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HEAT PUMP TO INCREASE THE EFFICIENCY OF THE GEOTHERMAL DISTRICT HEATING SYSTEM OF THE CĂLIMĂNEȘTI CITY

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This article continues a series of several works that deal with the possibilities of increasing the efficiency of the district heating system of Călimănești, which uses as heat source low enthalpy geothermal water provided by existing drilling on the right bank of the Olt River. The authors analyze the use of a heat pump to recover heat from hot wastewater discharged from the heat exchangers of the geothermal station. The optimal refrigerant and operating conditions were established, determining the effect of heat pump implementation on expanding the capacity of the district heating system.

Keywords: geothermal heat pump, geothermal district heating system, refrigerant R1233zd(E).

1. Introduction

Călimănești-Căciulata resort, documented since 1386, is considered the "pearl" of the resorts on the Olt Valley, due to the wonderful landscapes that can be admired in this area of Romania and the spa treatments that can be performed here. On the recommendation of Dr. Carol Davila, Emperor Napoleon treated himself with water brought from here. In addition to the waters with curative properties, the Călimănești-Căciulata area also has important low enthalpy geothermal water reserves. In the geothermal and use perimeter Căciulata - Călimănești the geothermal water is supplied from three wells, drilled at depths over 3000 m, located on the right bank of the river Olt at distances of about 1-1.2 km from each other, between the spa resort Călimănești and Cozia Monastery [1].

The geothermal resources in this perimeter have an attractive energetic potential, the water temperature produced by the wells having values of 90 ... 96 ^{o}C, and the artesian flows being in the range of 10… 23 l / s. At the same time, these waters are associated with a significant quantity of combustible gases, with a content of over 84% methane and a lower calorific value of 8.5 ... 8.8 kWh / m^3_N (Table 1).

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Since the commissioning of the wells (1983-1984), it has been found that the physicochemical properties of the water do not pose problems in terms of environmental impact, which has allowed (considering the characteristics of the geothermal reservoir) to be discharged the wastewater directly into the river Olt, after cooling to a temperature of 30 ^{o}C [2].

Table 1

Production characteristics and energy potential [2]

No.	Drilling	Production characteristics: Regime	Water flow / Gas ratio	Water temp. at the wellhead	Available thermal power: from water*	from gases	Total	Lower calorific value of gases
			l/s / m^3_N/m^3water	^{o}C	MW	MW	MW	kWh/m^3_N
1	1006 Căciulata	artezian	9.4 / 2.470	96	2.60	0.70	3.30	8.89
2	1008 Cozia	artezian	23 / 1.980	92	5.97	1.32	7.29	8.47
3	1009 Călimănești	artezian	18 / 2.645	92	4.07	1.39	6.06	8.61

*By cooling to 30 ^{o}C

The capitalization of this important energy potential was initially carried out by the direct use of geothermal water, for heating and preparation of domestic hot water, the cooling of wastewater to the discharge temperature being done in the thermal pools of the spa treatment units in the area. Since 2001, the centralized heat supply system in Călimănești has been operating using the geothermal water produced by well 1009 as a heat source. The modernization of this system was carried out with European funds and consisted in replacing the preparation of the thermal agent, made with hot water boilers with liquid fuel, with the preparation of the thermal agent in the heat exchangers using geothermal water produced by drilling 1009. The three previous thermal power plants, with liquid fuel, have been transformed into thermal distribution points, keeping the old hot water boilers to cover peak loads or emergency situations. With the expansion of the natural gas network in the perimeter of Călimănești - Căciulata, these reserve boilers were replaced with modern gas boilers. Combustible gases associated with geothermal water have never been used, as they are separated and released into the atmosphere [2], [3].

Fig. 1 shows the 1009 Calimănești borehole and Fig. 2 shows the operation diagram of the geothermal station. The water produced by the drilling is first introduced in the degassing tank (DT) in which the separation of the contained gases takes place and their evacuation in the atmosphere. The drilling flow is 64.8

m^3/h, of which 28.8 m^3/h are used by nearby hotel units, and the remaining 36.0 m^3/h feeds the geothermal station of Călimănești [4].

