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ARTICLE

Investigation of Combine Cycle Power Plants with Low-Grade Heat Utilization Working Methane-Hydrogen Mixtures

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ABSTRACT

The Russian energy sector remains heavily reliant on thermal power plants, with gas generation accounting for approximately 66% of the installed capacity. However, the industry faces challenges such as depletion of reserves, rising prices for hydrocarbons, and increasing concentrations of carbon dioxide in the atmosphere. This study focuses on developing new scientific and technical solutions to increase the efficiency and environmental safety of combined cycle power units. The research involves structural and parametric optimization of trinary cycle power plants operating on a methane-hydrogen mixture, as well as the development and optimization of turbine and heat exchange equipment for low-temperature power plants. The results show that the transition to trinary CCGT (Combine Cycle Gas Turbine) units with deep utilization and the use of hydrogen fuel can significantly reduce specific CO_2 emissions and increase energy efficiency up to 0.21% with also increases in capacity of turbine of approximately 17 MW. The aim of this research is to calculate the efficiency, cost effectiveness and environmental-friendly solution for power generation using mixture of hydrogen- methane as fuel in combine cycle power plant that includes ORC. Additionally, the efficiency of the organic Rankine Cycle (ORC) benefits from the increased moisture, with capacity improvements of 1–2 MW observed when the hydrogen proportion rises from 25% to 50%. Moreover, the potential for zero emissions, coupled with significant increases in power output and efficiency, underscores hydrogen's role as a pivotal component in the future of energy production.

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1. Introduction

The bulk of electricity production in Russia is carried out by burning fossil fuels. As of the end of 2022, more than 66% of the total installed capacity of power plants in the Unified Energy System (UES) of Russia belong to thermal power plants (TPPs)[1], and their share in the total volume of electricity production exceeds 60%. Despite the active development of generation based on renewable energy sources in Russia over 12 years, the share of TPPs in the total installed capacity has decreased by only 1.86%. According to the Forecast for the Development of the Energy Sector in Russia and the World, presented by ERI RAS in 2019, thermal power plants will retain a key role in providing electricity to Russia by 2040, accounting for from 62% in the conservative scenario to 55% in the low-carbon development scenario[2–4].

Gas generation accounts for approximately 50% of the installed capacity of all thermal power plants in the Russian Federation[5]. The key advantages of using gas fuel are its high combustion efficiency, relatively low number of toxic emissions and the possibility of using it in various types of power plants, including steam turbine, gas turbine and combined cycle plants, while for gas turbine and combined cycle plants the main heat input to the cycle is carried out not in the boiler, but in a less metal-intensive combustion chamber, which ensures low specific capital cost[6].

In the last decade, against the backdrop of climate change, many countries around the world have been actively implementing various organizational and economic mechanisms to reduce carbon dioxide emissions. The strategy for the socio-economic development of the Russian Federation with a low level of greenhouse gas emissions until 2050 adopted in Russia provides for a planned reduction in carbon emissions and the achievement of a balance between anthropogenic emissions and absorption of CO_2 by 2060, including through the introduction of carbon emission quotas[7]. The introduction of national and supranational carbon pricing is a challenge to the modern technological order in thermal power engineering, a response to which can be given in two ways: by further increasing the efficiency of existing generation facilities or by introducing fundamentally new low-carbon and carbon-free power plants[8–10].

Further development of gas thermal power engineering should take into account the challenges facing the industry, which include depletion of reserves and, as a result, rising prices for hydrocarbons, as well as rising concentrations of carbon dioxide in the atmosphere. Thus, it is necessary to improve the energy efficiency and environmental safety of the main generating equipment while maintaining an acceptable level of electricity prices[11].

The conversion of chemical energy of organic fuel into thermal and electrical energy is carried out mainly in installations operating on the Rankine, Brayton and Brayton-Rankine cycles. The prevalence of steam turbine, gas turbine and combined cycle technologies for the production of electrical energy is due to their advantages in the creation of high-power installations. The main sources of organic fuel in these installations are natural gas and coal, and air is the oxidizer[12].

The greatest thermal efficiency when operating on natural gas can be achieved by using combined-cycle power units, the efficiency of which reaches 60–64%, which is the best indicator among modern power plants[13]. As a result, CCGTs (Combine Cycle Gas Turbines) have the lowest specific emissions of harmful substances in comparison with other types of power plants, such as STU (Steam Turbine Unit) and GTU (Gas Turbine Unit). Over the past 10 years, intermittent renewable capacity has expanded dramatically and brought a major impact on markets, which has meant that combined cycle gas turbines (CCGTs) have had to change the way they operate[14].

Despite the fact that CCGTs are the most promising type of power plants today, their use does not allow for the transition to carbon-neutral energy. However, CCGTs have significant potential for increasing energy efficiency and environmental safety through the introduction of new circuit and technological solutions[15].

Reference[16] presents a comparison of the prospects for using ORC and hydrogen- methane mixture in CCGT cycles to recover heat from industrial gas turbines at thermal power plants. Carbon dioxide cycles can be more energy efficient, but due to the high capital costs for the construction of these plants, they are not widely used. In this regard, organic Rankine cycles are more promising for energy generation. Reference[17] considers the issue of the joint use of gas turbines and ORC for a more efficient electricity generation. The efficiency can be even higher by increasing the temperature of heat supplied for the cycle.

Reference[18] presents an analysis of a power plant operation with a gas turbine and ORC. It was found in the paper that the outdoor air parameters significantly affect the plant efficiency, primarily due to the impact on the air condenser operation.

The issue of selecting optimal schemes solutions for trinary power plants remains poorly studied today; a wide variety of low-grade heat recovery cycles necessitates their comparison for the case of their use in conjunction with CCGP plants. In this case, the selection of schemes parameters should be determined not only by the energy efficiency of power generation, but also by economic efficiency. Despite the large number of studies devoted to the use of cycles with low-boiling coolant for heat recovery, the issue of using such installations in the electric power industry, in particular in CCGT power plants, remains poorly explored. There is not enough information on the energy and economic efficiency of the trinary power plants operation; therefore, there are no recommendations on how to select flow schemes and design parameters.

