Method for determining the characteristics of a gas-steam chp with a steam turbo drive of a compressor

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Abstract. A new type of maneuverable combined-cycle plant with a steam-turbine drive of a low-pressure compressor is considered. The analysis of its technological processes in the generation of electric and thermal energy in heating and non-heating periods of the year is carried out. It is shown that in heating conditions, the burning of additional fuel in the afterburner between the stages of the evaporator of the utilizer boiler provides an increase in its steam production, maneuverability, thermal and electric power of the CCGT-CHP plant. The influence of the chemical composition and thermophysical properties of natural gas on obtaining the required excess air in the combustion chamber of a combined cycle gas turbine is considered. A mathematical model of technological processes is developed. The influence of deep utilization of the heat of the flue gases of the utilizer boiler on increasing the efficiency and improving the environmental characteristics of the gas-steam plant of a thermal power plant was recorded.
Key words: electric and thermal power, waste heat boiler, backpressure steam turbine, gas turbine installation, combined cycle gas turbine, compressor steam drive.

1. Introduction

One of the urgent problems of the energy sector is the creation of cogeneration CCGT-CHP with a high level of cogeneration generation of heat and electric energy [1]. To increase the thermal and electric power in the heating modes of their operation, in the patent [2] it was proposed to use a two-stage evaporator with a fuel afterburning chamber (*AC*) between its steps in a waste heat boiler (*HB*), to supply this *CCGT-CHP* with two counter-pressure turbines with network heaters. This allows, due to the combustion of fuel in the boiler house, to increase the production of steam in the boiler unit and significantly increase the electric and thermal power [3]. These measures contribute to improving the maneuverability of the combined cycle power plant. In the patent [4], a scheme is proposed for a gas-steam heat and power plant in which expanded steam in a counter-pressure steam turbine (*ST*) is mixed in the combustion chamber (*CC*) with the products of fuel combustion. The resulting mixture is expanded in a combined cycle gas turbine (*SGT*) driving an electric generator. The installation pro-

vides contact condensation of the vapor component of the gas-vapor mixture. The article [5] compares the characteristics of *CCGT* with steam and gas turbine drives of compressors. It was found that the steam drive helps to reduce the degree of increase in air pressure in the compressor and increase the gas temperature in front of the compressor unit. In the application for the invention [6], the authors proposed a new type of gas-steam *CCGT-CHP*, which has high mobility when working in both non-heating and heating periods of the year and increased thermal efficiency. *CCGT-CHP* (Fig. 1) contains a gas turbine unit with (gas turbine), a waste heat boiler with a two-stage evaporator and a chamber for afterburning fuel, and two counter-pressure transformer substations.

