Optimization of the thermal power plant operation by distributing exergy flows in the heat pump

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Abstract. The possibility of wide use in practice and implementation of optimization of a thermal and power plant on the basis of exergy analysis with the use of target functions are presented in the article. A tree of thermal economic costs of a solar thermal supply system with possible distribution of flows in a thermal and power plant with the representation of a block diagram is considered. The exergy characteristics of each element of the system are given in the form of dependence. It determines the exergy flux density of total solar radiation on the surface of solar collectors. The schedule of total solar radiation, average monthly air temperature and total exergy amount of the city of Krasnodar as well as the graphs of distribution of the price of 1kW of exergy for months of a year were taken into account. The distribution of exergy flows in the heat pump and its energy efficiency are estimated by the coefficient of conversion.

1. Introduction
The study of solar heating supply efficiency uses the discrete-element modular principle of exergy analysis, which involves the performance of the system in the form of groups of elements (modules). Their “internal” structure is known and “external” element properties that define the material and energetic interaction with other system elements are given.

The possibility of applying this principle to the analysis of efficiency of individual modules and also the possibility of coordination of modules during their aggregation in the system of solar heating show the causality between individual phenomena. It happens due to the main characteristics which reflect the essence of exergy concepts: universality and additivity [1].

2. Materials and methods
Empirical dependence of determining the exergy flux density of total solar radiation on the surface of solar thermal collectors is perpendicular to the solar rays:

$$E_{sl} = G_{sp}c_eT_0 \left\{ 1 + \frac{q_{absorb}}{K_{hl}T_0} - \frac{T_{in}}{T_0} \right\} \left[ 1 - \exp\left( -\frac{K_{hl}\eta_{hp}}{G_{sp}c_e} \right) - \ln\left( \frac{T_{in}}{T_{in} + \frac{q_{absorb}}{K_{hl}T_{in}}} \right) \right],$$  (1)

where \(q_{absorb}\) – density of total solar radiation, kW/m²; coefficient

$$E_{sl} = G_{sp}c_eT_0 \left\{ \frac{q_{absorb}}{K_{hl}T_0} \left[ 1 - \exp\left( -\frac{K_{hl}\eta_{hp}}{G_{sp}c_e} \right) - \ln\left( \frac{T_{in}}{T_{in} + \frac{q_{absorb}}{K_{hl}T_{in}}} \right) \right] \right\},$$  (2)

\(T_0\) – absolute temperature of environment, K.
In figures 1, 2 there are the graphs of total solar radiation intake, average monthly air temperatures, total exergy intake and exergy flows in the solar collector in the conditions of Krasnodar.

![Graph of total exergy intake for Krasnodar.](image)

**Figure 1.** Graph of total exergy intake for Krasnodar.

As can be seen from the graphs, the amount of exergy depends more on the intake of solar radiation than on the temperature of outside air.

For solar radiation entering the surface of the solar collector, the price of exergy is zero.

The exergetic efficiency of solar collectors is a measure of the reversibility of thermal and optical processes occurring in a given installation. In accordance with the basic provisions of the exergy analysis, the efficiency of solar collectors is determined by the ratio of the exergy flux density of the $E_{con}$ coolant in heat-removing channels from beam-absorbing panels to the exergy flux density of the $E_{full}$ of solar radiation coming to their beam-absorbing surface, [2]:

- for solar collectors with a liquid heat carrier:
  \[
  \phi = \frac{q_{con}}{G_{SPS}c_c},
  \]

- for solar collectors with an air heat carrier:
  \[
  \phi_{PER} = \frac{T_{con}}{T_{con} - T_e}.
  \]

In these equations there were the following symbols: $G_{SPS}$, $c_c$ is a heat carrier consumption through thermal channels of the beam-absorbing panel per unit of front surface of the solar collector, and its specific heat capacity; $T_0$ – absolute temperature of environment; $q_{absorb}$ – density of total solar radiation flux absorbed by the surface of beam-absorbing panel; $K_{hl}$ – coefficient of total heat losses per unit of frontal surface of the collector; $T_{in}$ – absolute temperature of the heat carrier in the entrance into a beam-absorbing panel of the solar collector; $\eta_{hp}$ – coefficient of the heat efficiency of a beam-absorbing panel of the solar collector.

