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Research Article

Thermoeconomical optimization of a regenerative air turbine cogeneration system

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ABSTRACT

The concept of optimization of operating and design parameters of the regenerative air turbine cogeneration (RATC) system is proposed. To determine the energy efficiency indicators of this system, its thermodynamic analysis was performed. At the same time, it is shown that it is not correct to search for the optimal parameters of such a system only by energy indicators, and modern thermoeconomic methods should be more actively involved in pre-design practice, which allow a comprehensive evaluation of the efficiency and economy of the energy-technological system as a whole and its individual elements. Therefore, on the basis of the data obtained during the thermodynamic analysis, exergy destruction and losses in the main system elements were calculated. Then, using the structural-variant method in combination with the graphical apparatus of C-curves, the pre-project thermoeconomic optimization of the RATC system was performed. This made it possible to choose the optimal operating and design parameters of the system and determine the minimum cost for its creation and operation throughout the entire life cycle, taking into account the thermodynamic perfection of the main system elements. Each variable operating mode parameter of the system serves as a kind of navigator when searching for the option of system parameters that is optimal in terms of energy and economic indicators, which is accompanied by graphic visualization. Therefore, the proposed approach makes the optimization of the system being designed convenient and clear. It can be used in the optimal design of various types of thermal transformers and other energy-technological systems.

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INTRODUCTION

Currently, much of the thermal and electrical energy both in Ukraine and in the world is generated by large thermal power plants (TPP) and combined heat and power plants (CHPP) whose power units have efficiency level of 35-37%, and the power units using low-calorie fuels have the efficiency that does not exceed even 15%. In addition to the relatively low energy efficiency, the operation of large TPPs and CHPPs is associated with large losses during energy transportation to the consumer, emissions of harmful substances and greenhouse gases into the environment.

One of the progressive solutions to existing energy problems is the construction of small TPPs with cost effective and environmentally friendly cogeneration systems (mini-CHPPs). Against this background, in 2005, Ukraine adopted the Law on Combined Heat and Power (Cogeneration) and the Use of Waste Energy Potential. Such cogeneration using a single primary energy source can increase fuel efficiency from 30–40% to 85–90% [1,2].

Thus, the development of the concept of creating small-sized high-efficiency environmentally friendly low-power cogeneration systems where renewable fuel sources can be used as fuel is an urgent scientific problem.

MODERN METHODS OF CREATING HIGHLY-EFFICIENT LOW-POWER COGENERATION SYSTEMS

Modern cogeneration (CHP) systems are complex energy-technological high-recovery energy complexes. The pre-project analysis of these systems should take into account both economic and thermodynamic components of energy processes. Meanwhile, in the cases where the number of energy subprocesses and technological elements is large enough, such an analysis can become significantly complicated. Therefore, we should more actively involve modern methods [3] in the pre-project practice, which will allow us to comprehensively evaluate the efficiency of the energy-technological system as a whole and its individual elements.

The scientific basis of these methods is the concept of system exergy [4,5], that is, the ability of this system to work in the conditions of a certain thermal state of the environment. Exergy serves as the sole basis for assessing the impact on the economic performance of the thermodynamic parameters of energy-transforming systems.

Depending on the purpose of the study of CHP systems, literary sources can be grouped into two categories. The first category includes works to determine the production costs and/or the cost of CHP losses. These studies focus on determining the velocity of cash flows through the elements [6–8]. If the goal is to optimize the energy system (second category), then the research focuses on choosing the best modes of system operation with taking into account the monetary value of its creation and operation [9–14].

So Sahoo [15] carried out an exergoeconomic analysis and optimization of the cogeneration system with an output of 50 MW of electricity and 15 kg/s of saturated steam at a pressure of 2.5 bar. The system was optimized using exergoeconomic principles and evolutionary programming. This, of course, a new modern research method is effective in determining and refining the costs of production and capital investments. However, it can be used at the stage of design analysis, with the already specified regime and design parameters of the scheme.

Oyedepo et al. [16] analyzed the costs of generating electricity and evaluating the performance of operating gas turbine power plants using a methodology based on the Specific Exergy Costing (SPECO) approach. With this approach, the fuel and product of a component are determined by systematically accounting for all additions and removals of exergy from all exergy flows of the system, and the costs are calculated using the basic principles of business administration [6]. The SPECO method allows you to analyze all the changes that occur with the exergy flow from the moment it is introduced into the system until the end product is obtained, taking into account the price of each internal exergy flow in each element of the system. This method can also be successfully applied both for diagnostics of energy systems and at the final design stage.

Seyyedi et al. [17] propose a new iterative approach to the optimization of complex thermal power plants, based on the exergoeconomic analysis and the method of structural optimization. It is demonstrated by the example of the optimization of a plant operating on a simple Brighton regenerative cycle. Exergoeconomic analysis is used to determine the sum of costs for the exergy destruction and investments for each element of the system. The advantages of this approach are that it can be applied to large real complex thermal systems while optimizing their operating modes. However, this method can also be applied only at the final design stage, when all the design parameters of the system have already been determined.

All of these methods have advantages and disadvantages, as well as their own field of application. However, in our opinion, the most versatile method is the structural-variant one [18,19], which allows us to carry out both the thermoeconomic analysis and optimization of the technological scheme already at the pre-design stage.

In this work, we propose to use the structural-variant method in combination with the graphical apparatus of C-curves to determine the minimum costs for the creation and operation of the system throughout the system entire life cycle. The graphical presentation of the dependencies of the operational characteristics of the system on capital investments does not require their mathematical description and makes it possible to present the results of optimization of the cogeneration system in a visual form [20].

This method is applicable to one of the most promising, in our opinion, schemes of the cogeneration plants, which is the regenerative air turbine cogeneration (RATC) system. The advantages of this scheme are described in [21]. The analysis of such a scheme was not performed by any of the above methods, unlike the known CGAM CHP system [22].

PURPOSE AND OBJECTIVES OF THE RESEARCH

The purpose of this study is to create the concept for the pre-project analysis and optimization of the RATC system with using the structural-variant optimization method.

