Energy Efficiency of Heat Pumps Heating Systems at Subsoil Waters for South-East Regions of Europe

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Abstract. This article is devoted to the methods for increasing the operational efficiency of the single-stage and two-stage schemes of heat pump installations for heating systems, based on the energy saving principles. The major aim of the study is the analysis of methods for increasing the efficiency of alternative low-potential heat pump installations based on the subsoil waters, which correspond to the requirements of the energy saving technologies. To achieve the aim a comparative analysis of efficiency of different schemes of heat pump heating systems was performed for the consumers of the South-East of Europe. The rational schematic and constructive solutions and the system operational modes that ensure the increase in the efficiency of the alternative heating system for various climatic conditions were grounded. The main ways of increasing the efficiency of the low-potential heating systems using the heat pump units based on the subsoil waters were determined. Recommendations for the practical application of the alternative heating system solutions, depending on temperature of the outside air were developed. The significance of the obtained results consists in justification of conditions, which make it possible to use the single- and two-stage HPI schemes on the subsoil waters in the South-Eastern Europe. The most significant results are those recommending the increase in the operational efficiency of the heat pump systems on the subsoil waters for the heat supply for the consumers of the South-Eastern Europe. The analysis results can be used for designing the heating systems based on the heat pumps using the low-potential energy of the subsoil waters.
Keywords: energy saving, efficiency, transformation coefficient, heat pump, subsoil waters, intermediate heat exchanger, subcooler, primary energy.
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Analiza eficienței energetice a instalațiilor cu pompă de căldură bazate pe apele subterane pentru regiunile sud-estice ale Europei

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Rezumat. În articol se discută modalitățile și metodele de îmbunătățire a eficienței schemelor cu un singur stadiu și cu două etape ale instalațiilor de pompare a căldurii în scopuri de încălzire, pe baza principiilor economisirii de energie. Scopul principal al studiului constă în analiza modalități de îmbunătățire a eficienței instalațiilor alternative cu pompă de căldură cu potențial termic redus bazate pe ape subterane care îndeplinesc cerințele tehnologiilor de economisire a energiei. Pentru a atinge obiectivul stabilit al studiului, a fost efectuată o analiză comparativă a eficacității diferitelor scheme din cadrul sistemelor de alimentare cu căldură prin pompare de căldură pentru consumatorii din regiunile sud-estice ale Europei. A fost dezvoltat un algoritm pentru calcularea sistemelor alternative de alimentare cu căldură pentru consumatori, luând în considerare condițiile climatice. Este propusă o metodă de evaluare a eficienței energetice a schemelor alternative de alimentare cu căldură eleborate. Implementarea metodei este prezentată pe un exemplu specific și se confirmă eficiența încălzirii alternative pentru consumatori. Importața rezultatelor obținute este de a fundamenta condițiile în care este recomandabil să se utilizeze schemele cu un etaj și două etaje a pompelor de căldură cu utilizarea apelor subterane din regiunile sud-estice ale Europei. Cele mai semnificative rezultate sunt recomandări pentru îmbunătățirea eficienței sistemelor cu pompe de căldură pe baza apelor subterane pentru alimentarea cu căldură a consumatorilor din regiunile sud-estice ale Europei. Rezultatele analizei pot fi utilizate pentru a proiecta rețele de căldură bazate pe pompe de căldură care utilizează energie cu potențial termic redus a apelor subterane.
***Cuvinte-cheie*:** economie de energie, eficiență, raport de transformare, pompă de căldură, apă subterană, schimbător de căldură intermediar, subrăcitor, energie primară.

Энергетическая эффективность теплонасосных установок на базе грунтовых вод для Юго-Восточных регионов Европы

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Аннотация. В статье рассмотрены пути и методы повышения эффективности работы одноступенчатых и двухступенчатых схем теплонасосных установок для целей отопления, основанные на принципах энергосбережения. Основной целью исследования является анализ способов повышения эффективности альтернативных низкопотенциальных теплонасосных установок на базе грунтовых вод, которые соответствуют требованиям энергосберегающих технологий. Для достижения поставленной цели исследования был проведен сравнительный анализ эффективности различных схем тепловых насосных систем теплоснабжения для потребителей Юго-Восточных регионов Европы. Разработан алгоритм расчета альтернативных систем теплоснабжения потребителей с учетом климатических условий. Предложена методика оценки энергоэффективности предложенных альтернативных схем теплоснабжения. Реализация метода показана на конкретном примере и подтверждена эффективность альтернативного отопления для потребителей. Проведено численное моделирование тепловых процессов в элементах одноступенчатой и двухступенчатой теплонасосной установки, коэффициентов преобразования теплоты, мощности, потребляемой электроприводом компрессора. Определены удельные затраты первичной энергии и проведено численное моделирование параметров циклов для предложенных схемных решений. Приведен анализ результатов численного моделирования тепловых процессов в элементах теплонасосной установки. Обоснованы рациональные схематические и конструктивные решения и режимы работы системы, обеспечивающие повышение эффективности альтернативной системы отопления для различных климатических условий. Определены основные пути повышения эффективности низкопотенциальных систем отопления за счет использования тепловых насосов на базе грунтовых вод. Разработаны рекомендации по практическому использованию альтернативной системы отопления в зависимости от температуры наружного воздуха. Значимость полученных результатов состоит обосновании условий, при которых в Юго-Восточных регионах Европы целесообразно использовать одноступенчатые и двухступенчатые схемы ТНУ на грунтовых водах. Наиболее значимыми результатами являются рекомендации по повышения эффективности работы теплонасосных систем на грунтовых водах для теплоснабжения потребителей в Юго-Восточных регионах Европы. Результаты анализа могут быть использованы для проектирования тепловых сетей на базе тепловых насосов, использующих низкопотенциальную энергию грунтовых вод.

