Using the steam turbines with bypass steam distribution at CCGT for primary frequency control

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Abstract. The system of bypass regulation of the steam turbine for the more effective participation in primary frequency control is examined. Power and relative internal efficiency ratio of turbine with bypass steam distribution for various steam consumptions are calculated. It is shown that the steam turbine with bypass steam distribution in the combined-cycle plant in the hot standby mode with reduced power (to provide power reserve) provides a higher efficiency than when the steam is throttled in the control valve without bypass steam distribution. In addition, power change of the steam turbine with bypass steam distribution provides greater acceleration compared to the operation at the sliding steam pressure and the opportunity of overloading the turbine to the required power level.

1. Introduction
The frequency of electric current is one of the indicators of the quality of electrical energy and the most important parameter of the power system regime. In accordance with the current standards, in all power companies of the Unified Power System (UPS) in Russia the frequency should be within 50.0 ± 0.2 Hz for at least 95% of the time, not exceeding the maximum permissible 50.0 ± 0.4 Hz [1].

There are three bound types of frequency regulation: primary, secondary and tertiary. Primary regulation is divided into total and rated and consists of the power plant reacting to frequency deviations in the power system (exceeding 0.002 Hz) by a proportional change in the active power. Secondary regulation is the process of changing the active capacity of power plants (specially allocated for this purpose) to compensate for the unbalanced power, to eliminate the overload of transit power lines, to restore the frequency and the external energy flows and (as a result) to restore power reserves spent in primary regulation. Tertiary regulation refers to the power change of specially allocated tertiary power plants to restore the secondary reserve and the optimal load distribution between the power system units [2].

In accordance with more stringent requirements to improve the quality of frequency regulation in the UPS in Russia, in the near future it is planned to involve all types of power units (except for power units with RBMK and BN reactors) in frequency regulation [1]. High efficiency (including partial loads) and good maneuverability make the combined-cycle plants indispensable for participation in eliminating unbalanced power and for frequency regulation.

The requirements for total primary frequency control for power units of various types, including combined-cycle plants, are determined by the regulatory document of the System Operator of the Unified Power System “Regulation of frequency and flows of active power in the UPS in Russia” [1].
power in the power system, the operation of combined-cycle plants is regulated by the standard of the System Operator “The rules of participation of combined-cycle and gas-turbine plants…” [3].

According to [3], when there is a sudden change in frequency deviation beyond the limits of “deadband” of primary regulation and it is necessary to change a primary power (for loading or unloading) of 5% of rated power or less within the control range, the main and auxiliary equipment of CCGT (GT) and technological automatic equipment should provide the dynamics of the change in primary power not worse than the following: 2.5% in 15 sec and 5% in 30 sec (i.e. 10% \( P_{\text{rated}} \)/ min).

When there is a sudden change in frequency deviation beyond the limits of “deadband” of primary regulation and it is necessary to change a primary power of more than 5%, the combined-cycle plant must give out a part of 5% of the required primary power in 30 seconds, and the remainder – with characteristics meeting the requirements for a common primary frequency control for combined cycle plants. That is, according to [1], the change in primary capacity should be 10% of the nominal power of the generating equipment in a period of no more than 2 minutes (which is 5% \( P_{\text{rated}} \)/ min).

In figure 1, based on the requirements of the System Operator for combined-cycle plants and gas turbine plants, the permissible areas of primary power variation with frequency drop and rise in the power system are shown.

![Figure 1. The permissible range for the change in the primary power of the CCGT (GT) with decreasing (a) and increasing (b) frequency.](image-url)

According to [3], to participate in automatic secondary regulation of frequency and of active power flows, the main and auxiliary equipment of the combined-cycle plant (gas turbine), its operation modes, technological automation should ensure: the power change of the combined-cycle plant (gas turbine) at a rate of up to 3% \( P_{\text{rated}} \)/ min; working out tasks that require the issuance of secondary power up to ±5% \( P_{\text{rated}} \), within the control range. In addition, the number of cycles of changes in the secondary power of the combined cycle gas turbine should not be limited, and the function of participation of the combined cycle gas turbine in the primary control should be retained while participating in automatic secondary regulation of frequency and of active power flows.