This paper is devoted to the feasibility study of a regenerator flow scheme and parameters for trinary energy cycles. To determine the optimal set of a flow scheme and design parameters, the paper proposes a method of energy-economic optimization based on the thermodynamic analysis of CCGT cycles with low-temperature heat recovery units and on the cost analysis of the main equipment for a power plant. Various known low-boiling cycles are considered as plants for recovering low-temperature heat from CCGT waste gases, while suggesting recommendations for choosing a circuit and type of coolant for a trinary cycle. The flow scheme and design solutions proposed in the paper will increase the energy and economic performance of combined cycle power plants.

2. Methodology

2.1. Nuclear Power Plant (NPP) to Produce Hydrogen

The use of a nuclear power plant to produce hydrogen (**Figure 1**) involves using the electricity generated by the plant during periods of low electrical demand to produce hydrogen. Hydrogen can be produced by electrolysis of water, in which an electric current is passed through water, breaking it down into hydrogen and oxygen. This process can be carried out efficiently using excess electricity, which is needed to convert water into hydrogen[19].

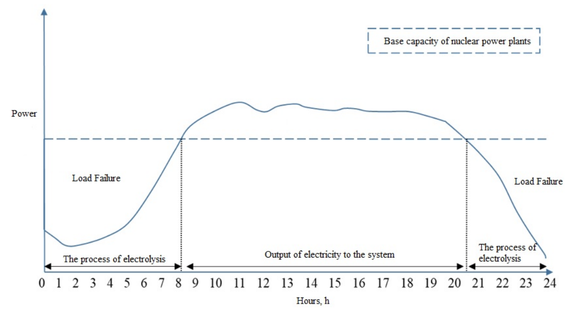


Figure 1. Concept of hydrogen production at a nuclear power plant using electrolysis during hours of power consumption failure.

Within the framework of this work, a concept will be considered which assumes joint operation of a CCGT on a mixture of natural gas with hydrogen, which is produced using "yellow" and "green" technologies. It is assumed that hydrogen will be produced using electricity obtained from renewable energy sources, as well as from nuclear power plants during periods of consumption load failure, with subsequent mixing of natural gas into pipelines and supply to the CCGT. Such a solution will increase the share of nuclear power plants in the electricity generation system together with stable operation in the nominal mode and will lead to a decrease in specific emissions of harmful substances into the atmosphere with a simultaneous increase in the efficiency of the CCGT[20–22].

2.2. Development of Mathematical Models and Thermodynamic Analysis of Combined Cycle Gas Turbines

The gas turbine unit operates as follows: air with an initial temperature of 15 °C and a humidity of 60%, having passed through the booster compressor, the aerodynamic resistance of which is 1 kPa, enters the compressor, in which the compression process occurs with an isentropic efficiency of 88%, after partial compression to a pressure corresponding to the pressure after the first and second stages of the gas turbine, it is partially bled for the cooling system, and the main flow is compressed and enters the combustion chamber. The flow rate between the two bled streams is distributed in a ratio of 30 to 70. Air and fuel enter the combustion chamber, the aerodynamic resistance of which is 3% of the inlet pressure, the pressure of which increases in the booster compressor from the pressure from the main gas pipeline, equal to 0.7 MPa, to a pressure that is 30% higher than the parameters in the combustion chamber, while the fuel consumption is selected based on considerations of ensuring the specified temperature of the combustion products. Pure methane is considered as a fuel, the higher and lower calorific values of which are equal to 55.5 MJ/kg and 50 MJ/kg respectively. In the case of using hydrogen fuel, it is considered that a mechanical mixture of methane and hydrogen enters the chamber. The higher calorific value of hydrogen is taken to be 142 MJ/kg, the lower - 120 MJ/kg. After the combustion chamber, the flow enters the gas turbine and expands, doing work, while the isentropic efficiency of the process is 89%. In the gas turbine, the cold air flow enters after the first and second stages, the pressure in which was calculated from the following considerations: it is assumed that the turbine uses 4 stages, and the enthalpy difference in each stage is the same. The exhaust pressure from the turbine is higher than atmospheric by the amount of aerodynamic resistance of the waste heat boiler and the outlet diffuser and is 106.7 kPa. The mechanical efficiency and the efficiency of the electric generator are taken to be 99%.

Table 1 shows the main parameters adopted in the gas turbine modeling. The parameters are taken from real gas turbines model number M501JAC unit. The parameters given in **Table 2** was adopted as a reference unit model number M701JAC.

Table 1. Main parameters of the mathematical model of the gas turbine unit.

Parameters	Values
Outside air temperature, °C	15
Outside air pressure, kPa	101.3
Air humidity, %	60
Resistance of the ACU (Air cooling unit), kPa	1
Relative internal efficiency of the compressor, %	88
Degree of pressure increase, shares	25
Distribution of refrigerant by stages, %	70:30
Fuel type	Methane
Fuel combustion heat (higher/lower), MJ/kg	55.5/50.03
Fuel pressure, MPa	0.7
Excess of fuel pressure after the fuel compressor over the pressure in the combustion chamber, %	30
Relative internal efficiency of the fuel compressor, %	88
Combustion chamber resistance, %	3
Temperature of combustion products before the nozzle apparatus, °C	1650
Relative internal efficiency of the turbine, %	88
Exhaust pressure, kPa	106.7
Mechanical efficiency, %	99
Efficiency of electric generator, %	99

Table 2. Parameters of the reference gas turbine.

Parameter	M701JAC [23]
Temperature after combustion chamber, °C	1650
Degree of pressure increase, shares	23
Exhaust gas flow rate, kg/s	865
Power, MW	448
Temperature of exhaust gases, °C	663

Total relative air flow for cooling the blades of the first two stages of the turbine ψ_{OXL} was determined using the correlation expression obtained in reference [24, 25], which relates the cooling flow rate to the temperature before the first stage shows in (Equation (1)) t_0:

$$\psi_{cold} = 0.0268 \cdot t_0 - 25.98 \tag{1}$$

Expansion and compression processes in turbomachines are calculated with constant isentropic efficiency using the equations given below.

