



Article Enhancing the Fuel Efficiency of Cogeneration Plants by Fuel Oil Afterburning in Exhaust Gas before Boilers

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Abstract: Cogeneration or combined heat and power (CHP) has found wide application in various industries because it very effectively meets the growing demand for electricity, steam, hot water, and also has a number of operational, environmental, economic advantages over traditional electrical and thermal systems. Experimental and theoretical investigations of the afterburning of fuel oil in the combustion engine exhaust gas at the boiler inlet were carried out in order to enhance the efficiency of cogeneration power plants; this was achieved by increasing the boiler steam capacity, resulting in reduced production of waste heat and exhaust emissions. The afterburning of fuel oil in the exhaust gas of diesel engines is possible due to a high the excess air ratio (three to four). Based on the experimental data of the low-temperature corrosion of the gas boiler condensing heat exchange surfaces, the admissible values of corrosion rate and the lowest exhaust gas temperature which provide deep exhaust gas heat utilization and high efficiency of the exhaust gas boiler were obtained. The use of WFE and afterburning fuel oil provides an increase in efficiency and power of the CPPs based on diesel engines of up to 5% due to a decrease in the exhaust gas temperature at the outlet of the EGB from 150 °C to 90 °C and waste heat, accordingly. The application of efficient environmentally friendly exhaust gas boilers with low-temperature condensing surfaces can be considered a new and prosperous trend in diesel engine exhaust gas heat utilization through the afterburning of fuel oil and in CPPs as a whole.

Keywords: water-fuel emulsion; heat exchange surface; recovery turbogenerator; diesel engine; heat recovery circuit; low-temperature corrosion

1. Introduction

Systems of centralized heat and power supply to consumers from steam turbine cogeneration power plants (CPP) with a unit capacity of up to 600 MW are widely used in energetics. The length of district heating pipelines often exceeds 25–30 km. The main advantage of centralized systems has been the high level of cogeneration energy generation. However, as a result of the long-term operation of CPPs and heat supply systems, which often exceed forty years of service life, there has been a significant moral and physical deterioration of CPP equipment and heat transport systems [1,2]. The use of combined cycle technology contributes to increasing the thermal efficiency of CPPs and district heating systems. Under these conditions, the construction of decentralized heat supply systems with a total electrical capacity of up to 100–150 MW in new microdistricts of cities in Ukraine becomes relevant. Their construction near consumers will significantly reduce the length of heating networks and heat losses [3,4]. There have been positive experiences regarding decentralized heat supply using combined cycle CPPs [4,5]. Unlike traditional combined cycle CPPs, its units include counter-pressure cogeneration turbines with network water heaters. The exhaust gas boiler (EGB) has two fuel oil afterburning chambers. The first



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Copyright: © 2023 by the authors. Licensee MDPI, Basel, Switzerland. This article is an open access article distributed under the terms and conditions of the Creative Commons Attribution (CC BY) license (https:// creativecommons.org/licenses/by/ 4.0/). of them is located in front of the superheater and ensures the maintenance of the steam superheated temperature both in non-heating and heating periods of operation. The second afterburning chamber is located in front of the gas-water heater of the system water. These are switched on during the heating periods of the year and ensure the regulation of the heat load in accordance with the outdoor temperature. The efficiency of the combined-cycle plant CPP-1 Nossener Bruke is ensured by the use of a counter-pressure steam turbine including stuffing box heaters for heating the water system and the absence of a circulating water supply system, which significantly reduces its cost compared to traditional CPPs [6,7]. At the same time, the main disadvantage of this installation is its incomplete adaptability to thermal loads that change during the year. Thus, in winter, at low air temperatures, the power of the steam turbine and system heaters decreases, and the specific generation of electricity for heat consumption deteriorates. Barbu, E. et al. described the results of the calculations of the electric and thermal power of the PSU-450T with fuel afterburning [8,9]. An increase in afterburning fuel consumption leads to a continuous increase in the power of the recovery turbogenerator (RTG), both due to the direct energy supplied by the fuel oil and due to the more efficient use of energy of exhaust gases due to increased steam cycle efficiency and EGB efficiency. According to Borovkov, V.M. et al., small amounts of fuel afterburning do not necessitate complicating the design of boilers with organization of a special furnace, but rather only require placing the burner in flue in the inlet of the EGB [10,11]. During the afterburning of additional fuel oil, the temperature of exhaust gases at the inlet to the EGB increases, which increases the temperature and pressure of superheated steam generated in the EGB as well as its flow rate in order to increase the capacity of the RTG [12]. A study by Shimanov, A.A. et al. showed that afterburning is especially effective using low-power CPPs [13]. For CPPs based on gas turbine units, which correspond to a sufficiently high steam temperature, an increase in efficiency during afterburning of 8% can be observed. Along with an increase in the electric power of the plant, an increase in the thermal efficiency of the CPP will take place. Biryuk, V.V. at. el. showed that, for low-economy steam turbines units (STUs) (with an efficiency of up to 30%), it is advantageous to use afterburning of fuel in a STU in the entire range of relative consumption of afterburning fuel oil β_a . For installations with an average degree of efficiency (the STU efficiency is 30–33%), it is advisable to perform the afterburning of the fuel only until the transition to high steam parameters and a more economical cycle of the STP is achieved. For highly economical gas turbines (with an efficiency of more than 35%), any afterburning of the fuel will affect the reduction in the efficiency of power generation by the plant [14]. Experimental studies conducted by Barbu, E. et al. have shown that afterburning additional fuel in a flue provides a reduction of NO_x emissions by 30–50% [15]. Eclipse, Coen, Pillard, John Zink Company, etc., exhibited improvements in afterburning units using CFD simulation [16].