Fig.1. Drilling head 1009 Călimănești

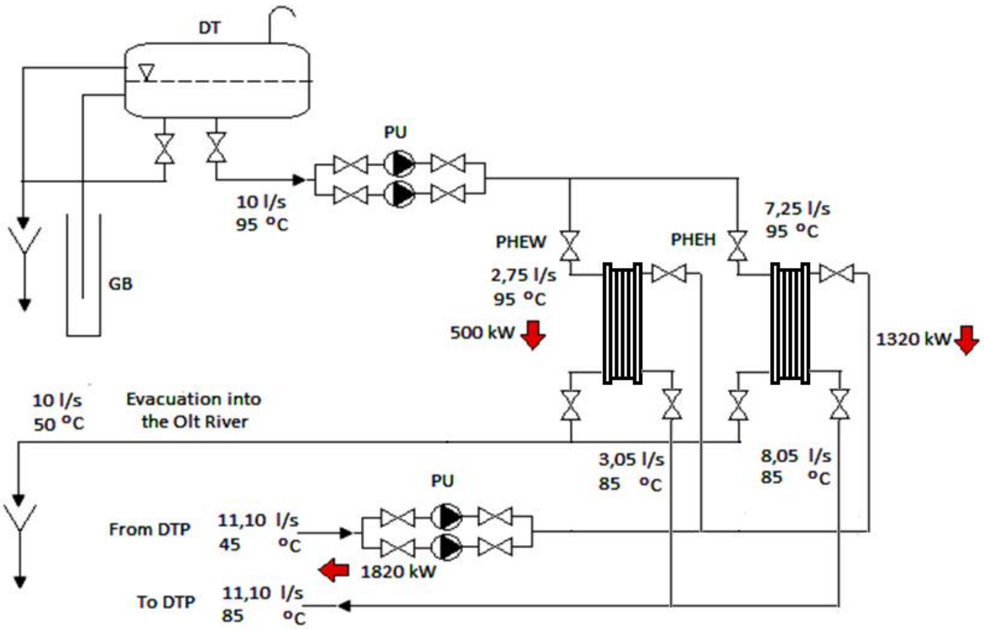

Fig.2. Operating diagram of the geothermal thermal station of Călimănești [5].
GB – Geothermal Borehole; DT – Degassing Tank; PU – Pumping Units; DTP – Distribution Thermal Points; PHEH – Plate heat exchanger for heating; PHEW – Plate Heat Exchanger for DHW

It comprises two plate heat exchangers: one with a thermal power of 500 kW and continuous operation to ensure the preparation of domestic hot water (PHEW) and a second, with a thermal power of 1320 kW, with seasonal operation, for the preparation of the primary heating agent sent in the district heating system (PHEH). Because the heating agent in the return of the district heating installation has a temperature of approx. 40 ... 45 ^{o}C, the hot water produced by the well, with a temperature of 95 ^{o}C, cannot be cooled in the heat exchangers of the station below the temperature of 50 ^{o}C, being then cooled in the open air to the temperature of 30 ^{o}C, with which is discharged into the Olt River.

For this reason, only about 2/3 of the thermal potential of geothermal water is used, the rest being losses. To make full use of the available thermal potential, several solutions have been proposed, one of them being the use of a heat pump that using the wastewater discharged from the heat exchangers as a heat source, supplements the flow of primary heating agent sent to the district heating network [5].