The enthalpy at the turbine outlet was calculated using the formula (Equation (2)):

$$h_{T.OUTLET} = h_{T.INLET} - (h_{T.INLET} - h_{T.OUTLET\_T}) \cdot \eta_T, \tag{2}$$

where $h_{T.INLET}$ and $h_{T.OUTLT}$ are the mass enthalpies of the working medium at the turbine inlet and outlet, kJ/kg;

$h_{T.OUTLET\ T}$—theoretical enthalpy at the turbine outlet, kJ/kg;

η_T—internal relative efficiency of the turbine, %.

The enthalpy resulting from compression in a compressor or pump was calculated using the formula (Equation (3)):

$$h_{comp.outlet} = h_{comp.inlet} + \left(h'_{comp.outlet} - h_{comp.inlet}\right)/\eta_{comp}, \quad (3)$$

where η_{comp} is the internal relative efficiency of the compressor or pump, %.

The combustion chamber is assumed to burn stoichiometrically in air. In the case of using a methane-hydrogen mixture or pure hydrogen as fuel in addition to methane, the mass heat of combustion of which is higher, the fuel supply to the chamber was reduced to ensure the required temperature. In the case of considering a fuel mixture, two independent reactions are assumed to occur shows in (Equation (4)):

$$\begin{gathered} CH_4 + 2O_2 \rightarrow 2H_2O + CO_2 \\ 2H_2 + O_2 \rightarrow 2H_2O \end{gathered} \quad (4)$$

The heat and mass balance of the combustion chamber are determined from the ratio shows in (Equation (5)):

$$\begin{gathered} G_{CH4} \cdot \left(H_{CH4} + Q_{HHV}^{CH4}\right) + G_{H2} \cdot \left(H_{H2} + Q_{HHV}^{CH4}\right) + \\ G_{air} \cdot H_{air} = G_{consumption} \cdot H_{consumption} \\ G_{CH4} + G_{H2} + G_B = G_{consumption} \end{gathered} \quad (5)$$

where G_{CH4}, G_{H2} and G_{air}—the flow rate of methane, hydrogen and air entering the combustion chamber, kg/s;

Q_{CH4} and Q_{H2}—the higher calorific value of methane and hydrogen, respectively, kJ/kg;

H_{CH4}, H_{H2} and H_{air}—physical heat of fuel and air, kJ/kg;

$G_{consumption}$ and $H_{consumption}$—consumption and enthalpy of combustion products, kg/s and kJ/kg.

The power of a group of turbomachine stages was determined by(Equation (6)):

$$N_{gr} = G_{gr} \cdot \left(h_{outlet} - h_{inlet}\right) \quad (6)$$

where G_{gr} is the mass flow rate of the working medium in a group of stages, kg/s;

h_{inlet} and h_{outlet}—mass enthalpy at the input and output of a group of stages taking into account internal efficiencies, kJ/kg.

The capacity of the gas turbine plant was determined by (Equation (7)):

$$N_{GTU} = \left(N_T - N_{comp}\right) \cdot \eta_{mech} \cdot \eta_{EG} - N_{FC} \cdot \eta_{mech} \cdot \eta_{ED} \quad (7)$$

where N_T and N_{comp} are the turbine and compressor power, determined as the sum of the power of a group of stages, kW;

N_{FC}—power of the fuel compressor, kW;

η_{mech}, η_{EG} and —η_{ED} mechanical efficiency of the electric generator and electric drive, respectively, %.

The efficiency of the gas turbine was determined by the lower calorific value of the fuel shows in (Equation (8)):

$$\eta_{GTU} = \frac{N_{GTU}}{B_{CH4}Q_{LHV}^{CH4} + B_{H2}Q_{LHV}^{H2}} \quad (8)$$

where Q_{RP}^{CH4} and Q_{RP}^{H2} is the working net calorific value of methane and hydrogen, kJ/kg.

Exhaust gases from the GTU enter the waste heat boiler, where their residual heat is used for additional generation of electric power in the steam turbine plant. In this paper, a single-circuit waste heat boiler with a condensing steam turbine plant is considered as a basic object (**Figure 2**). The waste heat boiler consists of the following heat exchange surfaces: a superheater (SH), an evaporator (EV), an economizer (EC) and a gas condensate heater (GCH). The exhaust gases from the GTU pass these surfaces and are released into the atmosphere. The steam produced in the superheater is sent to the steam turbine, which consists of high- and low-pressure turbines. From the HPT, some of the steam enters the deaerator through extraction, and the rest does work in the LPT and enters the condenser, where it is condensed due to cooling with circulating water. The resulting condensate enters the gas turbine unit via the condensate pump, where it is heated by the heat of the exhaust gases and enters the deaerator. Since the condensate temperature is about 30 °C, to prevent corrosion of the gas turbine unit surfaces, the condensate temperature is increased to 60 °C by recirculation. In the deaerator, non-condensable gases are removed and the water is heated, after which the feed pump supplies the feed water to the economizer, where it is heated to a temperature close to the saturation temperature. Then the heated feed water is directed to the evaporator drum with natural circulation, where the separation of the steam phase occurs. Saturated water from the drum passes the evaporator surfaces and part of the working medium is converted into steam. Saturated steam at the outlet of the evaporator drum enters the superheater, and then to the turbine.

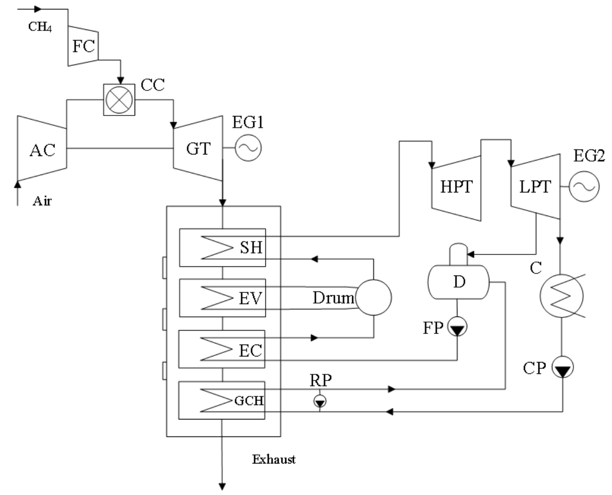


Figure 2. Scheme of a combined cycle gas turbine unit with a single-circuit waste heat boiler.