It is possible to increase the efficiency of the EGB not only by increasing the temperature of the gases at the inlet to the boiler, but also by reducing the temperature of the gases at its outlet. Nevertheless, the wall temperature of the boiler pipes should be 10–15 °C higher than the dew point temperature of the fuel combustion products in order to eliminate low-temperature corrosion (LTC) and extend the service life of these heat exchange surfaces (HESs) [16]. Flue gas dew point temperature is defined as the maximum wall temperature of the HES at which sulfuric acid vapor condenses [17]. Acid vapors also condense in the gas flow on the surface of ash particles [18,19], which then settle on the surface and increase the thermal resistance of the pollution layer [20,21] and the aerodynamic resistance of the gas flow [22] in the EGB, thus affecting its economical operation [23–25].

Water–fuel emulsion (WFE) combustion due to "microexplosions" will accelerate the formation of the mixture [26,27] and accelerate the combustion process due to a stronger turbulent flow in the combustion chamber [28]. The microexplosion of a separate drop of emulsion has been studied; in addition, the simulation of the growth of bubbles inside a drop of fuel has been carried out [29,30]. This allows the maintenance of a lower α compared to the combustion of anhydrous fuel. The efficiency of WFE combustion largely

depends on the dispersive characteristics of the [31,32] emulsions themselves. McMillin, R.E. et al. studied the effect of various surfactants (glycerol dioleate, glyceryl monooleate, and oleamide) on WFE stability using various experimental research methods [33,34]. According to Shafiei, M. et al., the combination of nanoparticles and surfactants resulted in higher stability and viscosity than when either was used alone [35]. The process of WFE spraying was studied depending on the size and speed of droplets, fuel, and substrate temperatures [36]. Another important factor characterizing the effectiveness of the use of WFE is the increase in the efficiency and durability of furnace equipment [37]. According to Mohd Tamam, M.Q. et al., the viscosity value of WFE with water concentrations of 5% and 10% is almost equal to the viscosity value of IFO-380 fuel oil [38]. In this case, it is possible to use such a WFE as a fuel without upgrading the fuel equipment [39]. According to Lee, T.-H. et al., WFE combustion in boilers with 3–14% water content reduces the NO_x output by approximately 40–60% [40,41]. Emissions of PM are also reduced [42,43]. Moreover, an optimal reduction in smoke of 45% has been described [44,45].

A rate of LTC of about 0.25 mm/year has been revealed when combusting WFE containing 30% water and when the temperature of the surface was less than 130 $^{\circ}$ C [46,47]. Thus, the application of gas boilers with such condensing surfaces enables an increase in the efficiency of cogeneration plants [48].

As previously mentioned, a deeper utilization of engine exhaust gas heat by applying the condensing surfaces in EGBs allows the improvement of the environmental and economical operations of power plants. In addition, heat potential can be converted to improve efficiency. Thus, the efficiency of CPP driving engines, such as diesel engines (DE) [49,50], gas engines (GE) [51,52], or turbines [53,54], can be enhanced due to realization of addition heat, for instance, through its conversion to refrigeration in waste heat recovery in refrigeration chillers of absorption type (ACh) [55,56] or refrigerant ejector chillers (Ech) [57,58]; in particular, for cooling engine cyclic air: intake air [59,60] or scavenge air in DE and GE [61,62] or intercooling compressed air turbines [63,64]. In addition, an increased available heat, gained due to deeper exhaust gas heat utilization, enables deeper engine intake air cooling by application of two-stage air coolers [65,66] and combined chillers [67,68]. With these various methods of heat transfer intensification and advanced heat exchanger circulation contours, including jet devices [69,70], increasing heat flux and reducing temperature differences in intake heat exchangers has been proposed in order to decrease the sizes of heat exchangers and corresponding power spent to cover their aerodynamical resistance [71,72] with lowering the leaving temperature of cooled air. They are used in stationary applications for combined cooling, heating, and the power supply of buildings and districts [73,74] as well as in ship [75] and railway applications [76]. Such combined in-cycle trigeneration technologies [77] provide maximum fossil fuel savings accompanied by minimal release of nitrogen oxides and sulfur oxides in exhaust gas. The efficient heat recuperation is realized in waste heat recovery chillers using low boiling refrigerants and the contours of their circulation [78,79] in highly efficient heat exchangers [80,81] due to intensive heat and mass transfer processes [82].

The aim of this research is to evaluate the possibilities of using WFE in DE and afterburning fuel oil in the exhaust gas at the inlet of the EGB to increase the electric and (or) thermal power of the CPP. The following points were investigated:

- Definition of the magnitudes of the LTC rate, the limited reliable operation of condensing surfaces, as well as the lowest temperatures of the exhaust gas from EGBs.
- Calculation of the thermal balance of the CPP, which includes a DE working on different types of fuel in order to evaluate the advantages of using WFE in the DE and afterburning of fuel oil in the exhaust gas at the inlet of the EGB.

2.1. Experimental Research of LTC Intensity

An original experimental set-up was designed (Figure 1). The main elements of the installation are the following: combustion chamber, burner [83,84], fuel preparation system, and gas pipeline, in which working sections with samples for studying LTC are installed.





