

New equipment layouts of combined cycle power plants and their influence on the combined cycle units performance[☆]

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Abstract

New layout diagrams of combined-cycle power plants, in which some of air compressor stages are rotated not by a gas turbine but by a steam one, are considered. It's showed that if the air compression ratio in the compressor is decreased, the combined-cycle plant efficiency and output power increase; and at some compression ratio increase the combined-cycle power plant (CCPP) can operate with a fully dedicated steam-turbocompressor unit.

Keywords: combined cycle power plant, steam turbine drive of the compressor, double flow gas turbine, two shaft compressor

1. Introduction

At present, the Russian energy sector development is based mainly on the combined-cycle technologies by means of commissioning of combined-cycle power plants constructed by the same typical process flowsheet developed in the 20 century.

Its' essence is combination of a gas turbine unit and a steam turbine unit interconnected by a waste heat recovery boiler.

As a result, even at a rather low efficiency of a high-temperature gas turbine unit (40%), the total efficiency of combined-cycle power plants reaches 60% due to an intensive heat recovery of waste gases.

However, in case of CCPP large-scale construction, the following disadvantages of these power plants become apparent:

1. Low efficiency of the steam-turbine cycle due to feed water regenerative heating absence and low steam conditions limited by an exhaust gas temperature;
2. Low power efficiency factor of the gas turbine as a half of its' output power is consumed by its' auxiliaries (air compressor drive at the expense of high energy).
3. Limited output power of the CCPP on the basis of one gas turbine unit (380 MW) since last stage blades of the gas turbines work at the ultimate stress. In addition, in order to rise the CCPP output power, it's necessary to increase the number of gas turbine units and, consequently, waste heat recovery boilers, electric generators and steam turbines which scarcely lead to decrease of CCPP metal requirement with accompanying a rise in output power.

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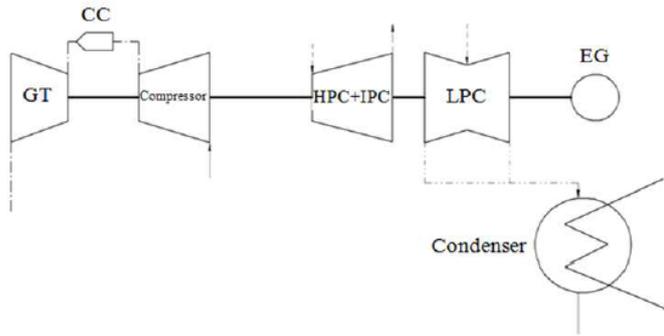


Figure 1: Equipment layout of single shaft CCPP of General Electric

4. With addition of CCPP shafts number, along with output power rise, there is increase of operating costs, the control system becomes more complex and the CCPP reliability decreases.
5. With application of the existing CCPP thermal diagram, the only way to increase the CCPP efficiency is to rise initial gas temperature at the gas turbine inlet. However, when a certain temperature level is reached (1,400–1,500°C), it's further rise is very difficult to accomplish.

It is impossible to eliminate the above disadvantages without fundamental changes in the process flow-sheets. Possible methods of increasing CCPP output power and reducing the metal requirement and the cost of power while maintaining the CCPP efficiency level are considered below.

2. Combined-cycle power plants with dedicated steam-turbocompressor unit

In existing CCPPs, additional power, relative to the gas turbine output power, is generated on the basis of a rather low-potential heat of exhaust gases. At the same time, 50% of high-potential gas energy is consumed by the air compressor drive (gas turbine unit auxiliaries). In terms of thermodynamics, the use of exhaust gases thermal energy for all or some auxiliaries is more efficient [1].