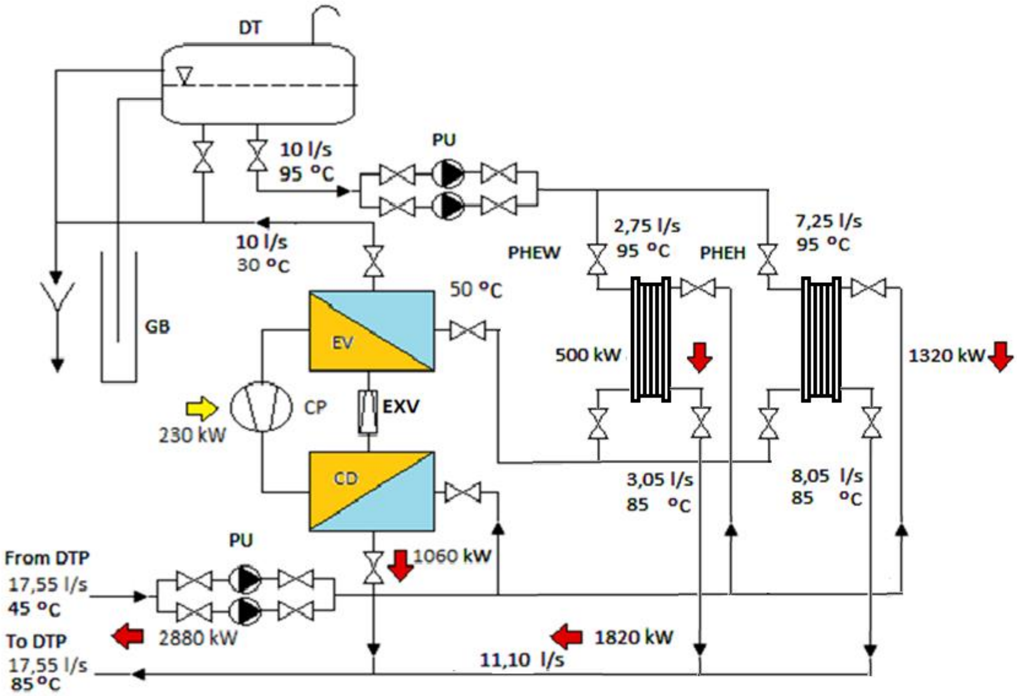

Fig.3. The operating scheme of the station coupled with a heat pump [5].
GB – Geothermal Borehole; DT – Degassing Tank; PU – Pumping Units; DTP – Distribution Thermal Points; PHEH – Plate heat exchanger for heating; PHEW – Plate Heat Exchanger for DHW; CP- Compressor; EXV – Expansion Valve; CD – Condenser; EV – Evaporator.

Fig. 3 shows the operating diagram of the geothermal station coupled with a heat pump with mechanical vapor compression. The authors of this paper analyzed this solution in [5], concluding that it has a high energy efficiency, increasing the thermal load of the system from 1820 kW to 2880 kW, which is an increase in its capacity by approx. 60%. In this paper the authors develop this idea, presenting a thermodynamic analysis of the solution using the heat pump and highlighting the optimal conditions in which this solution can be implemented.

2. Implementation of the heat pump in the heat supply system.

The heat pump is designed to operate in parallel with the heat exchangers from the geothermal station, supplementing the flow of the thermal agent, circulating in the centralized heat supply system of the city.

As the temperature of the thermal agent supplied to the heating system is 85 ^oC, the condensing temperature of the working agent must be at least 90 ^oC, which requires the use of high temperature agents, which have a critical temperature above 100 ^oC. In addition to meeting this requirement, the work agents must have an environmental impact in accordance with the provisions of the EU Regulation on fluorinated greenhouse gases [6] and have thermodynamic properties that allow obtaining a high efficiency of the plant. Table 2 shows a selection of some of the high temperature agents that meet these requirements and are currently used, being sorted by their GWP potential. The impact on the ozone layer is zero for all these agents.

Table 2

Usual working agents [7]

Agent	Group	t_0 [^oC]	p_0 [bar]	t_c [^oC]	t_{cr} [^oC]	ODP	GWP/ CO_2	Safety group
RE170	HC	25	6.908	90	127.2	0	1	A3
R1233zd(E)	HCFO	25	1.298	90	165.5	0	1	A1
R236fa	HFC	25	2.719	90	139.3	0	1.3	A1
R600a	HC	25	3.507	90	134.9	0	3	A3
R600	HC	25	2.433	90	154.2	0	4	A3
R1234ze(E)	HFO	25	4.985	90	109.4	0	6	A2
R152a	HFC	25	5.979	90	113.2	0	138	A2
R515b	HFO	25	4.974	90	108.9	0	293	A1
R245fa	HFC	25	1.486	90	154.0	0	858	B1
R134a	HFC	25	6.654	90	101.0	0	1300	A1
R123	HCFC	25	0.916	90	183.7	0.02	9100	B1

Most of the agents mentioned in Table 2 are of the “wet” type, with isentropic compression falling within the saturation range. To avoid this, a vapor overheating at the compressor suction by at least 10 degrees is required. Because the thermal agent from the heating system enters in the heat pump condenser at a temperature of 45 °C, the working agent of heat pump is possible to be subcooling to a temperature of around 60 °C, or even below, which increases the specific thermal load of the condenser and decreases the required working agent flow.