The characteristics of combined-cycle plants under variable loads are largely determined by the characteristics of the gas turbine. An important advantage of the gas turbine is its high maneuverability and wide load range. The power gas turbines are operable in any modes from idling to full load. In practice, their operating range in the combined-cycle plant is determined by economic and environmental considerations and possible restrictions of the steam circuit. Dependencies of the parameters of the gas turbine V-94.2 (in the composition of the PTU-450T North-West heat and power plant) on the load at close to the calculated external conditions (standard conditions ISO 2314: temperature of 15 °C, absolute pressure of 0.1013 MPas, relative humidity of 60%) are shown in figure 2 [4, 5].
In the high loads area, when the variable inlet guide vanes (VIGV) of the compressor is involved in regulating the power of the gas turbine, the power reduction to $0.6 \div 0.65 P_{\text{rated}}$ is accompanied by a decrease in the gas flow rate by $20 \div 25\%$ at a constant or little changing temperature behind the turbine. Due to the reduction in the degree of expansion in the turbine, proportional to the flow of gases, the temperature of the gases in front of the turbine is reduced by $110 \div 130 \degree C$. In the low loads area, the change in the gas turbine parameters is accompanied by an intense decrease in the temperature at the inlet and outlet of the turbine with a little changing gas flow rate [4, 5].

The steam part of binary combined-cycle plants is operated, as a rule, with sliding pressure of high-pressure steam, and steam turbines of such combined-cycle plants are performed without regulating stages with throttle steam distribution. An agreed reduction in temperatures and steam pressures at the inlet to the turbine avoids an increase in the relative humidity of steam at the end of expansion and risk of the blades erosion of the last stage of the turbine. In addition, the use of sliding pressure improves the reliability of the elements of the steam-water tract of the recovery boiler and the steam turbine due to the voltage reduction in their thick-walled parts with a reduction in steam pressure first at its constant temperature and then after completely covering the compressor inlet guide vanes with the steam temperature lowering [6].

To participate in the regulation of the frequency of the power system, an important characteristic of the maneuverability of a combined-cycle plant within the regulatory range of loads is the rate of load change. At that, the rates of load change of the gas and steam turbine are not the same. The dynamic characteristics of the combined-cycle plant are determined mainly by the change in the power of the gas turbine, which reaction to the fuel disturbance is fast and practically non-inertial, while the steam turbine perceives perturbations through the boiler-utilizer with some delay associated with its thermal inertia. The first reaction of the steam turbine becomes noticeable in about some seconds after the power change of the gas turbine, and the complete stabilization of the steam turbine power occurs $800 \div 900$ seconds after the onset of the disturbance [7].

When using the combined-cycle plant in the frequency control modes, it was originally intended to use only a gas turbine because of the significantly better characteristics of gas turbines maneuverability. However, they cannot fully ensure the participation of the combined-cycle plant in the modes of primary frequency control. The rate of change in the load of gas turbines is limited by severe temperature conditions and should be provided by a synchronous change in the position of the control valves of the gas turbine and the inlet guide vanes of the compressor. If movement speed of the fuel valves (as well as the steam turbine) can be quite high, the inlet guide vanes of the compressor often can not match the speed of the fuel valves. The exception is modern gas turbines equipped with an electrohydraulic drive of the inlet guide vanes. For such gas turbines, it is possible to change the load as quickly as the speed of the inlet guide vanes allows it, but still it is necessary to take into account additional dynamic temperature changes affecting the resource of the turbine metal [8].
In the PGU-45 the rate of change in the load of one gas turbine is 11 MW / min (respectively, for two – 22 MW / min), and for a steam turbine it is 5–7 MW / min due to the influence of the thermal inertia of the steam tract of the boiler-utilizer. The total rate of change in its power is 27 – 29 MW / min (6 – 6.4% \( P_{\text{rated}} / \text{min} \)). Averaging the allowable speed of load changes reduces the maneuverability of the combined-cycle plant, since at the beginning of the transient process, much higher speeds are allowed until the maximum permissible voltage in critical details is reached. After that, the load shall be increased at a rate at which these stresses remain at an acceptable level [6].

For the effective participation of steam turbines in the frequency control modes, the turbines must operate with a lowered capacity and have some reserve to open the regulating valves to increase power while reducing the frequency in the network. If the change in load of the steam installation is not related to participation in the primary frequency regulation and there are no special requirements for rapid change of the load, the change is effected only on the gas turbines, and the steam turbine with inertia of heat recovery boiler and steam turbine (with a constant time of several minutes) monitors the change of steam load of the heat recovery boiler and takes a new load [8, 9, 10].