Based on the fact that the average price of the solar collector (SC) is 5000 rubles/m², and the cost of installation and maintenance is 50% of the cost of the SC [10], it is possible to calculate the cost of one kW of exergy received from the solar collector. The schedule of distribution of price of 1 kW of exergy by months of the year is shown in figure 3.
Figure 2. Graph of exergy fluxes in solar collectors.

Figure 3. Distribution of the price of 1kW of exergy in months.

The cost of energy losses arising from the daily storage of 1 kW of energy in the storage tank when heating the coolant (water) at 20°C is shown in figure 4.

Figure 4. Cost of energy losses in the storage tank
3. Results

The distribution of the exergy flows in the heat pump was shown in figure 5, where the transmission of exergy from consumed fuel in the power plant to the heat consumer is represented.

![Figure 5. Distribution of the exergy flows in the heat pump.](image)

Energetic efficiency of the heat pump is evaluated by the coefficient transformation:

\[ e_T = q \left(1 - \frac{T_{env}}{T}\right), \]  

where \( q_{sp} \) – specific (on unit of working body’s mass) heat flow in a capacitor, kJ/kg; \( l \) – specific work in the cycle, kJ/kg. For ideal heat pump:

\[ \phi_{PER} = \frac{T_{con}}{(T_{con} - T_{ev})} \]  

where \( T_{ev}, T_{con} \) – temperature of the working unit in a capacitor and vaporizer respectively;

\[ q_{sp} = q_{ev} + l \]

It is clear from the equation that the conversion coefficient reflects only the first law of thermodynamics and does not take into account the second, which characterizes the qualitative side of the processes of energy conversion.

In compression heat pump plants (HPP), the supplied exergy \( E_{output} \) is equal to the actual consumption of electrical energy, and the effectively used exergy \( E_{output} \) is the exergy of generated heat, equal to the cost of work in the ideal heat production process.

In processes or systems, the total amount of exergy is not saved, but destroyed due to internal irreversibilities. In a thermodynamic system, exergy can be transferred to or withdrawn from the system in three forms: heat, work, and mass flow, which are recognized at the system's boundaries.

Exergy transmitted by the heat \( E \) is expressed as [3]:

\[ E_{muass} = \mu e \]

where \( Q \) – indicator of heat transmission where the boundaries are overlapped, kJ/s; \( T_{env} \) and \( T \) – temperature of environment and heat source, respectively, K.

Mechanical and electrical energies are indefinitely converted into other forms of energy, so they are completely transformed into exergy.

Exergy \( E_{work} \) (kW) equals to electrical or mechanical work \( W \) (kW). In the case of a mass flow crossing system boundaries, the exergy transfer by the mass \( E_{mass} \) can be calculated by the formula:

\[ l + eq_v = eq_p + eq_{w,v} + \sum de, \]
where $m$ – mass flow index crossing the system boundaries, kg/s; $e$ – exergy per mass unit, kJ/kg.

The energy in the form of heat flow consists of exergy and anergy - the lost part of the energy or the heat flow at the ambient temperature. $T_o$ - the exergy of the heat flow is less than the excess of its temperature level over $T_{env}$, when $T=T_{env}$ is zero.

At constant temperature $T$ ($T>T_{env}$), the heat flux, its exergy $q_e$ and anergy $q_a$ are linked by the following relations:

$$
\eta_{HPS} = \frac{e_{IN}}{e_{OUT}} = \frac{e_{q,p} + e_{q,k}}{l + e_{q,ev}} = \frac{(q_{q,p} + q_{q,k})}{l + q_{e}r_{input}},
$$

(10)

$$
T_0 < (e/\eta).TT, q=q_e+q_a, ,
$$

(11)

$$
Q = m \int_{T_0}^{T_f} C_{sp} dt ,
$$

(12)

$$
P_{nom} = 2\pi R(R + L) \cdot U(T_r - T_u)
$$

(13)

where $r_{input}$ – defines the amount of work which can be obtained from heat unit in an ideal direct cycle; $r_{input}$ is a function of thermodynamic system and environment condition and is called the exergic temperature function (the Carnot factor).

