OPTIMIZATION OF GAS-TURBINE COMBINED CYCLES

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Abstract

At the current stage of development of energy industry, major challenge for gas-turbine combined cycle units is improvement of maneuverability, that requires optimization of their thermodynamic parameters, technological schemes, management and regulation systems according to the requirements of their operation modes. Optimization task is based on a multifaceted analysis, which depends on various factors.

The study is dedicated to thermodynamic analysis and optimization of the parameters of the gas-turbine combined cycle. It includes a formulation of the cycle's thermal efficiency factor in using single and three-pressure steam loops as well as the cases with or without additional fuel combustion; the change of the efficiency factor depending on the changes of the efficiency factors of gas and steam cycles as well as heat recovery boilers and thermodynamic parameters is analyzed; key factors affecting the cycle efficiency are identified; the optimization task based on the decision of which the nature of the change of thermal efficiency factor of the combined cycle is determined is formulated - it increases significantly when initial parameters of the gas-turbine cycle increase and its increase is rather insignificant initial parameters of the steam-turbine cycle increase. Increase of the overheated steam pressure does not result in the significant increase of the efficiency factor of the combined cycle. It has an extreme point from where the increase of the initial pressure results in the reduction of the cycle's efficiency factor.

Specific properties of the heat recovery steam boiler as well as the change of the exhaust steam humidity as per initial parameters, etc. should be taken into consideration for optimization of the real cycle units. At this time, maximum values of the efficiency factor are achieved at the initial parameters that are different from the ideal cycles, and the optimization of the control and designing is conducted for specific conditions of the capacities and operation modes.

Key Words: *Grid, cycle, combined, optimization, extremum, gas turbine, power-generating unit.*

Introduction

Gas-turbine combined-cycle power-generating units are highly efficient units and are applied for generating base power. They are widely used in Eastern European and Post-Soviet Union countries where, till late 1990s, synchronous operation of the power systems and covering of deficit by power distribution was possible – there were proper technical capabilities in the systems (grids). As a result of the well-known geopolitical changes, in late 1990s, grids were isolated and converted into autonomous units. At the same time, there were less technical capabilities to connect them to each other, and this resulted in serious problems especially at small grids – it became hard to manage deficit, and maneuverability and sustainability decreased. Consequently, increase of the installed capacities, sustainability and flexibility became important, and this needs to be resolved by applying high maneuverability and efficiency power-generating units. Under such conditions, main challenge for the gas-turbine combined-cycle (cc) power-generating units is the improvement of the maneuverability which increases their participation in a daily regulation of the grid loads. This task requires optimization of thermodynamic parameters of the power-generating units, technological schemes, management and regulation systems according to the requirements of the operation modes.

Goal of the article authors is study of the dependence of the ideal gas-steam combined cycle efficiency factor on thermodynamic parameters and definition of the limits for the operation modes of such cycle units and their optimization tasks.

The study was conducted on the example of the scheme and parameters of GE206FA standard combined-cycle unit.

Thermal Efficiency factor of the Ideal Combined Cycle

Let's begin our analysis by formulating thermal efficiency factor of the ideal combined cycle. For simplicity purposes, let's review a single-pressure topping cycle – the cycle within which heat capacity of the exhaust gases in the gas turbines is sufficient for the operation of the steam turbine loop and it is not necessary to combust additional fuel in the heat recovery steam generator (combined gas-steam cycle). Ts cycle chart is shown on Fig.1, where the following symbols are used: Q_{1G} and Q_{2G} - heat supplied to and rejected from the gas-turbine cycle; Q'_{1S} and Q'_{2S} - heat supplied to and rejected from the steam-turbine cycle.

