Thermal energy recovery efficiency improvement of gas turbine technology

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Abstract. The research relevance on improving the thermal energy recovery efficiency of gas turbine technology is caused by the need to reduce the fuel and energy consumption when generating the electric energy. The research objective is to define the technically and economically feasible heat recovery ratio of exhaust gases after gas turbine in the heat regenerator. The optimization problem of the heat potential recovery process was solved. The algorithm to calculate the optimal temperature of heating the air supplied to the combustion chamber of the gas turbine unit was developed. Methods of mathematical modelling of heat exchange processes, methods of choosing the rational option in solving the optimization problem, and mathematical methods of solving the optimization problems were used in the paper. The influence analysis of the optimal air temperature required for burning the fuel in the combustion chamber on the technical and economic efficiency of the Siemens SGT-100 gas turbine equipment was carried out. The expediency of applying the performed studies results of practical application in designing and operating gas turbine machines was substantiated.

Key-words: heat regenerator, gas turbine equipment, discounted costs, efficiency, optimization.

1. Introduction

Gas turbine technologies with spent operating medium thermal energy regeneration are widely used to produce the electric energy in the industrial heat and power systems including the high-temperature processing units. High temperature fuel combustion products are used as the operating medium. Gas turbine technology includes both gas turbine equipment and heat recovery units. The uniqueness of the given technology involves gas thermal energy recovery after gas turbines and simultaneous production of the electric energy [1-3].

The objective of reducing the energy resources consumption in the industrial gas turbine technology, as well as improving and increasing the technical and economic efficiency of gas turbine equipment operation, reducing the harmful impact on the environment is relevant.

The ways of improving the cycle parameters of gas turbine equipment by increasing the compression ratio of the air supplied to the combustion chamber of the gas turbine machine and increasing its temperature due to compression are examined in papers [4, 5]. The given methods do not seem to be sufficiently effective, as they can result in the compressor operating costs increase.

In papers [6-8], it is proposed to heat the air by means of the recovery of the gas turbine exhaust gases heat energy and to reduce the consumption of the fuel and energy resources and operating costs. But at the same time, it is pointed out that the high-temperature air heating leads to the regenerator area increase and, as a consequence to the increase of the capital investments.
The need to improve the efficiency of regenerative systems is caused by the requirements to reduce the consumption of the world's fuel and energy resources and reduce operating costs when producing the energy products [1-14].

The disadvantage of the existing methods and solutions on recovery of the exhaust gas heat after gas turbine machine is a poor accuracy when determining the best relationship between the main parametric values, incomplete consideration of economic factors, and a sufficiently large amount of labour costs.

2. Problem statement

The efficiency improvement of energy saving in the gas turbine technology cycle is possible to be achieved by ensuring a more complete recovery of the exhaust gases thermal energy after the gas turbine machine for heating the air supplied to the combustion chamber. Heat recovery of the fuel combustion products can be carried out in the thermal regenerator installed behind the gas turbine. The recovery of the exhaust gases thermal potential for increasing the air enthalpy, when it is heated in a thermal regenerator, will reduce the energy resources consumption and increase the gas turbine technology efficiency value. However, the capital investments and operating costs of the exhaust gases heat recovery are increased. Consequently, it is necessary to provide the maximum technically possible and economically justified heat recovery ratio of the spent fuel combustion products in the gas turbine.

The efficient recovery of exhaust gases heat is ensured by achieving the minimum discounted costs, that is, at the lowest capital investments and operating costs.

The solution of the posed problem contributes to reducing the fuel and energy resources costs. The application of the exhaust gases thermal energy after heat and technological machines when heating the air in regenerators is the main way to improve the operating efficiency.

The task of developing the calculation method and algorithm of the air heating optimal temperature in a thermal regenerator, taking into account the fuel component of the electric energy production cost in the gas turbine technology cycle, as well as the changing operating costs and capital investments has been set in the paper.