The values of the exergy of the working agent at specific points of the process can be determined by the formula:

$$
U = \frac{1}{\frac{h}{C} + \mu}
$$

(14)

Change of exergy of the flow system:

$$
E = q - T_o(S - S_o)
$$

(15)

where $T_o$ - environment, K; $h$ – specific heat content, kJ/kg; $S$ – specific entropy, kJ/(kg.K). Index "o" indicates to the dead condition of environment, indices "1" and "2" indicate the other condition of the vapour flow.

**Figure 6.** Graph of dependence.

In all real (irreversible) processes exergy decreases, passing into anergy. In order to improve the exergetic efficiency of the installation ($\eta$), the amounts of exergies destroyed in the system and lost (leaving) through the outflow must be reduced.

The analysis of efficiency of separate processes and HPI as a whole is made by an exergetic method, so components of exergetic balance according to the equation are defined as:

$$
Q = m \int_{T_0}^{T_f} C_{sp} dt .
$$

(16)

Exergy losses can be divided into groups: 1) internal, associated with the irreversibility of processes occurring within the system; 2) external, related to the conditions of interaction of the
system with the environment and other sources and receivers of energy. In heat transformers, examples of internal losses can be losses associated with throttling, friction in machines.

External losses include those associated with different temperatures of a cooled refrigerant [4].

Exergic efficiency of the heat installation:

\[ \varepsilon_K = \frac{E_{P,K}}{E_{F,K}} = 1 - \frac{E_{D,K} - E_{L,K}}{E_{F,K}}. \]  

(17)

The distribution of energy flows in the heat pump is shown in figure 6.

The cost of the electrical energy needed to operate the heat pump will be less than the cost of buying natural gas or heat energy that could be applied to traditional heating systems if inequality is observed:

\[ T_e < \left( \frac{\varepsilon}{\eta} \right) T_T, \quad q = q_e + q_a, \]  

(18)

where \( T_e \) - tariff on electrical power; \( T_T \) – tariff on one of the traditional energy resources; \( \varepsilon \) – coefficient of heat pump transformation; \( \eta \) - coefficient of efficiency of traditional heat generator.

In order to be able to apply the formula (12), it is necessary that \( T_e \) and \( T_T \) tariffs are expressed in the same units of measurement. The gas tariff is usually expressed in rubles / m\(^3\), and the tariff for thermal energy in rubles/Gcal, while the tariffs for electric energy are always higher:

Figure 7. Dependence of coefficient of heat pump transformation on temperature of a heat carrier is expressed in rub/kW•h. It is convenient to use the following dependencies to compare tariffs: 1 rub/m\(^3\) = 0.106 rub/kW•h; 100 rub/Gcal = 0.086 rub/kW•h.

The dependence of the conversion coefficient of the heat pump on the temperature of the coolant at the outlet of the capacitor and on the temperature of outside air is shown in figure 7.

Figure 8. Dependence of the cost of 1 kW of energy generated by the heat pump depending on the conversion factor.

Thus, we obtain the dependence of the cost of production of 1 kW of thermal energy at the price of electricity 5.50 rubles/kW for heat pump installations on the conversion factor, which is shown in
figure 8. Let us consider the dependence of the cost of production of 1 kW of thermal energy on the time of year for the heat pump installation Air-Water, shown in figure 9. The joint operation of the heat pump and the solar collector can reduce the cost of creating a power installation by reducing the area of the solar space or using simpler solar collector designs.

Figure 9. The cost of producing 1 kW of thermal energy in different months.

Distribution of cost of production of 1 kW of thermal energy in the system HPP and SC are resulted in the figure.

Figure 10. The cost of producing 1 kW of thermal energy in different months for the heat pump unit (red) and the solar collector (blue).

The energy received in a solar collector or heat pump installation enters the storage tank, where it is accumulated and stored until used by the consumer.

To calculate the amount of heat accumulated by the coolant, a typical battery tank is adopted, consisting of a metal cylindrical tank equipped with two heat exchangers (from the solar collector and from the consumer), filled with a heat-accumulating liquid and covered with a heat-insulating material (figure 11).
Figure 11. Fundamental scheme of a battery tank: 1 – battery tank, 2 – heat exchanger of the solar collector, 3 – heat exchanger of the consumer.