The mathematical model of the waste heat boiler and steam turbine plant was developed in Excel, where the CoolProp package was used to determine the thermophysical properties of the working media. The main parameters adopted in the modeling are given in **Table 3**.

The calculation of the thermal scheme of the combined cycle plant was carried out according to the method described in reference [24]. The work considers schemes with an initial steam temperature equal to 580, 620, 660, 700 °C, while the minimum temperature difference at the hot end of the superheater should not be less than 20 °C.

The heat balance of the heat exchange surfaces of the waste heat boiler is determined as:

$$G_g \cdot (h_g{}' - h_g{}'') = \varphi \cdot G_w \cdot (h_w{}'' - h_w{}') \tag{9}$$

where $h_g{}'$ and $h_g{}''$ is the mass enthalpy of gases at the inlet and outlet of the waste heat boiler section, kJ/kg;

$h_w{}''$ and $h_w{}'$—mass enthalpy of water/steam at the outlet and inlet of the section, kJ/kg;

G_w water/steam consumption, kJ/kg;

φ—heat conservation coefficient characterizing heat loss to the environment, %.

The relative internal efficiency of the steam turbine was determined separately for the high-pressure and low-pressure turbines using the formulas (10) & (11).

For HPT:

$$\eta_{HPT} = \left(0.92 - \frac{0.2}{D_0 \upsilon_{cp}}\right)\left(1 + \frac{H_0 - 7 \cdot 10^2}{2 \cdot 10^4}\right) \tag{10}$$

where D_0 is the steam flow rate into the turbine head, kg/s;

υ_{cp}—average specific volume of steam in the turbine, m^3/kg;

H_0—available heat drop in the turbine, kJ/kg.

For LPT:

$$\eta_{LPT} = 0.87\left(1 + \frac{H_0 - 400}{10^4}\right) K_{sh} - \frac{H_{BC}}{H_0} \tag{11}$$

where K_{sh} is the correction factor for steam humidity, defined in (Equation (12)):

$$K_{sh} = 1 - 0.4\,(1 - \beta)\,(y_0 - y_z)\,\frac{H_B}{H_0} \tag{12}$$

β—the coefficient for taking into account the influence of average humidity on the internal efficiency depending on the design of the flow part, adopted as 0.1;

y_0 and y_z—steam humidity at the beginning and at the end of the group of stages;

H_B—heat drop in the wet steam region, kJ/kg;

H_{BC}—losses with output speed, depending on the turbine design, the value of 20 kJ/kg is taken.

capacity of the steam turbine circuit of the combined cycle plant was determined by (Equation (13)):

$$N_{STU} = (N_{HPT} + N_{LPT})\,\eta_{mech} \cdot \eta_{EG} - \frac{(N_{CP} + N_{FP} + N_{RP})}{\eta_{mech} \cdot \eta_{ED}} \tag{13}$$

where N_{HPT} and N_{LPT} are the power of the high-pressure turbine and the low-pressure turbine, kW;

N_{CP}, N_{FP} and N_{RP}—power of condensation, feed and recirculation pumps, kW.

The capacity of the combined cycle plant was determined by (Equation (14)):

$$N_{CCGT} = N_{GTU} + N_{STU} \tag{14}$$

The efficiency of the combined cycle plant was determined by (Equation (15)):

$$\eta_{CCGT} = \frac{N_{STU}}{B_{CH4}Q_{LHV}^{CH4} + B_{H2}Q_{LHV}^{H2}} \tag{15}$$

Today, single-circuit waste heat boilers, due to their relatively low efficiency, are less common than double-circuit ones. Therefore, to compare the technical solutions being developed with existing installations, the paper also considers a combined-cycle plant with a double-circuit boiler.

The operating principle of the circuit is similar to the single-circuit analogue, except that the waste heat boiler has a high-pressure circuit, represented by an economizer, an evaporator and a high-pressure superheater, as well as a low-pressure circuit, the generated steam in which goes directly

Table 3. Main parameters adopted in modeling the steam turbine circuit.

Parameters	Values
Temperature of live steam, °C	540/580/620
Minimum temperature difference in the superheater, °C	20
Minimum temperature difference at the cold end of the evaporator, °C	10
Deaerator pressure, MPa	0.12/0.6
Steam pressure losses in extractions, %	5
Condensation temperature, °C	30
Relative internal efficiency of pumps, %	85
Minimum condensate temperature at the inlet to the gas processing plant, °C	60
Heating of water in deaerator, °C	15
Underheating of water in the economizer to the boiling point, °C	10
Mechanical efficiency, %	99
Efficiency of electric generator, %	99
Maximum humidity of steam after LPT, °C	10
Pressure losses along the steam-water path SH, EC, GCH, %	5
Pressure loss in the stop valve, %	3
Losses in the LPT with output speed, kJ/kg	20
Heat losses in the surfaces of the waste heat boiler, %	0.4

to the LPT, partially being taken to the deaerator. In both circuits, feed water comes from a common deaerator, which receives condensate after the GCH. Such a circuit allows for deeper cooling of the GTU exhaust gases, which ensures an increased level of efficiency of the entire cycle.

Single-circuit waste heat boilers have a high potential for increasing the thermal efficiency of the cycle. First of all, a fairly large amount of heat leaves the cycle with exhaust gases, which makes the possibility of their useful utilization promising. Secondly, a large amount of heat is dissipated in the steam turbine condenser. Its useful use is hindered by the low temperature of the dissipated heat source - to ensure maximum power of the condensing STU, the pressure in the condenser is provided quite low (the condensation temperature of 30 °C is considered in the work). The main way to reduce the heat dissipated in the condenser is to develop a regenerative heating system for feedwater in the cycle: this will reduce steam consumption in the condenser. In traditional CCGT units, regeneration in the steam turbine part is rarely used, since the heat of the exhaust gases in the GCH is used to heat the condensate. If an additional utilization cycle is built after the steam turbine circuit, removing the GCH will increase the potential of waste heat and improve the efficiency of the cycle.