Let's write thermal efficiency factors of the gas-turbine, steam-turbine and combined cycle in a following manner:

$$\eta_G = (Q_{1G} - Q_{2G})/Q_{1G} \tag{1}$$

Steam-turbine cycle:

$$\eta_S = (Q_{1S} - Q_{2S})/Q_{1S}. \tag{2}$$

Combined cycle:

$$\eta_C = (Q_{1G} - Q_{2G} + Q_{1S} - Q_{2S})/Q_{1G}. \tag{3}$$

Let's introduce efficiency factor of the heat recovery steam generator (HRSG):

$$\eta_{SG} = Q_{1S}/Q_{2G}. \tag{4}$$

As a result of transformations, we will derive an equation of thermal efficiency factor for the single-pressure ideal combined cycle without combustion of additional fuel:

$$\eta_C = \eta_G + \eta_{SG}(\eta_S - \eta_G\eta_S)\ . \tag{5}$$

The equation in additional fuel combustion case will be get the following form:

$$\eta_C^{a} = \frac{1}{1+\alpha}\eta_C + \frac{\alpha}{1+\alpha}\eta_{SG}\eta_S\ , \tag{6}$$

where $\alpha = Q^{a}/Q_{1G}$ is the ratio of Q^{a} heat obtained by additional heat combustion with Q_{1G} heat supplied to the gas-turbine cycle.

The equation of the thermal efficiency factor for the ideal three-pressure combined cycle without combustion of additional fuel (Fig. 2) is given below. It shows dependence of the equation structure on the amount of the pressure loops in the steam cycle:

$$\eta_{C3} = \eta_C + \frac{1}{A_1}\eta'_{SG}(\eta'_S - \eta_G\eta'_S) + \frac{1}{A_2}\eta''_{SG}(\eta''_S - \eta_G\eta''_S) + \frac{1}{A_3}\eta'''_{SG}(\eta'''_S - \eta_G\eta'''_S)\ , \tag{7}$$

where A_1 , A_2 and A_3 are combination of ratios respectively for the first, second and third loops that are calculated in a following manner: $A_1 = (\alpha_{12}\alpha_{13} + \alpha_{13} + \alpha_{12})/\alpha_{12}/\alpha_{13}$, $A_1 = (\alpha_{12}\alpha_{13} + \alpha_{13} + \alpha_{12})/\alpha_{12}/\alpha_{13}$, $A_2 = (\alpha_{12}\alpha_{23} + \alpha_{23} + 1)/\alpha_{23}$, $A_3 = \alpha_{13} + \alpha_{23} + 1$. Here $\alpha_{12} = Q'_{1G}/Q''_{1G}$, $\alpha_{13} = Q'_{1G}/Q'''_{1G}$, $\alpha_{23} = Q''_{1G}/Q'''_{1G}$ are the ratio coefficients showing distribution of Q_{1G} heat supplied to the gas-turbine cycle proportionally to the steam loop capacities - $Q_{1G} = Q'_{1G} + Q''_{1G} + Q'''_{1G}$; η'_{SG}, η''_{SG}, η'''_{SG}, η'_S, η''_S, η'''_S - are, respectively, the efficiency factors of the first, second and third (high, medium and low pressure) loops of the heat recovery steam boiler and steam-turbine cycle.

Extremum of Thermal Efficiency Factor and Optimization Task

Below we will review dependence of the thermal efficiency factor of the ideal combined cycle on the efficiency factors of the constituent cycles and HRSG. For simplicity purposes, we will take single-pressure cycle without combustion of additional fuel (Fig.1). The results will apply to two and three-pressure cycles.

Initially let's assume, that η_G, η_{SG}, η_S efficiency factors included in (5) are independent variables. In this case, efficiency factor of the combined cycle, in an uncertain form, will be written as follows:

$$\eta_C = \eta_C(\eta_G, \eta_{SG}, \eta_S), \tag{8}$$

from where:

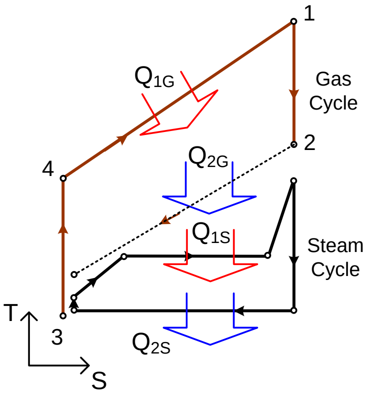


Figure 1. Single-pressure ideal combined cycle
(single-loop cycle without combustion of additional fuel)