Fuel oil consumption in the burner was 1-3 kg/h with a viscosity of 2-2.5 µm. The preparation of WFEs for combustion in the furnace of the experimental stand was carried out in a separate unit. Fuel and water were introduced into the mixing tank in certain quantities, depending on the value of the water content in WFEs required for research. A pump circulates the mixture of fuel and water through the disperser until an emulsion is obtained, in which water droplets have a diameter of 15–30 μm. As shown by preliminary adjustment tests of the experimental setup, with such water droplet diameters, the best quality of WFE combustion was achieved due to the torch turbulence due to microexplosions of emulsion droplets. The required quality of the WFE, determined by the diameter of the water droplets, was achieved by setting a certain pressure in front of the disperser. If the set pressure was exceeded, the spring safety valve was activated, and the emulsion was discharged into the mixing tank through a separate pipeline. The quality of the emulsion was checked using a microscope. The prepared WFE was sent to the tank of the experimental set-up, the fuel system of which provides for continuous circulation of fuel in front of the nozzle. It was maintained by a constant value of the obtained dispersion of water in the WFE during the entire experiment. Fuel oil and the WFE were heated to a temperature of 70–90 °C.

The temperatures of hot air varied from 150 to 180 °C. A smoke exhauster was applied to release the exhaust gas from the experimental installation. The surface temperatures t_w , the water W^r and sulfur S^r contents in the WFE, and the factor α of excess air influencing the LTC intensity were investigated. A working section of the pipe samples for LTC studies was designed for the temperatures of exhaust gas of about 350 °C. The pipe samples were simultaneously placed in the exhaust gas duct. Then, they were removed on elapsing the following intervals: 2, 4, 8, 12, and 100 h, sequentially. The wall temperatures t_w varied



within the range 70–140 °C [48]. Cooling of pipe-samples was carried out by supplying water through a hollow screw. The view of the set-up working space is shown in Figure 2.

Figure 2. View of samples for the research of LTC before the tests (a) and after the tests (b,c).

Determination of the velocity and flow rate of flue gases in the gas duct was carried out using a high-speed pipe and a flowmeter. The sample metal temperature was measured using thermocouples (Table 1).

Table 1. Equipment specifications.

Parameter	Equipment	Range	Unit
Flue gas velocity	High-speed pipe	0–25	m/s
Flue gas flow	Flowmeter	0–25	m ³ /h
Flue gas temperature	Resistance thermoconverter	−40−270 °C	°C
Flue gases composition	Gas analyzers	0–100 mL	mL
Sample length	Digital calipers	0-200	mm
Sample diameter	Micrometer	0–25	mm
Sample weigh	Analytical balance	0-200	mg
Sample temperature	Resistance thermoconverter	−40−270 °C	°C

Firstly, the samples were weighed before the tests on the analytical scale. The initial sample mass was marked as m_1 . The pipe samples containing the products of corrosion were also weighed (m_2). The sample processing procedure is presented in Figure 3. The deposits and products of the corrosion were removed, washed, dried, and weighed (mass m_3).



Figure 3. Method of processing pipe samples for the study of LTC.

The corrosion rate K of the condensing surfaces is determined according to the correlation:

$$K = \frac{m_1 - m_3}{F \times \tau} \tag{1}$$

where m_1 —initial mass of the sample before the test, g; m^3 —final sample mass after cleaning the products of corrosion and deposits of soot, g; F—the sample outer surface average area, m^2 ; τ —duration of experiment, h.

The approximation correlation for the corrosion rate depending on the wall temperature $K = f(t_w)$ at certain times of exposure to exhaust gas was determined based on the test data.

The corrosion process depends on a large number of factors that cannot be accurately taken into account [85]. It is necessary to determine their probable error to assess the accuracy of experiments. The probable error of a series of experiments is determined by the formula:

$$S = \theta \sqrt{\frac{\sum_{i=1}^{n} \delta_{i}^{2}}{n(n-1)}}$$
(2)

where δ_i —deviations of the individual measured values of the corrosion rate $K_1...K_n$ from the average value; $\sum_{i=1}^{n} \delta_i^2$ —sum of squares of all deviations from the mean; n—number of measurements; θ —coefficient depending on the number of measurements.

The heterogeneity of the samples has a great influence on the accuracy of the tests. The more samples differ from each other in composition, structure, surface and protective film conditions, random defects, etc., the greater the deviation of the corrosion rate from the average value and the lower the test accuracy. Thus, an increase in test accuracy can be achieved by improving the method and accuracy of work, as well as by increasing the number of samples. During the tests, three samples were installed in the gas duct at the same temperature, which is considered sufficient for corrosion tests. After the experiment, the samples were treated in a 5 and 10% hydrochloric acid solution with urotropin at the same time.

Analysis of the composition of flue gases and determination of excess air was carried out every 20–45 min, which made it possible, with a constant flow of air and fuel, to maintain a relatively constant composition and flow of flue gases and, accordingly, rate of washing the samples.

