Maneuvering Heat Combined Cycle Gas Turbine Engine Unit

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***Abstract*—The prospect of the decentralized combined cycle gas turbine-combined heat and power plant (CCGT-CHPP) creation for energy supplying of the city's neighborhoods, which are located far from centralized CHPP, is described. The new type of CCGT-CHPP is suggested with waste-heat boiler with mean steam parameters and afterburner chamber placed between its evaporation stages. Fuel afterburning influence on the electric and heat power and increasing and electric energy output on the heat source increasing is shown.**

***Index Terms*—decentralized combined-cycle gas turbine combined heat and power plant (CCGT-CHPP), gas turbines, afterburning chambers, cogeneration**

I. Introduction

Economics energy efficiency of many states is in 2.5-3 times less than energy efficiency of economically developed western states. Main reasons are decrease of industrial heat loads attached to combined heat and power plant (CHPP) with decrease of the cogeneration energy output, CHPP involvement in peak loads covering with operation of cogeneration turbines in condensation regime as well as high losses during generation, transportation and separation of electric and heat energy. Besides, if capital investments are not enough, modernization and technical improvement of energy is not executed which results in both physical depreciation and obsolescence of energy equipment. Ageing of the electric stations and networks are fastened and heat economic efficiency and durability of the energy equipment are decreased. Losses of electric energy during its transformation and transportation are increased, heat losses in long-distance and aged heat networks often reach the values of 20-25%. That's why tasks of new plants construction and modernization of existing plants with use of CHP technology are of high priority. But even powerful CCGT-CHPP, which COP reaches 55%, during non-heating season have a significant condensation electric energy generation with little heat loads for hot-water supply. Mounting of the steam cogeneration turbine, which operate during non-heating season with decreased heat loads, is not economically beneficial [1], [2].

It is necessary to reach economically rational combination of centralized and distributed energetics on combined generation of electric and heat energies generation on CCGT-CHPP. Achievement of high level of cogeneration of electric and heat energy and decrease of losses during energy transportation is simplified if CCGT-CHPP are located near customers. These questions are most significant during selection of the energy sources for new neighborhoods of expanding megalopolises which as a rule are distant from city centralized CHP. Their energy supply from their own CCGT-CHPP allows providing economy of natural gas in comparison with separate energy supply of neighborhood from roof boiler-room and from external electric networks. Modular configuration of CCGT-CHPP with use of GTE with power output equal to 20-25 MW allows to use it with full or partial decentralization with decrease of costs on utility connection with centralized electric networks [3].

At most European countries the part of the electric and heat energy produced on the decentralized mini-CHPP with high level of the cogeneration is up to 15-20%. Governments pass the laws and tax deductions, which stimulate wide inception of the mini-CHPP and represents fractional government funding during their construction, deduction of the ecological taxes and opportunity for supplying of the excessively produced energy to centralized electric networks [4].

Decentralized heat CCGT-CHPP must meet several requirements:

- Lesser specific capital investments than investments for powerful CHPP;
- Have high economic efficiency and be fully-variable with good adaptation to change of the heat load of heating and hot water supplying of customers;
- Have high level of cogeneration;
- To use preferably native energetic equipment.

Fully-variability of these plants can be achieved by using fuel afterburning on the waste-heat boilers as well as regulated guide vanes in gas turbine engines unit. It was found, that for heat CCGT-CHPP the most effective way is application of the negative pressure heat steam turbines with heating of the water from water facility in the steam network heaters. Rejection of the condensers does not require application of the complex circulation refrigeration system [5].

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II. CURRENT STATUS OF CCGT-CHPP

Among the heat CCGT-CHPP, which are used in different countries, the most effective CCGT-CHPP is Nossener Bruke, which has the high level of the cogeneration energy production and provide heat- and energy supplying of the Dresden (Germany). It consists of 3 gas turbine units V 64.3 with one-circuit waste-heat boilers of high pressure, negative pressure heat steam turbine, steam and gas-water heaters for the water from the water facility. Waste-heat boilers are equipped by two chambers for fuel afterburning. The first of them is mounted before the vapor superheater of the boiler and serves for keeping the vapor superheating temperature constant before the steam turbine. The second afterburning chamber is mounted after the boiler in the gas duct and is used during the heating season for increasing of the gas-water heater heat power. The burning of the fuel in afterburning chamber along with GTU guide vanes regulation allows increasing of its maneuvering ability and economic efficiency during the operation in the heat and non-heat seasons. The disadvantage of this CCGT-CHPP is connected with the fact that during the heat season operation regime for this plant the decreasing of the relative electric power output on the heat source is typical.