Ключевые слова: энергосбережение, эффективность, коэффициент трансформации, тепловой насос, грунтовые воды, промежуточный теплообменник, переохладитель, первичная энергия.

I. INTRODUCTION

Heat supply based on heat pumps (HP) is referring to the problems of the energy saving technologies and is found to be applied more extensively nowadays. One way to improve the energy supply systems is the problem solution of transition to the low-temperature heating systems based on the heat pump installations (HPI). This significantly expands the resources of the heat supply systems, making them less dependable on the use of the traditional primary energy resources (PER), which is of high priority in conditions of the resources exhaustibility and their increasing cost [1, 2]. The use of the energy saving tools is crucial for the innovation technical solutions aimed to improve the quality, reliability and economical efficiency of the systems of the alternative heat generation. For instance, the HPI application allows the use of the low-potential heat of the subsoil waters, which is not of a real value for its direct use in the heat supply systems [3]. In the heat pump cycle, the electrical energy is consumed for transition of the heat from a low-potential primary energy resource to the consumer at a high-potential level. Moreover, simultaneously, the thermal and power properties of the electrical energy are realized, due to which saving of traditional primary resources is reached [4].

The aim of the study is to improve the efficiency of the heat pump installations (HPI) based on the subsoil waters [5] for heating purposes at temperature of the ambient air of $t_0 = -18…8$ °C typical for the consumers of the South-East regions of Europe.

At present, the problem of energy saving can be solved both by the thermal losses reduction and assimilation of the innovation technologies of generating, distribution, regulation and consumption of the heat. One most efficient technology of the energy saving is the utilization of the heat pumps (HP), due to their possibility to use a renewable energy for the alternative heat supply [6—11]. However, the works presented in literature, which describe the peculiarities of the use of the heating tools for the low-temperature water heating, ventilation and heat water supply, are insufficient [12—16]. The foreign researches [17—20], lack the methods, which would describe the alternative HPI and conditions of their practical application in the heat supplying systems with different heating units for the environmental conditions of the South-Eastern Europe. In [21, 22], the effect on the replacement rate by the scheme-construction solutions and operational modes of the alternative heat-supply system is not considered. Therefore, the issue of conditions of the efficient use of the heat pump technologies needs a systematic approach.

Our work differs from those western papers presented earlier in that this article analyzes the efficiencies of various schemes of single-step and multi-step HPIs using a low-potential heat sources of subsoil waters for the heating systems with different heat supply units, which ensure the maximum replacement of the thermal load. This makes it possible to perform a rational choice of the conditions for the efficient operation of the heating system in winter period at the outside temperature of $t_0 = –18…8$°C typical for the South-Eastern Europe. This reduces the consumption of the hydrocarbon fuel in the structure of the heat balance of the regions and ensures the substantial energy saving and ecological effects in addition to those economical.

The attention payed by the foreign works [23—25] is insufficient, concerning the justification of the choice of the scheme-construction solutions in the alternative system of the heat supply, taking into account the effect of the basic elements of the system and the modes of its operation on the HPI replacing possibilities on the subsoil waters.

This work is aimed at the analysis of the energy efficiency of the HPI different schematic solutions with the low-potential heat sources in the form of the subsoil waters for the heating systems using different heating units (radiators and a warm floor), which operate at the temperature of the outside air of $t_0 = – 16…8$ °C, typical for the South-Eastern Europe.

II. EXPERIMENTAL

The development of the main heat schemes of the heat pump installations based on the subsoil waters is performed using the common methods of analysis and generalization. The rational HPI schemes for the purposes of the thermal supply for the consumers of the South-Eastern Europe are shown in Figs. 1—3 [26].

The scheme of the installation of the alternative heat supply using the low potential subsoil water energy (Fig. 1) suggests the presence of the intake well *1*, equipped with a feed pump *3*, which supplies subsoil water into the evaporator HP *4*, where the heat is delivered to the cooling agent, and then returns the subsoil water to the intake well *2*. In the evaporator, the HP working fluid boils. The vapor formed in the evaporator, is absorbed by compressor *5*; the vapor is compressed, and temperature and pressure of the cooling agent are increased. Then, the cooling agent enters the HP condenser *6* that is cooled with water, the latter becomes a high potential source of heat for the heating system *9*, equipped by circulator *8*. After the vapor heat transmission, the cooling agent is condensed, and in the liquid state it moves to expansion valve *7*, where its temperature and pressure drop, then it returns into the evaporator.

Figure 2 shows the HPI scheme with additional intermediate heat exchanger *10*, in which supercooling of the liquid cooling agent is combined with a superheating of its vapor, which allows the hot cooling agent after the condenser to heat the cool cooling agent after the evaporator. This increases the efficiency of operation of the present installation due to the temperature increase in the cooling agent at the compressor inlet.