Thus, the development of regeneration in a CCGT with a single-circuit boiler allows the removal of the gas turbine unit from the scheme, which will simultaneously increase the efficiency of the steam turbine part by reducing the steam consumption in the condenser and increase the heat content in the exhaust gases, which will allow it to be used more efficiently in an additional cycle.

The paper examines a CCGT system with two low-pressure mixing heaters, in which steam from the low-pressure turbine heat the condensate before the deaerator, and a high-pressure surface heater is installed between the economizer and the feed pump, where the extraction steam from the high-pressure turbine heats the feed water and partially (or completely) replaces heat absorption in the economizer (**Figure 3**). To ensure the operation of the mixing heaters, additional pumps are installed on the condensate heating line, increasing the pressure to the required extraction steam pressure.

Heating of the main condensate after the low-pressure regeneration system was performed at equal temperatures in each HPH. The extraction steam pressure was selected so that as a result of mixing the steam would be completely condensed, and the extraction flow rate determined the condensate temperature after mixing. The high-pressure heater is made with a drainage cooling section, while the extraction pressure from the HPT was selected so as to ensure condensation at a temperature 5 °C higher than the feedwater temperature, and the drainage was cooled to a temperature 5 °C higher than the feedwater temperature at the HPH inlet. A typical T-Q diagram of the HPH is shown in (**Figure 4**). The cooled drainage enters the deaerator, the extraction steam flow rate is determined to ensure a specified feedwater temperature at the economizer inlet.

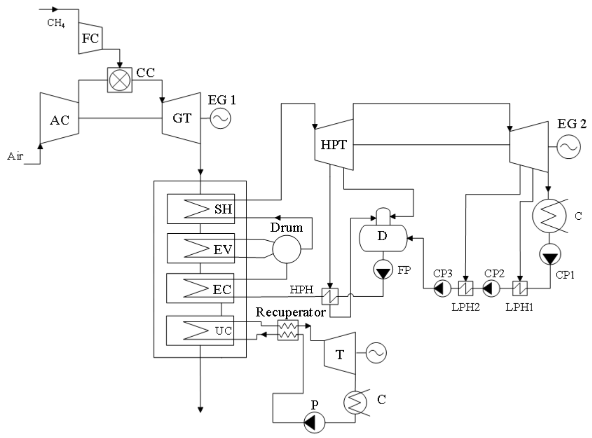


Figure 3. Scheme of a combined cycle power plant with a developed regeneration system.

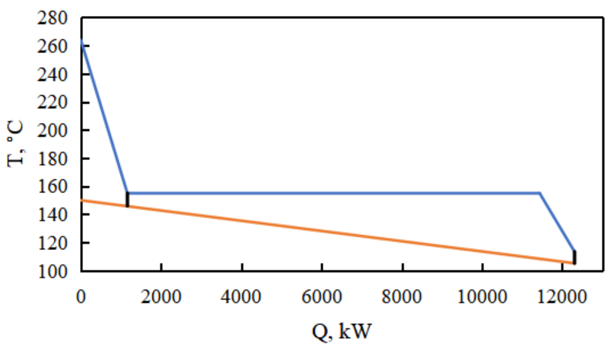


Figure 4. TQ diagram of the PVD.

As a utilization cycle or ORC cycle at a combined-cycle plant for additional generation of electric power. Since the temperature of the exhaust gases after the last surfaces of the waste-heat boiler is comparatively low (not higher than 250 °C) even despite the replacement of the GCH and EC by regeneration, efficient use of this heat is possible in cycles with a low-boiling working fluid, in particular an organic Rankine cycle is presented in **Figure 5**. The scheme of the organic Rankine cycle includes a heater, in which the heat of the exhaust gases is utilized to heat the working environment, a freon turbine, a condenser and a pump.

Today, new organic compounds are being developed for refrigeration and geothermal energy, which do not have a negative impact on the ozone layer and do not contribute to the acceleration of global warming processes. In this paper, the following organic compounds are considered as working fluids: R236ea, R123, R124, R134a, R 600a, R245fa, R22, R41, R12, R32. These freons have a low global warming potential (GWP index) and a low impact on the decomposition of the ozone layer (ODP index), and also belong to the flammability safety group A 1–A 2 (according to the ASHRAE scale)[26].

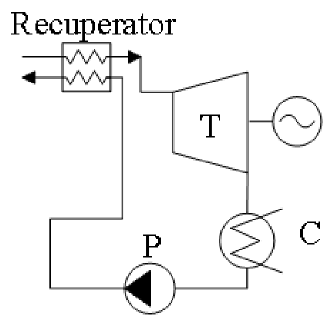


Figure 5. Schematic diagram of a simple organic Rankine cycle.

The calculation of the utilization cycle was carried out using a programming language Python, the thermophysical properties of freons were determined using the ctRefprop package. The main constants adopted in modeling organic Rankine cycles are given in **Table 4**.

In the utilizer, the exhaust gases were cooled to the minimum possible temperature determined by the temperature difference in the cycle and the freon temperature after the pump. The freon coolant flow rate was selected in such a way as to maximize the electric power of the utilization cycle while simultaneously ensuring a minimum temperature difference at the point of minimum temperature difference, as well as complete evaporation of the working medium. The expansion and compression processes in turbomachines were calculated using Equations (16) and (17). In the work, optimization of utilization cycles was carried out by the initial pressure and type of working fluid to achieve the maximum net electric power of the utilization cycle.

The net capacity of the utilization cycle was determined as:

$$N_{UC} = N_{FT} \cdot \eta_{mech} \cdot \eta_{EG} - \frac{N_{Fr.P}}{\eta_{mech} \cdot \eta_{ED}} - \frac{N_{Cr.P}}{\eta_{mech} \cdot \eta_{ED}} \quad (16)$$

where N_{FT}, $N_{Cr.P}$ and $N_{Fr.P}$ are the power of the freon turbine, circulation and freon pumps, respectively, kW.

The capacity of the power unit with a waste-to-energy cycle was determined by (Equation (17)):

$$N_{CCGT} = N_{UC} + N_{GTU} + N_{STU} \quad (17)$$

Table 4. Parameters adopted in the simulation of the recycling cycle.