Research objectives:

1. To conduct the thermodynamic analysis of RATC system in order to determine exergy flows and exergy losses in each of its elements.
2. To carry out thermoeconomic optimization of the system with using the structural-variant optimization method and graphical C-curves.

THERMODYNAMIC ANALYSIS AND THERMOECONOMIC OPTIMIZATION OF THE RATC SYSTEM

In [21], we performed a detailed thermodynamic analysis of the ideal and actual regenerative cycles with the determination of the optimal energy parameters of the RATC system. However, it is not enough to optimize the technological scheme based on energy characteristics only, as suggested in [23]. As shown above, it is necessary to take into account not only energy, but also economic indicators, that is, the monetary costs of the system.

Figure 1 shows the RATC system schematic heat diagram with the numbering of flows between elements.

The main advantages of the RATC system over traditional gas turbine units are as follows:

- Energy advantages – installation of a solid fuel boiler behind the air turbine makes it possible to use the heat of the turbine downstream air. This reduces the fuel consumption in the boiler and accordingly increases the system efficiency.
- Technological advantages – the air turbine operates with clean air and is protected from the formation of sediments on the surface of the blades or their erosion when using a "dirty" working substance. No external cooling systems are required for the air turbine, which greatly simplifies its design.
- Environmental advantages – the ability to operate the system with gas obtained as a result of thermal treatment of municipal solid waste. The boiler operates at almost atmospheric pressure with less emission of harmful substances into the atmosphere [21].

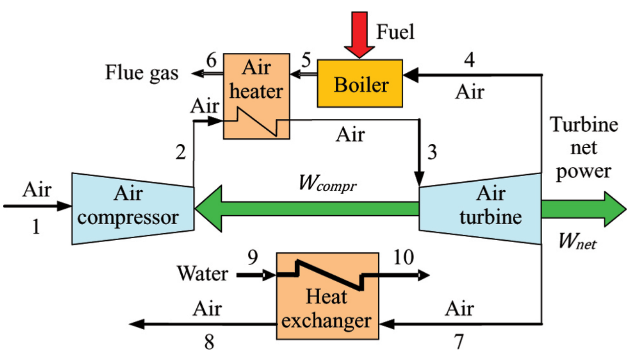

Figure 1. The schematic heat diagram of the RATC system.

First, consider the thermodynamic model of the RATC system, which is a component of the general thermoeconomic model of the RATC system, and is necessary to determine the parameters of the working substance in the main elements of the scheme:

Air Compressor

The air temperature after compression in the air compressor is:

$$T_2 = T_1\left(1 + \frac{1}{\eta_{ind_{compr}}}\left[\pi_c^{\frac{k-1}{k}} - 1\right]\right) \tag{1}$$

where $\eta_{ind_{compr}}$, π_c and k are the air compressor indicated efficiency, compressor ratio, and isentropic exponent respectively.

The air compressor inlet air pressure (P_1) and air compressor inlet air temperature (T_1) equal:

$$P_1 = P_0;\ T_1 = T_0 \tag{2}$$

where P_0, P_1 are the ambient temperature and pressure (T_0 = 298.15 K; P_0 = 0.1013 MPa).

The air compressor drive power is defined as:

$$W_{compr} = m_{air} c_{P_{air}} \left(T_2 - T_1\right)\frac{1}{\eta_{CD}} \tag{3}$$

where m_{air}, $c_{P_{air}}$ and η_{CD} are the air mass flow rate through the air compressor, the air specific isobaric heat capacity, and the air compressor drive efficiency respectively.

Air Heater

The functional purpose of the air heater is to increase in the air turbine upstream air temperature. The heat balance equation for the air heater is written as:

$$m_{air} c_{P_{air}} \left(T_3 - T_2\right) = m_{fg} c_{P_{fg}} \left(T_5 - T_6\right) \tag{4}$$

where m_{fg} and $c_{p_{fg}}$ are the flue gas mass flow rate, and the flue gas specific isobaric heat capacity respectively.

The air heater outlet air pressure (at the air turbine inlet) is determined as:

$$P_3 = P_2\left(1-\dot{P}_{AH_{air}}\right) \tag{5}$$

where $\dot{P}_{AH_{air}}$ is the generalized loss coefficient for the air flow pressure in the air heater ($\dot{P}_{AH_{air}}$ = 0.05).

The thermal-technical efficiency of the air heater is:

$$\eta_{AH} = \frac{T_3 - T_2}{T_5 - T_2} \tag{6}$$

Air Turbine

The temperature of the working substance (air) at the air turbine outlet equals:

$$T_4 = T_3\left(1-\eta_{ind_{turb}}\left[1-\left(\frac{P_3}{P_4}\right)^{\frac{1-k}{k}}\right]\right) \tag{7}$$

where $\eta_{ind_{turb}}$ is the air turbine indicated efficiency.

The air turbine power is determined by the equation:

$$W_{turb} = m_{air}c_{p_{air}}\left(T_3 - T_4\right)\eta_{mech_{turb}} \tag{8}$$

where $\eta_{mech_{turb}}$ is the mechanical efficiency of the air turbine, and the air turbine net power is:

$$W_{net} = \left(W_{turb} - W_{compr}\right)\eta_{EG} \tag{9}$$

where η_{EG} is the efficiency of the electric generator located on the air turbine shaft.

Boiler

The flue gas mass flow rate is calculated as:

$$m_{fg} = m_{air_4} + m_f \tag{10}$$

where m_{air_4} and m_f are the mass flow rate of air that enters the boiler, and fuel mass flow rate respectively.

The boiler heat balance equation is:

$$m_{air_4}i_4 + Q_b = m_{fg}i_5 \tag{11}$$

where i and Q_b are the specific enthalpy, and the boiler heat output respectively.

The boiler heat output is determined by the formula:

$$Q_b = m_f Q_L \eta_b \tag{12}$$

where Q_L and η_b are the lower heat of fuel combustion, and the boiler efficiency respectively.

