Applied Chemical Engineering (2024) Volume 7 Issue 4
doi: 10.59429/ace.v7i4.5564

RESEARCH ARTICLE

Low-grade heat utilization for combined cycle power plants using organic rankine cycle

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ABSTRACT

This paper presents a comprehensive thermodynamic analysis of trinary power plants, focusing on the efficiency of Organic Rankine Cycle (ORC) configurations and regenerative heating methods. Despite advancements in nuclear and renewable energy, fossil fuels still dominate electricity generation, necessitating improved efficiency in existing power plants. The study reveals that low-pressure mixing-type heaters provide higher efficiency compared to surface heaters, with net efficiencies of 0.099%, 0.227%, and 0.425% at deaerator pressures of 0.12, 0.3, and 0.7 MPa, respectively. The analysis highlights the impact of feedwater temperature on the thermal efficiency of steam turbine units (STUs), noting that while optimal feedwater temperatures enhance efficiency, they can reduce STU capacity. The study identifies configurations for regenerative heating that optimize exhaust gas temperatures, facilitating additional electricity production through a low-boiling working fluid in the ORC. The findings indicate that R245fa refrigerant is optimal for ORC without recuperative heater, achieving maximum net power at a feedwater temperature of 115°C. For ORC with a recuperative heater, R236ea is preferred for temperatures between 115°C and 154.5°C, while R245fa is optimal for higher temperatures. The results also demonstrate that trinary power plants with recuperators achieve greater efficiency and net capacity compared to double-circuit systems, with notable improvements in thermal efficiency attributed to effective regeneration schemes. This research underscores the potential for optimizing existing domestic power units to enhance their efficiency and performance without significant financial or technical burden, thereby contributing to more sustainable energy generation.

Keywords: Organic Rankine Cycle (ORC); trinary cycle; energy efficiency; CCGT; system of regeneration; optimization

ARTICLE INFO

Received: 04 November 2024
Accepted: 28 November 2024
Available online: 16 December 2024

1. Introduction

Despite the development of nuclear and renewable energy, the bulk of electricity in the world is generated by burning fossil fuel. The depletion of natural fuel reserves leads to an increase in prices[1-2]. In Western countries, more efficient equipment is being actively developed in order to reduce fuel consumption in power plants. For domestic turbine construction, achieving maximum efficiency required huge financial resources. Thus, the maximum achieved efficiency of foreign combined-cycle units is 65% versus 51% for domestic ones developed in Russia. The main method for increasing the efficiency of combined-cycle units is to increase the initial temperature of gases in front of the gas turbine. Thus, Mitsubishi Heavy Industries, Siemens, General Electric Energy, Alstom Power use innovative thermal barrier coatings and develop efficient cooling systems in order to increase the operating temperature of first-stage blades, and also use various high-alloy steel alloys. Today, the initial temperature level in gas turbines has reached 1650°C and developments are underway at 1700°C[3-5].

Such temperature levels cannot yet be achieved in domestic gas turbine units due to the peculiarities of the historical development of power engineering, where the main emphasis was placed on steam turbine units. Despite this, domestic science continues to conduct various studies that consider ways to improve the efficiency of traditional thermal power plants and nuclear power plants. One such example is the transition from a traditional water coolant to supercritical carbon dioxide[6]. Most often, such projects are very complex in technical implementation and require a large material and industrial base. Thus, it is necessary to look for ways to improve the efficiency of existing domestic power units without significant financial and time costs.

Despite their technological perfection, existing CCGT units still have a significant amount of waste heat. This amount of heat is not used to generate electricity and is classified as losses. The main losses in combined-cycle plants are losses in the condenser and losses with exhaust gases. Losses in the STU condenser are reduced by using a heat recovery system. This solution allows increasing the value of the average integral temperature of heat supply in the steam turbine unit, as well as reducing the mass flow rate of steam in the condenser. Due to the latter, the power removed from the cycle in the condenser decreases, which means that the efficiency of the steam turbine unit increases. However, this method has not found widespread use in CCGT units, since an increase in the feedwater temperature leads to a decrease in the efficiency of the waste heat boiler. This leads to an increase in the temperature of the exhaust gases and, as a result, to an increase in the losses of thermal energy.