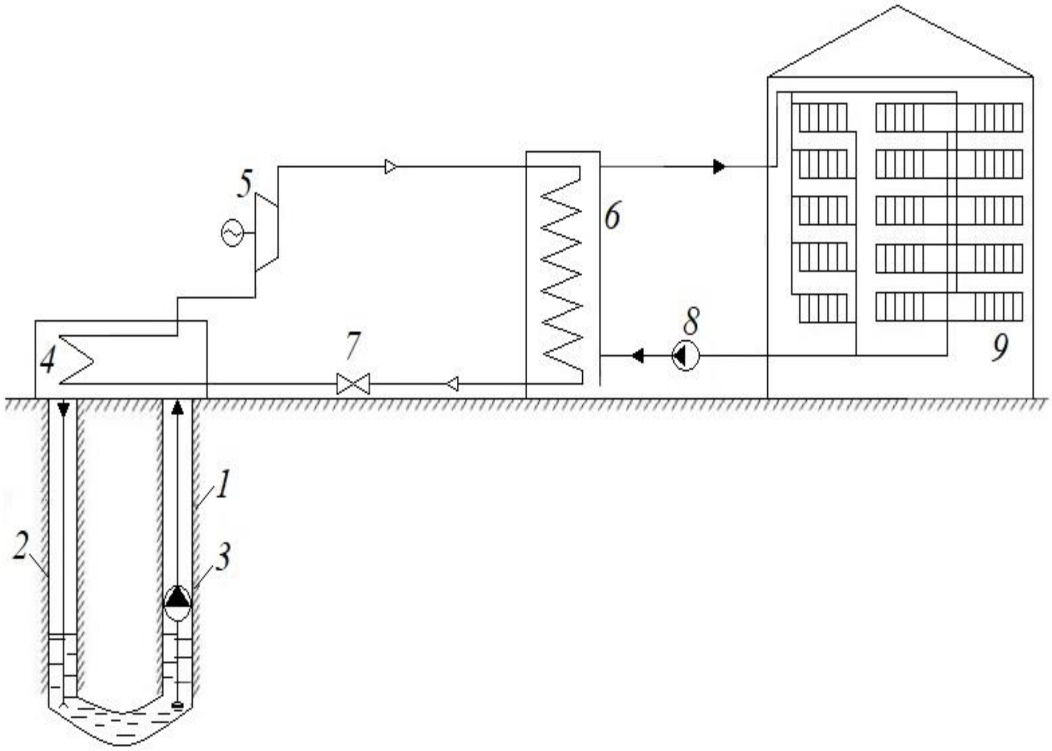
Figure 3 shows the HPI scheme with heat exchanger-supercooler *11*, which serves as the additional cooler for the high potential heat source.

The estimation of the replacing possibilities of the HPI schemes under study (Figs 1—3) takes place at a temperature of the outside air in winter period of $\boldsymbol{t_0 = -18}$ °C, which is typical for the environmental conditions of the South-Eastern Europe.

The low potential HPI source of energy is the subsoil water in the temperature range of $\boldsymbol{t_{n1} = 4 ... 12}$ °C.

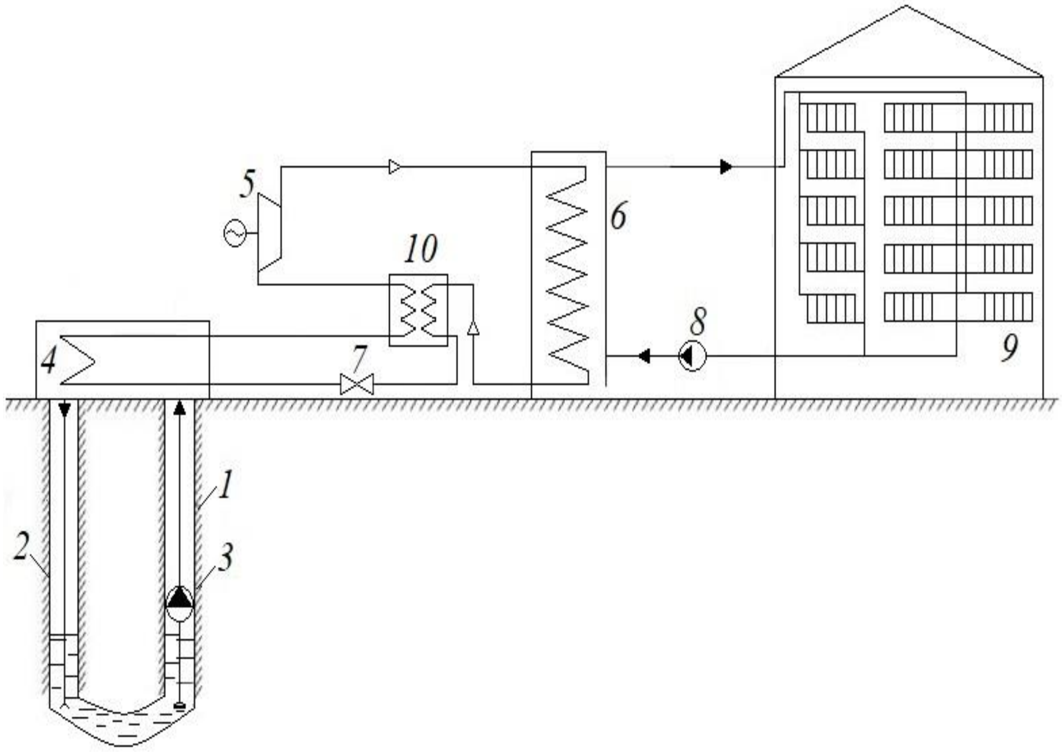
The estimation of the efficiency of the HPI application using the subsoil waters is performed using the method for the mathematic modeling of the HPI cycle with the use of the fundamental equations of thermodynamics, applying common dependences and expressions.

The CoolPack program was used during realization of the mathematic model of the heat pump cycle. For the analysis of the results of the energy efficiency modeling indices of the HPI schemes under consideration, the methods of [27] were used, applying the CoolPack program in *p*, *h*-diagram at the environment temperature of $t_0 = – 6$ °C (Fig. 4).