The boiler outlet flue gas temperature (T_5) was set at 50°C higher than the preset T_3. This was necessary to determine the mass flow rate of fuel (m_f). Taking into account that:

$$m_{fg} = \frac{m_{air}c_{p_{air}}\left(T_3 - T_2\right)}{c_{p_{fg}}\left(T_5 - T_6\right)} \tag{13}$$

where the air heater outlet flue gas temperature (T_6) is taken 20°C higher than the air heater inlet air temperature (T_2), and substituting Equations 10 and 12 into Equation 11 we get:

$$m_f = \frac{m_{fg}\left(i_5 - i_4\right)}{Q_L\eta_b - i_4} \tag{14}$$

Knowing both m_f and m_{fg}, we determine the required mass flow rate of air that enters the boiler (m_{air_4}) using Equation 10.

The boiler heat output is defined by the Equation 12.

Heat Exchanger

The heat exchanger is an important RATC system element producing the after product of the system. It uses the heat from the excessive hot air flow after air turbine to heat the water entering the hot water supply system.

The heat balance equation for the heat exchanger is written in the following form:

$$m_{air_7}c_{p_{air}}\left(T_7 - T_8\right) = m_{wtr}c_{p_{wtr}}\left(T_{10} - T_9\right) = Q_{HE} \tag{15}$$

where m_{air_7}, m_{wtr}, $c_{p_{wtr}}$ and Q_{HE} are the mass flow rate of air that enters the heat exchanger, the water mass flow rate, the water specific isobaric heat capacity, and the heat exchanger heat output respectively.

The mass flow rate of the air turbine downstream air that enters the heat exchanger m_{air_7} is found as the difference:

$$m_{air_7} = m_{air} - m_{air_4} \tag{16}$$

where the mass flow rate of air turbine upstream air m_{air} is defined as:

$$m_{air} = \frac{W_{net}}{\left(c_{p_{air}}\left(T_3 - T_4\right)\eta_{mech_{turb}} - c_{p_{air}}\left(T_2 - T_1\right)\frac{1}{\eta_{CD}}\right)\eta_{EG}} \tag{17}$$

The electrical efficiency of the RATC system is calculated as:

$$\eta_{el} = \frac{W_{net}}{Q_b / \eta_b} \tag{18}$$

The RATC system has two products (electricity and heat), so its total efficiency is calculated by the formula:

$$\eta_{tot} = \frac{W_{net} + Q_{HE}}{Q_b / \eta_b} \tag{19}$$

The RATC system with the air turbine net power of W_{net} = 300 kW was considered. The following parameters were chosen to be preset ones: the ambient parameters (T_0 = 25°C, P_0 = 0.1013 MPa); the air compressor inlet air parameters (T_1 = 25°C, P_1 = 0.1013 MPa); the air heater outlet flue gas temperature ($T_6 = T_2 + \Delta T$), where ΔT = 20°C; the heat exchanger outlet air temperature (T_8 = 25°C); the heat exchanger inlet water temperature (T_9 = 15°C); the indicated efficiencies of the air compressor ($\eta_{ind_{compr}}$ = 0.8) and air turbine ($\eta_{ind_{turb}}$ = 0.9); the boiler efficiency (η_b = 0.9).

The following parameters were chosen to be variable: the compressor ratio (π_c) in the range from 1.8 to 2.7; the air turbine upstream air temperature (T_3) in the range from 700°C to 850°C.

As the determining factors in choosing the optimal variant of the RATC system parameters, we chose the total efficiency of the system, the system total capital cost , as well as the operating costs and exergy destruction and losses in the main system elements.

Figure 2 shows the dependence of the RATC system efficiency on π_c with varying temperatures T_3.

Figure 2 shows that η_{el} increases with increasing π_c, while η_{tot}, on the contrary, decreases. This is explained by the fact that with increasing π_c, the temperature of the turbine downstream air (T_4) and the mass flow rate of the air turbine upstream air (m_{air}) decrease (Figure 3). The mass flow rate of the air that enters the heat exchanger (m_{air_7}) is also decreases. Therefore, Q_{HE} decreases at a fixed value of W_{net} = 300 kW. In addition, Figure 2 (b) shows that η_{tot} reaches 56% at T_3 = 850°C and π_c = 1.8. With a decrease in temperature T_3 to 700°C and an increase in π_c to 2.7, it decreases to 47.2%.

Further, for the thermoeconomic optimization of the RATC system, the capital cost (Z) in USD of the main system elements are calculated by the following dependencies [16,22]:

Air compressor capital cost:

$$Z_{compr} = \left(\frac{71.1 m_{air}}{0.9 - \eta_{ind_{compr}}}\right)\left(\frac{P_2}{P_1}\right)\ln\left(\frac{P_2}{P_1}\right) \tag{20}$$

Air heater capital cost:

$$Z_{AH} = 4122\left(\frac{m_{fg} c_{p_{fg}} \left(T_5 - T_6\right)}{18\bar{T}_{\ln_{AH}}}\right)^{0.6} \tag{21}$$

where $\bar{T}_{\ln_{AH}}$ is the mean logarithmic temperature head in the air heater.

Heat exchanger capital cost:

$$Z_{HE} = 231\left(\frac{A_{HE}}{0.0929}\right)^{0.639} \tag{22}$$

where A_{HE} is the heat exchanger surface area.