III. WAY TO INCREASE THE EFFICIENCY OF CCGT-CHPP

By our opinion, in the heat CCGT-CHPP production of the vapor with mean parameters with using the fuel afterburning between evaporation stages during the heat season is appropriate.

The influence of the produced vapor pressure on the heat absorption in the evaporator of the waste-heat boiler is shown on the Fig. 1.

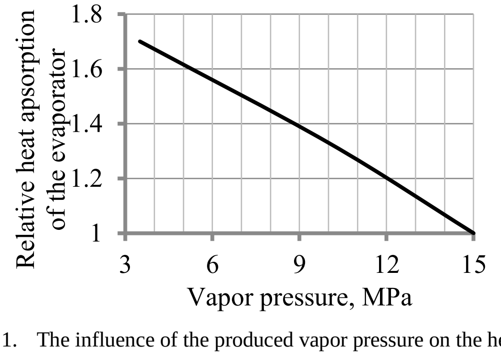

Figure 1. The influence of the produced vapor pressure on the heat absorption in the evaporator.

If vapor pressure is decreased from 15 MPa to 3,5 MPa, heat absorption of the evaporator is increased by 70%. Consequently, if CCGT-CHPP has mean vapor parameters and there are two stages of the evaporator with the second afterburning chamber AC2 placed between them, vapor output can be significantly increased by additional burning of the fuel between the evaporator stages. The new type of the modular decentralized heat CCGT-CHPP 1 is suggested by using these technical decisions.

Waste-heat boilers with two afterburning chambers (AC1 and AC2) are mounted in the exhaust gas ducts of the gas turbine unit and negative pressure heat steam turbine with network switches is used just like in the CCGT-CHPP Nossener Bruke. In contrast to Nossener Bruke, the overheated vapor with mean parameters with pressure 3,5 MPa and temperature 435˚C is produced in the waste-heat boiler. The boiler evaporator has two stages with second afterburning chamber AC2 placed between them while in the Nossener Bruke AC2 is mounted after the waste-heat boiler before the gas-water heater of the water from the facility.

During the CCGT-CHPP1 operation in the heat season regime with burning fuel in the AC2, waste-heat boiler vapor output, electric power output of the steam turbine unit and heat power of the network heaters are increased. During the atmospheric air temperature change, either electric or heat power of the plant can be regulated by changing the flow rate of the fuel burned in the AC2. It is important that during the CCGT-CHPP1 operation in the heat season the increasing of the electric energy production on the heat source (MW/Gcal) which is important coefficient characterizes CHPP heat efficiency [6], [7].

The principal heat scheme of the heat CCGT-CHPP1 is shown Fig. 2.

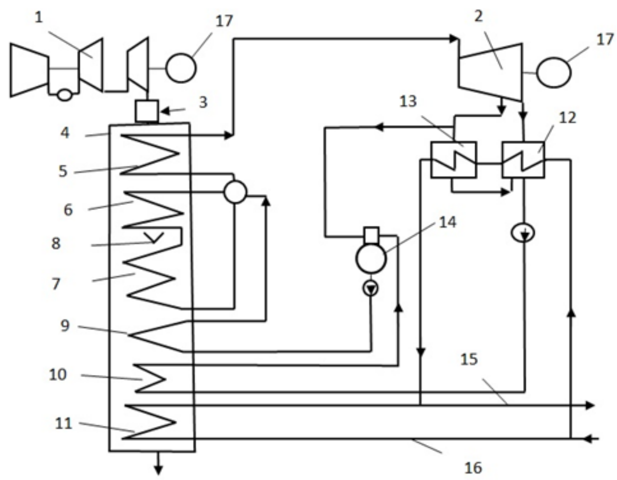

Figure 2. Heat scheme of the heat CCGT-CHPP1.