This source of waste heat can be used in an additional low-potential cycle[7]. One of the types of installations used to utilize heat from a low-temperature source is an organic Rankine cycle installation. The authors of[8-9] propose a scheme for utilizing the heat of exhaust gases after a gas turbine, provide information on variants of thermal schemes of ORC installations and their operation, both as the main generating equipment at power plants and as additional cycles for utilizing waste heat. The ORC technology is applicable to increase the energy efficiency of modern gas turbine driven GCU.

The thermodynamic efficiency of the ORC and the optimal parameters of the plant differ significantly when using different working substances. The power and energy efficiency of the ORC-module on the exhaust of GTU on Gas compressor unit significantly depends on the temperature of the atmospheric air[10]. In[11], the thermodynamic effect of using an ORC installation in a waste heat boiler operating on the organic Rankine cycle is studied. It was found that when using R134a freon, the efficiency of the combined cycle plant increases by more than 1.1%, when using R123 freon - by 1.16%.

The issue of the method of regenerative heating of the coolant, as well as the choice of the type of working fluid, its parameters and the thermal scheme of the ORC cycle for trinary power plants, is poorly understood. In this regard, the task of conducting optimization thermodynamic studies aimed at choosing the optimal parameters of the coolant both in the STU and in the utilization part of the trinary power plant becomes relevant.

This paper presents the results of a thermodynamic analysis of the methods of regenerative heating of the coolant in trinary power plants. Where, due to regeneration in the STU, losses in the condenser are reduced and an additional utilization circuit on a low-boiling working fluid is used to generate electricity due to the useful use of the heat of the exhaust gases.

2. Methodology

The objective is to study thermal diagram of a trinary power plant shown in **Figure 1**, consisting of a gas turbine unit, a waste heat boiler, a steam turbine unit with a high and low pressure regeneration system and a waste heat cycle using a recuperative heat exchanger for heat regeneration.

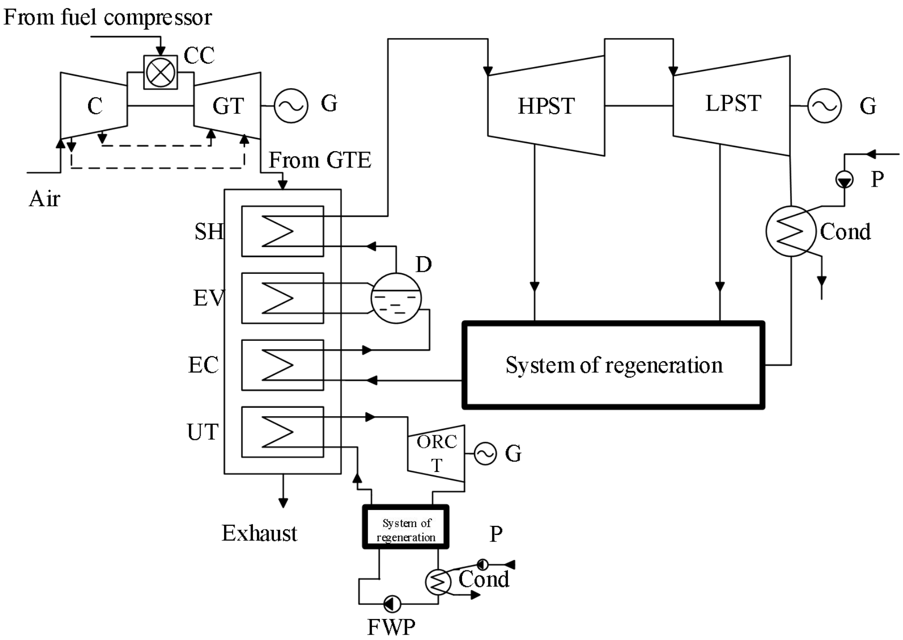

Figure 1. The considered scheme of a trinary power plant with heat regeneration in steam turbine and utilization units.

In order to reduce heat losses in the steam turbine condenser and, as a result, increase the efficiency of the steam turbine plant, a heat regeneration system is used, consisting of high- and low-pressure heaters of both mixing and surface types in order to increase the efficiency of the cycle. In addition to the heaters, the scheme also provides for the use of a deaerator to provide regenerative heating and purification of the main condensate from impurities. An increase in the feedwater temperature leads to an increase in the temperature of the exhaust gases at the outlet of the steam turbine heating surfaces, and, as a result, to a decrease in the efficiency of the waste heat boiler. To prevent an increase in heat losses emitted into the atmosphere with exhaust gases, a utilization cycle is used.