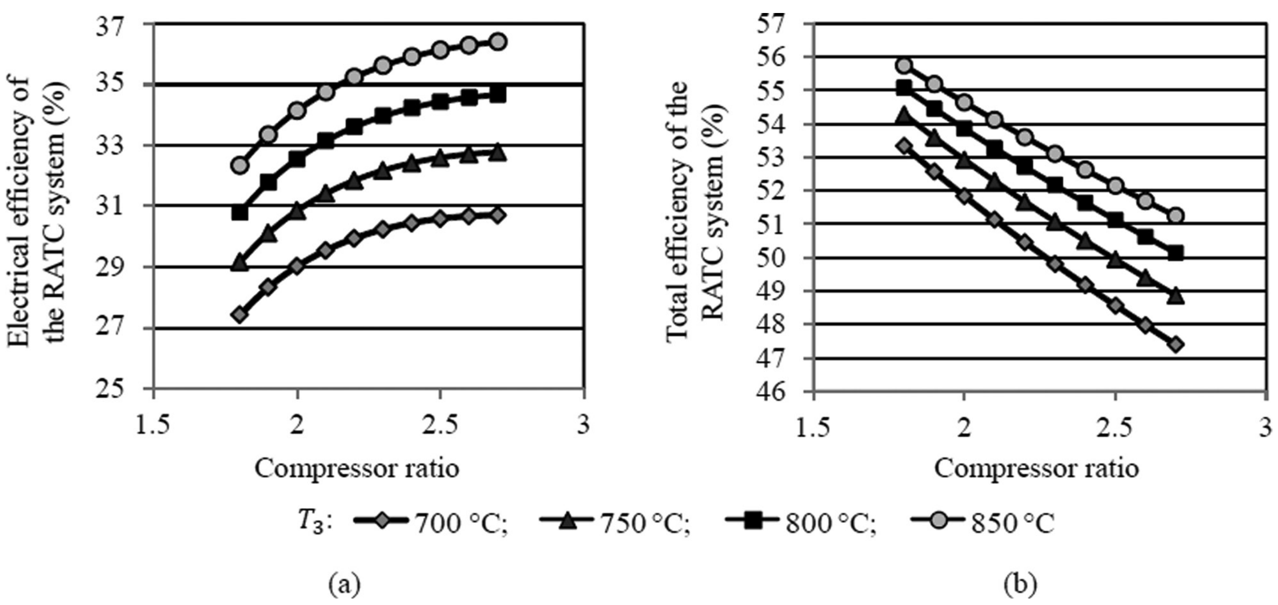

Figure 2. The dependence of the RATC system (a) electrical efficiency and (b) total efficiency on π_c.

Boiler capital cost:

$$Z_b = \left(\frac{46.08 m_{fg}}{0.995 - \frac{P_5}{P_4}} \right) \left(1 + e^{(0.018 T_5 - 26.4)} \right) \tag{23}$$

Gas turbine capital cost (it is taken equal to the air turbine capital cost):

$$Z_{turb} = \left(\frac{479.3 m_{air}}{0.92 - \eta_{ind_{turb}}} \right) \left(\frac{P_3}{P_4} \right) \left(1 + e^{(0.036 T_3 - 54.4)} \right) \tag{24}$$

Electric generator capital cost:

$$Z_{EG} = 60 W_{net}^{0.95} \tag{25}$$

As a result of summing Equations 20–25, we obtain the RATC system total capital cost (Z^{tot}).

Figure 4 shows the mutual influence of the total capital cost and the total efficiency of the RATC system when changing T_3 and π_c.

Figure 4 shows that the RATC system total capital cost significantly increases with decreasing the air turbine upstream air temperature T_3. This is due to the fact that with decreasing T_3, the mass flow rate of air turbine upstream air (m_{air}) increases (Figure 3). And with an increase in m_{air}, the costs of the system elements also increases. This especially affects the cost of the air turbine as the most expensive element of the RATC system. Also Figure 4 shows that the total efficiency of the RATC system operating in the mode with π_c = 1.8 and T_3 = 850°C is higher than that in the mode with π_c = 2.7 and T_3 = 700°C, while the total capital cost of the system is, on the contrary, lower. At first glance, we can conclude that the higher π_c and lower T_3, the more efficient and cheaper the RATC system. However, to increase the electrical efficiency of the system, on the contrary, the choice should be made in favor of increasing π_c, as can be seen from Figure 2 (a).

In cogeneration systems, a distinction is made between primary and secondary products. In this case, the electricity generated by the turbine is the primary product, because from an exergy position, this type of energy is of great value as pure exergy, i.e. energy without entropy component. Thermal energy is of lower quality from the standpoint of the exergy theory and has a large part of the entropy component. Consequently, the heat produced in the heat exchanger will be considered as a secondary product. In addition, as shown by Rusanov et al. [21], for systems operating on the Brighton air cycle, optimal modes should be determined taking into account the operation of this cycle, and not only by system electrical efficiency. This is due to the fact that when the electrical efficiency increases with increasing π_c, the cycle work decreases due to the increased consumption of electrical energy by the compressor. The conversion of electrical energy into mechanical energy of the compressor shaft rotation is accompanied by an increase in entropy (energy dissipation), which in turn leads to an increase in the operating costs of the system. Therefore, to evaluate all these factors, including the system products of the of different quality (from exergy positions), energy dissipation in system elements, operating costs for compensating the energy dissipation and capital costs for creating system elements, taking into account their efficiency, it is most correct to use the thermoeconomic approach to optimization of energy-technological systems.

For the final choose of the optimal operating and design parameters of the RATC system, we will use the method of

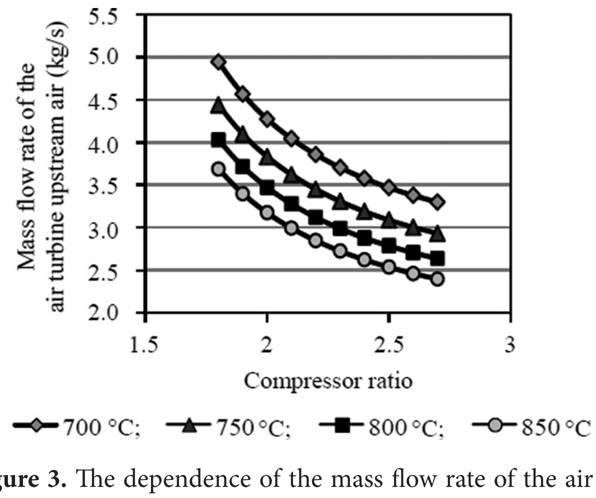

Figure 3. The dependence of the mass flow rate of the air turbine upstream air on π_c.