2. The principle of operation of the installation

The principle of its operation is as follows: atmospheric air is compressed in a low-pressure compressor (LPC) driven from the main counterpressure steam turbine PT1. The air is compressed in a high-pressure compressor (HPC) and then fed into the combustion chamber (CC). To increase the proportion of steam in the gas-vapor mixture, the combustion process in the CC is carried out at a small coefficient of excess air $\alpha \approx 1.2$. This ratio of fuel and air will ensure complete combustion at a low air flow through the compressor. In the mixing chamber CC serves steam expanded in PT1.
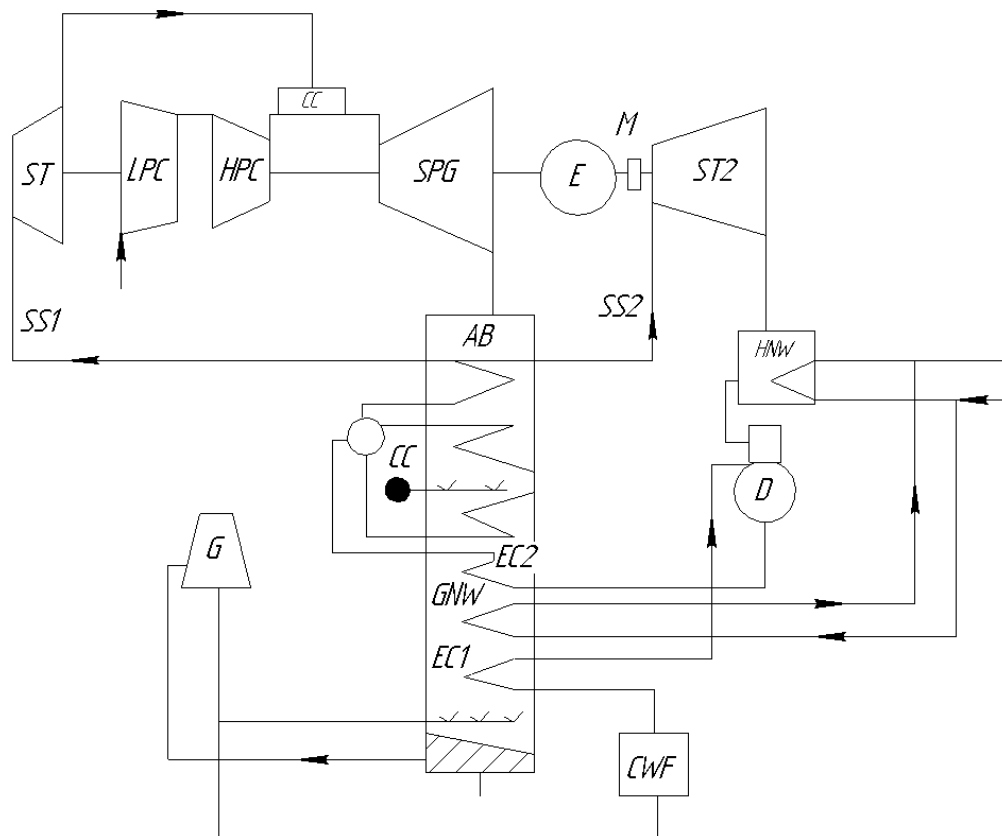

Figure 1. Schematic diagram of maneuverable gas and steam combined heat and power plants with steam compressor (*ST* - counterpressure steam turbine LPC drive, *ST2* - energy counterpressure steam turbine, *LPC* - low pressure compressor, *HPC* - high pressure compressor, *CC* - combustion chamber, *SGT* - gas and steam turbine, *E* - electric generator with flexible coupler, *SS1* - steam supply steam line to the main anti-pressure *ST*, *AB* - waste heat boiler, *SS2* - steam supply pipe for energy *ST2*, *HNW* - mains water heater, *CC* - afterburner with fan, *G* - cooling tower, *ECI* - economizer of the first stage, *EC2* - economizer of the second steps, *C* - steam condensate separator, *CWF* - chemical water treatment, *GWH* - gas-water heater, *M* - electric coupler).

The resulting gas-vapor mixture is expanded in a gas-vapor turbine (*GVT*). Her work is used to drive *HPC* and an electric generator (*E*). The steam-gas mixture expanded in the PGT is sent to a waste heat boiler (*HB*), where its heat is used: for generating superheated high-pressure steam in the economizer of the second stage (*EcII*), in the evaporator and steam superheater, for heating the heating network water, in gas and water heater (*GWH*) and for heating of feed water in the economizer of the first stage (*EcI*). As in [1], to increase the maneuverability of the unit during the heating period, additional fuel is burned in the fuel boiler and the steam production in the boiler is increased. This steam is fed to *PT2* to increase the production of electricity and heat network water in (*HNW*). When the temperature of the outside air changes, the steam capacity of the *HB* is regulated by changing the fuel consumption in the boiler.

A feature of this scheme is the low-pressure switch from the steam turbine *ST1*, and the high-pressure switch from the turbine. Introducing steam into the *CC* mixing zone allows increasing the work of expanding the gas-vapor mixture and increasing its power. *HB* provides for the use of heat of condensation of water vapor from a gas-vapor mixture and the return of condensate to the cycle. Its high potential heat is used to generate steam in the *HB*, and low potential heat is used in the hot water supply to heat the network water of the heating system.

