Combined installation of electric and heat supply for climatic conditions of Iraq

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Abstract. Electricity, heating and cooling are the three main components that make up the energy consumption base in residential, commercial and public buildings around the world. Demand for energy and fuel costs are constantly growing. Combined cooling, heating and power generation or trigeneration can be a promising solution to such a problem, providing an efficient, reliable, flexible, competitive and less harmful alternative to existing heat and cold supply systems. In this paper, scheme of the tri-generation plant on non-aqueous working substances is considered as an installation of a locally centralized electro-heat and cold supply of a typical residential house in a hot climate. The scheme of the combined installation of electro-heat (cold) supply consisted of the vapor power plant and heat pump system on low-boiling working substance for local consumers under the climatic conditions of Iraq is presented. The possibility of using different working substances in the thermodynamic cycles of these units, which will provide better efficiency of such tri-generation systems is shown. The calculations of steam turbine cycles and heat pump part on the selected working substances are conducted. It is proposed to use heat exchangers of plate type as the main exchangers in the combined processing. The developed method of thermal-hydraulic calculation of heat exchangers implemented in MathCad, which allows to evaluate the efficiency of plants of this type using the $\varepsilon$ - NTU method. For the selected working substances of the steam part the optimal temperature of heat supply to the steam generator is determined. The results of thermodynamic and technical-economic analysis of the application of various working substances in the "organic" Rankine cycle of the steam turbine unit and the heat pump system of the heat and cold supply system are presented.

1. Introduction
At present, questions of using heat-power plants working on the so-called "organic" Rankine cycle (ORC), for the utilization of heat of medium-and low-temperature potential of natural, technogenic and anthropogenic origin, are being intensively studied, with the purpose of generating electricity and providing loads of heat supply to household and industrial facilities in conditions of seasonal and daily fluctuations in energy and heat consumption. The Rankine cycle configuration is determined by the phase diagram of the state of the working substance.

There are two groups of such substances, differing in the nature of the dependence of the thermodynamic parameters of the state on the saturation line, the first group (A) of substances includes the traditional heat carrier - water and the thermodynamic cycle of the steam-turbine plant is called the Rankine cycle (figure 1a), fluorocarbons belong to the second group (B) of substances and the thermodynamic cycle of the steam-turbine plant is called the "organic" Rankine cycle (figure 1b).
If the heat input in a thermodynamic cycle ("organic" cycle) is carried out at temperatures and pressure smaller than temperature and pressure in a critical point and thermodynamic process in the phase diagram passes in two-phase area (liquid-vapor), then such cycle is called subcritical (the figure 1a, b) if the heat supply occurs at temperatures and pressure big, than temperature and pressure in a critical point, and, therefore, thermodynamic process does not take place in two-phase area, then such cycle call supercritical (figure 2). So-called cogeneration power stations can make two types of useful products, for example, the electric power and warmth for providing heat supply.

Figure 1. Thermodynamic cycles (Rankine cycle) of technical training college with the elementary thermal and circuit a working medium of group of A (a); group B (b)

Figure 2. Configuration of supercritical cycle on organic working substances

A steam turbine unit based on the organic Rankine cycle can use geothermal and solar energy, heat generated in the combustion of biomass, heat from the steam-water turbine plant, internal combustion engines to generate electricity as a source of heat, can be used as a second loop of gas turbine and combined-cycle plants, power plants. At the same time the excess thermal capacity of the line 2-4 (figures 1, 2) it can be useful it is used in a contour of heat pumping installation of heat and cold supply. At the same time increase in energy efficiency of such tri-generation installation in general is predicted. The high thermodynamic efficiency of application as working substances of technical training college of the non-aqueous working substances (WS) of a fluororganic class has been for the first time noted in work [1] in the middle of the XX century. However, calculations of cycles on
fluorocarbon WS became possible within the last twenty years in process of accumulation of information on their thermophysical properties [2].

2. Subject of investigation

The decentralized climatic systems consisting of self-contained (floor) air conditioners, electric heaters of water providing hot water supply and electric heaters are applied to apartment houses in Baghdad (figure 3) in most of which of the population lives for maintenance of comfortable conditions of accommodation in an individual order now, the system of water heating is absent. As a subject of inquiry the type 8th apartment house (figure 3) is chosen, and the climatic system existing now is chosen as system for comparison.