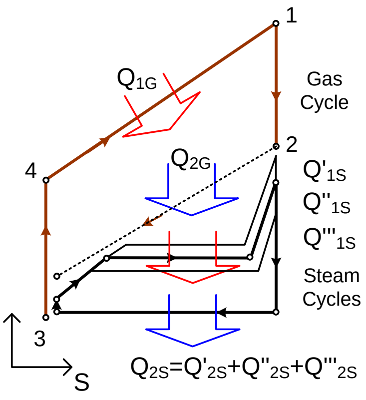


Figure 2. Three-pressure ideal combined cycle
(three-loop cycle without combustion of additional fuel)

$$d\eta_C = \frac{\partial\eta_C}{\partial\eta_G} d\eta_G + \frac{\partial\eta_C}{\partial\eta_{SG}} d\eta_{SG} + \frac{\partial\eta_C}{\partial\eta_S} d\eta_S \qquad (9)$$

If we consider, that based on (5) $\partial\eta_C/\partial\eta_G = 1-\eta_{SG}\eta_S$, $\partial\eta_C/\partial\eta_{SG} = \eta_S(1-\eta_G)$ and $\partial\eta_C/\partial\eta_S = \eta_{SG}(1-\eta_G)$, the following will be derived from (9):

$$d\eta_C = (1-\eta_{SG}\eta_S)d\eta_G + \eta_S(1-\eta_G)d\eta_{SG} + \eta_{SG}(1-\eta_G)d\eta_S\,. \qquad (10)$$

Let's review key parameters, that affect η_G, η_S and η_{SG} values. For the gas turbines that are: environmental pressure - p_a and temperature - T_a, initial pressure of gases - p_1 and temperature - T_1 (upon entering the first stage of the gas turbine); For the steam turbine: initial pressure - p_0, temperature - T_0 and pressure in the condenser - p_C. For HRSG: initial pressure - p_0, steam flow - D_0 and temperature gradient between the exhaust gasses (T_2) and initial temperatures (T_0). During analysis we will not consider the parameters (p_a, T_a, p_C) depending on environmental factors, and, as a constant value, we will get temperature gradient too. In such case, generally, we may write the following:

$$\begin{aligned} \eta_G &= \eta_G(p_1, T_1); \\ \eta_S &= \eta_S(p_0, T_0); \\ \eta_{SG} &= \eta_{SG}(p_0, D_0); \end{aligned} \qquad (11)$$

from where:

$$\begin{aligned} d\eta_G &= \frac{\partial\eta_G}{\partial p_1}dp_1 + \frac{\partial\eta_G}{\partial T_1}dT_1; \\ d\eta_S &= \frac{\partial\eta_S}{\partial p_0}dp_0 + \frac{\partial\eta_S}{\partial T_0}dT_0; \\ d\eta_{GS} &= \frac{\partial\eta_{GS}}{\partial p_0}dp_0 + \frac{\partial\eta_{GS}}{\partial D_0}dD_0\,. \end{aligned} \qquad (12)$$

Based on (12), the following will be derived from (10):

$$d\eta_C = (1-\eta_{SG}\eta_S)\frac{\partial\eta_G}{\partial p_1}dp_1 + (1-\eta_{SG}\eta_S)\frac{\partial\eta_G}{\partial T_1}dT_1 + (1-\eta_G)\left(\eta_S\frac{\partial\eta_{SG}}{\partial p_0} + \eta_{SG}\frac{\partial\eta_S}{\partial p_0}\right)dp_0 + \eta_S(1-\eta_G)\frac{\partial\eta_{SG}}{\partial D_0}dD_0 + \\ + \eta_{SG}(1-\eta_G)\frac{\partial\eta_S}{\partial T_0}dT_0 \qquad (13)$$

from where it is obvious, that in order to determine an extremum of the thermal efficiency factor of the combined cycle, it is necessary to know the dependences on p_1, T_1, p_0, D_0, T_0 parameters of the efficiency factors of the gas-turbine and steam-turbine cycle as well as the HRSG. On the other hand, the temperature of the exhaust gasses of the gas turbines which, specifically, depends on p_1 and T_1, defines the values and limits of p_0 and T_0. Respectively, η_G, η_{SG} and η_S are interdepending values that are intertwined with the above-mentioned parameters (although initially we mentioned that they were independent variables). Results of the computation study conducted by the authors are given on Figures

