223
	2	H_0, m	233	235	236	237	239
	3	H_0, m	247	249	250	252	254
	4	H_0, m	254	256	256	258	259
	1	N_{bor}, kW	55,5	57,3	59,2	61,1	63,0
	2	N_{bor}, kW	71,9	73,3	74,7	75,7	77,5
	3	N_{bor}, kW	85,1	87,0	88,9	90,8	92,7
	4	N_{bor}, kW	92,9	94,5	95,5	97,5	98,7
Two boreholes	1	Q_0, kW	277,7	283,9	290,0	296,1	302,2
	2	Q_0, kW	330,0	334,3	338,7	341,7	347,2
	3	Q_0, kW	369,4	374,9	380,4	385,8	391,1
	4	Q_0, kW	391,7	396,1	399,0	404,5	407,7
	1	H_0, m	151	153	155	156	158
	2	H_0, m	165	166	167	168	169
	3	H_0, m	174	176	177	178	180
	4	H_0, m	180	181	181	183	183
	1	N_{bor}, kW	39,2	40,5	41,9	43,2	44,5
	2	N_{bor}, kW	50,8	51,8	52,8	53,5	54,8
	3	N_{bor}, kW	60,2	61,5	62,9	64,2	65,6
	4	N_{bor}, kW	65,7	66,8	67,6	68,9	69,8
Three boreholes	1	Q_0, kW	277,7	283,9	290,0	296,1	302,2
	2	Q_0, kW	330,0	334,3	338,7	341,7	347,2
	3	Q_0, kW	369,4	374,9	380,4	385,8	391,1
	4	Q_0, kW	391,7	396,1	399,0	404,5	407,7
	1	H_0, m	124	125	126	128	129
	2	H_0, m	135	136	136	137	138
	3	H_0, m	142	144	145	146	147
	4	H_0, m	147	148	148	149	150
	1	N_{bor}, kW	32,0	33,1	34,2	35,3	36,4
	2	N_{bor}, kW	41,5	42,3	43,1	43,7	44,8
	3	N_{bor}, kW	49,1	50,2	51,3	52,4	53,5
	4	N_{bor}, kW	53,6	54,6	55,2	56,3	57,0

Notes: 1 – single-stage HPP with section radiators; 2 – two-stage HPP with section radiators; 3 – single-stage HPP with underfloor heating; 4 – two-stage HPP with underfloor heating.

By calculation, it was determined that more efficient is the heating system based onto low-potential groundwater heat source using the two-stage HPP, rather than a system based onto single-stage HPP that is demonstrated with average productivity increase of 19 %, as evidenced by the heat transformation coefficient μ. The two stages HPP can almost 1,3 times reduce the average energy consumption, that represent their indisputable advantage when compared to single stage installations. Thus, the conventional heat supply can be replaced by alternative methods based on the two stage HPP use. This is of high actuality for the climatic conditions, when necessary is to completely cover the heating load, particularly at ambient temperature below $t_0 = -18°C$. The use of two stages heat pump allows to increase the amount of heat for heating system and consumed electrical energy.

Results of numerical simulation shows that improving efficiency of HPP is possible by using underfloor heating instead traditional sectional radiators. This result is achieved through lower temperature coolant supplied to the intrahouse system, thereby reducing the power consumption by the compressor drive for compression of refrigerant to the condensing pressure. This is because the underfloor heating system cowers the whole surface of the floor, and thus, the heat transfer occurs from the larger surface at relatively less than low temperature.

In addition, to maintain the required evaporator capacity must be rational distribution and the amount of intake boreholes. For example, using less than three deep wells compared with one deep borehole can be reduced energy consumption by 1.7 times, resulting in faster payback HPP.

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