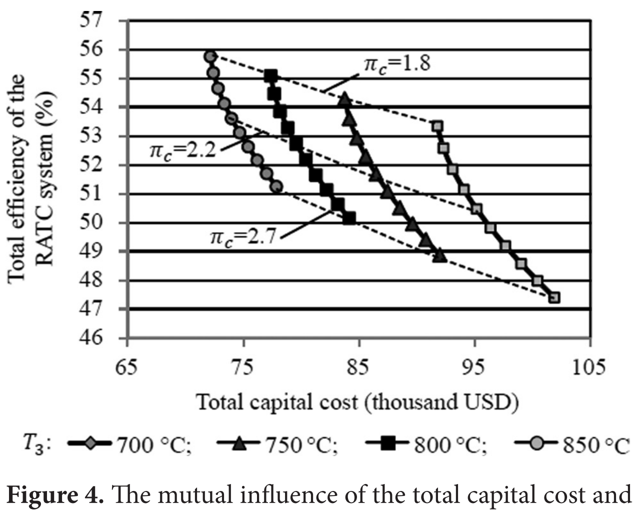

Figure 4. The mutual influence of the total capital cost and the total efficiency of the RATC system when changing T_3 and π_c.

thermoeconomic structural-variant optimization [18,19]. The advantage of this method is that the thermoeconomic model here is not associated with the specific technological scheme. Thus, it is possible to break the scheme and optimize each element individually using techno-economic parameters.

The purpose of the thermoeconomic structural-variant optimization of the RATC system is to determine the minimum total costs for the creation and operation of the system with a given capacity.

The exergy balance of the system as a whole can be written as:

$$E_D^{tot}(x) + E_{loss}^{tot}(x) = E_{in}(x) - E_{out} \tag{26}$$

where E_D^{tot} and E_{loss}^{tot} are the total exergy destruction in the all system elements and total losses of exergy to the environment respectively, $E_{in}(x)$ and E_{out} are the system inlet and outlet exergy flows respectively, and x is a variable parameter.

The total costs are related to the system service life:

$$\Xi_{tot}(x) = \tau_{oper}\tilde{c}_{in}E_{in}(x) + a_r Z^{tot}(x) + \tilde{b} \tag{27}$$

where τ_{oper} is the RATC system operating time during service life, $\tilde{c}_{in}$ is the cost factor of the system inlet primary flow (cost of fuel, since the boiler should be considered as inlet element to the RATC system), α_r is the recoverable amount ratio, $Z_{tot}(x)$ is the RATC system capital cost, and $\tilde{b}$ is the costs of maintenance, which does not affect the optimization.

Today, the world has adopted a methodology for the economic assessment of energy conversion systems, in which the contribution of the capital component to the cost of the target product is determined based on the return of bank investments to the project. Thus, the contribution of the capital cost component to the target product cost becomes nullified, which in general should contribute to the more intensive introduction of expensive energy-saving technologies. The investment component of the product cost is determined on the grounds that, during the RATC system service life, the loan should be returned to the bank with its interest taken into account [20].

To take this into account in Equation 27, we use the recoverable amount ratio, which is calculated by the equation:

$$a_r = \frac{r(1+r)^n}{(1+r)^n - 1} \tag{28}$$

where r and n are the discount factor and the system service life respectively.

With the thermoeconomic approach, exergy as a measure of the practical suitability of energy serves as the only basis for assessing the influence of the thermodynamic parameters of energy conversion systems on economic indicators characterizing the inefficiency of thermodynamic processes by additional financial costs. This requires an explanation.

Exergy, unlike energy, is consumed in the actual process of converting energy. This consumption can be divided into exergy destruction $\left(E_D^{tot}(x)\right)$ in system elements and exergy losses to the environment $\left(E_{loss}^{tot}(x)\right)$. To compensate for exergy consumption, fuel exergy is spent, which has a certain price. On the other hand, exergy consumption are influenced by each element, namely its efficiency, which is directly related to capital cost for its. Therefore, for a more correct analysis of the RATC system $E_{in}(x)$ in Equation 27, taking into account Equation 26, should be replaced by the sum $E_D^{tot}(x) + E_{loss}^{tot}(x) + E_{out}$. Differentiating Equation 27, we obtain:

$$\frac{\partial \Xi_{tot}(x)}{\partial x} = \tau_{oper}\tilde{c}_{in}\frac{\partial\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)}{\partial x} + a_r\frac{\partial Z^{tot}(x)}{\partial x} \tag{29}$$

In Equation 29, the value $\partial\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)/\partial x$ determines the effect of the variable parameter on the losses from the irreversibility of thermal hydraulic processes in each system element and losses to the environment. The value $\partial Z^{tot}(x)/\partial x$ takes into account the effect of changing some parameter x on the cost of the element.

The minimum total costs are determined if the left side of Equation 29 is equated to 0, then we get:

$$\tau_{oper}\tilde{c}_{in}\frac{\partial\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)}{\partial x} = -a_r\frac{\partial Z^{tot}(x)}{\partial x} \tag{30}$$

Exergy flow at the i-th point of the cycle is defined as follows:

$$E_i = m\left(c_p T_0\left[\frac{T_i}{T_0} - 1\right]\left[1 - \frac{\ln\left(T_i/T_0\right)}{T_i/T_0 - 1}\right]\right) + RT_0\ln\left(\frac{P_i}{P_0}\right) \tag{31}$$

where R is the universal gas constant.

Figure 5 shows a diagram of exergy flows in the RATC system, indicating the percentage of exergy destruction and losses in each element of the system from the total value of exergy destruction and losses. It should be noted that for each variation of the system parameters, this ratio changes. Therefore, Figure 5 shows the average ratio of exergy destruction and losses for all considered options.

In Figure 5, the exergy destruction E_{D_k} in each k-th element of the RATC system is the difference between the exergy flows at the system inlet (E_i) and outlet (E_{i+1}).

Figure 6 shows that the maximum exergy destruction is observed in the boiler. In this case, in the variant with T_3

= 850°C, the exergy destruction in the boiler is noticeably lower than in the variant with T_3 = 700°C. This is due to the fact that in the first variant, the air mass flow rate (m_{air}) required to provide of W_{net}= 300 kW is less. The exergy destruction in the air compressor and air turbine increases with increasing π_c, while the exergy destruction in the air heater decreases. However, at the same time, the exergy losses to the environment with flue gases increase.

To find the optimal system parameters corresponding to the minimum total costs, for convenience, we will use a graphical tool for the thermoeconomic optimization (method of constructing C-curves) [20].