![Figure 3. A subject of inquiry - a typical 4th floor house.](image-url)

The house has four floors, on each floor is located on two apartments, in each apartment lives on one family consisting on average of five people.

2.1. Features of climate of Iraq

Climate of Iraq subtropical in the north and tropical in the south, hot and dry. In table 1 annual dynamics of average monthly temperatures of the air-and-water pool is presented on width Baghdad.

| Month | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 | 9 | 10 | 11 | 12 |
|-------|---|---|---|---|---|---|---|---|---|----|----|----|
| Daytime temperature (°C) | 17 | 25 | 30 | 36 | 42 | 44 | 48 | 49 | 41 | 35 | 26 | 20 |
| Night time temperature (°C) | 4 | 7 | 11 | 15 | 19 | 21 | 25 | 26 | 23 | 18 | 8 | 6 |
| Average water temperature (°C) | 16 | 22 | 26 | 32 | 38 | 41 | 46 | 46 | 39 | 32 | 24 | 19 |

Comfortable conditions of accommodation in the considered house, namely: maintenance of required level of temperature and relative air humidity in rooms, and also year-round hot water supply, it is reasonable to provide by means of combined installation with the heat pump of vapor-compression type. Follows from table 1 that within calendar year in the conditions of Iraq two seasons are most expressed: long hot summer and shorter cool, and sometimes cold winter:

- winter within 4 months (November-February)
- summer within 8 months (March-October)
2.2. Rated conditions

The maximum load of heating (without HWS) corresponding to the minimum night time temperature of the winter period of normal year falls on January and makes 43 kW, minimum (November) – 34.4 kW, average – 38.17 kW. (figure 4). The maximum cooling load corresponding to the maximum daytime temperature of the summer period of normal year falls on August and makes 49.58 kW, minimum (March) – 27.25 kW, an average – 39.74 kW. (figure 4)

During the period from November to February the cooling load is absent. During the period from March to October there is no heating load. Hot water supply (HWS) is necessary all the year round at the same time a design heat load during the winter period makes 15 kW, and during the summer period – 12 kW. The rated histogram heating and cooling consumption of a house within a year is presented on the figure 4.

![Figure 4](image-url)

**Figure 4.** Annual dynamics of heating load and HWS, rated for 100%

On the basis of the analysis of regular data on the average daytime and night time temperature of air during the summer and winter period of normal year and the corresponding loadings of cooling, heating and HWS (table 1, the figure 4) rated initial conditions for the summer and winter periods of work of the combined installation (table 2) are accepted.

| Table 2. Initial conditions for the combined installation. |
|-----------------------------------------------------------|
| Initial conditions                                      | Summer period (march - october) | Winter period (november - february) |
|----------------------------------------------------------|
| Average daily temperature air, °C                       | 30                            | 14                                  |
| Maximum daytime temperature of air, °C                  | 49                            | 26                                  |
| Minimum night time temperature of air, °C              | 15                            | 4                                   |
| Average temperature of water, °C                       | 38                            | 21                                  |
| Indoor airs, °C                                         | 24                            | 21                                  |
| Coolant, °C on an entrance                              | -                             | -                                   |
| at the exit                                              | 20                            | 8                                   |
| Heat carrier, °C on an entrance                         | -                             | 35                                  |
| at the exit                                              | 8                             | 45                                  |
| HWS, °C on an entrance                                  | 27                            | 22                                  |
| at the exit                                              | 55                            | 60                                  |
| Refrigerating power, kW maximum                        | 50                            | -                                   |
| minimum                                                 | 27                            | -                                   |
| average for the period                                  | 39                            | -                                   |
| Thermal power, kW maximum                               | -                             | 43                                  |
| minimum                                                 | -                             | 35                                  |
| average for the period                                  | -                             | 39                                  |
| Thermal power of HWS, kW maximum                       | 12                            | 15                                  |
| Electric loadings, kW max                               | 30.255                        | 24.809                               |
3. Scheme of combined installation

The scheme of the considered combined installation and thermodynamic cycle of different parts (by analogy with [3]) this installation are presented on figures 5 and 6.

Figure 5. The scheme of the combined installation for electricity generation and warmth.

A – subsystem of the intermediate heat exchanger; B - electricity generation subsystem; C – a subsystem of heat pump unit (HPU).