3. Materials and Methods

Natural gas (*NG*) is used as a fuel for CCGT, which differs in its chemical composition in the content of hydrocarbons and other gases depending on the field. Therefore, the *NG* has different values of the lower heat of combustion of the fuel Hu and the stoichiometric coefficient L_0. In [7, 8], an analysis was made of the influence of the composition of the working fluid on the parameters of *GTU* efficiency. Failure to take into account the composition of the fuel gas for a mixture with $\alpha=1...4$ reduces the accuracy of engineering thermodynamic calculations, then it decreases from 7% to 3% [9]. When compiling a mathematical model for calculating the parameters of combined cycle steam turbine units with *LPC* steam drives, the air humidity at the inlet to the gas turbine and the humidity of the fuel gas were taken into account. The coefficient of excess air in the compressor station is determined taking into account the properties of the components of the gas mixture: air and combustion products of gaseous fuels N_2, CO_2, O_2, H_2O, depending on the parameters of the gas turbine based on the heat balance equation:

$$\alpha = \frac{\eta_G \cdot Hu - h_{\text{г}G}^* \left(1 - \frac{1}{L_0}\right) + h_{\text{в}C}^*}{h_{\text{в}G}^* - h_{\text{в}C}^*}, \tag{1}$$

where: η_{CC} – the coefficient of completeness of fuel combustion in the CC, $h_{\text{в}G}^* = V^O C_p^{\text{в}} T_G^*$ - is the enthalpy of air at the theoretically necessary amount for combustion at a temperature T_G^* in front of the turbine, $h_{\text{г}G}^* = \left(V_{CO_2}^0 \cdot C_p^{CO_2} + V_{N_2}^0 \cdot C_p^{N_2} + V_{H_2O}^0 \cdot C_p^{H_2O}\right) T_G^*$ is the enthalpy of the products of combustion with the coefficient of excess air $\alpha=1$ and temperature T_G^*, $h_{\text{в}C}^* = V^O C_p^{\text{в}} T_C^*$ - enthalpy of air at a theoretically necessary amount at temperature T_C^*, $C_p^{CO_2}, C_p^{N_2}, C_p^{H_2O}, C_p^{\text{в}}$ - mass heat capacities of carbon dioxide, nitrogen, water vapor and air, respectively, at the temperature of the combustion products T_G^* and air at the entrance to the combustion chamber T_C^*; $V_{RO_2}^0$ - is the volume of triatomic gases of combustion products (without water vapor at $\alpha=1$), V_{H_2O} - is the volume of water vapor taking into account the moisture content in fuel and in air, V_{R_2} - is the volume of diatomic gases of combustion products (without water vapor at $\alpha=1$): $V^O = V_{RO_2}^0 + V_{R_2} + V_{H_2O}$ - is the total volume of combustion products.

The coefficient of excess air in the combustion chamber α depends on the enthalpy of gas in front of the turbine h_G and the enthalpy of air behind the compressor h_C, which is determined by the degree of increase in pressure in the compressor π_C. For the temperature range in front of the turbine from *1373* K to *1673* K, the dependence of α on the degree of pressure increase in the compressor π_C.
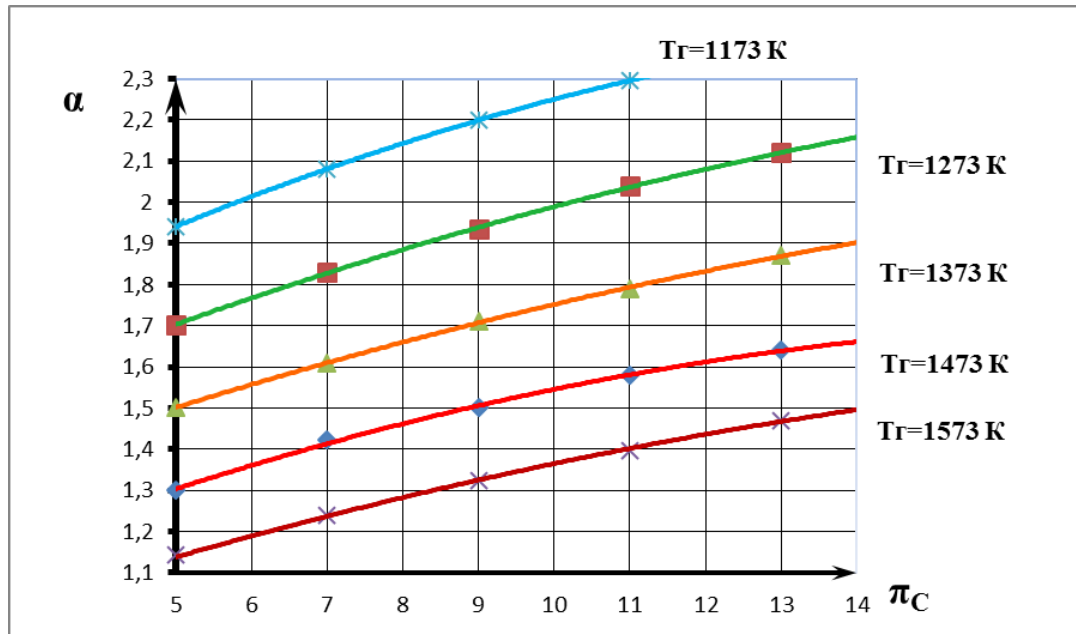

Figure 2. The dependence of α on the degree of pressure increase in the compressor π_C.