3. Theory

The research on improving the efficiency of the exhaust gases thermal regeneration is conducted by examining the classical heat and power system with gas turbine equipment (figure 1) [5-8]. In the given flowchart, the required thermal power is provided due to the operation of the thermal regenerator at the given temperature of the gas turbine cycle exhaust gases. Gas turbine operation contributes to the reliable supply of the electric energy, both to the heat and power system internal consumption and to the outside consumers.
Figure 1. The flow chart of the gas turbine technology heat and power system with thermal energy recovery. 1 is the air flow of the compressor; 2 is the compressor; 3 is the heat regenerator; 4 is the gas turbine combustion chamber; 5 is the fuel path of the combustion chamber; 6 is the gas path; 7 is the gas turbine; 8 is the generator; 9 is the exhaust flue gas path out of the gas turbine.

The heat regenerator is installed for heating the air behind the gas turbine in the exhaust pipe (figure 2):

- \( d_r \) is the internal diameter;
- \( \delta \) is the wall thickness of the regeneration heat exchange;
- \( l_r \) is the length of the active heat transfer surface of regenerator;
- \( d_g \) is the diameter of the inlet and outlet regenerator nozzles;
- \( t_{a1}, t_{a2} \) are the air temperatures at the inlet and outlet of the thermal regenerator;
- \( t_{g1}, t_{g2} \) are the gas temperatures at the inlet and outlet of the thermal regenerator, °C.

Figure 2. Thermal regenerator.
Thermal operation of gas turbine equipment with thermal energy recovery along the gas flow is based on the interaction between the «compressor – combustion chamber – gas turbine – thermal regenerator» systems. Complex heat exchange is performed in the system. In the process of heat exchange, heat is transferred by radiation, convection, and thermal conductivity.

Heat transfer and heat energy balances of the gas turbine technology are considered under the main assumptions: the heat exchange and energy transfer are stationary processes; the heat flux densities on the heat exchange surfaces are constant; the heat energy in the heat exchange systems is transferred at the constant temperature. Single-valued conditions include the following: the initial conditions: gas turbine power and gas temperature behind it are defined, the geometric conditions: the dimensions and geometry are set, the physical conditions: the general physical properties are viscosity, density, heat capacity, thermal conductivity, the boundary conditions are the supplementary conditions for the operating space of the units.

The criterion of the capital investments economic efficiency is characterized by the lowest costs taking into account the operating costs and capital investments [8-10]:

\[ E = C + I r K \]  \hspace{1cm} \text{(1)}

where \( E \) is the total discounted expenditures for the heat regenerator and combusted fuel, RUB/year; \( C \) is the annual operating costs, RUB/year; \( I r \) is the investments discount rate 1/year; \( K \) is the capital investments, RUB;

When using gas turbine equipment, operating costs are composed of the energy costs of the fuel combustion process in the combustion chamber and costs of implementing the gas turbine technology, as well as other production costs for the electric energy generation. For the purpose of minimizing the time and labour costs when finding the most profitable solution of the optimization problem of gas turbine unit thermal operation, the costs associated with reducing the harmful impact on the environment, and maintenance staff are assumed to be constant.

When installing a thermal regenerator in a gas turbine technology the variable costs include the costs for the fuel, injection machines (compressor, smoke exhauster, etc.). The cost components related to the deductions for the renovation and maintenance of gas turbine equipment are assumed to be proportional to the capital investments.

Using the classical mathematical optimum seeking method when the minimized function partial derivatives are equal to zero [9], for calculating the most optimal temperature value of the heated air at the heat regenerator outlet, the equation (1) providing the achievement of the minimum capital investments and operating costs is transformed into the following form:

\[ \frac{dE}{dt_a} + C_f \frac{dB}{dt_a} + C_r \frac{dF}{dt_a} = 0 \]  \hspace{1cm} \text{(2)}

where \( t_a \) is the air temperature in the burner device of the combustion chamber, °C; \( C_f, B \) are the annual cost and fuel consumption (RUB/m³)·(s/year), m³/s; \( C_r \) is the cost of the regenerator heat transfer area unit RUB/(m²·year); \( F \) is the area of the thermal regenerator heat transfer surface, m².