The organic Rankine cycle on a low-boiling coolant, which consists of a utilization heat exchanger, a feed pump, a turbine, a recuperator, a condenser and an electric generator, is considered as a utilization cycle in the work.

The choice of a heat carrier for the organic Rankine cycle directly determines the efficiency of waste gas heat recovery, however, it should be noted that some of the heat carriers are toxic and flammable. In[12], based on such indicators as the global warming potential (GWP) and the impact on the decomposition of the ozone layer (ODP), it was concluded that freons R22, R124, R134a, R236ea are promising for use in utilization cycles. In addition to the above-mentioned heat carriers, freons R245fa, R32 can also be classified as safe and non-flammable. All these freons have a high safety rating A1-B1 and a low ODP value, and within the framework of this work, the above-mentioned heat carriers are considered for the utilization cycle of a trinary unit.

The developed model of the thermal scheme of the trinary power plant consists of several main blocks: GTU, STU, waste heat boiler and waste heat superstructure.

When modelling the GTU as part of the trinary power plant, the parameters of the GTE-160 as part of the CCGT were taken as reference parameters. The values and parameters of the flow rates for cooling were calculated according to the method described in[13]. The calculation of combustion processes in the combustion chamber was carried out under the condition of the stoichiometric combustion reaction of methane, the temperature in the combustion chamber was 1060 ° C.

The initial data for modelling the thermal schemes of combined-cycle plants are given in **Table 1**.

Table 1. Initial data for modelling thermal circuits.

Parameter	Value
Consumption of exhaust gases of GTU, kg/s	509
Underheating at the hot end of the superheater, °C	20
Maximum steam temperature at the outlet of the waste heat boiler, °C	560
Underheating at the cold end of the evaporator, °C	10
Underheating at the economizer inlet, °C	10
Underheating at the deaerator inlet, °C	10
Temperature in the condenser, °C	30
Deaerator pressure, MPa	0,12 / 0,3 / 0,7
Pressure loss in superheater, %	5
Hydraulic losses of steam extraction pipelines of heaters, %	5
Excess of pressure in the deaerator selection over the pressure in the deaerator, %	40
Hydraulic losses in the waste heat boiler economizer, %	10
Hydraulic losses in heaters, %	5
Hydraulic losses of the condenser, kPa	80
Internal efficiency of feed, condensate, circulation pumps, %	85
Dryness level at the steam turbine outlet, %	90
Heat losses in the heating surfaces of the waste heat boiler, %	0.004
Heat losses in heat exchangers, %	1
Underheating in the high-pressure heater, °C	1.5
Underheating in the low-pressure heaters, °C	55
Cooling water temperature before the feed pump, °C	15
Mechanical efficiency, %	99
Efficiency of electric generator, %	99
Temperature in ORC condenser, °C	30
Temperature of cooling water at the inlet to the circulation pump, °C	15
Hydraulic losses of the condenser of the utilization cycle, kPa	80
Minimum temperature difference in the utilizer, °C	10
Minimum temperature difference in the regenerator, °C	5
Internal relative efficiency of the pump/compressor, %	85
Internal relative efficiency of the turbine, %	85
Efficiency of electric generators and electric motors, %	99
Mechanical efficiency, %	99
Minimum degree of steam dryness at the outlet of the turbine, %	90

The net power of the gas turbine plant was determined as:

$$N_{GTU} = \left(N_T - N_{comp}\right) \cdot \eta_{mech} \cdot \eta_{EG} - N_{FC} \cdot \eta_{mech} \cdot \eta_{ED} \quad (1)$$

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;

capacity of the steam turbine circuit of the combined cycle plant was determined as:

$$N_{STU} = (N_{HPT} + N_{LPT})\eta_{mech} \cdot \eta_{EG} - \frac{(N_{CP}+N_{FP}+N_{RP})}{\eta_{mech} \cdot \eta_{ED}} \quad (2)$$

where N_{HPT} and N_{LPT} is the power of the groups of stages of the HPT and LPT, Kw; N_{CP}, N_{FP} and N_{RP}– power of condensation, feed and recirculation pumps, Kw.