IN – the delivery pipeline (heat supply pipeline); OUT – a return pipeline; H1 – pump (pumping the return water); H2 – pump (pumping the return water); H3 – pump (pumping the return water); H4 – pump (giving the heat carrier); H5 – pump (working substance pump); K1, K2 – valves (on the line of supply of the return water); K3 – the valve (on the line of giving of the heat carrier); T – turbine; PG – steam generator; Con 1 – the electricity generation subsystem condenser; EG – the electric generator; Con 2 – the condenser of a subsystem of the heat pump; Evap - the evaporator of a subsystem of the heat pump; K – compressor.

Figure 6. T-s diagram for the scheme (figure 5).

4. Calculation of a part of power generation

Calculation of thermophysical parameters of a subsystem of production of electricity was made by means of the REFPROP [4] program. In table 3 the initial data for calculation are presented. Thermodynamic cycle is presented in the figure 7.
Table 3. Basic data (R218)

| Name and designation                              | Numerical value |
|-------------------------------------------------|-----------------|
| Temperature on an entrance - T1                 | 90 °C           |
| Temperature of the return network water - T5    | 35 °C           |
| Molecular mass of R218, kg/kmol                 | 188.02          |
| Efficiency of the turbine: ηt                    | 0.8             |
| Efficiency of the pump of working fluid: ηrp     | 0.8             |
| Efficiency of the electric generator: ηgt        | 0.96            |
| Efficiency of the pump of low heat source: ηwp  | 0.8             |

Figure 7 – T-s diagram of a subsystem of production of electricity

Output power is defined on unit of a mass flow rate of a hot source (T1) (figure 6) as:

\[ P_{\text{net}} = \frac{N_{\text{net}}}{G_1}, \]  

(1)

where: \( G_1 \) - a mass flow rate of a hot source, \( N_{\text{net}} \) – power generation plant net output, W:

\[ N_{\text{net}} = N_t - N_{H5} - N_{H4} - N_{H2}. \]  

(2)

\( N_t \) – power on electric generator plugs; \( N_{H5} \) – the power of the pump of working substance of a cycle; \( N_{H4} \) – power on pumping of the heat carrier of the steam generator; \( N_{H2} \) – the power of the pump of network water.

Details of calculations of values in formulas (1, 2) and thermal-hydraulic calculations of the heat-exchange apparatus of installation are provided in [5]. Dependence of \( P_{\text{net}} \) (1) on boiling temperature in the steam generator is presented on the figure 8.

Figure 8. Power output power of installation from boiling temperature in PG (figure 5).

5. System of heat-cold supply (SHCS) on the basis of heat pump unit (HPU)

The selected equipment of SHCS should provide the maximum refrigerating and thermal power. Taking into account considerable seasonal fluctuation of loadings, SHCS should provide regulation of
refrigerating and thermal power ranging from 100 to 20% of the maximum values. For providing a year-round cooling, heating and HWS of a standard house, the scheme of the reversible machine of heat-cold supply (MHCS) [6] (figure 9) is presented.

![Figure 9. Schematic thermal diagram of reversible MHCS.](image)

Installation consists of the following basic elements: the heat exchanger working substance – the heat carrier (network water) 1, working as the evaporator during the summer period and as the condenser during the winter period; single-stage compressor 2; the heat exchanger working substance – water for HWS 3 of year-round action; the heat exchanger working substance - air 4, working as the air condenser during the summer period and as the evaporator during the winter period, the regenerative heat exchanger 5, a receiver 6, the regulating and shutoff valves 7, 9.

In the figure 9 the version of the scheme "summer" (continuous lines) is represented. In this operating mode of MHCS water is cooled in the evaporator 1 due to boiling of coolant (freon), vapor of which on the line 1 come to a regenerator 5 where overheat and come to the compressor 2. After compression in the compressor 2 the superheated steam is cooled in the HWS heat exchanger 3. Then steam comes to the air-cooled condenser 4 where it is condensed, transferring warmth of phase change to an atmospheric air. Condensate comes to a receiver 6 and further overcools in a regenerator 5, giving warmth to vaporous coolant. After throttling in a throttle valve 7, working substance comes to the evaporator 1 and working process repeats.

5.1. Evaluation of power efficiency of SHCS with HPU.

Application of such indicators as conversion factor and refrigerating coefficient for cases of the combined power generation is obviously not enough. At evaluation of energy efficiency of this or that installation providing needs of SHCS it is necessary to consider that the electric drive of the piston compressor is carried out at the expense of thermal power plant (figure 10).