This solution can be simply implemented on the single-shaft CCPPs of General Electric. The equipment layout for these units on the single-shaft is shown in Fig. 1. In this case the conventional gas turbine is connected with the heat recovery steam turbine and both units drive the electric generator. In

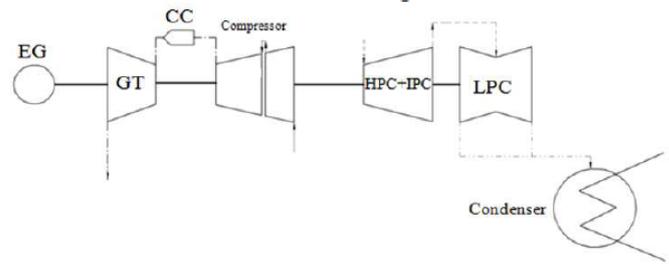


Figure 2: Thermal diagram of two-shaft CCPP with steam turbine drive of the low pressure compressor

comparison with two-shaft and three-shaft CCPPs, the single-shaft units are more space-saving, they have only one electric generator, their metal requirement and power cost decrease as the power increases, they require lower investment and maintenance costs and are more flexible and reliable.

All the mentioned advantages of the single-shaft CCPP are also maintained when all output power of the steam turbine is used for driving of a part of compressor stages. As the steam turbine power of existing CCPPs is considerably lower than the required one to power the air compressor drive, then only a part of first compressor stages may be connected with a steam turbine. Thereby we have a low-pressure compressor (LP compressor) connected with a steam turbine and a high-pressure compressor (HP compressor) connected with a gas turbine.

At introduction of changes under consideration, the single-shaft unit (the layout given in Fig. 1) is modified into the two-shaft one which has the same equipment in-line arrangement and differs from the layout shown in Figure 1 only in the electric generator location which is arranged on the gas turbine side. The thermal diagram of the two-shaft CCPP with partial compressor driving by the steam turbine is shown in Fig. 2.

Naturally, the considered changes of the thermal diagram don't influence on the thermal calculation results of the initial units. The output power and efficiency of the CCPPs are about the same.

The equipment structure is also unchanged except for the air compressor structure as it's required to divide the initial compressor into two parts and connect as much first compressor stages to the steam turbine as having the cumulative total power equal to

the steam turbine output power. The specified conditions allow us to find the pressure value P behind the LP compressor last stage and calculate the number of LP compressor stages to be connected with the steam turbine. The equivalence of the compressor and steam turbine output power result in the following apparent expression:

$$N_{ST} = N_{LPC} = G_{air} \cdot C_{p_{air}} \cdot T_{\alpha} \cdot \left(1 - \frac{1}{\varepsilon_{b1}^m}\right) \cdot \frac{1}{\eta_C} \quad (1)$$

where: G_{air} – design air flow, T_{α} – ambient air temperature, $C_{p_{air}}$ – air specific heat at constant pressure, ε_{b1} – air compression ratio in the low pressure compressor.

It's follows from the expression 1 that

$$\varepsilon_{b1} = \sqrt[m]{\frac{G_{air} \cdot C_{p_{air}} \cdot T_{\alpha}}{G_{air} \cdot C_{p_{air}} \cdot T_{\alpha} - N_{ST} \cdot \eta_C}} \quad (2)$$

In this case air compression ratio in the HP compressor is defined by the following formula:

$$\varepsilon_{b2} = \frac{\varepsilon}{\varepsilon_{b1}} \quad (3)$$

where ε is the total air compression ratio in the gas turbine unit compressor.

In comparison with conventional two-shaft layout the offered changes in the CCPP equipment layout give the following advantages:

1. At separation of some stages from the gas turbine, the starter power of the gas turbine unit is decreased by about 40%
2. Now it is possible to regulate the air flow and its' compression ratio by variation of the steam turbine rotary speed instead of air throttling at the compressor inlet section.
3. In case of rather low air flows, the low pressure compressor can be constructed as a high-speed one which leads to decrease of equipment overall dimensions and mass.
4. Use of a single generator reduces capital expenditure during construction of a CCPP.
5. Fraction of power which the gas turbine spends on the compressor drive is reduced from 50–55% to 20–25% and all the CCPP's useful electrical output is produced on the basis of the

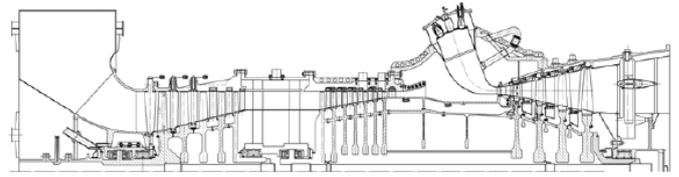


Figure 3: Longitudinal section of gas turbine unite with two shaft compressor in a single casing