CCGT-CHPP1 includes gas turbine unit 1, negative pressure heat steam turbine 2, AC1 – 3, waste-heat boiler 4 with vapor superheater 5, evaporators of the second and the first stages 6 and 7, economizers of the second 9 and the first stages 10, gas-water heater of the network water 11, network heaters of the first 12 and second 13 stages, deaerator 14, ducts of the direct 15 and reverse 16 lines of the hot water supply system, electric generator 17.

In first afterburning chamber fuel is burned in amount of:

$$B_{AC1} = G_F \frac{h_5 - h_4}{Q_{\text{H}}^{P} \eta_{AC1} - h_5} \tag{1}$$

where G_F - gas-flow rate after gas turbine plant, h_4 - enthalpies of the gas after gas turbine plant, h_5 - enthalpies of the gas after the gas afterburner, Q_{H}^{P} -

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calorific value of natural gas, η_{AC1} - efficiency of the afterburning chamber.

Either heat or non-heat season. The increasing of the enthalpy after the vapor superheater is:

$$h_5' = h_5 - \frac{(1-\alpha_{WG})D_{WHB}(h_{VS}-h^{"})}{(G_F+B_{AC1})\eta_{VS}} \quad (2)$$

where α_{WG} -blowdown factor, D_{WHB} - steam heat boiler, h_{vs} - enthalpy of superheated steam, $h^{"}$ - enthalpy of dry saturated steam, η_{vs} – efficiency of superheater.

The vapor output in the waste heat boiler is

$$D_{WHB} = (G_F + B_{AC1})\frac{h_5'-h_8}{h^{"}-h'}\eta_{WHB} \quad (3)$$

where h_8 - enthalpy after first stage evaporator, h' - enthalpy of boiling water, η_{WHB} - efficiency of heat boiler.

During the heat season, flow rate in the AC2 is calculated by

$$B_{AC2} = (G_F + B_{AC1})\frac{h_7-h_6}{Q_H^P\eta_{AC2}-h_7} \quad (4)$$

where h_6 - enthalpy after second stage of the evaporator, h_7 - enthalpy after second afterburning chamber, η_{AC2} – efficiency of second afterburning chamber.

Due to additional fuel burning in AC2 and temperature t_7 increasing before the first stage of the evaporator either heat load on its first stage or heat load of the both stages are increasing that in turn causes increasing of the waste-heat boiler vapor output

$$D_{WHB} = \frac{[(G_F'+B_{AC1})(h_5-h_8)+(G_F+B_{AC1}+B_{AC1})(h_7-h_6)]\eta_{WHB}}{(h^{"}-h')+(1-\alpha_{WG})(h_{VS}-h^{"})} \quad (5)$$

where G_F' is gas flow rate after the gas turbine during the heat season operation regime.

From the given dependencies it is followed that superheated vapor flow rate in the waste heat boiler depend on the fuel flow rate in the AC2 B_{AC2}, gas temperature t_7 before the first stage of the evaporator, and also on the value of minimal temperature difference $\Delta t_{\min}$ on the gas exit from the second stage of the evaporator and on the value of heat absorption in the evaporator first stage Δq_{ev1}.

During the heat season regime the heat load of the CCGT-CHPP1 is regulated correspondingly with the temperature of the environment and temperature chart of the hot water supply system. For this purpose when the environment temperature t_{EN} is decreased, fuel flow rate in the AC2 is increased, gas temperature t_7 before the first stage, heat load of the first stage and entire evaporator are increased. To determine a maximal vapor output of the waste heat boiler, power capacity of the negative pressure steam turbine and heat capacity of the network heaters, we should choose the optimal values of the relative and exact values of the heat absorption λ in the first stage of the evaporator Δq_{EV1} and also the total heat absorption Δq_{EV} in both stages of the evaporator.

$$\lambda = \frac{\Delta q_{EV1}}{q_{EV}} = \frac{(G_F'+B_{AC1})(h_7-h_8)}{D_{WHB}^{HS}q_{EV}}\left\{1+\frac{h_7-h_6}{Q_H^P\eta_{AC2}-h_7}\right\}\eta_{EV1} \quad (6)$$

where $G_F^{'}$ is gas flow rate after the gas turbine during the heat season operation regime.