The annual fuel cost and the cost of the regenerator heat transfer surface area unit can be defined according to the following formulas [3, 8-10]:

\[ C_f = A_f \tau \]  \hspace{1cm} \text{(3)}

\[ C_r = K_r (I_r + O) + N_{d.u.} Z A_{d.u.} (I_r + O) + C_e \tau \]  \hspace{1cm} \text{(4)}

where \( A_f \) is the annual costs of 1 kg of gaseous or liquid fuel, RUB/m³, RUB/kg; \( \tau \) is the annual operation time of the thermal regenerator, s/year; \( K_r \) is the nonrecurring expenditures of the manufacturing and installing the heat transfer surface unit of the thermal regenerator, RUB/m²; \( O \) is the renovation deductions share, 1/year; \( N_{d.u.} \) is the supplied electric power of the heat exchanger heat transfer surface unit operation, W; \( Z \) is the electric power reserve coefficient; \( C_{d.u.} \) is the cost of discharge units, RUB/W; \( C_e \) is the electric energy cost, RUB/ (W·s).
Gas turbine fuel consumption can be defined according to the heat balance equation of the gas turbine technology with thermal energy recovery [1-4]:

\[
B = \frac{N_e + Q_{env} + Q_{cool} + Q_{ac}}{Q_a (1-R_2) + C_{f1} t_{f1} - V g (C t_{g} + R_1) + V g C_{g2} (t_{g2} - \delta t_{g})},
\]

where \(N_e\) is the rated power of the gas turbine, \(W\); \(Q_{env}\) is the heat losses from the thermal regenerator to the atmosphere, \(W\); \(Q_{cool}\) is the heat losses with cooling medium, \(W\); \(Q_{ac}\) is the heat losses during periods of run-up and shutdown of the thermal regenerator, \(W\); \(Q_a\) is the net calorific value of the operating fuel, \(J/m^3, J/kg\); \(R_2\) is the losses proportion of the mechanical incompleteness of combustion; \(C_{f1}, t_{f1}\) are the heat capacity and fuel temperature, \(J/(m^3 \cdot K), 0^\circ C\); \(V g\) is the obtained volumetric component of the natural gas combustion products calculated per unit of its quantity, \(m^3/m^3\); \(t_{g2}, C_{g2}\) are the temperature and heat capacity of the gas turbine exhaust gases, \(J/(m^3 \cdot K), 0^\circ C\); \(R_1\) is the thermal energy of the unburned carbon monoxide in the flue gases, \(J/m^3\); \(V a\) is the required air volume for the complete fuel combustion, \(m^3/m^3\); \(C_{a2}\) is the heat capacity of air at the thermal regenerator outlet, \(J/(m^3 \cdot K), 0^\circ C\); \(\delta t_g\) is the air temperature decrease on the way out of the thermal regenerator to the combustion chamber, \(0^\circ C\).

The heat exchange area of the heat exchanger (\(F\)), the average temperature difference between the heat-transferring and heat-receiving media in the thermal regenerator (\(\nu\)), the gas temperature values at the inlet (\(t_{g1}\)) and outlet (\(t_{g2}\)) of the heat regenerator are defined according to the equations (6)–(9) [9]:

\[
F = \frac{B n a V a (C_{a2} t_{a2} - C_{a1} t_{a1})}{K \epsilon F u} \quad (6)
\]

\[
\nu = A (t_{g2} - t_{a1}) + B (t_{g1} - t_{g2}) \quad (7)
\]

\[
t_{f1} = \frac{C_{g1} t_{g1} + \Theta C_{a1} t_{a1}}{(1+\Theta) C_{g1}} \delta t_g \quad (8)
\]

\[
t_{g2} = \frac{C_{g1}}{C_{g2}} (t_{g1} - \frac{V a n a (C_{a2} t_{a2} - C_{a1} t_{a1})}{V g n g C_{g2}(1+\Theta)}) \quad (9)
\]

where \(n_a\) is the coefficient showing the air losses share in the heat regenerator; \(C_{a1}\) is the heat capacity of air at the inlet of the heat regenerator, \(J/(m^3 \cdot K)\); \(K\) is the heat transfer coefficient, \(W/(m^2 \cdot K)\); \(\epsilon\) is the correction coefficient of the complex heat exchange circuit; \(\nu\) is the average temperature difference, \(0^\circ C\); \(A\) and \(B\) are the coefficients depending on the ratio \((t_{g2} - t_{a1})/(t_{g1} - t_{a2})\); \(C_{g1}, C_{g2}\) are the heat capacities of gases at the inlet and outlet of the heat exchanger, \(J/(m^3 \cdot K)\); \(n_g\) is the coefficient showing the heat losses through the building envelopes of the heat exchanger walls into the environment; \(\Theta\) is the coefficient indicating the exhaust gases dilution with air on the pathway to the heat regenerator; \(\delta t_g\) is the gas temperature loss on the way to the heat regenerator.