Based on the mathematical model, a graphical dependence of the coefficient of excess air in the compressor station on the parameters of the gas turbine unit was constructed (Figure 2), which allows one to estimate the range of the choice of the degree of increase in pressure π_C and the gas temperature in front of the turbine T_T for a mixture with an excess air coefficient $\alpha \approx 1.15...1.25$.

From the choice of the total degree of increase in pressure in the compressor $\pi_{\Sigma C} = \pi_{LPC} \cdot \pi_{HPC}$ depends on the amount of useful work of the cycle of gas turbines Le and the obtaining in the *CC* value α at the level of *1.15 .... 1.25*. The power consumed by the compressor N_C was calculated, depending on the air flow rate G_V through it at different $\pi_{\Sigma C}$. The compressor power increases with increasing $\pi_{\Sigma C}$ and the flow rate of compressed air G_V. The steam parameters at the *CC* inlet are set taking into account the steam parameters in front of typical backpressure turbines and the level of hydraulic losses in the steam path. Accepted: $R_{S.S.} = 9.0$ to *13.0 MPa*; $M_{S.} = 818$ *K*; $R_{S.S.} = 1.6$ *MPa*; $T_{S.S.} = 513$ *K*.

4. Calculation method

Steam flow through the PT is determined by a given G_V and π_{LPC} [10, 11]. In Figure 3 shows the dependence of the relative steam flow rate through PT1 on the distribution of πK over the compressor stages, where:

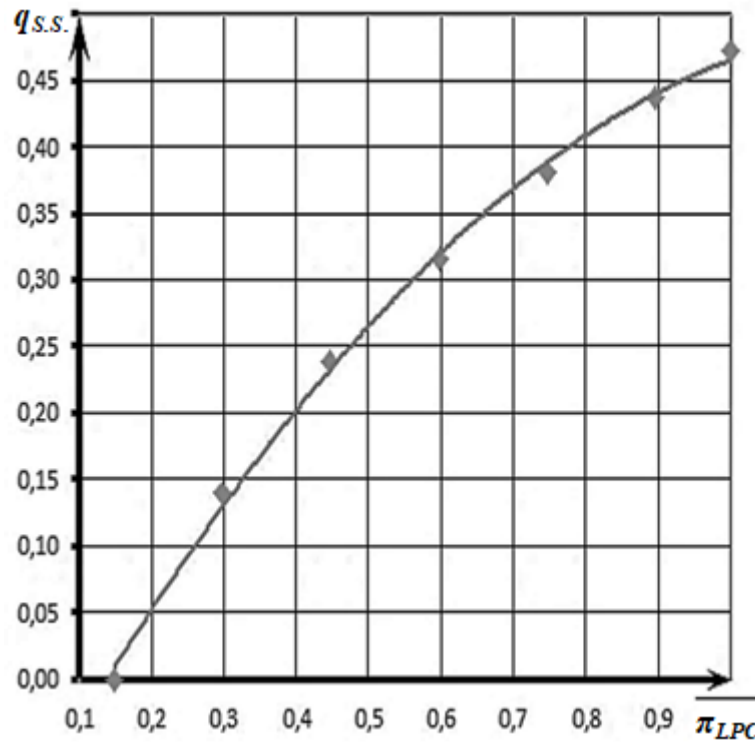$$\overline{\pi_{LPC}} = \frac{\pi_{LPC}}{\pi_{\Sigma C}} \qquad (2)$$

Figure 3. The dependence of $q_{S.S.}$ on the relative $\overline{\pi_{LPC}}$.