The relative error in obtaining the corrosion rate is:

$$\frac{\Delta K}{K} = \pm \left(\frac{\Delta(\Delta m)}{\Delta m} + \frac{\Delta F}{F}\right)$$
(3)

The relative error in measuring the area of the pipe sample is:

$$\frac{\Delta F}{F} = \frac{\Delta d}{d} + \frac{\Delta L}{L} \tag{4}$$

The samples were weighed with an accuracy of 0.0001 g. In order to determine Δm it is necessary to weigh the sample twice. The absolute error in determining the mass loss of the sample was $0.0001 \times 2 = 0.0002$ g. Therefore, the relative error in determining Δm with a minimum weight loss of 0.04 g was:

$$\frac{\Delta(\Delta m)}{\Delta m} = 0.0002 \times \frac{100}{0.04} = 0.5\%$$

The outer and inner diameters of the sample were measured with a micrometer in two mutually perpendicular planes with an accuracy of 0.01 mm. The sample length was measured with a vernier caliper with an accuracy of 0.05 mm. The minimum internal diameter of the sample tube was 20 mm and the minimum length was 23 mm. The deviation

from the average diameter value in the measured sections was 0.22 mm. The relative error in determining the area of the corroding surface of the sample was:

$$\frac{\Delta F}{F} = \left(\frac{0.01}{25} + \frac{0.01}{25} + \frac{0.22}{25} + \frac{0.05}{23}\right) \times 100 = 1.18\%$$

The total relative error in determining the corrosion rate is:

$$\frac{\Delta K}{K} = 0.5 + 1.18 = 1.68\%$$

It becomes possible to determine the probable error of this series of experiments when the number of samples at temperatures that differed by 1...2 °C in one combustion mode was more than three.

The systematic error in determining the corrosion rate is:

$$\Delta \mathbf{K} = \sqrt{\left(\frac{\Delta \mathbf{m}}{\tau \times \mathbf{F}}\right)^2 + \left(\frac{\Delta \mathbf{m}}{\tau^2 \times \mathbf{F}} \times \Delta \tau\right)^2 + \left(\frac{\Delta \mathbf{m}}{\tau \times \mathbf{F}^2} \times \Delta \mathbf{F}\right)^2} \tag{5}$$

The systematic error in obtaining the area of the pipe sample is:

$$\Delta F = \sqrt{\left(\frac{\partial F}{\partial d_{ex}} \times \Delta d\right)^2 + \left(\frac{\partial F}{\partial F_{av}} \times \Delta d\right)^2 + \left(\frac{\partial F}{\partial l} \times \Delta L\right)^2} \tag{6}$$

With the accepted geometric dimensions of the sample, the value of systematic error was $\Delta F = 1.84 \times 10^{-4} \text{ m}^2$. The surface of the pipe sample was $F = 0.082 \text{ m}^2$.

Systematic errors were determined with the definition of the corrosion rate for a certain time of experiments for the duration of corrosion tests $\Delta \tau = 10 \text{ min } (0.167 \text{ h})$, during which the processes in the experimental setup were stabilized after the installation of the sample assemblies. When conducting corrosion tests, the limiting relative systematic error in determining the corrosion rate was assumed to be 10%. In this case, the corrosion rate K should be at the level of $10.\Delta K$ for each time point of the experiments (Table 2).

Table 2. Results of calculations of ΔK and K values.

Parameter	Operating Time of the Experimental Setup							
Exposure time of the flue gas flow τ , hours	1	2	4	8	10	50	88	100
The value of the systematic error in measuring the corrosion rate ΔK , g/(m ² ·h)	0.56	0.27	0.138	0.069	0.056	0.012	0.0063	0.0056
Minimum corrosion rate K, g/($m^2 \cdot h$)	5.6	2.7	1.38	0.69	0.56	0.12	0.063	0.056

2.2. Determination of Energy Indicators of CPPs

The energy efficiency of fuel use in various schematic solutions, which include the CPP, can be evaluated using a heat balance. On the basis of such an analysis, it is possible to establish the main indicators of CPP efficiency.

When carrying out calculation studies of the technical and economic indicators of CPP schemes due to the use of exhaust gas heat in the EGB and additional afterburning of fuel oil in the gas flow at the inlet of the EGB, a unit consisting of a DE, EGB, and RTG was adopted as the basic scheme of the CPP.

Previous works [86,87] present the main indicators of the energy efficiency of the CPP:

Electrical efficiency of the unit;

$$\eta_e = \frac{N_e}{BQ_1^r} \tag{7}$$

– Thermal efficiency of the unit;

$$\eta_t = \frac{N_t}{BQ_1^r} \tag{8}$$

- CPP efficiency, which determines the integral energy efficiency of the unit;

$$\eta_{CPP} = \frac{N_e + N_t}{BQ_1^r} \tag{9}$$

where N_e —electrical power, kW; N_t —thermal power, kW; B—fuel consumption, kg/s; Q_l^r —lower calorific value of fuel, kJ/kg.

For calculations of the steam turbine part of the CPP, the raw data were: the parameters of fresh steam of high (p₁, t_{s1}) and low (p₂, t_{s2}) pressures, pressure in the steam turbine condenser—6 kPa, degree of steam dryness—0.892 at the exit from the STU, the internal efficiency of the STU—0.85. The following designations were adopted: ϑ_{in} —the temperature of exhaust gases at the inlet of the EGB; ϑ_{out} —the temperature of the exhaust gases at the outlet of the EGB; B_{DE} and B_a—fuel consumption in the DE and afterburning; D₁, D₂—steam productivity of high- (HPHRC) and low-pressure heat recovery circuits (LPHRC).