In order to avoid the delay of the reaction of the steam tube when the frequency in the power system changes, it is possible to partially open the steam turbine control valve and, as a result, the fresh steam is throttling during normal operation, and while changing the frequency, the opening or closing of this valve. In figure 3, based on model studies, the possibility of CCGT participation in frequency regulation [10] shows the changes in the electric power of the PGU-39 power unit (as part of the GT10C gas turbine unit and the T-10/11-5.2/0.2 steam turbine unit) in a combined way in mode of sliding pressure with an increase in the load of a gas turbine by 5 MW and a change in the position (opening) of the control inlet valves of a steam turbine.

![Figure 3. The change in the electric power of the CCGT (PGU-39) with the change in the load of the gas turbine and the opening of the steam turbine control valve (according to [10]).](image)

However this method has the disadvantage of reducing the efficiency of the steam turbine under normal operating conditions due to losses from throttling in the control valves. The article discusses using of steam distribution with external steam bypass on a condensing turbine. The article also considers usage of a steam turbine with a bypass steam distribution on a condensing turbine that works in conjunction with a heat-recovery steam generator of two pressures to participate in the normalized primary frequency control. It’s assumed that initially the gas turbines operate at partial load during the power generator unit's participation in the frequency and power control. Consequently, the steam turbine also operates at partial load with a closed bypass control valve. It is possible to increase the power of the gas turbine and the steam turbine overload while reducing the frequency by increasing the flow of steam into the chamber between the second and third stages of the steam turbine. The turbine will be able to pass more steam and provide an increase in power due to the larger cross section. In the initial period, the increase in steam-production capacity will be due to the accumulated heat in the heat-recovery steam generator, and then the level of steam-production capacity will maintain due to the increased consumption of combustion products.
This method of participation in regulation is relevant for power plants where during the design and selection of the main equipment it is taken into account that at the maximum power of the gas turbine the heat-recovery steam generator must provide a steam-production capacity covering the maximum load of the steam turbine. Figure 4 shows a scheme of a steam turbine with bypass steam distribution.

![Figure 4. The scheme of steam turbine with bypass steam distribution: 1 – main control valve; 2 – bypass control valve; 3 – mixing chamber.](image)

To the first stage, water vapor is brought through the valve 1, which operates as a throttle until the pressure in front of the stage becomes equal to the pressure. Then the valve 2 starts to open. Part of the steam bypasses the first group of stages through this valve and is directed to the turbine. Opening valve 2 allows more steam to pass through the turbine, which will increase its power. The steam flow rate \( G_s \) increases as the bypass valve opens, but the pressure \( p_x \) in the overload chamber increases and the flow of steam \( G_1 \) decreases through the bypass group of stages. The arc of the ellipse \( ab \) divides the total steam pass into two streams and is plotted by calculating the relative steam flow \( G_1/G_0 \) through the first group of stages at different total steam consumption through the turbine, figure 5 [11].

![Figure 5. Steam distribution in case of bypass steam distribution (opening of the bypass valve at 0.8 \( G/G_0 \)).](image)

2. The methodology
To determine the power of the turbine, it is necessary to find out the steam parameters at the characteristic points of the expansion process. The calculation was carried out for the accepted range of the ratio of the actual steam flow through the turbine \( G \) to the maximum steam flow through the turbine \( G_0 \) from zero to one. The ratio of pressures and steam flows to the head of the turbine and to the overload chamber is determined using figure 5. Further, the flow rate and pressure into the turbine head and into the overload chamber are determined.
The efficiency ratio of the group of bypass stages was calculated by the following formula [11]:

\[
\eta_{\text{oi}} = \left( 0.925 - \frac{0.5}{G_{\text{average}}V_{\text{average}}} \right) \times \left( 1 + \frac{H_0^{\text{group}} - 600}{20000} \right) \times (1 - \varepsilon_{\text{ex.los}}),
\]

where \( G_{\text{average}} \) and \( V_{\text{average}} \) are the average steam flow through the group of stages, kg/s and the specific volume of steam, m³/kg; \( H_0^{\text{group}} \) is the available heat energy of the group of stages, kJ/kg; \( \varepsilon_{\text{ex.los}} = \frac{1}{z} \sin^2 \alpha_z \) is the coefficient of the turbine exit losses; \( z \) is the number of stages in the group.