Calculation of heat amount stored in the battery tank produced by the formula:

$$Q = m \int_{T_0}^{T_1} C_{sp} \, dt,$$

where $m$ – mass of heat storage substance, $C_{sp}$ - specific heat capacity of accumulating environment at constant pressure, $J/(kg \cdot K)$, $T_0, T_1$ – temperature of heat storage material in cold and heated condition.

The amount of energy stored in the storage tank directly depends on the temperature difference of the heat storage material before and after heating. Therefore, the higher the boiling point of the coolant, the greater the amount of energy it can potentially store. However, it should not be forgotten that the rate of heat loss is directly proportional to the temperature difference between the coolant and the environment, and is determined by the formula [1, 5, 6]:

$$P_{\text{heat}} = 2\pi R(R + L) \cdot U (T_s - T_a),$$

where $R$ and $L$ – radius and height of the cylinder, $T_a$ – environmental temperature, $T_s$ – temperature of the heat carrier in the storage. $U$ – coefficient of proportionality:

$$U = \frac{1}{x \lambda + \mu}.$$

Share of heat lost per unit of time is:

$$\eta = \frac{P_{\text{heat}}}{Q} = \frac{2U \left(1 + \frac{R}{L}\right)}{R c_{sp} \rho},$$

where $c_{sp}$ and $\rho$ – heat capacity and density of a heat carrier.

Amount of exergy stored in the storage tank determined by the formula:

$$E = q - T_0 (S - S_0),$$

where $q$ – heat of the heat carrier, $T_0(S - S_0)$ – energy of heat.

4. Discussion

Summarizing the above, it can be concluded that the use of the exergy method can be an effective means to solve a number of optimization problems associated with the design and calculation of the parameters of the energy system based on renewable energy sources. The exergy method has a number of advantages in comparison with other methods of evaluating efficiency in the energy sector, which make it a universal means of choosing the optimal parameters of the system, both with a linear and branched scheme of joining elements. This method allows you to compare not only the efficiency of
the system as a whole, but also the use of a particular scheme, or its elements. When determining the exergy efficiency of the solar collector, the following data were obtained (Table 1).

5. Conclusion
The distribution of the amount of exergy coming to the ray-receiving surface of the solar collector directly depends on the amount of incoming solar radiation, and to a lesser extent, depends on the air temperature, so the maximum exergy comes in summer, and the minimum in winter.

The amount of exergy for the heat pump depends on the ambient temperature, so the maximum exergy comes in summer and the minimum in winter. The price of exergy depends on the cost of the equipment, its service life and the amount of incoming exergy and its cost, thus the highest cost of exergy for the solar collector and heat pump is obtained in winter, and the lowest-in summer.

Table 1. Data for determining the exergy efficiency of the solar collector

| Month  | Distribution of energy passing into the surface absorbing rays |  |  |  |
|--------|---------------------------------------------------------------|---|---|---|
|        | $q_{fall}$ | $t$ | $T$ | $\psi$ | $ex$    |
| January| 39.6       | 0.6 | 273.6 | 0.937 | 37.093 |
| February| 57.2      | 1.1 | 274.1 | 0.937 | 53.572 |
| March | 100.1      | 5.5 | 278.5 | 0.936 | 93.649 |
| April | 133.1      | 12.2 | 285.2 | 0.934 | 124.316 |
| May | 183.7      | 17.2 | 290.2 | 0.933 | 171.364 |
| June | 200.2      | 21.3 | 294.3 | 0.932 | 186.566 |
| July | 204.6      | 24.1 | 297.1 | 0.931 | 190.534 |
| August | 176      | 23.7 | 296.7 | 0.931 | 163.916 |
| September | 130.9 | 18.5 | 291.5 | 0.933 | 122.070 |
| October | 85.8      | 12.3 | 285.3 | 0.934 | 80.136 |
| November | 40.7      | 6.1 | 279.1 | 0.935 | 38.071 |
| December | 24.2      | 2.1 | 275.1 | 0.936 | 22.659 |
| Year | 1376.1 | 12.1 | 285.1 | 0.934 | 1285.316 |

5. References
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