According to these conditions, a heat pump with mechanical compression of vapors was designed, in a single stage, having included in the scheme a regenerative heat exchanger to achieve superheating of the vapors at the compressor suction and a subcooler for the warm liquid agent, coming from the condenser. The operating diagram of the designed heat pump installation is shown in Fig. 4 and in Fig. 5 the thermodynamic cycle in the *p-h* diagram.

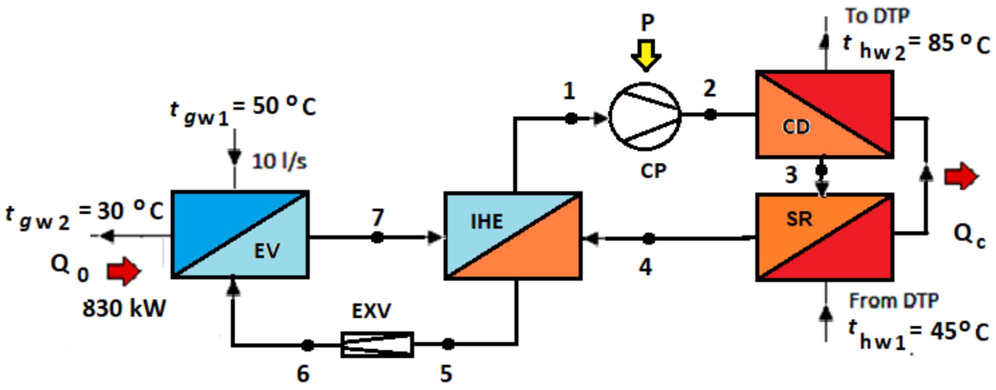

Fig. 4 The operating diagram of the heat pump installation.
EV – Evaporator; IHE-Internal Heat Exchanger; CP – Compressor; CD – Condenser, SR – Subcooler; EXV – Expansion Valve; DTP – Distribution Thermal Point.

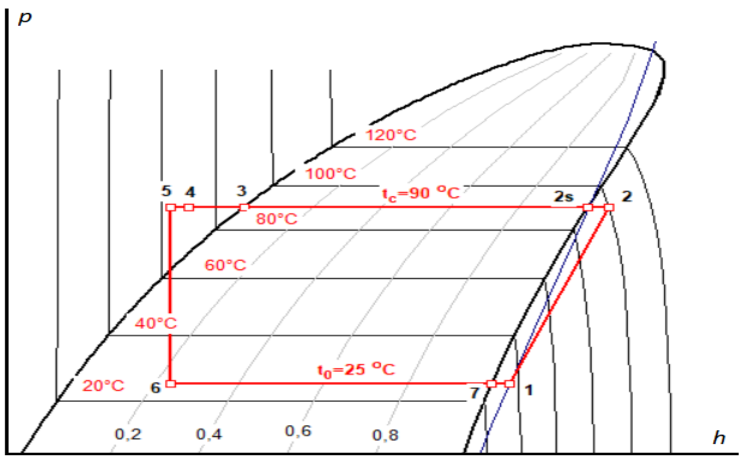

Fig.5 The thermodynamic cycle of the heat pump

Considering in the evaporator and condenser a temperature difference for the pinch-point of at least 5 degrees, results, in the given conditions, the vaporization temperature of the working agent $t_0 = 25$ °C and the condensation temperature $t_c = 90$ °C, whatever the working agent used. The vaporization and condensation pressures, respectively, are the saturation pressures corresponding to the mentioned temperatures and depend on the agent considered:

$$p_0 = p_{sat}(t_0); \quad p_c = p_{sat}(t_c) \tag{1}$$

Compression of the working agent is considered irreversible adiabatic; the isentropic efficiency of the compressor being set to $\eta_c = 0.8$. The degree of superheating of the vapors at the suction of the compressor is set to the value of $\Delta t_{sh} = 10$ degrees, and the degree of subcooling at the exit of the condenser to the value of $\Delta t_{sc} = 20$ degrees. The superheating of the vapors at the suction of the compressor is done with the help of the hot liquid coming from the condenser, by means of a heat exchanger. Because the same fluid flow flows on either side of the heat exchange surface, the energy balance of this apparatus is in the form of:

$$h_4 - h_5 = h_1 - h_7 \tag{2}$$

relationship that causes the condensation to cool depending on the degree of overheating required.