Parameters	Values
Minimum temperature pressure in the waste heat recovery unit, °C	10
Relative internal efficiency of the turbine, %	85
Relative internal efficiency of the pump, %	85
Condensation temperature of the working fluid, °C	30
Heat energy losses in the waste heat recovery unit, %	1
Pressure loss in the waste heat recovery unit, %	5
Temperature head at the cold end of the waste heat recovery unit, °C	10

The sequence for a trinary unit includes: calculation of a gas turbine, calculation of a steam turbine circuit at optimal initial pressure with varying feedwater temperature, calculation of a utilization cycle for different temperatures of exhaust gases after the steam turbine unit with optimization of the initial pressure and type of coolant, determination of parameters that ensure the maximum net electrical power of the power unit.

This trinary cycle scheme allows for deep cooling of the exhaust gases, but in traditional combined cycle plants the exhaust gas temperature is usually maintained at no less than 80 °C to prevent condensation and subsequent corrosion of the heat exchange surfaces. However, in the case of using corrosion-resistant heat exchange surfaces, deeper cooling of the exhaust gases in the utilization cycle will allow the latent heat of condensation of water vapor to be used usefully.

Moisture in the exhaust gases is present due to the humidity of the source air, and is also formed as a result of fuel combustion. If there is enough moisture in the gases, the dew point may be higher than the minimum temperature of the freon in the cycle, and the useful use of the heat of the falling dew becomes possible. In this case, the additional perceived power will be higher, the higher the moisture content in the combustion products.

Thus, the concept of hydrogen energy is promising, within the framework of which one of the methods of using hydrogen is its combustion in three -phase combined cycle gas turbines, while hydrogen is supplied through separate pipelines or in the existing gas transportation system in a mixture with natural gas (**Figure 6**).

The use of a methane-hydrogen mixture in combined-cycle power units leads to an increase in the moisture content of the exhaust gases, since the only product of hydrogen combustion is water vapor. Therefore, within the framework of hydrogen energy, the concept of power units with low-temperature cycles for condensing water vapor from combustion products is becoming promising. This will not only reduce the level of harmful emissions into the atmosphere, but also increase the efficiency of electricity production.

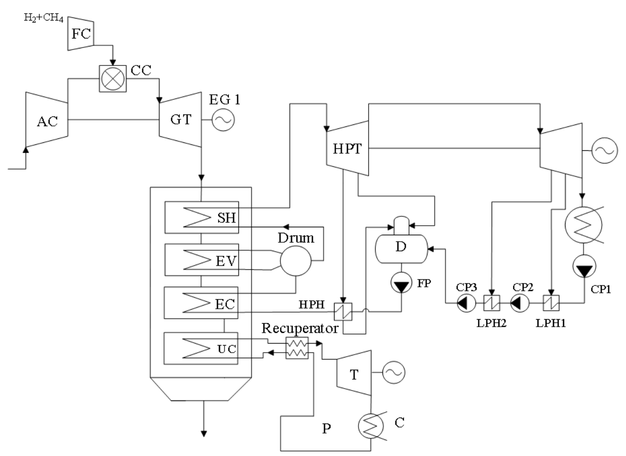


Figure 6. Schematic diagram of the thermal circuit of the studied trinary CCGT unit.

The dew point of water vapor in the exhaust gases was determined as the temperature at which steam condensation occurs at a given partial moisture pressure $t_s = f(P_B)$, depending on the pressure of the exhaust gases and the molar composition $P_B = P_{exahust.gas.} V_B$.

The additional heat from condensation of water vapor from the exhaust gases was calculated as follows. Before the dew point, the thermal power from cooling the gases was determined by (Equation (18)):

$$Q_{cold.mos} = G \cdot \bar{c_P} \cdot (t_{inlet} - t_S) \tag{18}$$

where $\bar{c_p}$ is the isobaric heat capacity of the exhaust gases at the average temperature of the gases up to the dew point, $(t_{inlet} + t_s)/2$, kJ/kg · °C;

After the dew point, the thermal power consists of (Equation (19)):

$$Q = Q_{cond} + Q_{cold.w.} + Q_{cold.mos} \tag{19}$$

where Q_{cond} is the heat generated from the condensation of water vapor, kW, defined as (Equation (20)):

$$Q_{cond} = G_{cond} \cdot r\left(P_B\right) \tag{20}$$

where G_{cond}—condensate flow rate, kg/s;

r—mass heat of condensation, determined by the partial pressure of vapor P_B, kJ/kg;

$Q_{cond.mos}$—heat from cooling condensed moisture, kW, defined as (Equation (21)):

$$Q_{cold.mos} = G_{cond} \cdot c_P \cdot \Delta t \tag{21}$$

c_p—isobaric heat capacity of water at a given temperature, kJ/kg °C;

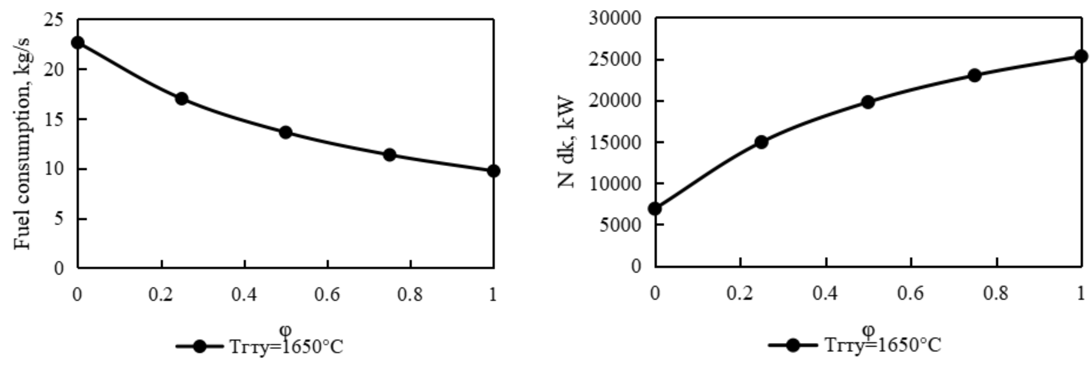
Δt—the difference between the temperature at which a given moisture flow occurs and the temperature at the cold end of the waste heat recovery unit, °C.