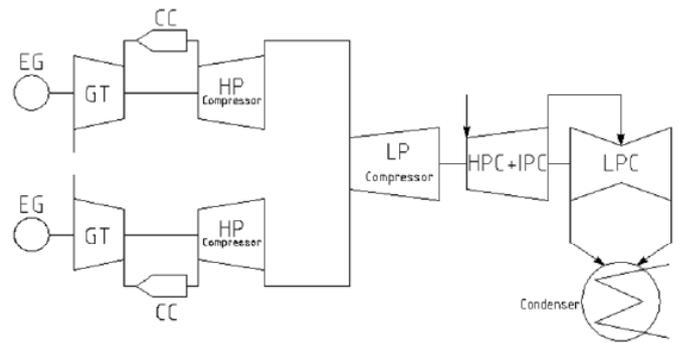


Figure 4: Equipment layout of three shaft CCPP with common low pressure compressor

high-potential energy of gases leaving a gas turbine unit combustion chamber.

In the end, within the initial basic single-shaft CCPP, a structural transition to two-shaft CCPP of the type under consideration comes down to installation of two additional bearings inside the compressor casing only since if the design turbo-compressor unit speed is not changed, then not only the total number of stages but also their profiles remain unchanged.

The longitudinal section of a gas turbine unit with two-shaft compressor in a single casing, intended for operation in a combined-cycle power plant with a partial steam turbine drive of the compressor, is given in Fig. 3.

The proposed solution can also be used for multi-shaft (particularly three-shaft) CCPPs but in this case a low pressure compressor common for all turbines is separated into a self-contained turbo-compressor unit as shown in Fig. 4.

In this layout two gas turbines are connected with their high-pressure compressors and electric generators and the steam turbine drives a low-pressure compressor common for both gas turbines. The gas turbine exhaust gases are directed either to a common waste heat boiler or to individual boilers in case of large cumulative flows of the exhaust gases.

Usage of two or more gas turbines in CCPP layouts enables us to increase their output up to 800–1,000 MW and maintain all the above-mentioned advantages of the dedicated steam-turbocompressor unit. At that, however, a serious problem of designing of a low-pressure compressor with very large air flow rates arises.

In the case under consideration, when a steam turbine remains a drive for a low-pressure compressor only, the above-mentioned problem can be solved by using a low-speed compressor. As the result, due to speed reducing, open flow area of the compressor blade rows can be increased significantly and, accordingly, high air volume rates sufficient for two-three gas turbines can be provided.

3. Combined cycle power plants with steam turbine drive of the compressor

The gas turbine exhaust heat recovery problems have been being considered very attentively since the moment when large gas turbine power plants were developed. Different thermal circuits with regenerative heating of air from compressor by exhaust gas heat were considered as the main way to increase the gas turbine unit efficiency.

Theoretical studies and numerous calculations have shown that usage of air regenerating heating leads to abrupt decrease of the optimum compression ratio $\bar{T}_C = \frac{T_C}{T_a} = 0.287$. So, for example, if at temperature ratio in the simple gas turbine cycle the maximum efficiency value is achieved at $\varepsilon \approx 30$, then at the regeneration ratio of $\sigma = 0.7$ the maximum efficiency value is not only increased by 7–8% but is achieved at $\varepsilon = 6$ [2]. So, the exhaust gas heat recovery by means of transferring a part of heat of these gases to the air supplied to the gas turbine combustion chamber is accompanied with abrupt decrease of the optimum air compression ratio in the gas turbine unit compressor.

Evidently, this principle is common for all ways of exhaust gases heat recovery. Then, this principle shall also be observed in the combined-cycle power plants in which the considered heat is intensively recovered. However, if we exclude study [3] from the consideration, this problem is not considered in the scientific publications.