Table 3

Algorithm for determining the state parameters

State	Description	Pressure	Status parameters
1	Superheated vapors at the compressor suction	p_0	$t_1 = t_0 + \Delta t_{sh}$ $h_1 = h(p_0, t_1)$ $s_1 = s(p_0, t_1)$
2s	Superheated vapors at the end of the isentropic compression	p_c	$s_{2s} = s_1$ $t_{2s} = t(p_c, s_{2s})$ $h_{2s} = h(p_c, s_{2s})$
2	Superheated vapors at the end of the real, irreversible, compression	p_c	$h_2 = h_1 + (h_{2s} - h_1)/\eta_c$ $t_2 = t(p_c, h_2)$
3	Saturated liquid at the exit of the condenser	p_c	$x_3 = 0$ $t_3 = t_c$ $h_3 = h(p_c, x_3)$
4	Subcooled liquid at the exit of the subcooler	p_c	$t_4 = t_3 - \Delta t_{sc}$ $h_4 = h(p_c, t_4)$
5	Subcooled liquid at the inlet to the expansion valve	p_c	$h_5 = h_4 - (h_1 - h_7)$ $t_5 = t(p_c, h_5)$
6	Two-phase mixture after expansion valve	p_0	$h_6 = h_5$ $t_6 = t_0$ $x_6 = x(p_0, h_6)$
7	Saturated vapors at the exit of the evaporator	p_0	$x_7 = 1$ $t_7 = t_0$ $h_7 = h(p_0, x_7)$

Table 3 shows the algorithm for determining the state parameters in the points of the operation diagram presented in Fig. 4 and *p-h* diagram from Fig. 5, respectively.

The characteristic sizes of the heat pump installation are:

- specific thermal load of the condenser:

$$|\mathbf{q_c}| = \mathbf{h_2} - \mathbf{h_4} \quad \text{[kJ/kg]} \tag{3}$$

- specific cooling power of the evaporator

$$q_0 = h_7 - h_6 \quad \text{[kJ/kg]} \tag{4}$$

-specific mechanical compression work

$$|l_c| = h_2 - h_1 \quad \text{[kJ/kg]} \tag{5}$$

The cooling capacity of the analyzed heat pump depends on the available waste geothermal water flow $\dot{V}_{gw}$ and the temperature difference on which it can be cooled:

$$\dot{Q}_0 = \dot{V}_{gw}\ \rho_{gw}\left(t_{gw1} - t_{gw2}\right) \quad \text{[kW]} \tag{6}$$

where ρ_{gw}, the density of waste geothermal water, must be considered at average temperature $t_{gw} = 0.5\left(t_{gw1} + t_{gw2}\right)$. The flow of working agent is determined by the heat flow taken from the waste geothermal water:

$$\dot{m} = \frac{\dot{Q}_0}{q_0} \quad \text{[kW]} \tag{7}$$

The heat flow given to the condenser and the driving power of the compressor are given by the relations:

$$\dot{Q}_c = \dot{m}\ |q_c| \quad \text{[kW]} \tag{8}$$

$$P_c = \dot{m}\ |l_c| \quad \text{[kW]} \tag{9}$$

resulting in the value of the coefficient of performance:

$$COP = \frac{\dot{Q}_c}{P_c} \quad \text{[kW]} \tag{10}$$

3. Results

To determine which of the working agents presented in Table 2 is more suitable in terms of properties and performances for the heat pump installation, the state quantities were determined for each of the agents, at the points of the thermodynamic cycle in Fig. 5, according to the calculation algorithm presented in table 3. The compressor driving power and the heat flow yielded at the condenser were also calculated, resulting in the performance coefficient of the installation. The initial conditions were considered the same for each working agent: vaporization temperature: $t_0 = 25°C$; condensation temperature:

$t_c = 90$ ℃; degree of superheating at the compressor suction: $\Delta t_{sh} = 10$ degrees, subcooling degree of the condense: $\Delta t_{sc} = 20$ degrees. The isentropic efficiency of the compression process was set to 0.8. The results obtained under these conditions, sorted by the value of the coefficient of performance are presented in Table 4.