The fullest and generalizing indicator for the choice of the energy strategy, is the coefficient of useful use of warmth of fuel (KIT). KIT is the universal indicator characterizing efficiency of the combined consumption (production) of heat and electric power.

Power efficiency of HPU is characterized by energy conversion factor \( \mu = \frac{Q_k}{N_e} \), where \( Q_k \) – the heat output of heat pump [W], \( N_e \) - the electric power of compressor’s drive unit [W], which is delivered from thermal power plant at a heat expense on the turbine [W]:

\[
\mu = \frac{Q_k}{N_e}
\]
Figure 10. Schematic energy diagram.

\[ Q_{tu} = \frac{N_e}{\eta_{pe}} \times \frac{byt}{3600} \times Q_{low}, \]  \hspace{1cm} (3)

here \( Q_{low} = 29308 \) – the lowest combustion heat of equivalent fuel [kJ/kg yt], \( byt \) - a specific consumption of equivalent fuel [kg yt / (kW*h)], \( \eta_{pe} \) - effectiveness of electricity transmission.

For calculation of coefficient of useful use of warmth of fuel the formula is received:

\[ KIT = \frac{Q_k}{\frac{N_e}{\eta_{pe}} \times \frac{byt}{3600} \times Q_{low}} = \frac{3600 \times \mu \times \eta_{pe}}{byt \times Q_{low}} \]  \hspace{1cm} (4)

Dependence of \( KIT = f (byt, \mu) \) is presented on the figure 11a, at \( byt=0.2 \ldots 0.4, \mu=1 \ldots 10 \), the \( KIT=1 \) level surface is shown in the same place. From the graph (figure 11a) it is possible to draw a conclusion on availability of area of the inefficient modes of application of HPU (at \( KIT <1 \), more right from line of crossing), and areas where application of HPU is reasonable (at \( KIT> 1 \), more left from line of crossing). The equation for calculation of the line of crossing in function \( \mu = f (byt, \eta_{pe}) \):

\[ \mu = \frac{byt \times Q_{low}}{3600 \times \eta_{pe}} \]  \hspace{1cm} (5)

is presented graphically on the figure 11b that allows to estimate quickly efficiency of regional application of HPU with this conversion factor at the known prevailing quality of production and transmission of electricity to the end user in this area.

Figure 11. a) Dependence of \( KIT \) on a consumption of equivalent fuel (byt) and conversion factor (\( \mu \)); b) Dependence of boundary conversion factor (\( \mu \)) from a consumption of equivalent fuel (byt) and efficiency of a power transmission system (\( \eta_{pe} \)).

The hierarchical structure of SHCS is presented on the figure 12.

Characteristics of the compressor system (CS) depending on boiling temperature (T0) and condensation (Tk) of working substance form as a result of interaction of characteristics of a thermodynamic cycle with characteristics of the elements entering CS. Characteristics of MHCS depending on temperature of low-temperature (Ts) and high-temperature (Tw) of heat carriers form as a result of interaction of characteristics of CS with characteristics of the main heat-exchange apparatus – evaporators and condensers. Characteristics of installation for heat-cold-supply (IHCS) form as a
result of interaction of characteristics of MHCS with characteristics of the external heat-exchange apparatus for cooling and heating of an object and heat exchanging processes with an environment.

### Figure 12. Hierarchical structure of SHCS

The maximum design factors of transformation $\mu$ and refrigerating $\varepsilon$ arise when using specific thermal and power characteristics of a thermodynamic cycle. Lower values of these indicators arise when accounting irreversibility of processes of compression and throttling, i.e. at the level of CS, the valid values of coefficients arise when accounting irreversibility of heat exchanging processes in the main heat-exchange apparatus – the evaporator and the condenser.

5.2. $\varepsilon$ - NTU method of calculation of HPU

At the level of MHCS $Q_0(\text{MHCS}) = f (ts2, toc)$, $Q_k(\text{MHCS}) = f (ts2, toc)$, $N_e(\text{MHCS}) = f (ts2, toc)$: $\mu = \frac{Q_k(\text{MHCS})}{N_e(\text{MHCS})}$, $\varepsilon = \frac{Q_0(\text{MHCS})}{N_e(\text{MHCS})}$, or

$$
\mu = \frac{1}{1 - \frac{Q_0}{Q_K}}, \quad \varepsilon = \frac{1}{\frac{Q_K}{Q_0} - 1} \quad (6)
$$