After cooling the gases below the dew point, it is assumed that the moisture in the combustion products is always on the saturation line and the humidity is 100%. The volume of moisture that has condensed is removed from the exhaust gases, and the partial pressure of the vapors is reduced to a level at which the moisture is on the saturation line at a given temperature.

3. Result and Discussion

An additional reduction in carbon dioxide emissions can be achieved by switching to promising hydrogen fuel. On one hand, this will reduce the use of carbon fuel. On the other hand, it will lead to a change in the energy characteristics of the thermodynamic cycle.

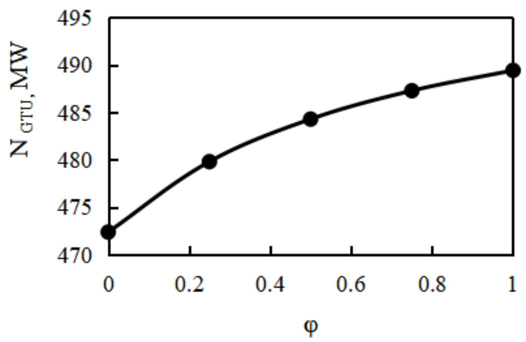
The admixture of hydrogen which gets into the fuel for the combustion chamber of a gas turbine unit at a constant temperature before the nozzle apparatus and the consumption of combustion products will lead to a change in the total fuel consumption, the total calorific value of which will be determined by the composition of the methane-hydrogen mixture, as well as to a change in the composition of the combustion products and a change in their thermophysical properties. (**Figure 7**) shows a graph of the change in the total fuel consumption and the cost of compression in the booster fuel compressor from the proportion of hydrogen in the mixture. With an increase in the proportion of hydrogen in the mixture, the total consumption decreases, since to ensure a constant temperature before the nozzle apparatus, it is necessary to supply a smaller amount of fuel, the heat of which increases with an increase in the proportion of H_2. At the same time, the costs of compression of the mixture increase significantly with an increase in the proportion of hydrogen.



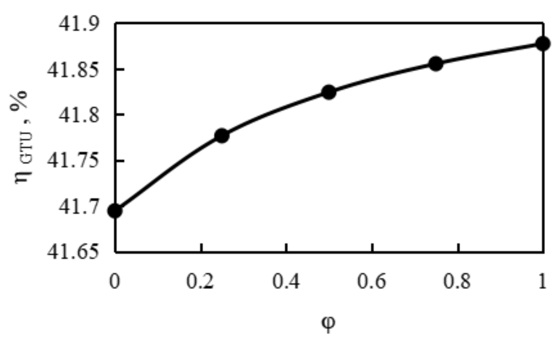
(**a**) Fuel consumption (**b**) Booster compressor power

Figure 7. Dependence of fuel consumption and costs for its compression on the proportion of hydrogen in the mixture.

The capacity of a gas turbine increases with the increase of hydrogen in the mixture. Thus, for a turbine, a complete transition to hydrogen will increase the capacity by 17.07 MW, despite the increase in the cost of fuel compression, the net capacity increases due to a change in the composition of combustion products: the content of carbon dioxide decreases and the moisture content increases, which in total leads to an increase in the total heat capacity of the mixture (**Figure 8**). The efficiency and power of the gas turbine by the lower calorific value also increases with an increase in the proportion of hydrogen in the mixture: with a complete transition to hydrogen, it increases by 0.18%.



(**a**) Power of gas turbine unit



(**b**) Efficiency of gas turbine unit

Figure 8. Dependence of the power and efficiency of the gas turbine on the proportion of hydrogen in the mixture.

The main reason for the change in the characteristics of the gas turbine is the increase in the mole fraction of moisture in the composition of the combustion products. With an increase in the proportion of hydrogen in the mixture, the moisture content increases: with a complete transition to hydrogen, the molar content of water vapor in the exhaust increases from 9.8% to 15.4%. At the same time, the exhaust temperature decreases from 655.25 °C to 651.37 °C.

As the moisture content in combustion products increases, the condensation point also increases. (**Figure 9**) shows the dependence of the dew point temperature on the proportion of hydrogen in the GTU fuel. The precipitation of condensate from the exhaust gases begins at their temperature of 45.4 °C, and when methane is replaced by pure hydrogen, 54.5 °C.

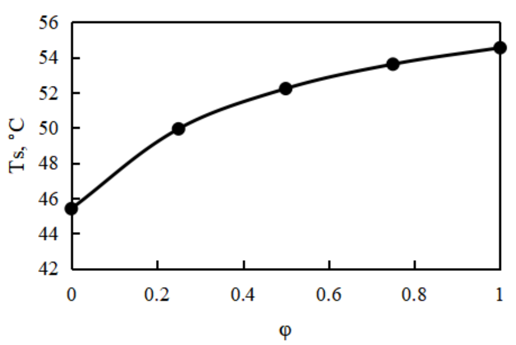


Figure 9. Dependence of dew point temperature on the proportion of hydrogen in fuel.

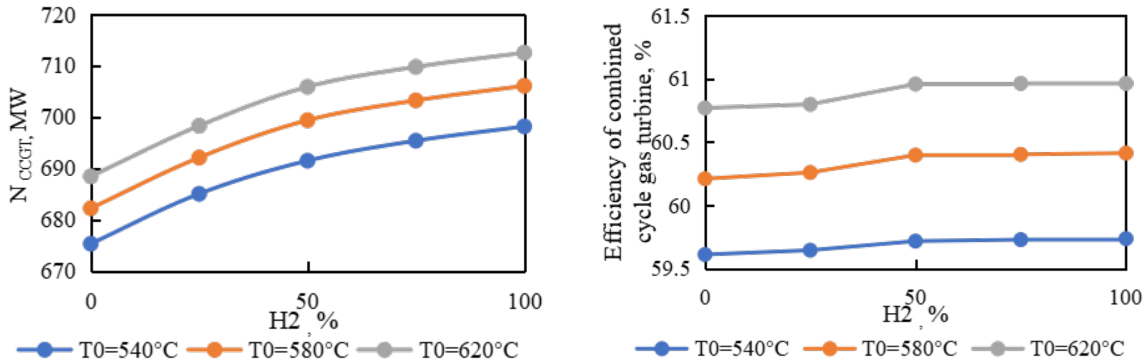
An increase in the moisture content in the combustion products increases the total heat content of the exhaust gases heating the water coolant. This leads to the fact that the capacity of the steam turbine circuit, despite the drop in gas temperature before the superheater, increases. Thus, with a complete transition to hydrogen fuel, the increase in capacity is on average 4 MW, and the change in the thermal efficiency of the cycle practically does not occur with a change in the proportion of hydrogen in the mixture.