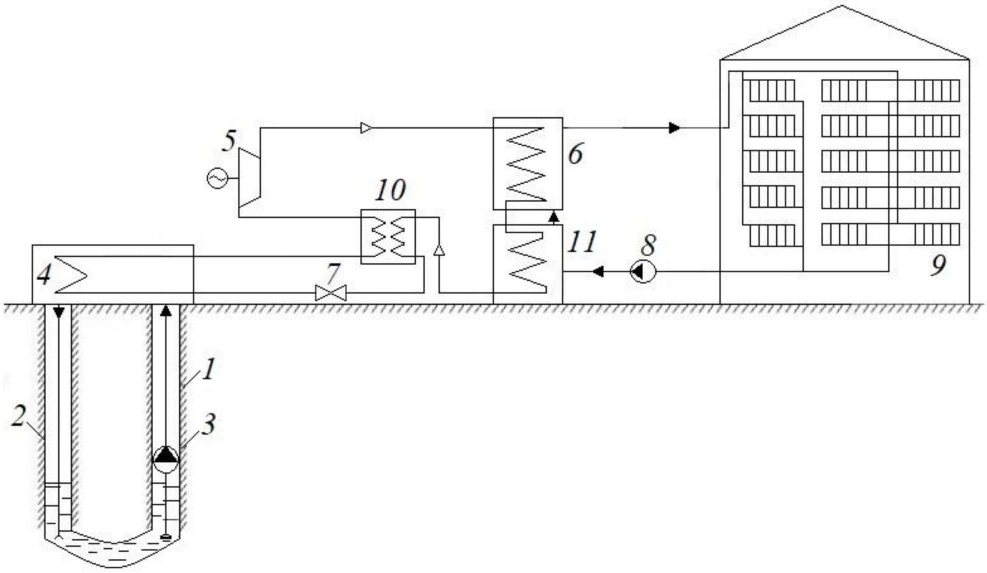
1– intake well; 2– absorption well; 3– feed pump; 4 – evaporator; 5 – compressor; 6 – capacitor; 7 –throttle; 8 – circulation pump; 9 – heating system

Fig. 1. Scheme of HPP based on groundwater.



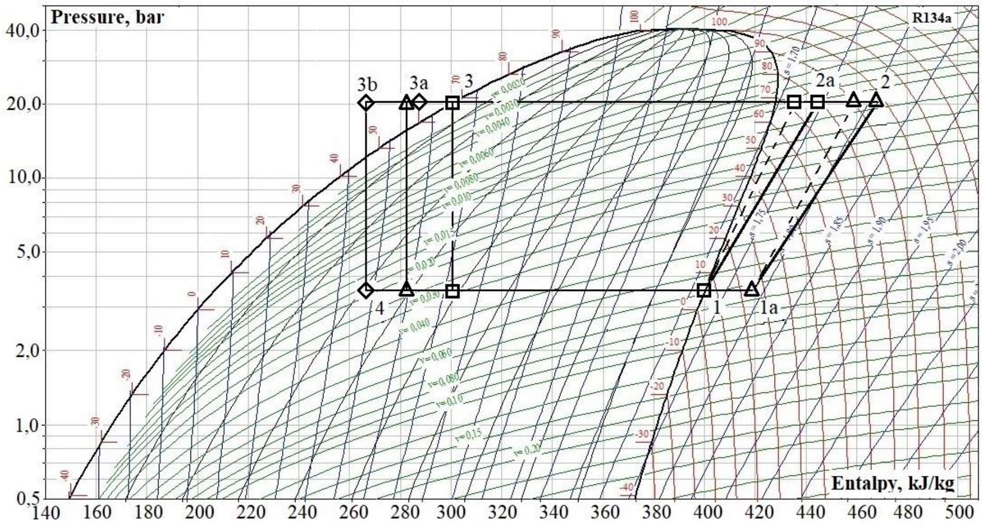
1– intake well; 2– absorption well; 3– feed pump; 4 – evaporator; 5 – compressor; 6 – capacitor; 7 –throttle; 8 – circulation pump; 9 – heating system; 10 – intermediate heat exchanger

Fig. 2. Scheme of HPI based on groundwater with intermediate heat exchanger.



1– intake well; 2– absorption well; 3– feed pump; 4 – evaporator; 5 – compressor; 6 – capacitor; 7 –throttle; 8 – circulation pump; 9 – heating system; 10 – intermediate heat exchanger; 11 – heat exchanger-supercooler

Fig. 3. Scheme of HPI based on groundwater with intermediate heat exchanger and subcooler



☐ *– HPI (Fig. 1);* Δ *– HPI (Fig. 2);* ◊ *– HPI (Fig. 3);*
1a – parameters of the refrigerant at the compressor inlet (Figs. 2, 3); 2a – parameters of the refrigerant at the compressor outlet (Figs. 2, 3); 3a – after the condenser (Fig. 2); 3b – at the inlet to the intermediate heat exchanger (Fig. 3); 4 – parameters of the refrigerant at the inlet to the evaporator

Рис. 4. Cycles of different schemes of HPI based on ground energy (for R134a freon).

III. РЕЗУЛЬТАТЫ ИССЛЕДОВАНИЯ

Let us use the methods of the heat pump cycle calculation [26] to analyze the energy efficiency of the application of the HPI three schemes under study (Figs. 1—3) for the South-Eastern Europe in winter period $(t_0 = -18^0 C)$.