In Russia CCGT are built commonly with using import gas turbines with power capacity 75-150 MW. In the same time, machinery companies in the Moscow, Samara, Perm and Ufa produce aeroderivative gas turbine units. The most competitive one among them is NK-37 gas turbine unit, which have power capacity 25 MW and COP 36,4% and is produced by the publicly-traded corporation “Kuznetsov” [8], [9]. The four of such gas turbine units were used for CHPP modernization in Samara, Kazan and Lida (Belarus) [10]. Suggested modular CCGT-CHPP1, applied for heat and electric supplying of metropolises neighborhoods and large communities, equipped by aeroderivative gas turbine unit NK-37, steam waste-heat boiler with mean vapor parameters, two afterburning chambers AC1 and AC2 and gas-water heater of the water from the facility, negative pressure heat steam turbine and network heaters NH1 and NH2. Additional fuel is burnt in AC1 either during the heat season or during non-heat season for keeping the required vapor superheating temperature constant. The gas temperature before the vapor superheater is equal to 500°C. The influence of the relative specific heat load λ of the first stage on the minimal temperature difference with the temperature of the environment t_{EN}=-6°C and the gas temperature after AC2 t_7= 440°C is presented of Fig. 3 (a).

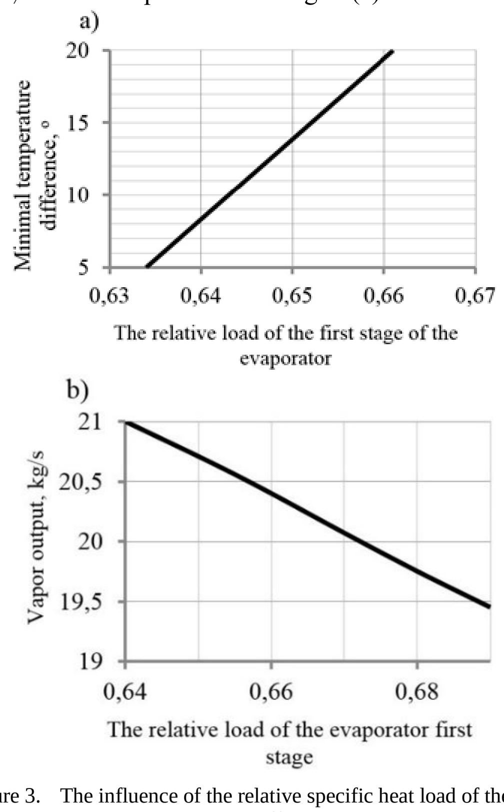

Figure 3. The influence of the relative specific heat load of the first stage on the minimal temperature difference (a) and waste-heat boiler vapor output (b).

The curve of the Fig. 4 (b) shows the dependency of the waste-heat boiler vapor output DWHB on the value of first stage relative load λ.

For comparison of the two types of the heat CCGT-CHPP which contain 25 MW aeroderivative gas turbine units, waste-heat boilers, negative pressure heat steam turbines of mean parameters (3,2 MPa, 435 ^{o}C), network and gas-water heaters of the water from the how water supply system. These types of CCGT-CHPP differ from the original one because there are two stages of the evaporator with afterburning chamber AC2 placed between them in the CCGT-CHPP1 and there is afterburning chamber AC2 in the gas duct before the gas-water heater in CCGT-CHPP2. The characteristics of these plants were determined for heat season (t_{EN} +8, -6, -15 ^{o}C, $\Delta t_{\min}$ =15 ^{o}C) with the same total heat capacity $Q_{T\Sigma}$ of the network and gas-water heaters. With these values of the t_{EN} gas temperature t_7 after AC2 in CCGT-CHPP1 is 350, 440 and 470 ^{o}C correspondingly. The vapor output in the waste-heat boilers D_{WHB} dependency on the t_{EN} is presented on Fig. 4 (a).