When carrying out calculation studies, the following assumptions were made:

- Electric generator: efficiency of the electric generator—98%; the efficiency of the reduction gear for engines—98%, with a rotation frequency of the power shaft n > 3000 rpm;
- EGB: temperature pressure at the inlet to the superheater—20 °C; temperature pressure of the evaporating surface—10 °C; underheating of water in the economizer—10 °C; coefficient of heat preservation—0.97;
- RTG: relative internal efficiency of the flow part—0.8 (humidity correction was introduced in the wet steam zone); steam pressure in the condenser—6 kPa; pressure in the feed water deaerator—0.12 MPa.

The following indicators were used in the analysis of the efficiency of the afterburning of fuel oil taking into account the rise of steam parameters [88,89]:

Efficiency of CPP;

$$\eta_{CPP} = \frac{N_{DE} + N_t}{\left(B_{DE} + B_a\right)Q_l^r}$$
(10)

Efficiency of EGB;

$$\eta_{\text{EGB}} = \frac{T_{\text{EGB}_{\text{in}}} - T_{\text{EGB}_{\text{out}}}}{T_{\text{EGB}_{\text{in}}}} \tag{11}$$

Efficiency of the steam cycle;

$$\mathfrak{Y}_{sc} = \frac{N_e}{[B_{DE}(1-\mathfrak{Y}_{DE}) + B_a]Q_l^r\mathfrak{Y}_{EGB}}$$
(12)

– Efficiency of the afterburning of fuel oil;

$$\eta_a = \frac{N_{STU} - N_{STU}^0}{B_a Q_1^r} \tag{13}$$

The relative power increase of the STU in the CPP scheme with the afterburning of fuel oil was determined by the equation:

$$\Delta N_{STU} = \frac{N_{STU} - N_{STU}^0}{N_{STU}^0}$$
(14)

0

where N_{STU} , N_{STU}^{0} —the value of the STU capacity in the CPP with afterburner and in the binary CPP.

The relative consumption of afterburning of fuel oil was determined as:

$$\beta_a = \frac{B_a}{B_{DE}} \tag{15}$$

The following versions of the calculations of the two-pressure scheme were performed with gradually increased consumption of additional fuel oil and correspondingly raised parameters of the fresh steam of the HPHRC.

After reaching the HPHRC steam parameters of 9 MPa and 520–540 °C, further increasing β_a was accompanied by an increase in the heat drop at the above-mentioned p_1 , t_1 values. In all variant calculations, the values of the temperature pressures on the elements of the EGB and the internal efficiency of the RTG were kept unchanged. The degree of dryness of the steam at the exit of the RTG was kept constant, determining the required pressure by the method of successive approximations. The following parameters were calculated and specified according to existing methods [9,90]: the amount of heat that needs to be obtained in the afterburner chamber in order to increase the steam productivity of EGB, which will ensure its maximum value, permissible under the conditions of increasing steam parameters; fuel consumption in the afterburner chamber; the maximum electrical and thermal capacity of the CPP with the afterburning of fuel oil; and change in the efficiency of CPP when afterburning fuel oil.

When analyzing the results of CPP calculations with DE, it was assumed that there is no redistribution of steam consumption into thermal and electric power. Thus, when calculating the maximum electrical power, all the heat introduced by afterburning is spent on increasing only the electrical power (without changing the thermal power), and when calculating the maximum thermal power—on increasing the thermal power only (without changing the electrical power).

Two options are possible: (1) With unchanged initial steam parameters, when additional fuel use increases the steam consumption and, therefore, the power of the STU (increasing the efficiency of the STU cycle in this case does not occur). When WFE combustion with a water content of 20–30% occurs, it is possible to achieve an increase in the efficiency of the EGB due to a decrease in the temperature of the waste gases ϑ_{out} . (2) When switching to new, higher parameters in the steam turbine part, when there is a change in both the steam consumption and the heat transfer, which is triggered in the STU, in connection with which the electric power and efficiency of the STU change.

Additional afterburning of fuel oil in the exhaust gas at the inlet of the EGB without supplying additional air under the condition of stable combustion is possible with a sufficient concentration of the oxidizer in the flow of exhaust gases for RTG and DE-O₂ > 12–14% (excess air coefficient α > 2). Therefore, the supply of additional air for fuel oil afterburning is not required. For the majority of modern RTGs and DEs, the excess air coefficient in the exhaust gases is α = 3–4, with a temperature of 450–580 °C, and for modern ships, MSEs—350 °C, LSEs—400 °C, α = 2–3.

3. Results and Discussion

3.1. Determination of Admissible Values of LTC Rate

The results of this research show that the contamination of the heating surface during the combustion of WFE occurs more intensively than during the combustion of conventional fuel oil (taking into account the lower corrosion rate). This is explained by the fact that, due to the greater impermeability of the sulfate film, the deposits on the heating surface during combustion of the emulsion are wetter than during the combustion of non-watered fuel oil, which contributes to the adhesion of ash and soot particles on this surface. It is necessary to note the differences in the appearance of the deposits, which is due to their composition. Deposits in the "acid peak" region are black and wet when fuel oil combustion occurs (Figure 2c). When the emulsion is burned, the deposits are moist and multi-layered: the

upper layer is gray (Figure 2c), presumably due to the presence of nitrosis; under this there is a layer of black soot deposits, and directly on the metal surface there is a dense layer of corrosion products.