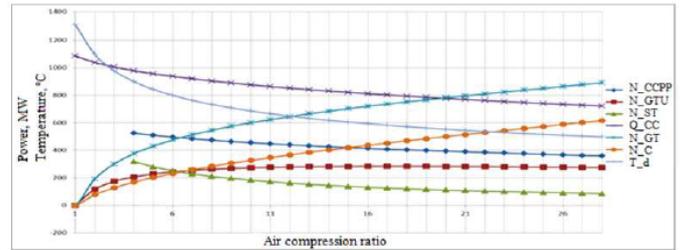


Figure 5: Diagram of CCPP parameters dependence from air compression ratio

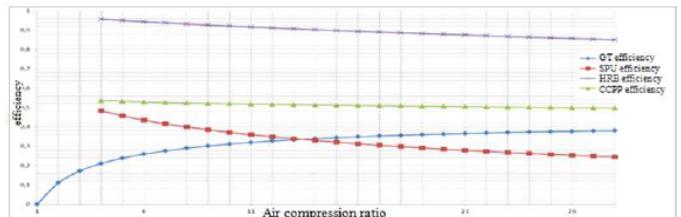


Figure 6: Diagram of CCPP and its' main components efficiency from air compression ratio

As the result, up to now all CCPPs have been built based on gas turbine units, the air compression ratio of which is determined by autonomous operation conditions. As the result, has a sufficiently higher value and varies within the range $\varepsilon = 17\text{--}25$ for the most typical gas turbine units.

When a gas turbine unit operates as a part of the CCPP the optimum air compression ratios shall be essentially lower.

Let's demonstrate this fact by an example of calculation of a simple CCPP built on the basis of gas turbine unit Siemens SGT5-4000F (V94.3A) with a double-pressure heat recovery boiler. The input data for this calculation are the ratings of the gas turbine unit [4]: $N_{GTU} = 265$ MW, $G = 656$ kg/s, $\varepsilon = 17$, $T_C = 1,315^\circ\text{C}$.

The CCPP efficiency equal to 52% obtained during calculation of economic parameters of the CCPP on the basis of this turbine are lower than the efficiency 54–55% provided in publications.

However, as this study considers mostly the tendency of CCPP efficiency ratio change due to change of the air compression ratio, the calculated difference of the efficiency values cannot change the behavior of the specified value at decrease of the initial air compression ratio in the compressor. Results of the calculations are given in Fig. 5 and 6.

Fig. 5 shows how the power of the gas turbine, compressor, heat recovery steam turbine and CCPP is changed at the change of air compression ratio ε and constant air flow through the compressor, and constant initial gas temperature. The same figure shows air compression ratio dependence of heat energy amount delivered in the combustion chamber.

Behavior of all specified dependencies is quite obvious as when air compression ratio is decreased, both power required to drive the compressor and gas turbine output power are also decreased. At the same time, when is being decreased temperature T_d is increased continuously, and at $\varepsilon < 10$ this increase is quite intensive, as well as amount of heat energy delivered to the air within the combustion chamber is increased.

Temperature T_d rise enables us to increase considerably the steam initial conditions before the steam turbine and thereby to increase its' output power and efficiency. At the same time, temperature T_d rise leads to increase of the heat recovery boiler efficiency.

Since the combined-cycle power plant efficiency is determined by the well-known expression [4]:

$$\eta_{CCPP} = \eta_{GTU} + (1 - \eta_{GTU}) \cdot \eta_{HRE} \cdot \eta_{STU} \quad (4)$$

where η_{CCPP} is combined-cycle power plant efficiency, η_{GTU} is gas turbine unit efficiency, η_{STU} is steam turbine unit efficiency, η_{HRB} is heat recovery boiler efficiency, then the cumulative economic effect depends on intensity of changing of all parameters of the expression 4.

These dependencies shown in Fig. 5 display that when ε value is decreased, the increase of steam turbine unit and waste heat recovery boiler efficiencies not only compensate the gas turbine unit efficiency decrease but also result in some increase of the CCPP efficiency. At balanced ε -value, equal to 6, when the steam turbine power becomes equal to the power required to drive the compressor, the CCPP efficiency is increased by 1.9% in comparison with the basic value. At that, as per data provided in Fig. 5, the CCPP power capacity is increased by 50 MW and makes 450 MW in comparison with basic 400 MW.