The use of low pressure π_{LPC} in low pressure steam pumps allows to reduce the consumption of superheated steam and to reduce the power of the drive *PTl* and the steam productivity of *HB* in nonheating operating modes of the *CCGT-CHP*. The characteristics of the gas-vapor mixture in the *SGT* and in the *HB* — the flow rate, pressure, and enthalpies of the mixture were determined taking into account the shares of the products of combustion and steam [12]. The electric power of the gas-steam *CCGT-CHP* in non-heating modes of its operation is determined:

$$N_{E\Sigma} = G_V \left[\left(1 - q_C + q_T + q_{ST}\right) L_{SGT} \cdot \eta_M - L_{HPC} \right] \tag{3}$$

where: L_{SGT} is the specific operation of a combined cycle gas turbine, L_{HPC} is the specific operation of *HPC*, η_M is the mechanical efficiency, G_V is the mass flow rate of gas turbine units, q_i is the relative flow rate of the working fluid (cooling air, fuel, saturated steam). Enthalpy of gas-vapor mixture at the outlet of the steam generator *HB*:

$$h_{outAB} = \frac{\left(1 - q_C + q_T\right) h_{SGT}^G - q_{ST} \cdot h_{SGT}^P}{\left(1 - q_C + q_T + q_{ST}\right)} \tag{4}$$

where: h_{SGT}^G - is the enthalpy of the vapor-gas mixture, h_{SGT}^P - s the enthalpy of the vapor-gas mixture at the outlet of the steam section of the *HB*.
Steam production *HB*:

$$G_{P\Sigma} = \frac{\left(1 - q_C + q_T\right) h_{SGT}^G - q_{NP} \cdot h_{SGT}^P}{\left(h_{outAB} - h_{PV}\right)} . \tag{5}$$

5

In heating modes, additional fuel is burned in the boiler to increase steam production in the boiler for *ST2*, its operation is used to drive the electric generator. The total electric power of *SGT* and *ST2* in heating modes of the gas-steam *CCGT-CHP*:

$$N_{E\Sigma} = G_{SGmix}L_{SGT} + G_{SS}L_{ST2} - G_V L_{HPC} ; \tag{6}$$

$$L_{ST2} = \left(h_{SS} - h_{BP}\right)\eta_{SGs} \tag{7}$$

where: L_{ST2} is the specific operation of the energy steam turbine, h_{SS} is the enthalpy of superheated steam, h_{BP} is the adiabatic back pressure enthalpy, and η_{STs} is the efficiency of the steam turbine 2. Additional electrical power of the installation due to expansion of steam in *ST2*:

$$N_{EST2} = \left(G_{V\Sigma} - G_{ST1}\right)L_{ST2} . \tag{8}$$

The total electric power of gas-steam CCGT-CHP in heating modes:

$$N_{E\Sigma} = G_V\left[\left(1 - q_C + q_T + q_{SS}\right)L_{SGT} \cdot \eta_M - L_{HPC}\right]. \tag{9}$$

Relative fuel consumption in AB:

$$q_{fAB} = \frac{G_{fAB}}{G_{V2}} = \frac{h_{inAB}^* - h_{outAB}^* - h_T + h_V}{\eta_{AB} \cdot H - h_T^* + h_T} , \tag{10}$$

where: h_{inAB}^*, h_{outAB}^* are the enthalpies of the mixture in front of and behind the afterburner, G_{fAB} is the fuel consumption in AB. h_T is the enthalpy of fuel gas, h_V is the enthalpy of air.

Thermal power of the installation in heating conditions:

$$Q_\Sigma = Q_{NW} - Q_{GWH} , \tag{11}$$

$$Q_{NW} = G_{P\Sigma} \cdot r , \tag{12}$$

$$Q_{GWH} = \left(G_{P\Sigma} + G_{AB}\right)\left(h_{2HB} - h_{CU}\right)\eta_{GWH} \tag{13}$$

where: Q_{NW} - heat generated in the *HNW*; - specific heat of vaporization, Q_{GWH} - thermal energy of G_{WH}.