3, 4, 5 showing that in realistic interval of the change of the initial parameters, the efficiency factors of the gas-turbine cycle and HRSG have no extreme points however the extremum of the efficiency factor of the steam-turbine cycle exists and it depends on the initial pressure values. This means that the extremum study for the equation (13) should be carried out only based on the initial pressure p_0 and, at the same time, the participant which depends on p_0 and which has the extremum should be singled out in the equation. In our case, such participant is $\partial\eta_S/\partial p_0$ (dependence of η_S efficiency factor of the steam-turbine cycle on p_0 initial pressure in constant T_0 - Fig. 5). In this case, (13) will be simplified and will get the following form:

$$d\eta_{C(p_0)} = (1-\eta_G)\eta_{SG}\frac{\partial\eta_S}{\partial p_0}dp_0. \tag{14}$$

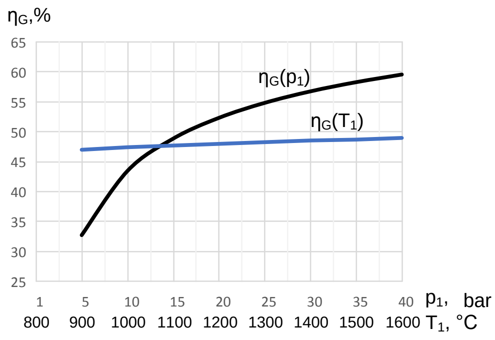


Figure 3. Dependence of thermal efficiency factor of the ideal gas-turbine cycle on initial pressure and temperature
(calculations have been made for the following conditions: environmental pressure and temperature - 1 atm, 15°C; initial parameters - p_1 =14 bar, T_1 = 1213 °C).

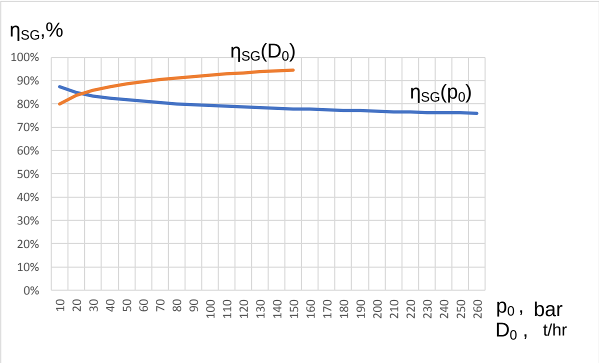


Figure 4. Dependence of efficiency factor of the HRSG on pressure and delivery of the overheated steam

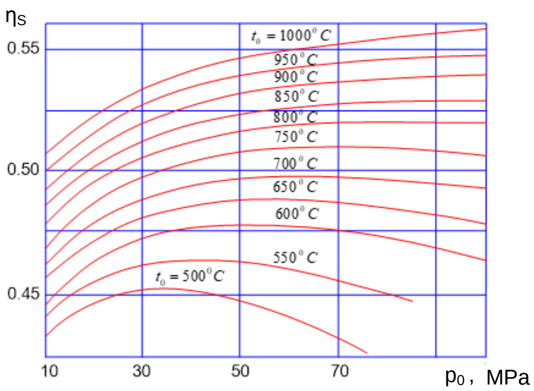


Figure 5. Dependence of thermal efficiency factor of the ideal steam-turbine cycle on the initial pressure and temperature
(calculations have been made for single steam overheating and constant pressure in the condenser).

Let's go back to the interaction of the parameters. As mentioned above, initial parameters (p_1, T_1) of the gas-turbine cycle determine p_0 and T_0 values and change intervals. Therefore, it is possible to research the equation (14) extremum through optional calculation for fixed values of the initial pressure (p_1) and temperature (T_1). Calculation results are shown in Figures 6 and 7.