Considering that $Q_0 = Q_{MAX}^{EV} \cdot \varepsilon_{EV}$ and $Q_K = Q_{MAX}^{K} \cdot \varepsilon_{K}$, where $Q_{MAX}^{EV}, Q_{MAX}^{K}$ - maximum heating capacity of the evaporator and condenser, respectively; $\varepsilon_{EV}, \varepsilon_{K}$ - efficiency of the evaporator and condenser, respectively, one can be written:

$$
\mu = \frac{QQ}{QQ - 1}, \quad \varepsilon = \frac{1}{QQ - 1}, \quad \text{where} \quad QQ = \frac{Q_{MAX}^{K} \cdot \varepsilon_{K}}{Q_{MAX}^{EV} \cdot \varepsilon_{EV}} \quad (7)
$$

Dependence $\mu = f(QQ)$, $\varepsilon = f(QQ)$ is presented on the figure 13a, it is visible that the area of the effective modes is in QQ interval $(1, 2]$, this interval is presented on the figure 13b more precisely for $\mu \in [2, 7]$. 


Figure 13. a) $\mu = f(QQ)$, $\varepsilon = f(QQ)$, b) – inverse relationship

The area of the effective modes (the dark planes) for function $QQ = f\left(\frac{Q_{K}^{\text{MAX}}}{Q_{EV}^{\text{MAX}}}, \frac{\varepsilon_{K}}{\varepsilon_{EV}}\right)$ is presented on the figure 14a, in the figure 14b this area is enlarged. Thus, for area of the effective modes a ratio $\left(\frac{Q_{K}^{\text{MAX}}}{Q_{EV}^{\text{MAX}}}, \frac{\varepsilon_{K}}{\varepsilon_{EV}}\right)$ have quite determined values. From figures 14 it is visible that at $\left(\frac{Q_{K}^{\text{MAX}}}{Q_{EV}^{\text{MAX}}}\right) \in [1; 100]$, $\left(\frac{\varepsilon_{K}}{\varepsilon_{EV}}\right) \in [0.012; 2.05]$.

Figure 14. a) $QQ = f\left(\frac{Q_{K}^{\text{MAX}}}{Q_{EV}^{\text{MAX}}}, \frac{\varepsilon_{K}}{\varepsilon_{EV}}\right)$; b) $QQ = f\left(\frac{Q_{K}^{\text{MAX}}}{Q_{EV}^{\text{MAX}}}, \frac{\varepsilon_{K}}{\varepsilon_{EV}}\right)$ more detailed

For heat exchangers with phase change of one of heat carriers, efficiency $\varepsilon$ is connected with number of transferred units (NTU) by ratio: $\varepsilon_{K} = 1 - \exp(-NTU_{K})$ - for the condenser; $\varepsilon_{EV} = 1 - \exp(-NTU_{EV})$ - for the evaporator. Their relation $\left(\frac{\varepsilon_{K}}{\varepsilon_{EV}}\right) = \frac{1 - \exp(-NTU_{K})}{1 - \exp(-NTU_{EV})}$ is
presented in the figure 15a in the function form \( \left( \frac{E_K}{E_{EV}} \right) = f \left( NTU_K, NTU_{EV} \right) \). The surface 

\[
\left( \frac{E_K}{E_{EV}} \right) = f \left( NTU_K, NTU_{EV} \right)
\]
is presented on the figure 15b.

**Figure 15.** a) \( \left( \frac{E_K}{E_{EV}} \right) = 1 - \exp(-NTU_K) = \left( \frac{E_K}{E_{EV}} \right) = f \left( NTU_K, NTU_{EV} \right) \); b) 3-D surface

\[
\left( \frac{E_K}{E_{EV}} \right) = f \left( NTU_K, NTU_{EV} \right)
\]

From figure 15 availability of area of the effective modes and area of functional independence is obvious.

5.3. Evaluation of the maximum heating capability of the heat-exchange apparatus

For HPU generally it is possible to consider three main types of the heat-exchange apparatus on thermodynamic parameters of a condition of heat carriers in the device:

- devices without change of phase of heat carriers;
- devices with change of phase of heat carriers (the evaporator, the condenser);
- devices with pseudo change of phase of heat carriers (for example, at supercritical parameters of one of heat carriers)

Thus, four cases limiting heat exchanging process in the heat-exchange apparatus [7] are possible:

- The beginning of boiling of the coolant at \( \min \{Q_1, Q_2, Q_0\} = Q_1 \)
- The beginning of condensation of the hot heat carrier at \( \min \{Q_1, Q_2, Q_0\} = Q_2 \)
- On an entrance to the heat-exchange apparatus
- \( \) at \( \min \{Q_1, Q_2, Q_0\} = Q_0 \)
- At the exit from the heat-exchange apparatus

In single-phase area parameters are defined by pressure and temperature \((p, t)\), in two-phase – pressure, temperature and steam content \((p, t, x)\).