Using the proposed mathematical model, an analysis of the characteristics of the gas-steam CCGT-CHPP was carried out during its operation in the non-heating and heating periods of the year according to recommendations [14]. In the calculations, the following were taken: air flow through compressors G_A=25 kg/s, the degree of increase in pressure in the low-pressure switch and in the high-pressure switch $\pi_{\Sigma C}$=6.4; excess air coefficient $\alpha = 1.2$. The temperature of the mixture before *SGT* T_{SG}=1100K. The parameters of superheated steam produced in the *HB* - 13 MPa, 540 [11]. The regulation of the characteristics of the units is carried out in accordance with the operating conditions of the CCGT unit by changing the fuel consumption in *CC* and *AB*. In fig. Figures 3 and 4 show the effect of fuel consumption in *AB* on *HB* steam production and on the generation of additional electric power in ST2. The use of the afterburner with an additional fan gives a double increase in the electric capacity of the CCGT unit.

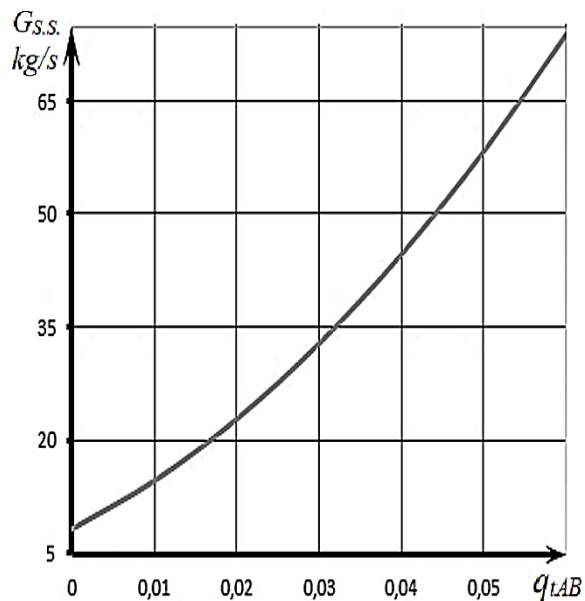**Figure 4.** The dependence of the *HB* steam productivity on the relative consumption of fuel gas in the gas compressor $G_{S.S.}=f(q_{tAB})$
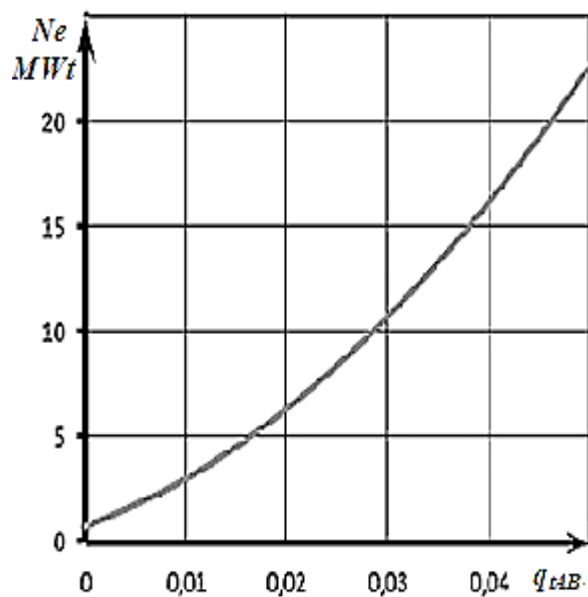
Figure 5. The dependence of the additional electric power N_E on the relative fuel consumption q_{tAB} in *AB*.

5. Conclusion.

The proposed scheme of a gas-steam installation with a low-pressure steam drive with an excess air coefficient α=1.2 in the compressor station and supplying steam from the *ST1* to the mixing chamber makes it possible to reduce the metal consumption of the design of compressors and *ST1* at a low level of G_V and $\pi_{\Sigma C}$. The expansion of the gas-vapor mixture in the gas-turbine *SPT* contributes to an increase in temperature before the *CC* increases its steam production. The installation between the stages of the *KD* evaporator with a fan for supplying air to it makes it possible to increase the combined heat and power capacity of the *CCGT* unit with an increase in the indicators of joint cogeneration generation of heat and electric energy [15]. The technological scheme of the gas-steam *CCGT-CHP* is relatively simple and has lower capital costs compared to a binary cycle *CCGT*. According to recommendations [16], the introduction of steam into the gas turbine combustion chamber improves the environmental performance of the *CCGT-CHP*

References

[1]   Veselov F V, Erokhina I V, Makarova A S and Khorshev A A 2017 Comprehensive Assessment of the Effective Scope of Modernization of Thermal Power Plants to Substantiate the Rational Structure of the Generating Capacities for the Future until 2035 *Thermal Eng.* **64(3)**