Freon evaporation temperature t_0, which can be used to determine enthalpy h_1 and pressure P_0 of freon after the evaporator:

$$t_u = t_{n2} - \Delta t_u, \quad (1)$$

where t_{n2} is the temperature of the low-potential heat source, °C; and Δt_u is the temperature difference at the outlet from the evaporator, K.

Temperature of freon condensation t_k, which can be used to determine enthalpy h_3; and pressure P_k of freon after the condenser is as follows:

$$t_k = t_{w2} - \Delta t_k, \qquad (2)$$

where t_{w2} is the temperature of the high-potential heat source at the outlet from the evaporator, °C; and Δt_k – перепад температур на выходе из конденсатора, К.
Adiabatic efficiency coefficient of the compressor is the following:

$$\eta_a = 0,98\left(273 + t_0\right) / \left(273 + t_k\right), \qquad (3)$$

where t_0 is the temperature of the outside air, °C.
Freon enthalpy after the compressor is as follows:

$$h_2 = h_1 + \left(h_{2a} - h_1\right) / \eta_a, \qquad (4)$$

where h_{2a} is the freon enthalpy after the process of the adiabatic compression, kJ/kg.
The condenser specific heat load is the following:

$$q_k = h_2 - h_3, \text{ kJ/kg.} \qquad (5)$$

The HPI specific heat load is:

$$q_{hp} = h_k, \text{ kJ/kg.} \qquad (6)$$

The work of compression in the compressor is as follows:

$$l_k = h_2 - h_1, \text{ kJ/kg.} \qquad (7)$$

The specific energy, which is consumed by the electric motor is the following:

$$W = l_k / \left(\eta_{em} \cdot \eta_e\right), \text{ kJ/kg,} \qquad (8)$$

where η_{em} is the electromechanical coefficient of efficiency of the compressor; and η_e – КПД электродвигателя.
Coefficient of heat performance is:

$$\mu = q_{hp} / l_k. \qquad (9)$$

Compression coefficient in the compressor is:

$$\varepsilon = P_k / P_u. \qquad (10)$$

Mass flow of freon:

$$G_f = Q_{hp} / q_{hp}, \text{ kg/s,} \qquad (11)$$

where Q_{hp} is the HPI heat load, kW.
Electrical energy consumption for the compressor drive:

$$N = W \cdot G_f, \text{ kW.} \qquad (12)$$

Specific consumption of the primary energy:

$$PER = 1 / \left(\eta_{em} \cdot \eta_e \cdot \eta_{eu} \cdot \eta_{per} \cdot \mu\right), \qquad (13)$$

where η_{eu} is the installation coefficient of efficiency; and η_{per} is the coefficient of efficiency of the electrical energy supply system.
For the HPI schemes (Figs. 2—4), the cooling agent temperature at the inlet into compressor t_{1a}:

$$t_{1a} = t_u - \Delta t_{pt}, \qquad (14)$$

where Δt_{pt} is the temperature of the vapor superheat in the intercooler.
Cooling agent enthalpy after the compressor:

$$h_2 = h_{1a} + \left(h_{2a} - h_{1a}\right) / \eta_a. \qquad (15)$$

The cooling agent enthalpy at the inlet into the intercooler (Figs. 3, 4):

$$h_{3b} = h_3 - \left(h_{1a} - h_1\right). \qquad (16)$$

The work of compression in the compressor:

$$l_k = h_2 - h_{1a}, \text{ kJ/kg.} \qquad (17)$$

For the HPI scheme (Fig. 3), the cooling agent temperature after the condenser:

$$t_{3a} = \left(\frac{c_{p3}^{/} \cdot t_k + c_w \left(\Delta t_{pt} + t_{w1}\right)}{c_{p3}^{/} + c_w} \right), \qquad (18)$$

where $c_{p3}^{/}$ is the freon heat capacity after the condenser, kJ/(kg·K); c_w is the heat capacity of water, kJ/(kg·K); t_{w1} is the temperature of the high-potential source of heat at the inlet into the evaporator.
The specific heat load of the super cooler

$$q_{po} = h_3 - h_{3a}, \quad \text{kJ /kg.} \qquad (19)$$

The HPI specific heat load:

$$q_{hp} = q_k - q_{po} \text{ , kJ/kg.} \qquad (20)$$

To calculate the energy efficiency of the HPI schemes (Figs. 1—3) the following basic data were used:

$t_{n1} = 20^o C$ is the temperature of the low-potential source of heat at the inlet of the HPI evaporator;

$t_{n2} = 10^o C$ is the temperature of the low potential source of heat at the outlet of the HPI evaporator;

$\Delta t_k = \Delta t_u = \Delta t_{po} = 5K$ is the difference in temperatures at the outlet from the heat exchanges;

$\Delta t_k = 20$ K is the temperature of the vapor superheat in the intercooler;

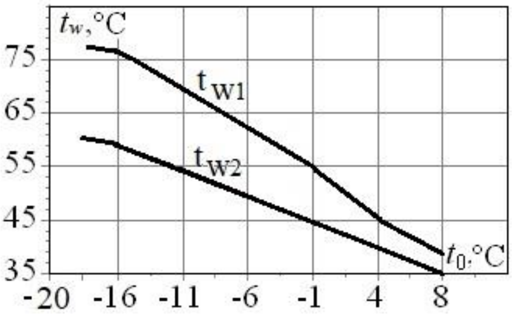
$\eta_e = 0,85$ is the electric motor coefficient of efficiency;

$\eta_{em} = 0,95$ is the compressor electromechanical coefficient of efficiency;

$\eta_{TTP} = 0,4$ is the thermal power plant coefficient of efficiency;

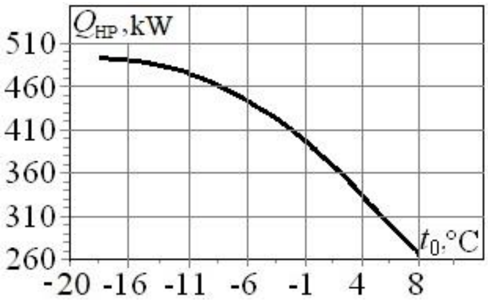
$\eta_{PER} = 0,95$ is the energy supply systems coefficient of efficiency.