The efficiency ratio of the group of stages of low pressure is calculated by the following formula:

\[
\eta_{\text{oi}} = 0.870 \times \left( 1 + \frac{H_0^{\text{low pres}}}{10000} \right) - \frac{\Delta H_{\text{ex.los}}}{H_0^{\text{low pres}}},
\]

where \( H_0^{\text{low pres}} \) is available heat energy of the low-pressure part, kJ/kg; \( \Delta H_{\text{ex.los}} \) is approximate turbine exit losses at the last stage, kJ/kg.

The correction for humidity is calculated by the following formula:

\[
k_{\text{humidity}} = 1 - a_{\text{humidity}} \cdot \frac{y_1 - y_2}{2},
\]

where \( y_1 \) and \( y_2 \) are humidity at the entrance to the group of stages and the exit from the group of stages, respectively; \( a_{\text{humidity}} \) is the coefficient, which is assumed to be approximately equal to 0.8 if the steam path is made with peripheral moisture removal.

Then the parameters of steam in the overload chamber were calculated after mixing two flows, one of which (\( G_1 \)) passed through the first group of stages and has the enthalpy \( h_1 \), and the other (\( G \)) goes through the bypass valve with the enthalpy \( h_0 \):

\[
h_{\text{mix}} = \frac{G_1 h_1 + G h_0}{G_1 + G}.
\]

The calculation of the group of stages between the overload chamber and the introduction of steam from the low-pressure circuit was estimated in a similar way. The saturation temperature and the pressure in the condenser are determined [12].

Then the internal power of the turbine is calculated:

\[
N = \sum G_i \cdot H_i.
\]

Average efficiency ratio of turbine:

\[
\eta_{av} = \frac{G_i H_{0-\text{x,i}} + (G_1 + G_k) H_{0-1p,i} + (G_1 + G_k + G_{lp}) H_{1p-\text{c,i}}}{G_i H_{0-\text{x,i}} + (G_1 + G_k) H_{0-1p,i} + (G_1 + G_k + G_{lp}) H_{1p-\text{c,i}}},
\]

where \( G_1 \), \( G_k \) and \( G_{lp} \) – steam flow to the head of the turbine, through the bypass and from the low-pressure circuit, respectively; \( H_{0-\text{x,i}}, H_{0-1p,i} \) and \( H_{1p-\text{c,i}} \) – available enthalpy drop between the first stage and the overload chamber, between the overload chamber and the low-pressure mixing chamber, a low-pressure mixing chamber and the end of the expansion process, respectively.

3. Results

The results of calculations for two variants of turbines based on K-110-6.5 as a component of PGU-325 if the bypass valve is opened at 0.8 and 0.9 G/Go are shown in figure 6. Figure 6 shows that during the opening of the bypass valve, the power changes almost as linearly as before opening it.
Figure 6. Dependence of turbine power variation on the relative steam flow through the turbine.

Figure 7 shows the dependence of the change in the average efficiency ratio of turbine on the steam consumption for the three variants of the steam distribution. There is a slight decrease in the efficiency ratio of turbine due to losses from the throttling of steam in the bypass valve during the opening of the bypass valve. Figure 7 shows that the operation of a steam turbine with bypass steam distribution at part load in the standby mode occurs with greater efficiency than with steam throttling in the steam turbine control valve without bypass steam distribution. The turbine efficiency with bypass steam distribution is higher by 1.4% compared to the turbine without bypass steam distribution if the control valve is opened at 0.8 G/G₀. The turbine efficiency with bypass steam distribution is higher by 0.48% if the bypass valve is switched on at load of 90%. The efficiency of turbines with bypass steam distribution is lower by 0.5% than the turbine without bypass steam distribution at 100% load, figure 7. In the steam turbine of K-110-6.5 type as part of PGU-325 it is possible to obtain an additional power of 1.35 MW at a load of 80% with a fully open steam inlet valve and a closed bypass control valve and 0.52 MW at a load of 90% in the standby mode with participation in the primary frequency control.

4. Conclusions
1. The operation of a steam turbine with bypass steam distribution at partial load occurs with a higher efficiency than with steam throttling in the steam turbine control valve without bypass steam distribution. Besides, power change of the steam turbine with bypass steam distribution provides greater acceleration compared to work in the sliding-pressure mode.

2. The available capacity increase of the steam turbine with bypass steam distribution as part of PGU-325 is 1.35 MW at a load of 80% (with a fully opened vapor inlet valve and closed bypass valve) and
0.52 MW at a load of 90% in the standby mode with participation in the primary frequency control (with other things being equal for a steam turbine based on K-110-6.5).

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