The solution of equation (2) after differentiation and substitution of formulas (3)-(9) for heat exchange equipment has the following form:

\[
\begin{align*}
\dot{a} & = C_r n_a C_{a2} V a W - K \epsilon A t C_{a2} C_t W^2 \\
\dot{b} & = 2 (K \epsilon A t C_t U - C_r n_a C_{a1} t_{a1} V_a C_{a2} W) \\
\dot{d} & = C_r n_a \left[ C_{a2} E U + C_{a1} t_{a1} (C_{a2} V a U - W E) \right] - K \epsilon A t C_{a2} C_t U^2 \\
U & = (A \frac{C_{g1}}{C_{g2}} + B) t_{g1} + (\frac{V a n a C_{a1}}{V g n g C_{g2}(1+\Theta)}) At_{a1}
\end{align*}
\]

\[(10)\]
\[ W = \frac{AV_a \eta_a C_2}{V_g \eta_g C_g (1 + \Theta)} + B \]  
\[ E = Q_a (1 - R_a) + C_n t_1 - V C_t - V R - V C_{\delta a} \delta t_a \]  

where \( a, b, d, U, W, E \) are the ranges of values.

4. Experimental results
Based on the obtained equation (10) and formulas (11-16), the algorithm to define the optimal temperature of heating the air \( t_{a,\text{opt}} \) supplied to the combustion chamber of the gas turbine has been developed.

The numerical studies were carried out for the thermal power system using the gas turbine Siemens SGT-100 [14]: the nominal power is 5.05 MW; the air temperature outside the compressor is 340 °C; the exhaust gases temperature at the outlet of the gas turbine is 545 °C.

The graphical dependences of the total discounted costs for the thermal regenerator and combusted fuel, fuel consumption and heating surface of the thermal regenerator on the air temperature at the burner inlet of the combustion chamber (figures 3, 4), as well as the dependences of the air heating optimal temperature in the regenerator on the annual fuel cost and the area unit cost of the regenerator heat transfer surface (figures 5, 6) have been obtained. These dependencies are valid for the industrial enterprises of the Northern regions of Russia. The fuel used is natural gas with a net calorific value of 35.4 MJ / m³.

![Figure 3. The dependence of \( E \) on \( t_a \).](image)
Figure 4. The dependencies of B and F on $t_a$.

Figure 5. The dependence of $t_{a,\text{opt}}$ on $C_f$. 
5. Results discussion
The numerical studies obtained results analysis of the dominant factors influence on the value of the optimal heating temperature of the air supplied to the combustion chamber of the gas turbine revealed that the increase in the fuel costs and annual operation time of the gas turbine up to 13% results in the \( t_{a,\text{opt}} \) rising. The increase of the heat regenerator heat transfer surface cost leads to the decrease of \( t_{a,\text{opt}} \) to 11%. The specific fuel consumption and discounted costs decrease to 15 % at the calculated optimal temperature of heating the air supplied for the fuel combustion of \( t_{a,\text{opt}} = 416 \) °C.

6. Conclusions
The regenerator optimal air heating temperature value calculation according to the proposed algorithm improves the efficiency of the gas turbine equipment thermal operation while minimizing the capital investments and operating costs.
The presented algorithm makes it possible to define the optimal air heating temperature in the regenerator taking into consideration the operating conditions, capacity of the gas turbine depending on the cost of the thermal regenerator, type of the fuel, annual operating life of the gas turbine unit.
The conducted numerical studies results substantiate the expediency of using the given algorithm in designing and operating gas turbines and thermal power system of gas turbine technology.

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