Figures 5, 6 show the graphs of temperature change of the high potential source of heat at the inlet t_{w1} and outlet t_{w2} of the heating system and the graphs of change in the HPI heat load, correspondingly. For a 50-floor building with 60 apartments, with the outside air temperature change in the range of $t_0 = -18...8^0 C$, using sectional heating appliances (radiators), the heat load of the heat supply is $Q_{hp} = 502$ kW.



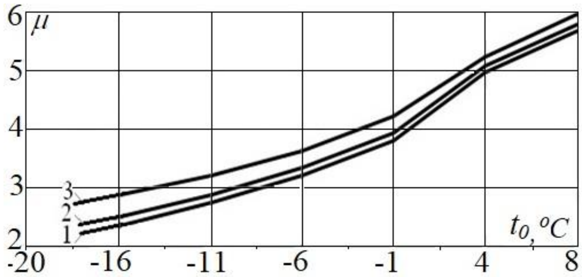
t_{w1} *– inlet temperature;* t_{w2} *– outlet temperature;*
t_0 – outdoor temperature

Fig. 5. Changing of the temperature of the high-potential heat source in inlet and outlet of the heating system.



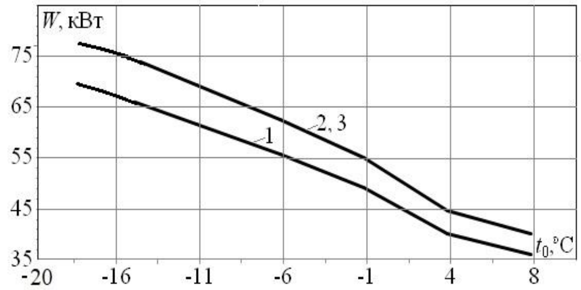
t_{HP} *– heat load of HPI;* t_0 *– outdoor temperature*

Fig. 6. Changing of heat load of the system.



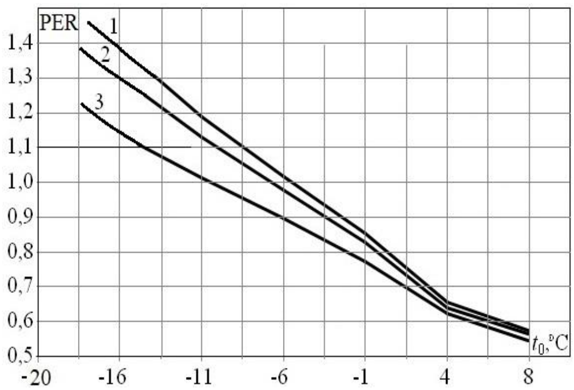
1– HPI (Fig. 1); 2 – HPI (Fig. 2); 3 – HPI (Fig. 3)

Fig. 7. Changing of coefficient of performance of heat.



1– HPI (Fig. 1); 2 – HPI (Fig. 2); 3 – HPI (Fig. 3)

Fig. 8. Changing of specific energy for drive of compressor of HPI.



1– HPI (Fig. 1); 2 – HPI (Fig. 2); 3 – HPI (Fig. 3)

Fig. 9. Changing of specific consumption of the primary energy.

In order to determine the advisability of a two-stage HPI on the subsoil waters for the consumers of the South-Eastern Europe we shall analyze the conditions of its application for the efficiency increase in the system of the alternative heat supply.

In a two-stage heat pump installation (Fig.10), the water from the heating system 9 is supplied by pump 8 for heating into the HP condensers (6, 7) mounted in series along the system water. In the HP condensers there are two separate zones for cooling the superheated vapor and condensation. The counterflow scheme of motion of the working carrier of the cycle and the heating water, increases the water temperature at the condenser outlet and decreases the energy losses. In condenser 7 of the first stage, the water is heated from temperature t_{w1} to certain intermediate temperature t_{pr}. After that, the water moves to condenser 6 of the second stage, where it is heated to temperature t_{w2}. The low potential heat of the subsoil waters is transferred to the evaporator 3 to the boiling HP working medium, whose vapors at a pressure of P_0 move into compressor 1 of the low stage, where they are compressed up to P_{pr}, after which, the waters are divided into two flows. The first flow shifts into condenser 7, where it is condensed in the process of the heat delivery to the heated water. The second flow travels to compressor 2 of the upper stage, where it is compressed up to P_k pressure and then it moves into condenser 6, where the water is heated up to temperature from t_{pr} to t_{w2}. After that, the condensate of the working media via throttle valve 5 moves to condenser 7, and a total condensate flow shifts to evaporator 3 from condenser 7 via throttle valve 4.

Let us use methods of [26] to evaluate the possibilities of covering the heat load of heating by the two-stage HPI based on the subsoil waters, whose temperature is $t_{n2} = 4...12^oC$, with temperature of the outside air $t_0 = -18^oC$ typical to the South-Eastern Europe winter season.