Table 4

Heat pump installation performance for different agents

Agent	q_0 [kJ/kg]	$\lvert q_c \rvert$ [kJ/kg]	$\lvert l_c \rvert$ [kJ/kg]	$\dot{m}$ [kg/s]	$\dot{Q}_c$ [kW]	P [kW]	COP	Safety group
R515b	106.4	144.9	38.5	7.801	1130	300.3	3.76	A1
R134a	117.5	159.5	42.0	7.064	1127	296.7	3.80	A1
R245fa	106.3	143.4	37.1	7.808	1120	289.9	3.86	B1
R1234ze(E)	110.7	149.1	38.4	7.498	1118	287.7	3.88	A2
R236fa	95.8	128.3	32.5	8.664	1112	281.6	3.95	A1
R152a	205.3	274.3	69.0	4.043	1109	279.0	3.98	A2
R600a	230.0	306.1	76.1	3.609	1105	274.7	4.02	A3
RE170	301.3	399.9	98.6	2.755	1102	271.7	4.06	A3
R600	262.9	347.2	84.3	3.157	1096	266.0	4.12	A3
R1233zd(E)	138.4	181.9	43.5	5.997	1091	260.9	4.18	A1
R123	131.0	171.4	40.4	6.336	1086	255.8	4.24	B1

From the analysis of the data contained in Table 4, under the given conditions there are several agents that allow to achieve a high efficiency, above the value 4, and have a zero impact on the environment: hydrocarbon class agents (RE170 - Dimethylether, R600 - Butane, R600a - Isobutane) and high temperature agents (R123 - HCFC, R1233zd (E) - HCFO). Hydrocarbons, although having good thermodynamic properties, were eliminated for use in the heat pump plant designed, due to their high flammability (safety group A3), remaining in discussion the agents R123 and, recently placed on the market, R1233zd (E).

R1233zd (E) refrigerant is chemically a hydrochlorofluoroolefin (HCFO). Although this refrigerant is an HCFC and therefore carries chlorine that affects the ozone layer, it is not on the list of fluorinated greenhouse gases that will be eliminated because the life in the atmosphere is very short (26 days). The properties of this agent are close to the characteristics of the ideal refrigerant: adequate operating pressures, zero GWP potential, zero ozone depletion potential (ODP), non-flammable and non-toxic (safety group A1), adequate volumetric capacity [8].

R123 refrigerant is an HCFC, seen as an ideal alternative to the CFC R11 freon replacement. Extremely efficient, it has a very low ODP (0.02) and, although not considered significant at the time, a GWP of about 9100. Major problems, which put this agent on the replacement list worldwide, are related to its toxicity (safety group B1). Its vapors are heavier than air and can cause suffocation by reducing the oxygen available for respiration. It causes heart disease and an increased incidence of benign tumors in the liver, pancreas, and testis [9].

Based on these arguments, it was decided to use the refrigerant R1233zd (E) in the designed heat pump installation. For the chosen working agent, it was investigated how the efficiency of the installation and the heat flow given to the condenser depend on the variable parameters: the degree of subcooling and the degree of overheating at the compressor suction. The results are sown in Fig. 6.

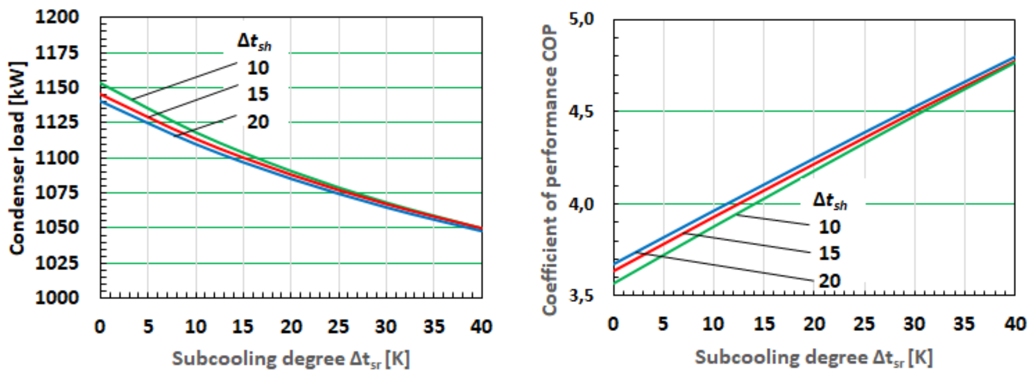

Fig. 6. Changing condenser load and efficiency in relation to the degree of subcooling and the degree of overheating