The results of experimental studies with a duration of 100 h during the fuel oil M100 and WFE combustion with $\alpha = 2.9$ with practically constant parameters for conducting experiments in the form of dependencies K = f (t_w) are shown in Figures 4 and 5 (determination coefficient R² = 0.99).



Figure 4. Relationship of corrosion rate K with wall temperature t_w when the fuel oils combusted.



Figure 5. Relationship of corrosion rate K with wall temperature tw when the WFE combusted.

Figure 4 represents the experimental and calculated values for K with fuel oil (W^r = 2%, S^r = 1.5%, α = 2.9) combustion.

The approximation equation for calculation of the corrosion rate K dependent on t_w while combusting the fuel oil for 100 h is accepted as [48]:

$$\mathbf{K} = 370.16 - 16.582 \mathbf{t}_{\mathbf{w}} + 0.2905 \mathbf{t}_{\mathbf{w}}^2 - 2.4863 \times 10^{-3} \mathbf{t}_{\mathbf{w}}^3 + 1.0411 \times 10^{-5} \mathbf{t}_{\mathbf{w}}^4 - 1.7108 \times 10^{-8} \mathbf{t}_{\mathbf{w}}^5 \tag{16}$$

The Equation (16) was used to study the peculiarities of the corrosion rate: $t_w = 70-150$ °C, $W^r = 2\%$.

Figure 5 exposes the experimental and calculated magnitudes of K with WFE (W^r = 30%, S^r = 1.5%, α = 2.9) combustion.

The following equation for approximating the corrosion rate K dependent on t_w while combusting the WFE for 100 h is accepted as:

$$K = 27.342 - 0.9715t_w + 0.013t_w^2 - 7.7595 \times 10^{-5}t_w^3 + 1.7254 \times 10^{-7}t_w^4$$
(17)

The Equation (17) was used to study the peculiarities of the corrosion rate: t_w = 80–140 $^\circ\text{C}, W^r$ = 30%.

The variable nature of the LTC speed is explained by the transient nature of the condensation of H_2SO_4 vapor at t_w above 140 °C and the condensation of H_2O vapor at t_w below 80 °C with the simultaneous effect of H_2SO_4 vapor fog formation due to the supersaturation of H_2SO_4 vapor at t_w in the range of 70–100 °C and the action of adsorption processes (chemisorption and physical adsorption). At t_w below 70 °C the increase in the LTC intensity is due to the condensation of H_2O vapor, which reduces the H_2SO_4 concentration and increases the LTC rate.

In [91], the dependencies $K = f(\alpha)$ are presented at $\tau = 100$ h when burning standard fuel oil with sulfur content $S^r = 2.07\%$. This makes it possible to compare the results of the predicted and experimental data obtained over the same time at the experimental plant when burning fuel oil with the same sulfur content in practice. It follows from the data that the dependence of the LTC on α is also close to a power function of α , as well as the dependence of the SO₃ content in flue gases on the same excess of air α [91]. Since there are no literature data on LTC at such α , when constructing the dependence K = f (α), one has to rely on the values of the corrosion rate obtained in studies of the LTC kinetics. Approximating the values presented in [91], we obtained the value K = 1.6 g/(m²·h) at $\alpha = 2.9$, K = 1.2 g/(m²·h) at $\alpha = 2$, which practically coincides with our experimental data.

In real conditions, variable combustion modes and variable t_w of heating surfaces are observed, which leads to an increase in corrosion rates. The corrosion rate of 0.25 mm/year at $\alpha = 1.05$ [92] was adopted to obtain comparative characteristics K = f (t_w) during WFE and fuel oil combustion at different α and W^r (Figure 6).



Figure 6. Relationship of corrosion rate $K = f(t_w)$ with different water contents.

According to these dependencies, it is possible to determine the permissible corrosion rate K_{per} at different t_w and the minimum magnitudes of the condensing surface temperatures according to the corrosion intensity acceptable for the reliable operation of the EGB condensing HES. The obtained dependences showed that, with water content $W^r = 30\%$, the accepted allowable value of the corrosion rate in the area of the "acid peak" is provided, as well as the minimum wall temperature t_{min} in the area of the second "peak" of corrosion, at which the accepted allowable value of the corrosion rate is provided. This is due to the formation of a dense salt layer and metal passivation [93,94]. This justifies the fact that the condensing HES service time is the same as for a dry surface while combusting WFE.

3.2. The Efficiency of the Afterburning of Fuel Oil in the Exhaust Gas at the Inlet of EGBs of CPPs

Calculation studies and comparisons of the effectiveness of different HRCs were performed for the CPP using a Wärtsilä corp. 16V32 DE with a nominal power of 9280 kW, a specific fuel consumption of 0.195 g/(kW·h), and an exhaust gas temperature of 350 °C.

In the considered versions of the SPP, two-pressure EGBs with multiple forced circulation are used during fuel oil combustion and HWSS, are used during WFE combustion. Calculations of the thermal diagrams of the specified heat recovery circuits (HRCs) [47] show that, due to the possibility of reducing the value of ϑ_{in} from 160 °C to 90 °C during WFE combustion with a water content of 30%, the thermal capacity of the EGB increases by 40%. When fuel oil is afterburning at the inlet of the EGB, each value of β_a corresponds to the value of the initial pressure of the superheated steam, which increases with the increase of β_a . Currently, low-power steam turbines are produced for the limit parameters of superheated steam of about 9 MPa, 550 °C [88]. Such parameters can be estimated as marginal for turbines of a similar class. The limitation of the further increase in the efficiency of the EGB occurs when the minimum permissible value of ϑ_{out} is reached at the corresponding value of Δt at the last surface of the EGB, which is taken to be equal to 20 °C:, provided that corrosion of the condensing HES of the EGB is prevented when liquid gas turbine fuel combustion at S^r = 0.5%—120 °C [88], when fuel oil combustion with a sulfur content of ~2%—150–160 °C, and when WFE combustion based on fuel oil—90–100 °C, due to a significant reduction of LTC due to passivation of the metal of condensing HESs.