The intermediate pressure of compression is as follows:

$$P_{pr} = \left(P_u \cdot P_k\right)^{0,5}. \qquad (21)$$

During the calculation of the two-stage HPI (Fig. 6), the freon consumption in the circuits of low $G_{f.low}$ and high $G_{f.high}$ pressure is determined from the energy balance:

$$G_{f.low}\left(h_4 - h_9\right) = G_{f.high}\left(h_4 - h_8\right), \qquad (22)$$

where h_4, h_8, h_9 is the enthalpy at the working points (4, 8, 9) of the HP cycle, kJ/kg.

The ratio of the working media consumption in the circuit of low pressure $G_{f.low}$ to the consumption in the circuit of high pressure $G_{f.high}$ is:

$$\frac{G_{f.low}}{G_{f.high}} = \left(h_4 - h_8\right) / \left(h_4 - h_9\right) = 1 / \left(1 + \delta\right), \qquad (23)$$

where δ is the share of freon vapors from the condenser of the first stage with regard to the vapors of the first stage of the compressor.

The enthalpy of freon vapors that shift to the second stage of the high pressure compressor is as follows:

$$h_3 = (h_2 + \delta \cdot h_4)/(1+\delta), \text{ kJ/kg}, \qquad (24)$$

where h_2 is the enthalpy of Freon after the first stage of the compressor, kJ/kg.

The freon consumption in the high circuit pressure is:

$$G_{f.high} = Q_{hp} / \left(h_5 + h_7\right), \text{ kg/s} \qquad (25)$$

where h_5 is the enthalpy of freon after the second stage of the compressor, kJ/kg. Freon consumption in the low pressure circuit is:

$$G_{f.low} = G_{f.high} / \left(1 + \delta\right), \text{ kg/s} \qquad (26)$$

The consumption of the mechanic energy for the drive of the compressor of two stages, kW:

$$N_m = G_{f.high}\left(h_5 - h_3\right) + G_{f.low}\left(h_2 - h_1\right). \qquad (27)$$

The electric energy consumption for the compressor drive is:

$$N = N_m / \eta_{em}, \text{ kW} \qquad (28)$$

The coefficient of heat performance:

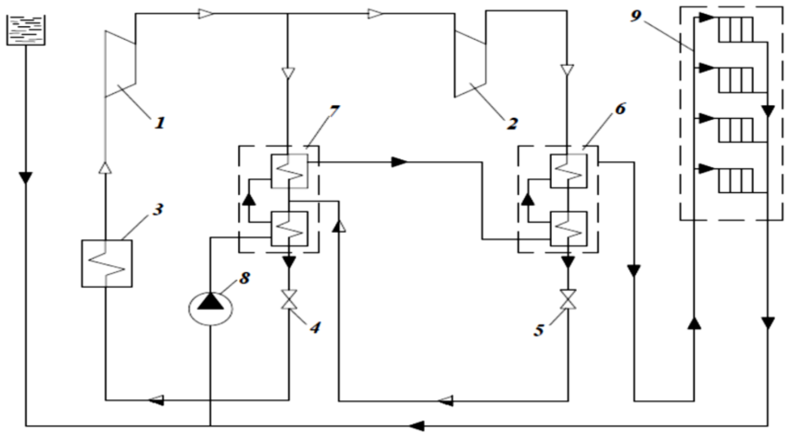
$$\mu = Q_{hp} / N. \qquad (29)$$

The research of the energy efficiency for the use of the HPI two-stage schemes is performed for the cycles with the application of ecologically safe R134a, R152a, R290 freon.

The calculation cycle of the HPI two-stage scheme is built in *p*, *h*-diagram for the working

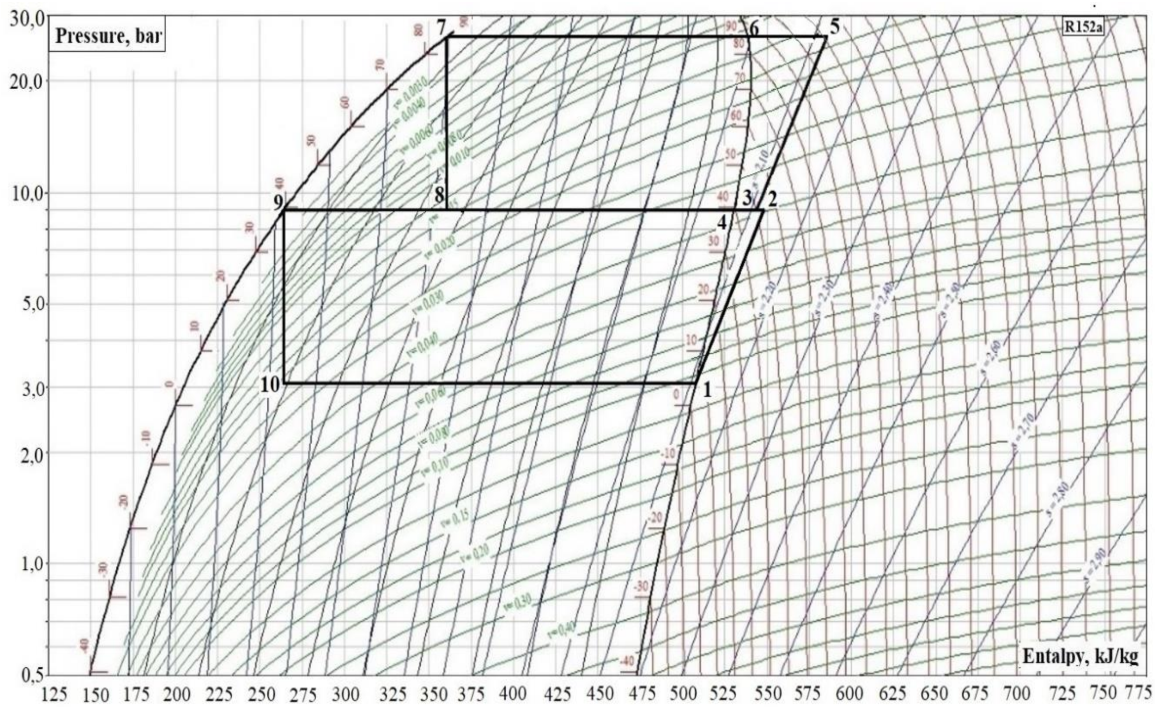
medium R152a and environmental temperature of $t_0 = -6$ °C using the CoolPack program [26].
For the analysis of the energy efficiency of the application for the consumers of the South-Eastern Europe of the schemes under consideration of the single-stage HPI (Figs. 1—3) and the two-stage HPI (Fig. 10), at other conditions being equal, the following results were obtained (Table 1, Figs. 5, 6).