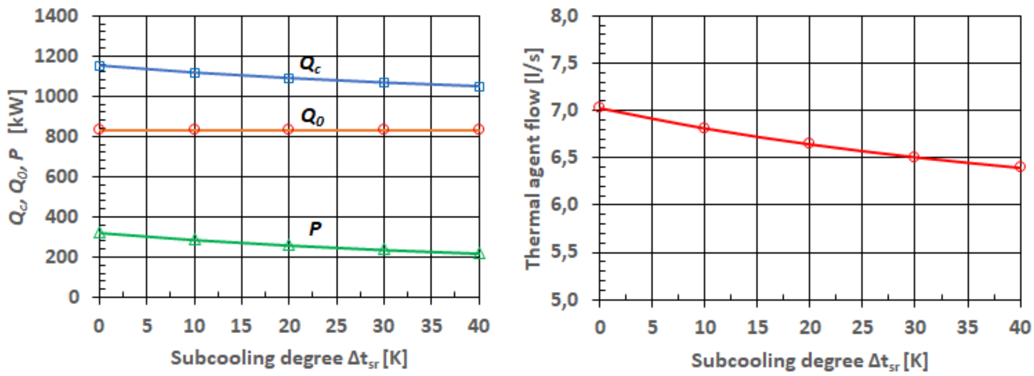

Fig.7. Evaporator cooling power, condenser thermal power and compressor drive power for the overheating degree of 10 degrees and maximum flow of geothermal water.

It is found that the greatest influence upon the heat pump performances has the subcooling degree of the condensate, while the influence of the degree of overheating at the compressor suction is practically insignificant. For this reason, the minimum value of 10 degrees was considered for the degree of overheating, which ensures that the state corresponding to the compressor discharge does not enter in the wet area and was investigated the way in which only the degree of subcooling influences the performance of the heat pump. The results are shown in Fig. 7.

6. Conclusions

Given the planned elimination in the next decade of type F gases, manufacturers of refrigerators and heat pumps have developed equipment that works with "ultra-green" agents. Natural refrigerants and hydrofluoroolefins (HFOs) are the two types of such agents, due to their zero ODP and GWP potentials. Natural refrigerants are natural and non-synthetic substances wich include hydrocarbons (propane, butane and cyclopentane), CO2, ammonia, water and air. Hydrofluoroolefins are unsaturated organic compounds of hydrogen, fluorine and carbon, offering a much more environmentally friendly alternative to conventional CFC, HCFC and HFC refrigerants. For this reason, the authors comparatively analyzed the performance of the proposed installation for the classic agents in use and the latest generation agents, finding that in addition to their zero environmental impact, similar or even better performance is obtained.

Considering a certain scenario for the operation of the heat pump, it turned out that the refrigerant R1233zd (E) is the most suitable, offering a high energy efficiency, comparable to that of the refrigerant R123, but unlike it is non-toxic and has virtually zero GWP potential. The chosen refrigerant is of the “wet” type, requiring an overheating at the compressor suction. The study showed that the performance of the heat pump depends insignificantly on the degree of overheating and as a result was considered a minimum overheating (10 degrees), avoiding the entry of compression in the wet area.

The parameter that most influences the performance is the subcooling degree of condense. By subcooling of the condense to the minimum possible temperature under the given operating conditions, a 33% increase in energy efficiency and a 10% increase in the heat delivered at the condenser is obtained, compared to the case of operation without subcooling.

By recovering the heat from the hot wastewater discharged from the heat exchangers of the geothermal station, with the help of the heat pump running with refrigerant R1233zd (E), under the conditions highlighted by the study, the capacity of the district heating system of the city can be expanded by about 60 %.

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