The net capacity of the ORC plant was determined as:

$$N_{ORC} = N_{FT} - N_{P} \quad (3)$$

where $N_{FT}$and $N_{P}$are the power of the freon turbine and pump, respectively, Kw.

The net efficiency of binary and trinary cycles was calculated as:

$$\eta = \frac{N_{GTU}+N_{STU}+N_{ORC}}{BQ_{LLV}} \quad (4)$$

The CoolProp database[14] was used to determine the thermophysical properties of freons and other working fluids.

3. Results of thermodynamic analysis

3.1. Efficiency of regeneration systems

The results of the thermodynamic analysis demonstrated that the regeneration system using low-pressure mixing-type heaters is more efficient than surface heaters. The net efficiency of the turbo plant utilizing mixing-type heaters was found to be:

- 0.099% at a deaerator pressure of 0.12 MPa
- 0.227% at a deaerator pressure of 0.3 MPa
- 0.425% at a deaerator pressure of 0.7 MPa

3.2. Impact of feedwater temperature

The presence of an optimal feedwater temperature is attributed to an increase in the thermal efficiency of the steam turbine unit (STU). However, as the feedwater temperature increases, the capacity of the STU decreases. Additionally, an increase in deaerator pressure necessitates an increase in the number of low-pressure regenerative heaters. The maximum allowable heating of the main condensate in the heater was set at 50°C.

As feedwater temperature rises, the temperature difference between the heated and heating coolant diminishes, leading to decreased efficiency of the waste heat boiler and an increase in exhaust gas temperature.

3.3. Optimization of regenerative heating

During the optimization of the regenerative heating composition for the STU, a configuration with a deaerator pressure of 0.12 MPa, one high-pressure heater, and two low-pressure mixing-type heaters was selected. With these parameters, the exhaust gas temperature after the economizer surface of the waste heat boiler ranged from 195°C to 245°C, depending on the feedwater temperature, making it feasible to utilize the cycle for additional electricity production.

3.4. Working Fluid Selection for ORC

Further analysis focused on selecting the working fluid type and its initial parameters to maximize net power in the Organic Rankine Cycle (ORC). The maximum net power of the ORC is achieved at a feedwater temperature of 115°C using R245fa refrigerant at a pressure of 5 MPa. In the temperature range of 115°C to

190°C, R245fa remains the optimal working fluid, with the initial pressure increasing by an average of 1-2 MPa for every 25°C rise in temperature.

For the ORC utilizing a recuperative heat exchanger, the maximum net power of 14 MW is achieved in the feedwater temperature range of 115°C to 154.5°C using R236ea refrigerant. In the range of 154.5°C to 190°C, the maximum net power of 17.16 MW is achieved using R245fa.

3.5. Performance of trinary CCGT units

The study also examined the influence of feedwater temperature on the efficiency of a trinary combined-cycle gas turbine (CCGT) unit. A trinary power plant with an advanced heat recovery system in the steam turbine section, without using a recuperative heater in the ORC with R245fa refrigerant, achieved maximum efficiency and net power of 50.97% and 222.9 MW, respectively, at a feedwater temperature of 132°C.

In contrast, using a recuperative heater in the utilization cycle with R236ea refrigerant increased the optimal feedwater temperature to 142°C, resulting in efficiency and net power of 51.2% and 223.89 MW, respectively.

3.6. Comparative analysis

Table 2 presents a comparison of the trinary power plant with an advanced heat recovery system in the steam turbine and the ORC against single-circuit and double-circuit combined-cycle plants. Key findings include:

- The net efficiency of a trinary plant with a recuperator in the ORC is 0.37% higher than that of a double-circuit plant and 0.23% higher than that of a trinary plant without a recuperator in the ORC.
- The net capacity of a trinary plant with a recuperator in the ORC is 1.59 MW higher than that of a double-circuit plant and 0.99 MW higher than that of a trinary plant without a recuperator in the ORC.

These improvements can be attributed to the enhanced thermal efficiency of the steam turbine plant, resulting from a more efficient regeneration scheme with low-pressure mixing heaters and a high-pressure heater, as well as the utilization of exhaust gas heat in the utilization cycle using a low-boiling working fluid.

Table 2. Results of calculation of thermal schemes of combined-cycle plants.