As we see on Figure 6, by increasing initial pressures of the gas-turbine and steam-turbine cycles, thermal efficiency factor of the combined cycle increases more significantly in the increase of the initial pressure (p_1) of the gas-turbine cycle than the one of the steam turbine (p_0). However, by increasing the initial pressure of the gas-turbine cycle, the temperature of gases (t_{2t}) decreases to such level when it is impossible to get in the HRSG overheated steam the pressure of which will be higher than certain pressure. For example, according to the charts of Figure 6,b, when p_1 initial pressure of the gas-turbine cycle is 35 bar, the temperature of the exhaust gases is 343°C at which it is impossible to get the overheated steam of more than 102 bar pressure (point m), but when p_1=25 bar, it is possible to increase the overheated steam pressure to ~200 bar (point n). As for p_1=15 and p_1=5 bar cases, here the overheated steam pressure is in fact unlimited and for the steam-turbine cycles it is even possible to receive super critical pressure overheated steam. Regardless of such possibilities, it is clear from the charts, that the increase of the overheated steam pressure does not result in the significant increase of the thermal efficiency factor of the combined cycle, moreover, starting from certain values (points k and l), pressure increase reduces the thermal efficiency factor of the combined cycle. Therefore, in terms of thermal efficiency, key role in the combined cycle is still played by the initial pressure of the gas-turbine cycle. This circumstance is taken into the account when selecting optimum parameters in designing the units.

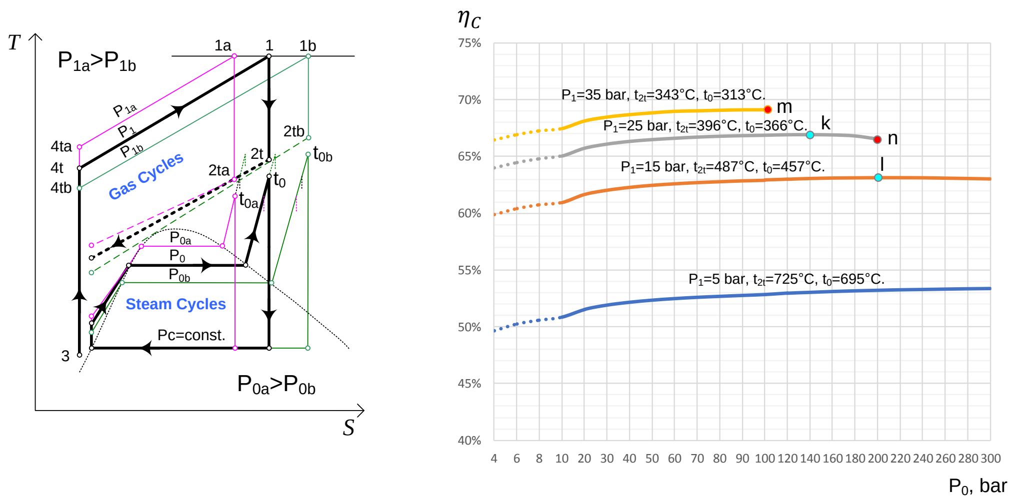
It is clear from the charts of Figure 7, that the thermal efficiency factor of the combined cycle increases by increasing initial temperature of the gas-turbine cycle (t_1). At this moment, as in the previous case, pressure values of the overheated steam are limited at low initial temperature. For example, if the initial temperature of the gas-turbine cycle is t_1=800°C, then the boundary pressure of the overheated steam is 30 bar, and if t_1=1000°C, then the boundary pressure is 160 bar. Such restrictions do not exist at higher gas temperature (t_1=1200°C/1400°C/1600°C). Due to the fact that the increase of the initial temperature of the gas-turbine cycle explicitly increases the thermal efficiency factor of the combine cycle, and that by increasing the initial pressure of the steam-turbine cycle, initially it goes up and then goes down (k_1 and l_1 maximum points), the initial gas temperature takes the leading role in terms of affecting the thermal efficiency factor of the combined cycle.

Ways of Increasing the Thermal Efficiency Factor of the Combined Cycle

Above analysis show that it is possible to increase the thermal efficiency factor of the combined gas-steam cycle through:

- Increasing initial pressure and temperature of the gas-turbine cycle;
- Increasing efficiency of the HRSG;
- Increasing initial pressure and temperature of the steam-turbine cycle.