Despite the minor change in the cycle energy efficiency, the transition to hydrogen fuel at the CCGT allows to reduce the level of carbon dioxide emissions into the atmosphere to zero. On the other hand, the increased moisture content in the gases can be usefully used in the utilization cycle, where this moisture can be condensed, and the heat hidden in it can be usefully used.

Combustion of hydrogen fuel leads to an increase in the dew point temperature, which significantly affects the parameters of the utilization cycle. At the point where condensate begins to fall out, there is a sharp increase in the heat removal required to cool the combustion products. In surface heat exchangers, this leads to the formation of a pinch point, where the temperature difference becomes minimal. To ensure complete evaporation of the freon coolant, an amount of high-temperature heat (at the freon evaporation temperature) is required equal to or even greater than in the economizer freon zone. Therefore, it is not always possible to implement the efficient use of low-temperature condensation heat. For greater utilization of low-temperature heat, it is necessary to lower the average temperature of heat supply in the utilization cycle, which can be achieved by reducing the working pressure in the cycle and, as a consequence, the boiling temperature of freon. The most efficient freon at a lower supply temperature is R 245 fa. In the work for deep utilization, the use of this coolant is considered for all considered configurations of the trinary cycle.

The dependence of the ORC power on R 245 fa on the proportion of hydrogen in the fuel for a gas turbine with T0 = 1650 °C. With an increase in the proportion, the capacity of the utilization cycle grows, with the greatest increase observed with an increase in the proportion from 25% to 50% and is 1–2 MW depending on the temperature of the exhaust gases at the inlet to the ORC. The increase in the ORC power is due to the possibility of deep utilization of low-temperature heat. Thus, with an increase in the proportion of hydrogen from 25% to 50%, the dew point reaches 52 °C, which allows partial condensation of moisture from the gases.

Figure 10 shows a graph of the dependence of the trinary cycle power on the proportion of hydrogen in the GTU fuel. The cycle power increases with the increase in the proportion of hydrogen, in particular, in the GTU, a complete transition to hydrogen fuel leads to an increase in power by an average of 15 MW. Due to partial condensation of moisture from the exhaust gases, an increase in the net efficiency of the trinary CCGT is observed by an average of 0.2% due to an additional increase in the ORC power.



(**a**) Power with different % of H2 in fuel mixture (**b**) Efficiency with different % of H2 in fuel mixture

Figure 10. Dependence of the trinary cycle power on the proportion of hydrogen in the GTU fuel.

A comparative analysis of the study results showed that the increase in capacity and efficiency of a trinary power plant using pure hydrogen as fuel is 24.08, 24.07, 22.86 MW and 0.21, 0.2, 0.18% for 540, 580, 620 °C of the initial temperature of the CCGT, respectively. Thus, the transition to trinary cycles of CCGTs operating on hydrogen fuel allows not only to increase the efficiency of power units, but also to reduce emissions of harmful substances into the atmosphere to zero, while further increase in efficiency is possible due to an increase in the temperature of moisture precipitation by increasing the partial pressure of water vapor in the combustion products or using air cooling in the utilization part of the CCGT below 30 °C.

4. Conclusion

The transition to hydrogen fuel in gas turbine units and combined cycle gas turbines (CCGTs) signifies a critical step toward achieving sustainable energy solutions. By replacing traditional carbon-based fuels with hydrogen, it is possible to eliminate carbon dioxide emissions entirely, aligning with global climate goals.

The analysis shows that increasing the proportion of hydrogen in the fuel mixture can lead to a decrease in total fuel consumption, with a notable increase in gas turbine capacity by approximately 17.07 MW when fully transitioning to hydrogen. Additionally, the efficiency of the gas turbine improves by about 0.18% with a complete switch to hydrogen. The increased moisture content in the combustion products enhances the overall heat capacity, resulting in further benefits for the energy cycle. Moreover, the Organic Rankine Cycle (ORC) benefits from the increased moisture, with capacity improvements of 1–2 MW observed when the hydrogen proportion rises from 25% to 50%. A comparative analysis of trinary cycles using pure hydrogen demonstrates potential capacity increases of 24.08 MW, 24.07 MW, and 22.86 MW for initial CCGT temperatures of 540 °C, 580 °C, and 620 °C, respectively, alongside efficiency gains of 0.21%, 0.20%, and 0.18%.

In conclusion, while the transition to hydrogen fuel necessitates careful operational adjustments and system redesign, it offers a pathway to cleaner, more efficient energy systems. The potential for zero emissions, coupled with significant increases in power output and efficiency, underscores hydrogen's role as a pivotal component in the future of energy production. Continued research and innovation will be essential to address existing challenges and fully leverage the advantages of hydrogen as a sustainable fuel source.

Author Contributions

Conceptualization, M.M.S. and M.I.A.; methodology, R.E.Z.; software, R.E.Z.; validation, M.M.S., M.I.AA. and M.A.O.; formal analysis, M.M.S.; investigation, D.V.P.; resources, M.I.A.; data curation, D.V.P.; writing—original draft preparation, R.E.Z.; writing—review and editing, M.M.S.; visualization, D.V.P.; supervision, M.I.A.; project administration, M.A.O.; funding acquisition, M.I.A. All authors have read and agreed to the published version of the manuscript.

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Institutional Review Board Statement

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Informed Consent Statement

Not applicable.

Data Availability Statement

Data sharing is not applicable to this article as no new data were created or analyzed in this study.

Conflicts of Interest

The authors declare no conflict of interest.

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