The use of the HRC of CPPs with two-pressure EGBs in the steam part of the CPP allows the increase of its efficiency both by reducing the heat to the minimum permissible level when installing a low-pressure evaporative surface and by increasing the parameters of the steam in the HPHRC. In a two-pressure EGB (Figure 7), the maximum number can be seven HESs. In this case, individual drums and feed pumps of HPHRC and LPHRC are required.



Figure 7. Thermal scheme of the HRC of the CPP with a two-pressure EGB, operating on fuel oil: DE—diesel engine; AS—afterburning system; S1—superheater; E1, E2—evaporator high- and low-pressure; EC1, EC2—economizer high- and low-pressure; RTG1—recovery turbogenerator; G—generator; C—condenser; CP—condensing pump; Fp1, Fp2—feed pump high and low pressure.

In two-pressure EGBs, due to the generation of steam in the LPHRC, the temperature ϑ_{out} decreases to the minimum allowable calculated level of 153 °C during fuel oil combustion. At the same time, an increase in the steam pressure of the boiler p_1 does not lead to an increase in ϑ_{out} and, as a result, to a decrease in the efficiency of the EGB. Therefore, for two-pressure RTGs and CPPs based on MSE, increasing the initial steam parameters of the HPHRC under other constant conditions clearly leads to an increase in economy due to an increase in the efficiency of the steam turbine cycle and of the steam turbine power. The amount of low-pressure steam and the share of power generation by this flow are lower than in the HPHRC. The parameters of the HPHRC are decisive in the design of the SPP. The parameters of the LPHRC are usually chosen in such a way that, when mixing in the steam turbine of both steam flows, their temperatures and enthalpies are close, and when mixing the steam of the LPHRC, there is no influence on the parameters of the main flow. During sulfuric fuel oil combustion, it is necessary to maintain $t_w = 130$ °C; the minimum value of p_2 is 300 kPa, which corresponds to $t_{s2} = 133$ °C (minimum value of t_w under the condition of preventing LTC).

During WFE combustion, it becomes possible to reduce the temperature of exhaust gases ϑ_{out} to 90 °C, which allows installing a third evaporative surface and (or) a section of hot water supply (Figure 8).



Figure 8. Thermal scheme of the HRC of the CPP operating on WFE: DE—diesel engine; AS—afterburning system; S1, S2—superheater; E1, E2—evaporator high and low pressure; EC1, EC2—economizer high and low pressure; RTG1, RTG2—recovery turbo generator high and low pressure; G—generator; C—condenser; CP—condensing pump; Fp1, Fp2—feed pump high and low pressure, HWSS—hot water supply system.

For increasing the thermal capacity of the EGB while maintaining a constant value of p_1 and p_2 with an increase in β_a , there is an increase in ϑ_{in} at the inlet of the EGB, which leads to an increase in N_e or N_t (Figure 9a). At the same time, the steam productivity of the HPHRC is constantly increasing, and the LPHRC remains almost constant. The total value of steam productivity ensures the increasing of N_e and N_t (Figure 9b).



Figure 9. The influence of the degree of fuel oil afterburning in the exhaust gas at the inlet of the EGB on: (a) the temperature of the gases at the outlet of the heat exchanger N_t (at $N_e = 0$) and N_e (at $N_t = 0$); (b) steam consumption D_1 , D_2 of the HPHRC and LPHRC, and total steam productivity D_{Σ} .

Since, in the two-pressure EGB scheme, the temperature of the exhaust gases outlet of the EGB is at the minimum level, it is possible to increase the efficiency of the EGB with an increase in β_a . There is a reserve for increasing the parameters and efficiency of the steam turbine part due to the increase in the operation of the heat drop in the turbine, as well as lowering the share of steam generated by the LPHRC with a relatively low potential. For two-pressure EGBs, the gas temperature at the inlet of the LPHRC is preserved with an increase in β_a at $p_1 = \text{const}$, which ensures a constant value of D_1 (Figure 9b).

An increase in β_a provides an increase in the values of η_a and η_{CPP} at each optimal value of pressure p_1 . When reaching the maximum possible value in terms of the economy of the first stages of RTG and its strength, a fracture occurs when the accepted limit pressure is reached, after which η_{CPP} = const and the afterburning efficiency η_a decreases (Figure 10). The EGB efficiency is continuously increasing.



Figure 10. CPP efficiency indicators.

The dependence of the flow rate of high- and low-pressure steam (Figure 11a) has fractures at the point of reaching the limit parameters (9 MPa, 545 °C). With a further increase in β_a , the decrease in D₂ continues (Figure 11b). After a certain critical β_a , the EGB scheme with only one circuit is more acceptable.