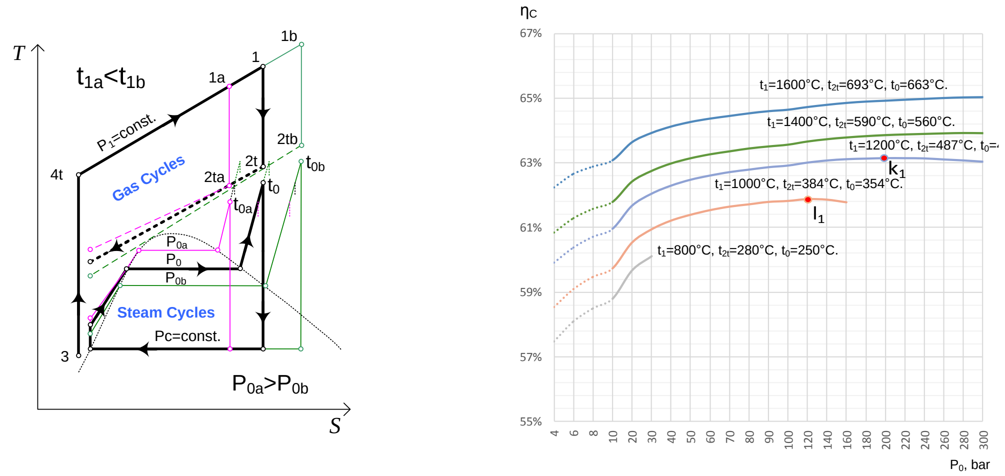
At the current stage of the technology development, it is impossible to completely realize these possibilities due to the well-known technical limitations. For instance, increase of the initial pressure of the gas-turbine cycle makes the conditions of the air compressor operation harder, increases the ability of reverse flow generation in the transition modes and makes it difficult to equalize axial forces and increases demand on the auxiliary electric power, etc. Increase of the initial temperature results in the overheating of the first stages of the gas turbine and this reduces their strength and durability. Resolution of this issue is the most complex task, and special attention is paid to it in the studies and production of units.



a) Ts chart in initial pressure change b) Thermal efficiency factor

Figure 6. Dependence of thermal efficiency factor of ideal combined cycle on the initial pressures of the gas-turbine and steam-turbine cycles: charts are made for initial gas-turbine cycle temperature - t_1=1200°C

Heat recovery steam generator operating without combustion of the additional fuel is a heat exchanger unit in which effective heat rejection from low temperature (600-750°C) exhaust gases should occur. Accomplishment of this task without using unjustified large area the heat transfer surfaces is only possible when the heat receiving operating body undergoes phase transformation during the process of the heat transfer. Besides, the lower the water pressure, the higher its hidden evaporation heat and possibility of heat rejection from the exhaust gases. However, except for positive sides, low pressure has



a) Ts chart in the change of initial temperature of the gas-turbine cycle

b) Thermal efficiency factor

Figure 7. Dependence of thermal efficiency factor of ideal combined cycle on the initial temperature of the gas-turbine cycle and initial pressure of the steam-turbine cycle: charts are made for the initial pressure - p_1=15 bar of the gas-turbine cycle

negative sides too – low pressure steam received at this moment is not optimal for conducting work in the steam turbine. Therefore, usually an interim option is chosen, in particular, the exhaust steam boiler is supplied with feeding water at two or three different pressures and quantities through two or three feeding pumps. These streams evaporate at various temperatures and accumulate in various pressure drums. From drums the steam transfers to the respective steam turbine compartments through the overheaters. According to such division the HRSG can be dual-drum (double pressure) or triple-drum (three pressure).

Due to the heat transfer related problems, the steam having supercritical parameters requiring significant overheating for reducing humidity at the final steam turbine stages is not applied in the combined cycle at this stage. It is hard to achieve this in moderate size HRSGs. Supercritical parameters also make water-chemical modes stricter and cause undesired transition processes in the water and steam tracts of the HRSGs.

Specific properties of the HRSG as well as change of humidity of the exhaust steam according to the initial parameters should be considered for optimization of the real combined-cycle units. At this time, maximal efficiency factors are achieved at the initial parameters that are different from the ideal cycles. Respectively, optimization of the control and designing should be implemented for specific requirements of the capacities and operation modes. Currently, initial pressures and temperatures of gases and steam p_1=15-20 bars, p_0=70-110 bars, t_1=1200-1400°C, t_0=550-590°C are considered to be moderate parameters of standard combined-cycle units.

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