The idea of analysis with the help of C-curves for many years remained only an idea, an illustrative material for

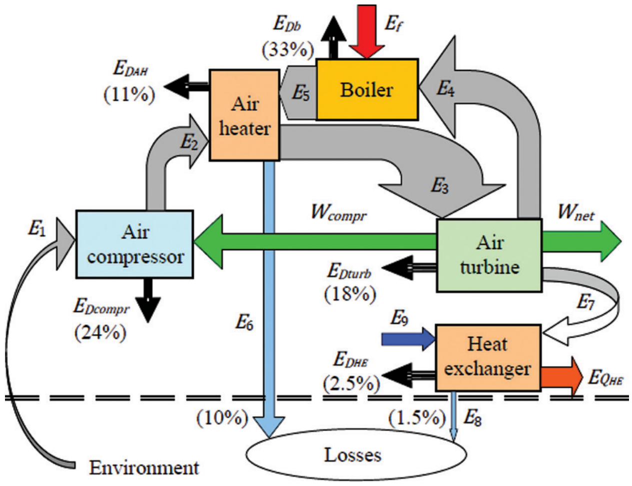

Figure 5. The exergy flow diagram of the RATC system.

textbooks on exergy analysis. With the widespread introduction of thermoeconomic analysis methods into the practice of designing energy conversion systems, graphic interpretation has acquired more significant information value [20]. The graphical apparatus of C-curves clearly shows the ratio of exergy consumption with other optimization factors. In thermoeconomic analysis such factors are the capital and operating costs for the system.

In this work, we proposed to use in a complex the structural-variant approach to optimization and the graphical apparatus of C-curves. The idea is to plot the graph $Z^{tot}(x) = f\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)$ that looks like the letter C and find the arc of choice on it (Figure 7). The arc of choice is a segment on the C-curve bounded by two points corresponding to the minimum values of $Z_{tot}(x)$ and $\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)$. Therefore, the arc of choice for each C-curve, which corresponds to a certain temperature T_3, in Figure 7 is located between the points of intersection of the C-curve with the lines π_c = 1.8 and π_c = 2.2. The options that lie on the right side of the intersection of the C-curves with π_c = 2.2 line are not included in the arc of choice (gray line in Figure 7), since with an increase in $Z_{tot}(x)$, an increase in the sum $\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)$ is observed.

The segments of the C-curve outside the arc of choice show the overruns of both exergy and capital costs. Therefore, in further analysis, only the arc of choice will be considered, each point on which may correspond to a compromise decision between the economic and exergy parameters of the RATC system.

Figure 7 shows that the minimum values of $Z_{tot}(x)$ and $\left(E_D^{tot}(x) + E_{loss}^{tot}(x)\right)$ correspond to the RATC system options that lie on the lower arc of choice. This is a part of the C-curve plotted for the RATC system operating modes at T_3 = 850°C and at π_c in the range from 1.8 to 2.2. Thus, first

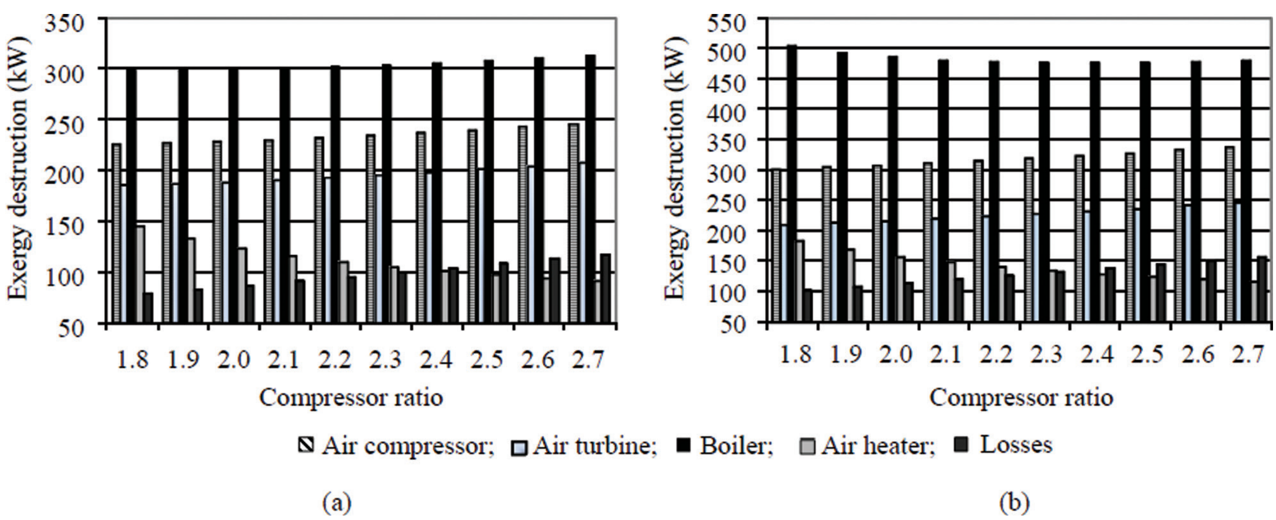

Figure 6. The dependence of exergy destruction in the main RATC system elements and losses on π_c at (a) T_3 = 850°C and (b) T_3 = 700°C.

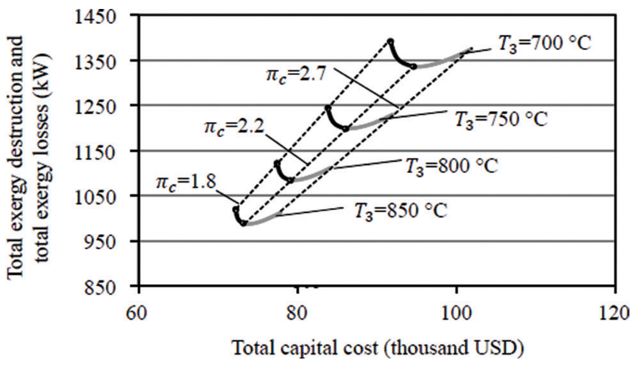

Figure 7. The dependence of $(E_D^{tot}(x) + E_{loss}^{tot}(x))$ on $Z^{tot}(x)$.