The calculation results listed in Table 1 show that using the two-stage HPI reduces the consumption of the electric energy for the compressors' drive and the coefficient of heat performance increases. Therefore, the use of the two-stage HPI allows increasing the amount of the generated heat at a similar consumption of the electric energy compared to the single-stage HPI.



1, 2 – compressor of the first and second stages, respectively; 3 – evaporator; 4, 5 – throttle valve; 6, 7 – capacitor of the second and first stages, respectively; 8 – pump; 9 – heating system

Fig. 10. Scheme of two-stage HPI.



1 – parameters of the refrigerant at the inlet to the compressor of the first stage; 2 – parameters of the refrigerant at the outlet of compressor of the second stage; 3 – parameters of the refrigerant at the inlet to evaporator; 4, 5 – parameters of the refrigerant after throttle valves, respectively; 6, 7 – parameters of the refrigerant after condensers of the second and first stages, respectively; 10 – parameters of the refrigerant at the inlet to the evaporator

Fig. 11. Cycle of two-stage HPI based on ground energy (freon R152a).

Table 1.

HPI indices of energy efficiency

$t_{n1}, {}^{o}C$	Electric energy consumption for the compressor drive, N, kW		Coefficient of heat performance μ	
	Single-stage	Two-stage	Single-stage	Two-stage
10	221,2	174,7	2,4	2,8
14	208,7	168,9	2,5	2,9
18	195,7	155,4	2,7	3,3
22	183,2	149,4	2,9	3,4
26	170,7	137,1	3,1	3,7

IV. RESULTS AND DISCUSSION

The analysis of the energy efficiency of the application of the HPI schemes under study for the South-Eastern Europe during the winter period of the year, shows that the heat pump heating system (Fig. 1) is advisable to be used, when the subsoil water temperature is $t_{u1} = 20^{o}C$ at the inlet to the evaporator in the south regions of Europe, where temperature of the outside air is up to $t_0 = -5,5^{o}C.$ The heat supply systems with the intermediate heat exchanger (Fig. 2) is recommended to be also used in the south regions of Europe at the outside air temperatures lower than $t_0 = -6,5^{o}C.$ The heat supply systems with an additional supercooler is suggested to be applied in the South-Eastern Europe at the outside air temperatures no lower than $t_0 = -10^{o}C$.

The analysis of the calculation results of the energy efficiency of the alternative systems of heat supply show that the less the temperature difference between the low potential heat source and that of high potential, the more efficient the HPI.

From the viewpoint of energy, the heat supply using HPI is more profitable compared to burning the nature fuel used for the electricity production provided that the specific consumption of the primary energy is $PER < 1$.

The analysis of the dependences shows that the higher the environmental temperature, the less the requirements for temperature the heat carrier supplied to the heating system. This results in an increase in temperature difference of the low potential source of heat of the subsoil waters and high potential source heat of the heating system.

With an increase in the outside air temperature, the specific energy W consumed by the electric motor is decreased intensively, which attests to a decrease in the work of compression l_k of the compressor, while the coefficient of heat performance μ grows, which allows us to infer a more efficient operation of the system.

V. CONCLUSIONS

The study performed allowed us to establish the validity of the use of the HPI single-stage schemes for the South-Eastern regions of Europe at the temperatures of the environmental air up to $t_0 = -10^{o}C$. At other conditions being equal, the most perspective is the system of the HPI heat supply with an intermediate heat exchanger and additional super cooler, which is supported by such main indices of the energy efficiency as the specific energy that is consumed by the electric motor, coefficient of heat performance and specific consumption of primary energy.

In the Eastern regions of Europe with a more severe climate at the outside air temperatures lower than $t_0 = -10^{o}C$, it is advisable to use as a low potential source of heat the subsoil waters in the temperature range of higher than 20 °C.

If in the heat pump systems, the temperature range between the low potential source of heat and the heat carrier of the heat supply system is fairly high, the heating load cannot be ensured entirely. In this case, it is reasonable to apply the two-stage HPI based on the subsoil waters, in which due to a lesser consumption of the electric energy for the compressor drive, it is possible to increase the coefficient of heat performance. The application of the two-stage HPI allows a greater heat amount to be generated for the heat supply system at the same level of the electric energy consumption.

It is particularly urgent at a maximum heating load in a winter period, when the environmental temperature is the lowest. The two-stage HPI make it possible to decrease the annual average consumption of the energy, which is the advantage compared to the single-stage HPI.

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