Returning to the CCPP layout with the dedicated low-pressure steam-turbocompressor unit (Fig. 4),

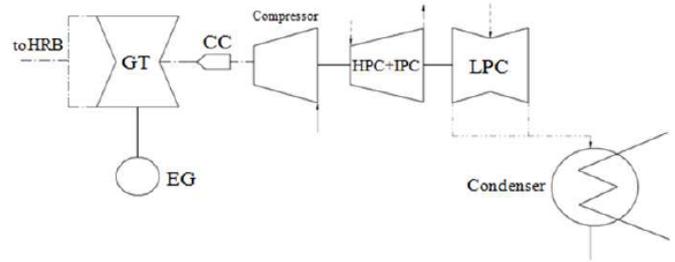


Figure 7: Equipment layout of two-shaft CCPP with double flow gas turbine

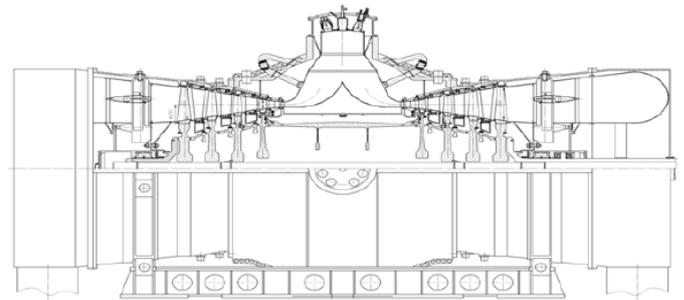


Figure 8: Longitudinal section of double flow gas turbine

we can conclude that at the balanced compression ratio $\varepsilon = 6$ the whole output power of two gas turbines is consumed completely by the electric generators drive. In this case the high pressure compressor is absent and the required for gas turbines operation air compression ratio is achieved in the dedicated low pressure compressor driven by steam turbine.

The specific amount of metal for the considered three-shaft CCPP can be reduced sufficiently if one new double-flow gas turbine is used instead of two gas turbines. In this case, the three-shaft CCPP becomes a two-shaft turbine with one more powerful electric generator and its output power is increased up to 900 MW at higher efficiency.

The layout of a new two-shaft CCPP is given in Fig. 7.

The longitudinal section of the double-flow gas turbine with output power of 900 MW and gas flow equal $G = 1,320$ kg/s is given in Fig. 8.

The design parameters of the turbine are given in Table 1.

Naturally, transition to double-flow gas turbine increases its axial dimension mainly due to the enlarged inlet and additional outlet manifold. In comparison with the basic Siemens SGT5-4000F (V94.3A) gas turbine, the total number of turbine

Table 1: Characteristics of double-flow gas turbine

Initial gas temperature at gas turbine inlet, T_C , °C	1315
Inlet pressure, P_C , bar	5.7
Gas flow, G , kg/s	1,320
Turbine output power, N_{GT} , MW	900
Compression ratio, ε	6
Gas temperature at turbine outlet, T_d , °C	763
Number of stages in one flow	4
Length without electric generator, L , mm	7,900

stages is increased by 4 stages and turbine-generator unit length is reduced due to absence of the compressor in the unit. Also, the last stage blade length is significantly reduced.

Due to transition to the double-flow turbine, the turbine thrust bearing loads are reduced at variable loads.

4. Conclusions

1. In terms of thermodynamics in the CCPP it is reasonable to use low-potential turbine exhaust gas energy for the compressor drive instead of high-potential gas turbine power.
2. When the first stages of the GTU air compressor are combined with a heat recovery steam turbine, starting unit power is decreased and there is a possibility to regulate the CCPP load by changing the steam turbine speed as well as to reduce a specific amount of metal of the turbo-compressor unit at moderate air flows due to increase of the steam turbine speed.
3. When the air compression ratio of the GTU compressor is reduced, the CCPP output power and efficiency are increased. Correspondingly, at some air compression ratio the heat recovery steam turbine output power becomes equal to power required to drive the compressor which allows to realize a complete transition to the steam turbine drive of the compressor.
4. Transition to the steam turbine drive allows us to eliminate one generator without loss of the two-shaft CCPP efficiency, significantly reduce

unit dimensions as well as to increase the total CCPP output power by 15–20% on the basis of one gas turbine unit.

5. At transition to compressor steam turbine drive it is possible, due to the steam turbine speed reduction, to built a compressor with extreme air flow rates and, correspondingly, by transiting to double-flow gas turbine, to create the two-shaft CCPP with output power 800–900 MW.

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