On an entrance to the heat exchanger it is known:

\[
p_1, t_1', x_1' \rightarrow h_1',
p_2, t_2', x_2' \rightarrow h_2'
\]

Assuming \( p_1 \approx \text{const} \) and \( p_2 \approx \text{const} \) on the heat exchanger, we will allocate generally four characteristic temperatures:

- \( t_1' \) and \( t_2' \) – on an entrance
- \( t_1s \) and \( t_2s \) – on a saturation line at \( p_1 \) and \( p_2 \)
If $t_1s$ and/or $t_2s \in [t_2', t_1']$, then it determines a possibility of phase change in one or in both heat carriers at this $t_1s \notin [t_2', t_1']$, and $t_2s \in [t_2', t_1']$ - boiling; $t_1s \in [t_2', t_1']$, and $t_2s \notin [t_2', t_1']$ - condensation.

5.3.1. Algorithm of determination of the highest possible heating capability of the main HPU heat-exchange apparatus

I. $t_1s \notin [t_2', t_1']$, $t_2s \in [t_2', t_1']$, (boiling of the coolant), then

$$Q_{\max} = \min(Q_0, Q_1),$$

where $Q_0 = \min[G_1 \cdot (h_1^* - h_1^*)], \ G_2 \cdot (h_2^* - h_2^*)], \ Q_1 = G_1 \cdot (h_1^* - h_1^*) + G_2 \cdot (h_2^* - h_2^*),$

where $h_1^* = h_1 (p_1, t_2', x_1=1)$, similarly for the second heat carrier we will write down $h_2^* = h_2 (p_2, t_1', x_2=0)$; $h_1^* = h_1 (p_1, t_2, x_1=1)$ - an enthalpy of the hot heat carrier at $p_1$ and $t_2s(p_2)$; $h_2s1$ – an enthalpy of the cold heat carrier on the left boundary curve at $p_2$.

II. $t_1s \in [t_2', t_1']$, $t_2s \notin [t_2', t_1']$, (condensation of the hot heat carrier), then

$$Q_{\max} = \min(Q_0, Q_2),$$

where $Q_0 = \min[G_1 \cdot (h_1^* - h_1^*)], \ G_2 \cdot (h_2^* - h_2^*)], \ Q_2 = G_1 \cdot (h_1^* - h_1^*), \ G_2 \cdot (h_2^* - h_2^*),$

where $h_2^* = h_2 (p_2, t_1s, x_2=0)$ - an enthalpy of the cold heat carrier at $p_2$ and $t_1s(p_1)$, $h_1s2$ – an enthalpy of the hot heat carrier on the right boundary curve at $p_1$.

Further one can define $Q = \varepsilon \cdot Q_{\max}$, $h_1' = h_1 - \frac{Q}{G_1}, \ h_2' = h_2 + \frac{Q}{G_2}$, then using $(p, h) \rightarrow t, x$, etc.

Thus, to evaluate and predict conversion factor and refrigerating coefficient in varying duties of work of HPU perhaps having information on efficiency of the evaporator and the condenser and calculating the maximum heating capacity of each device on a limited set of thermophysical properties and parameters of a thermodynamic cycle.

5.3.2. Calculation of characteristics of CS

Calculation of characteristics of CS is possible after selection of the compressor, for example, by means of the program Bitzer [8], where the compressors providing heat- cold- supply - MHCS in a winter and summer operating mode are picked up. For R134a coolant – the semi-hermetic piston compressor of brand 6HE-28Y was selected.