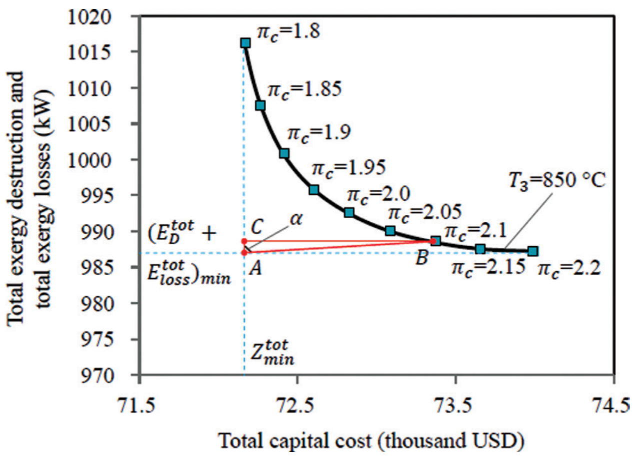

Figure 8. Choosing the best option of the RATC system with the air turbine net power of 300 kW.

the variable parameter T_3 (Figure 7), and then the variable parameter π_c (Figure 8) serve as a kind of navigator when searching for the optimal version of the RATC system.

As a result, the agreed optimum can be found in Figure 8 using the linear relationship between changes in $Z_{tot}(x)$ and $\left(E_D^{tot}(x)+E_{loss}^{tot}(x)\right)$:

$$\Delta Z^{tot}(x)=\tan\alpha\Delta\left(E_D^{tot}(x)+E_{loss}^{tot}(x)\right) \quad (32)$$

where the variable parameter (x) is π_c.

Drawing tangents to the arc of choice, as shown in Figure 8, we obtain the minimum values of Z^{tot} and $\left(E_D^{tot}+E_{loss}^{tot}\right)$. From point A, obtained at the intersection of these tangents, we draw a straight line at an angle α to the vertical. At the intersection of this line with the arc of choice, we find the agreed optimum (point B). Drawing a horizontal line from point B to a vertical tangent, we get point C. Thus, the segment CB in Figure 8 corresponds to the $\Delta Z^{tot}(x)$ value, and the segment AC corresponds to the $\Delta\left(E_D^{tot}(x)+E_{loss}^{tot}(x)\right)$ value (see Equation 32).

Accordingly, the tangent of angle α of a right-angled triangle ABC is:

$$\tan\alpha=\frac{CB}{AC} \quad (33)$$

To determine the value of the angle α, we proposed using Equation 30 to write α as:

$$\tan\alpha=\frac{\tau_{oper}\tilde{c}_{in}}{a_r} \quad (34)$$

The cost of the RATC system inlet exergy (cost of fuel) $\tilde{c}_{in}$ in USD/(kW·h) was calculated as follows:

$$\tilde{c}_{in}=\frac{\tilde{c}_{sf}3600}{1000}\frac{1}{Q_L} \quad (35)$$

where $\tilde{c}_{SF}$ is the cost of standard fuel, which today in the world is approximately 48 USD/t.

Under the given conditions, if we take, for example, natural gas as a fuel, the price of which is calculated by Equation 35 as $\tilde{c}_{in}$ = 0.00345 USD/(kW·h), and τ_{oper} = 8,000 h, then using Equation 34, α = 27.648 was calculated, which corresponds to the angle α = 88°. Forming this angle α in Figure 8, we can see that the agreed optimum (point B) corresponds to the RATC system operating at $\pi_{c_{opt}}$ = 2.1, which is therefore the best option.

Thus, at the given fuel cost, the choice should be made in favor of the RATC system with lower exergy destruction and losses but more expensive capital cost. However, if the cost of fuel is less, for example, when burning municipal solid waste, then the angle α, respectively, will be smaller. This will lead to the choice of the RATC system with a lower capital cost, but with greater exergy destruction and losses in the system.

Figure 9 shows the dependences of the exergy destruction in RATC system some elements depending on their capital cost with varying π_c and at the given values of W_{net}= 300 kW and T_3 = 850°C.

Figure 9 (b) shows that exergy destruction in the boiler decrease with increasing π_c, while the boiler capital cost increases. There is no contradiction in this, since the more efficient the system element, the more expensive it is. However, with the presented combination of thermodynamic and mass characteristics of the RATC system, with an increase in the exergy destruction in the air turbine and air compressor, their capital cost also increases (Figure 9 (a) and (b)). This is due to the fact that although an increase in π_c leads to a decrease in mass flow rate of the air turbine upstream air (m_{air}) (Figure 3), it also leads to an increase in compressor outlet air pressure (P_2) and air turbine upstream air pressure (P_3). In equations 20 and 24, increasing P_2 and P_3 has a greater effect on increasing the air compressor and

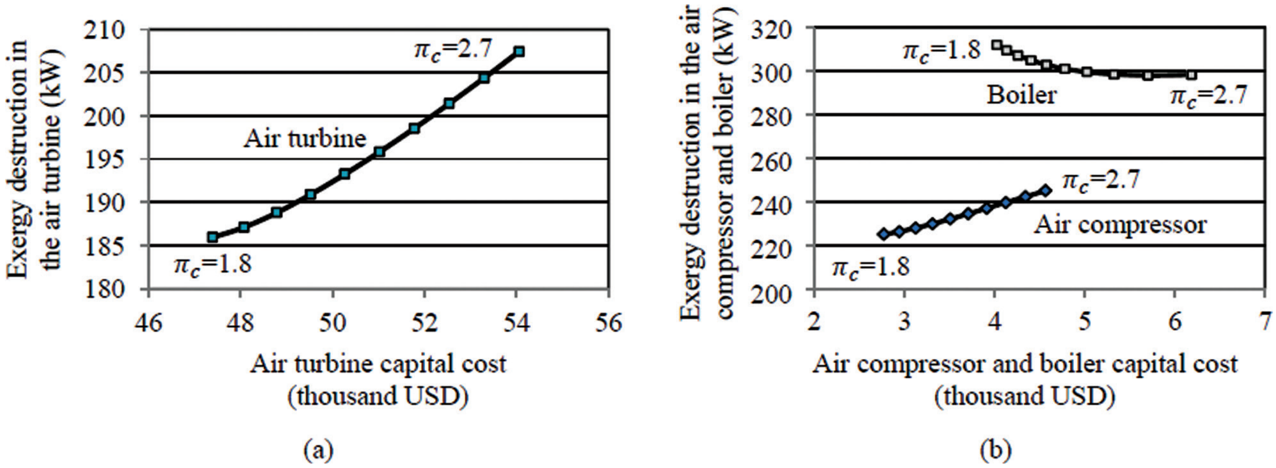

Figure 9. The dependence of the exergy destruction in the (a) air turbine, (b) air compressor and boiler of the RATC system with the air turbine net power of 300 kW at T_3 = 850°C on the capital cost of these elements.

turbine capital costs than decreasing m_{air} has an effect on decreasing them.