Type of CCGT	Net capacity of CCGT, MW	Net efficiency of CCGT, %
Double-circuit	222.29	50.83
Trinary plant without a recuperator in the ORC	222.9	50.97
Trinary plant with a recuperator in the ORC	223.89	51.2

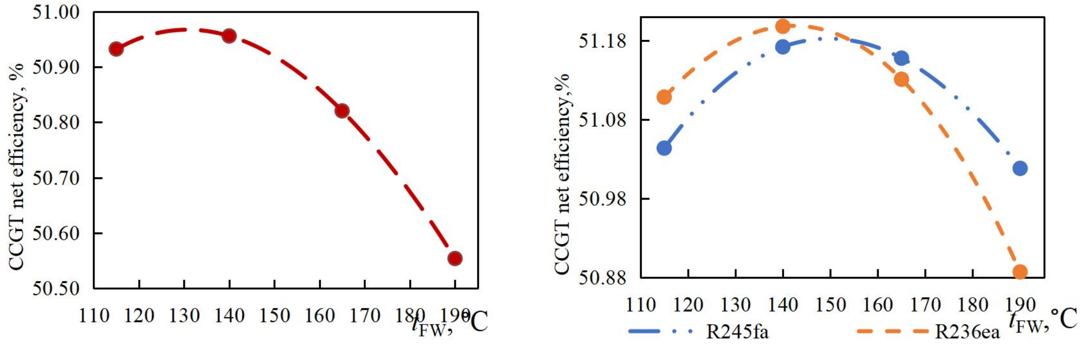

Figure 2. Dependences of the net efficiency of a trinary CCGT without a recuperator in the ORC (a), with a recuperator (b) on the feedwater temperature.

4. Conclusions

This paper analyses the thermodynamic performance of combined-cycle power plants operating on trinary cycles, focusing on Organic Rankine Cycle (ORC) configurations.

1. ORC without Recuperative Heater: The maximum net power is achieved using R245fa refrigerant across the entire range of feedwater temperatures. As feedwater temperature increases, the optimal initial pressure of the working fluid rises by 1-2 MPa.
2. ORC with Recuperative Heater: The maximum net power is realized with R236ea refrigerant for feedwater temperatures between 115°C and 154.5°C. For temperatures from 154°C to 190°C, R245fa refrigerant yields the highest net power.
3. Trinary Power Plant (No Recuperator): An optimum feedwater temperature of 132°C results in a maximum efficiency of 50.97% and a net power output of 222.9 MW. At this temperature, the initial pressure of the working fluid in the low-boiling cycle is 5 MPa, with the ORC contributing 11.4 MW and the steam turbine unit 68.5 MW.
4. Trinary Power Plant (With Recuperator): At an optimum feedwater temperature of 142.5°C, the efficiency and net power output reach 51.2% and 222.89 MW, respectively, using R236ea refrigerant at an initial pressure of 8 MPa. The ORC's net capacity is 13.1 MW, while the steam turbine unit's capacity is 67.7 MW.
5. Comparison of CCGT Systems: The efficiency and net capacity of a triplex CCGT with a recuperator are higher than those of a double-circuit CCGT by 0.37% and 1.59 MW, respectively.

Acknowledgement

This study conducted by Moscow Power Engineering Institute was financially supported by the Ministry of Science and Higher Education of the Russian Federation (project No. FSWF-2023-0014, contract No. 075-03-2023-383, 2023/18/01).

Abbreviations

TPPs	Thermal Power Plants	EG	Electric Generator
GTU	Gas Turbine Unit	EV	Evaporator
ORC	Organic Rankine Cycle	EC	Economizer
GW	Giga Watt	G	Generator
ORCT	Organic Rankine Cycle Turbine	GTE	Gas Turbine Exahust
HRSG	Heat Recovery Steam Generator	FWP	Feed Water Pump
STU	Steam Turbine Unit	CP	Condensor Pump
CCGT	Combine Cycle Gas Turbine	D	Deaerator
HPSH	High Pressure Super Heater	C	Condensor
CO2	Carbon Dioxide	HPST	High Pressure SteamTurbine
SH	Super Heater	LPST	Low Pressure Steam Turbine
GE	General Electric	UT	Utilization
C	Compressor	LHV	Low Heating Value
GT	Gas Turbine		

Conflict of interest

The authors declare no conflict of interest.

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