Figure 11. Parameters of the two-pressure EGB scheme with an increase of degree of fuel oil afterburning: (a) gas temperature and values of pressures p_1 and p_2 ; (b) steam productivity generated by the HPHRC D_1 and LPHRC D_2 and the total steam productivity D_{Σ} .

An increase in the degree of fuel oil afterburning β_a primarily causes the rise in gas temperature at the inlet of the EGB. There is an increase in the steam consumption of the HPHRC and an increase in the initial parameters of the RTG cycle (t_{s1} and p₁). This makes it possible to increase the electric power N_e of the CPP at the expense of the RTG power of two pressures, or the thermal power N_t, if the maximum amount of saturated steam is produced in the EGB (Figure 12a). In addition, this causes a faster rate of increase in the efficiency of the RTG (Figure 12b).



Figure 12. The influence of the degree of fuel oil afterburning on: (**a**) electrical and thermal power of the CPP; (**b**) efficiency of the CPP.

The expediency of the afterburning of fuel oil in the exhaust gases in the DE for the CPP should be determined in combination with detailed optimization calculations of the parameters of the selected CPP circuit pair. The use of fuel oil afterburning makes it possible to increase the efficiency and power of the CPP, taking into account the significant decrease in the exhaust gas temperature in the outlet of the EGB during WFE combustion, which under certain conditions can cause some increase in the efficiency of the CPP as a whole.

4. Conclusions

The use of combined cycle technology contributes to increasing the thermal efficiency of CPPs and district heating systems. The construction of decentralized CPPs with a low electrical capacity in new microdistricts of cities becomes relevant. Their construction near consumers will significantly reduce the lengths of heating networks and heat losses.

In order to increase the steam capacity of CPPs, the afterburning of fuel oil in the exhaust gas at the inlet of the boiler was carried out. This enabled an increase in the efficiency of the EGB, not only due to increasing the temperature of the gases at the inlet of the boiler, but also by reducing the temperature of the gases leaving the boilers, id est, reducing heat waste.

Investigations of corrosion intensity when combusting fuel oils and WFEs were carried out at wall temperatures less than sulfuric acid vapor dew point values in a specially designed set-up.

The wall temperatures of condensing surfaces varied within the interval of 70–140 $^{\circ}$ C and the intensity of LTC within a suitable range from 0.15 to 0.25 mm/year was determined. Corresponding reductions in leaving boiler gas temperature, waste heat, and exhaust emissions by almost two times is a testament to the enhancement of the gas boiler efficiency and CPP as a whole. This allows for the widened application of efficient environmentally friendly exhaust gas boilers with low-temperature condensing HESs and can be considered as a new and prosperous trend in diesel engine exhaust gas heat utilization through the afterburning of fuel oil and in CPPs as a whole.

The calculation results proved the advantages of the application of exhaust gas boilers with low-temperature condensing HESs in CPPs with diesel engines working on different types of fuel.

The use of WFEs and afterburning fuel oil provides an increase in the efficiency and power of the CPPs based on diesel engines of up to 5% due to the decreased exhaust gas temperature at the outlet of the EGB from 150 $^{\circ}$ C to 90 $^{\circ}$ C and waste heat, accordingly.

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Nomenclature and Units

ACh	absorption chiller		
СР	condensing pump		
CPP	cogeneration power plant		
DE	diesel engine		
DF	diesel fuel		
Ech	ejector chiller		
EGB	exhaust gas boiler		
HES	heat exchange surface		
HPHRC	high-pressure heat recovery circuit		
HRC	heat recovery circuit		
HWSS	hot water supply system		
LPHRC	low-pressure heat recovery circuit		
LSE	low-speed engine		
LTC	low-temperature corrosion		
MSE	medium-speed engine		
RTG	recovery turbo generator		
STU	steam turbine unit		
WFE	water-fuel emulsion		
Symbols and units			
B _{DE} , B _a	fuel consumption in the diesel engine and afterburning	kg/s	
$D_1 D_2$	steam productivity of high- and low-pressure heat	ko/s	
<i>D</i> ₁ , <i>D</i> ₂	recovery circuit	KG/ 5	
F	average area of the outer surface of the sample	m ²	
K	corrosion rate	g/m²∙h	
m_1	mass of sample before experiment	g	
m ₂	mass of sample after experiment	g	
m_2	mass of sample after cleaning of corrosion products and	σ	
	soot deposits	8	
N _e , N _t	electrical and thermal power	kW	
N _{DE}	diesel engine power	kW	
N _{STU}	capacity of the steam turbine unit	kW	
p ₁ , p ₂	steam pressure	kPa	
Sr	sulfur content of fuel oil	%	
t_{s1}, t_{s2}	steam temperature	°C	
t _w	wall temperature of heating surface	°C	
Ql	lower caloritic value of fuel	kJ/kg	
Wr	water content of emulsion	%	
α	excess air coefficient	-	
β _a	relative consumption of afterburning fuel oil	%	
$\vartheta_{\rm in}, \vartheta_{\rm out}$	exhaust gas temperature at the inlet and outlet of the	ilet and outlet of the °C	
	exhaust gas boiler		
τ	duration of experiment	h	
ıj _a	efficiency of afterburning	-	
¹ J _e , ¹ J _t	electrical and thermal efficiency	-	
^{IJ} CPP	etticiency of cogeneration power plant	-	
^{IJ} EGB	etticiency of exhaust gas boiler	-	
J _{SC}	efficiency of steam cycle	-	

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