DISCUSSION OF THE RESULTS OF THERMOECONOMIC OPTIMIZATION OF THE RATS SYSTEM

The possibility of using the structural-variant method in combination with the graphical apparatus of C-curves for pre-design analysis and thermoeconomic optimization of the regenerative air turbine cogeneration (RATC) system is shown. This made it possible to choose the optimal operating and design parameters of the RATC system in terms of exergy and economic indicators.

Each variable operating mode parameter of the RATC system serves as a kind of navigator when searching for the optimal system parameters, which is accompanied by graphic visualization. Therefore, the proposed approach makes the optimization of the system being designed convenient and clear.

On the other hand, due to lack of data, the problem of the capital cost of the air turbine operating at air temperatures from 700°C to 850°C remained unconsidered. In this work, Equation 24 was used, obtained for the capital cost of gas turbines, the cost of which varies in the range of USD 400-600 per kW depending on the manufacturer. However, when designing an air turbine operating at lower temperatures of the working substance, it is possible to use cheaper materials and not use a cooling system. This may significantly (~30%) reduce its cost. This problem can be the subject of further research.

But, despite the fact that the determination of the capital cost of the air turbine can be a disputable issue and requires clarification, this does not reduce the practical value of the presented approach.

The area of practical application of the structural-variant method of thermoeconomic optimization using C-curves is not limited only to cogeneration systems. The proposed approach can be used in the optimal design of various types of thermal transformers [20] and other energy-technological systems [24–26].

CONCLUSION

The concept of optimization of operating and design parameters of the regenerative air turbine cogeneration (RATC) system is proposed.

To determine the energy efficiency indicators of the RATC system with the air turbine net power of 300 kW, its thermodynamic analysis was performed. The following parameters were chosen to be variable: the compressor ratio (π_c) in the range from 1.8 to 2.7 and the air turbine upstream air temperature (T_3) in the range from 700°C to 850°C.

It has been shown that an increase in T_3 leads to an increase in both the electrical and total efficiency of the RATC system while decreasing total capital cost of the system. This is due to the fact that with an increase in T_3, the mass air flow rate of the air turbine upstream air (m_{air}) decreases at the same air turbine net power. With decreasing m_{air}, the capital cost of RATC system elements and losses with flue gases decreases. Decreasing π_c also increases the total efficiency and decreases the total capital cost of the system. However, the main product of the cogeneration system is electricity, which has a greater exergy value than heat. And in order to increase the electrical efficiency of the system, on the contrary, a choice should be made in favor of increasing π_c.

Therefore, on the basis of the data obtained during the thermodynamic analysis, exergy destruction and losses in the main system elements were calculated. Then, using the structural-variant method in combination with the graphical apparatus of C-curves, the pre-project thermoeconomic optimization of the RATC system was performed. As the determining factors in choosing the optimal variant of the RATC system parameters in terms of exergy and economic indicators, we chose the total efficiency of the system, the system total capital cost , as well as the operating costs and exergy destruction and losses in the main system elements.

As a result of the pre-project thermo-economic optimization of the RATC system with the air turbine net power of 300 kW, the following optimal mode variable parameters were chosen: the compressor ratio (π_c) is 2.1 and the air turbine upstream temperature (T_3) is 850°C. These parameters provide the compromise decision between the economic and exergy parameters of the RATC system. In this case, the RATC system heat output (Q_{HE}) is 167 kW.

The purpose of further research will be a deep element-by-element thermoeconomic analysis of the RATC system with the choice of its optimal design characteristics.

NOMENCLATURE

A	Surface area, m^2
a_r	Recoverable amount ratio
$\tilde{b}$	Cost of maintenance, USD
c_p	Specific isobaric heat capacity, kJ/(kgK)
$\tilde{c}_{in}$	Cost factor of the system inlet primary flow (cost of fuel), USD(kW h)
$\tilde{c}_{sf}$	Cost of standard fuel, USD/t
$\dot{E}$	Exergy flow, kW
E_D	Exergy destruction, kW
E_{loss}	Exergy losses, kW
i	Specific enthalpy, kJ/(kg)
$\underline{k}$	Isentropic exponent
m	Mass flow rate, kg/s
n	System service life, year
p	Pressure, Pa
$\dot{P}$	Generalized loss coefficient for the flow pressure
Q	Heat output, kW
Q_L	Lower heat of fuel combustion, kJ/(kg)
R	Universal gas constant, J/(mol K)
r	Discount factor
T	Temperature, °C
$\bar{T}_{\ln}$	Mean logarithmic temperature head, °C
W	Power, kW
x	Variable parameter
Z	Capital cost, USD

Greek symbols

η	Efficiency
Ξ_{tot}	Total costs, USD
π_c	Compressor ratio
τ_{oper}	Operating time, h

Subscripts

0	Refers to environment
1	Refers to air at air compressor inlet
2	Refers to air at air compressor outlet
3	Refers to air turbine upstream air
4	Refers to air at boiler inlet
5	Refers to flue gas at air heater inlet
6	Refers to flue gas at air heater outlet
7	Refers to air at heat exchanger inlet
8	Refers to air at heat exchanger outlet
9	Refers to water at heat exchanger inlet
10	Refers to water at heat exchanger outlet
AH	Air heater
air	Air
b	Boiler
CD	Air compressor drive
compr	Air compressor
EG	Electric generator
el	Electrical
f	Fuel
fg	Flue gas
HE	Heat exchanger
in	Inlet
ind	Indicated
mech	Mechanical
net	Net power
out	Outlet
tot	Total
turb	Air turbine
wtr	Water

AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. Raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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