The general view of approximating dependence for calculation of characteristics of CS for a technique of Bitzer is provided by a formula:

$$y(t_0, t_K) = c_1 + c_2 \cdot t_0 + c_3 \cdot t_K + c_4 \cdot t_0^2 + c_5 \cdot t_K^2 + c_6 \cdot t_0 + c_7 \cdot t_K + c_8 \cdot t_0^2 + c_9 \cdot t_K^2 + c_{10} \cdot t_0^3 (8)$$

Values of coefficients $c_1 \ldots c_{10}$ to within the second (third) sign after a comma are given in table 4.

| Table 4. Approximation coefficients |
|------------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|------------------|
| Koef  | c1 | c2 | c3 | c4 | c5 | c6 | c7 | c8 | c9 | c10 |
|-------|----|----|----|----|----|----|----|----|----|----|
| Qo [W] | 86086.9 | 3532.4 | -669.6 | 54.49 | -26.9 | 1.94 | 0.27 | -0.409 | -0.029 | 0.0103 |
| Ne [W] | 3362.58 | -250.05 | 371.3 | -7.709 | 13.01 | -1.817 | -0.058 | 0.099 | -0.037 | -0.0029 |

In the figure 16 the three-dimensional surfaces of characteristics of CS calculated by a formula (8) are given below.

5.3.3. Technical results of comparison

For option of basic data (table 5) and design variables of cycle HPU (table 6) indicators of heat-cold-supply of the building by means of the developed SHCS are compared with use option for the same purpose of self-contained (floor) air conditioners and electric heaters table 7.
Figure 16. Calculated dependence of a) $Q_0(t_0, t_k)$ [kW]; b) $N_e(t_0, t_k)$ [kW]

Table 5. Basic data (R134a)

| Name and designation                             | Numerical value |
|-------------------------------------------------|-----------------|
| Temperature of direct water: $T_6$               | 45 °C           |
| Temperature of the return water: $T_5$          | 35 °C           |
| Efficiency of the water pump, $\eta_{wp}$       | 0.8             |
| Internal relative efficiency of the compressor, $\eta_{os}$ | 0.85          |
| Boiling temperature in the evaporator: $T_e$    | 5 °C            |
| Condensation temperature in the condenser: $T_c$| 60 °C           |
| Minimum difference of temperatures: $\Delta T_p$| 3 °C            |

Table 6. Calculation of thermodynamic parameters in characteristic points of a cycle of a part of SHCS by REFPROP (R134a)

| point No. | $p$, MPa | $t$, °C | $h$, kJ/kg | $s$, kJ/(kg * K) | $v$, m³/kg |
|-----------|----------|---------|------------|-----------------|-------------|
| 8         | 0.34966  | 5       | 401.49     | 1.7245          | 0.058034    |
| 5t        | 1.6818   | 65.55   | 434.04     | 1.7245          | 0.012064    |
| 5         | 1.6818   | 70.072  | 439.784    | 1.7413          | 0.012526    |
| 6''       | 1.6818   | 60      | 426.63     | 1.7024          | 0.011444    |
| 6         | 1.6818   | 60      | 287.5      | 1.2848          | 0.0009497   |
| 7         | 0.34966  | 5       | 287.5      | 1.3146          | 0.024663    |
| 7''       | 0.34966  | 5       | 206.75     | 1.0243          | 0.00078     |

Table 7. Comparative results

| Heat-cold-supply option     | Summer period | Winter period |
|-----------------------------|---------------|---------------|
|                             | Energy consumption, kWh | kWh |
| SHCS                        | 658168        | 263808        |
| Conditioners, electric devices | 658560    | 368640        |
6. Conclusion
In the work, the efficiency of the combined electric power plant operating according to the organic Rankine cycle and the heat-cold generation on the basis of a heat pump unit were investigated.

The possibility of using octafluoropropane (R218) as a working substance in the ORC was considered, and freon R134a was considered as the working substance of the heat pump unit.

As a result of the technical and economic analysis, it is established that the scheme in question does not have significant advantages over the existing climate system (air conditioners + electric heaters) during the summer period of operation, and undoubtedly is preferable in the winter.

The use of R218 in the part of electricity generation in the framework of subcritical ORC can be considered small effective, however, the tendency of an improvement in the situation with increasing boiling point makes it possible to recommend supercritical cycles for consideration. The use of combined air conditioning and heating systems for residential and public buildings is a worldwide trend. However, their performance, composition, operating parameters, depend on the climatic conditions of the region where they are used.

In countries where the ambient temperature does not fall below zero degrees (atmospheric t > 0), the use of the cold and heat supply system for year-round cooling and heating is possible and preferable. In these conditions, the most economical source of low-temperature heat for the operation of the system in the heat pump mode is ambient air. The cold and heat generation unit is a reversible vapor compression machine capable of operating in cooling modes (in summer) and a heat pump (in winter).

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