[2]   Biryuk V V, Kirsanov Yu G, Livshits M Yu, Tsapkova A B and Sheludko L P 2016 *Teplofikacionnaya parogazovaya ustanovka* [Combined-cycle heat and gas installation] Patent RF No 2650232 [In Russian]

[3]   Teplov B D and Radin Yu A 2019 Improving Flexibility and Economic Efficiency of CCGT Units Operation in the Conditions of the Wholesale. Electricity Market *J. of Thermal Eng.* **66(5)** 323-30

[4]   Nikishin V A, Peshkov L I, Ryzhinsky I N and Sheludko L P 2017 *Parogazovaya ustanovka s paroturbinny`m privodom kompressora i regenerativnoj gazovoj turbinoj* [Combined-cycle plant with a steam-turbine compressor drive and a regenerative gas turbine] Patent RF No 2409746 [In Russian]

[5]   Zaryankin A E, Zaryankin V A, Mager A S and Noskova M A 2015 Parogazovy`e ustanovki s dopolnitel`ny`m e`nergeticheskim kotlom i paroturbinny`m privodom kompressora [Combined-cycle plants with an additional energy boiler and a steam-turbine compressor drive] *Ga-zoturbinny`e texnologii* **3** 40-45[In Russian]

[6]   Gulina S A, Tyan V K, Livshits M Yu and Sheludko L P 2019 *Manevrennaya gazoparovaya teploe`lektrocentral` s parovy`m privodom kompressora* [Maneuverable gas-steam heat and power plant with a steam compressor drive] Zayavka na izobretenie RF No 2019127680 [In Russian]

[7]   Avdeev K A, Agureev I E, Elagin M Y and Hmelev R N 2017 Determination of thermodynamic and thermophysical properties of alter fuels *J. of Eng. Physics and Thermophysics* **90(6)**

[8]   Sheludko L P, Gulina S A and Biruk V V 2018 Efficiency increase of the technologic electric plant on oil-pumping stations of the main oil pipe line *J. of Physics: Conf. Series* **1111(1)** 012064

[9]   Gulina S A and Sheludko L P 2016 Modelirovanie termodinamicheskogo cikla GTD, rabotayushhego na gazoobraznom toplive proizvol`nogo sostava [Modeling the thermodynamic cycle of a gas turbine engine running on gaseous fuel of arbitrary composition] *Sbornik trudov XXIX mezhdunarodnoj nauchnoj konferencii «Matematicheskie metody` v texnike i texnologiyax MMTT – 29»* Vol 12 (Saratov) pp 67-73 [In Russian]

[10]   Zamaleev M M, Gubin I V, Sharapov V I and Bushuev E N 2018 Calculation methods of power efficiency of combined heat and power plant at change of equipment operating modes and thermal schemes *J. of Physics: Conf. Series* **1111(1)** 012039

[11]   Astvatsaturova A A, Zorin V M and Trukhnii A D 2015 Assessment of steam work efficiency as applied to a turbine being designed *J. of Thermal Eng.* **62(1)** 26-33

[12]   Biryuk V V, Shimanov A A, Larin E A, Livshits M Y and Sheludko L P 2018 Unitized Power and Heat Generating Steam–Gas Plant *J. of Eng. Physics and Thermophysics* **91(4)** 1029-37

[13]   Shase D L 2001 Combined-Cycle Development Evolution and Future GER-4206 (GE Power Systems)   URL   https://www.ge.com/content/dam/gepower-pgdp/global/en_US/documents/technical/ger/ger-4206-combined-cycle-development-evolution-future.pdf

[14]   Ankudinova M S, Larin E A and Sandalova L A 2018 Influence of ambiguity of reliability indices information on configuration shaping of combined cycle plant *J. of Physics: Conf. Series* **1111(1)** 012011

[15]   Mikhailov V E, Khomenok L A and Kovalev I A 2018 Development of High-Powered Steam Turbines by OAO NPO Central Research and Design Institute for Boilers and Turbines *Thermal Engineering* **65** 1-10

[16]   De D K, IkonoU I and Akinmeji S O 2018 Health, Environmental Effects, the Control of Emission from Power Plants and the Need for a New Emission Capture Technology *J. of Physics: Conf. Series* **1072(1)** 012019