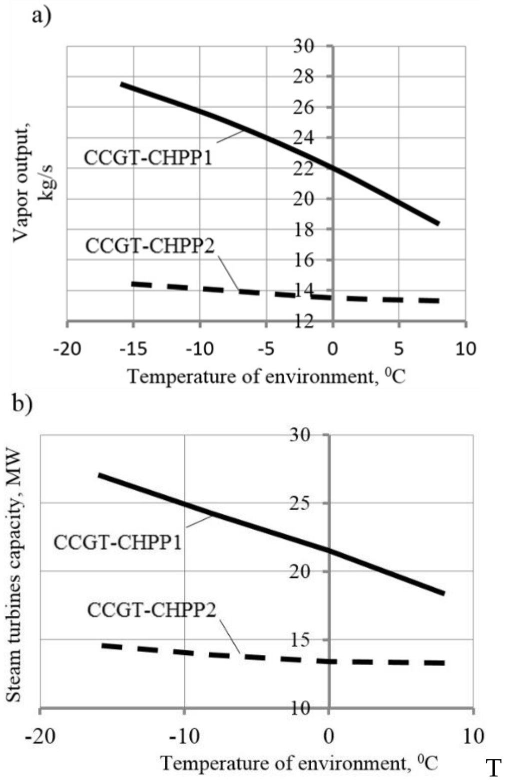

Figure 4. The vapor output in the waste-heat boilers (a) and electric power of CCGT-CHPP1 and CCGT-CHPP2 steam turbines (b) during the heat season.

If temperature of environment is decreased, vapor output in the waste-heat boiler of CCGT-CHPP2 is slightly increased which is caused by increasing of the air flow rate in the gas turbine unit.

The influence of the environment temperature on the steam turbines capacity and heat power of the network heaters of the CCGT-CHPP1 and CCGT-CHPP2 during the heat season is shown on the Fig. 4 (b) and Fig. 5 (a).

Curves presented on Fig. 5 (b) display the influence of the environment temperature during the heat season operation regime of the compared CCGT-CHPP on the most important coefficient which indicate their economic efficiency – electric power output on the heat source W (MW/Gcal). The continuous line corresponds to CCGT-CHPP1 and the dashed one corresponds to CCGT-CHPP2.

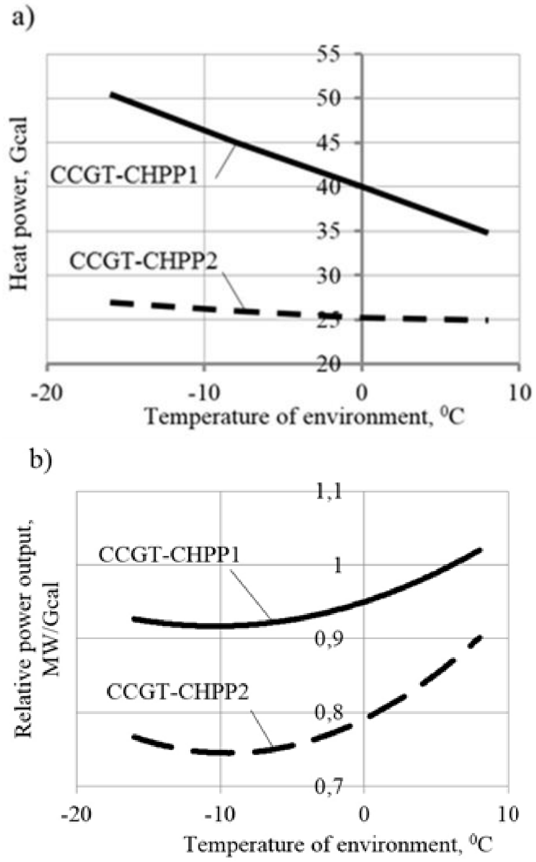

Figure 5. Increasing of the CCGT-CHPP1 and CCGT-CHPP2 network heaters heat power (a) and relative power output on the heat source W (b) during environment temperature decreasing

IV. Conclusions

Based on the above it follows that during the heat season operation regime with the same heat loads of the compared plants, cogeneration electric energy output on heat source of the CCGT-CHPP1 is more than the value of CCGT-CHPP2 in a factor of 1,2. Thus, application of the evaporator two stages with AC2 mounted between them in waste-heat boiler of CCGT-CHPP1 allows achievement of higher level of cogeneration energy output in comparison to CCGT-CHPP2 where AC2 is mounted in exit gas duct of the waste hat boiler before the gas-water heater of the water from the facility.

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