Energy and exergy evaluation of co_2 closed-cycle gas turbine

Lorencin Ivan, Štifanić Daniel, Musulin Jelena, Mrzljak Vedran
Faculty of Engineering, University of Rijeka, Vukovarska 58, 51000 Rijeka, Croatia
E-mail: ilorencin@riteh.hr, dstifanic@riteh.hr, jmusulin@riteh.hr, vedran.mrzljak@riteh.hr

Abstract: *This paper present energy and exergy evaluation of CO_2 closed-cycle gas turbine process. The most important operating parameters of the whole observed cycle, as well as of each of its constituent components are presented and discussed. In the observed process, produced useful mechanical power for the power consumer drive is equal to 5189.78 kW, while the energy efficiency of the whole cycle is equal to 36.6%. Heat Regenerator is a crucial component of the observed process – without its operation energy efficiency of the whole cycle will be equal to only 16.91%. From the exergy aspect, Turbocompressor (TC) and Turbine (TU) shows good performances because its exergy efficiencies are higher than 90%. Regenerator exergy efficiency could be increased by lowering the temperature of the ambient in which analyzed CO_2 closed-cycle gas turbine operates.*

KEYWORDS: CLOSED-CYCLE GAS TURBINE, CO_2, ENERGY ANALYSIS, EXERGY ANALYSIS

1. Introduction

Gas turbines are nowadays widely used in a power production as a stand-alone devices [1], as a part of a complex combined-cycle power plants [2], in cogeneration systems [3], etc. Gas turbines also found its implementation in marine power and propulsion systems as a primary propulsion element [4], part of a complex ship energy system [5] or as an auxiliary element in additional power production [6].

In the literature can be found many researches of gas turbines with various upgrades [7, 8] or without any upgrade [9]. However, the majority of researches are related to open-cycle gas turbines and its performances, closed-cycle gas turbines are exploited in the literature much lower.

Closed-cycle gas turbines offers various benefits in comparison to open-cycle gas turbines. They can use various operating mediums, which are non-corrosive, which offers better thermodynamic characteristics (when compared with air and combustion gases) and they did not have any connection with the environment. Also, produced useful power in such processes can be easily regulated by changing of operating medium mass flow rate.

Disadvantages of closed-cycle gas turbines can be found in a fact that used operating mediums are usually expensive and cannot be always found in the market, such cycle requires additional plant for operating medium storage, heaters and coolers inside the process have huge dimensions, etc.

In this paper is analyzed closed-cycle gas turbine which operating medium is CO_2. It is investigated operation performances, transferred heat through the process as well as efficiencies and losses (destructions) from the energy and exergy aspect. Heat Regenerator is an essential component of this closed-cycle gas turbine, so it is also presented a change in its exergy efficiency and exergy destruction during the ambient temperature change.

2. Description and operating characteristics of the analyzed CO_2 closed-cycle gas turbine

Scheme of the analyzed CO_2 closed-cycle gas turbine along with operating points required for the analysis is presented in Fig. 1. Turbocompressor (TC) increases CO_2 pressure and delivers it to Heater (through the Regenerator). In the Heater occurs main CO_2 heating and after Heater CO_2 with the highest temperature in the process is delivered to Turbine (TU). For the heating purposes in the Heater can be used any fuel or any heat type (as for an example, it can be used waste heat from some other processes). After Heater, CO_2 expands inside the Turbine and after the Turbine is delivered to Regenerator.

The Regenerator is a vital part of the whole process – without it, the efficiency of this process will be unacceptable low (shown later in the analysis) [10]. In the Regenerator CO_2 with the higher temperature (after Turbine) is used for heating of CO_2 after Turbocompressor (which has lower temperature). Therefore, Regenerator decreases the amount of heat transferred to CO_2 in the Heater. After heat transfer in the Regenerator, CO_2 is delivered to Cooler which decreases its temperature to the Turbocompressor inlet temperature. For the cooling purposes can also be used any type of cooling medium. After Cooler, the whole process is continuously repeated. As can be seen already from the presented scheme, the whole process did not have any connection with the environment. However, the ambient parameters (dominantly the ambient temperature) can significantly influence exergy efficiencies and exergy destructions of all the components from the observed process.

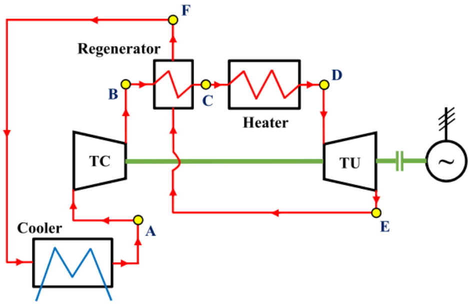

Fig. 1. *Scheme of the analyzed CO_2 closed-cycle gas turbine along with operating points required for the analysis*

General overview of the analyzed CO_2 closed-cycle gas turbine process (*T-s* diagram) is presented in Fig. 2. It should be noted that for the observed process pressure drops in Regenerator, Heater and Cooler (process losses) are neglected, but they are usually lower in comparison with the same components from open-cycle gas turbines [11]. Therefore, the whole process operates between two constant pressures (p_A and p_B). For the Turbocompressor, ideal (isentropic) compression is marked with operating points A-Bis, while real (polytropic) compression is marked with operating points A-B. Identical principle is used for the Turbine where ideal (isentropic) expansion is marked with operating points D-Eis, while real (polytropic) expansion is marked with operating points D-E, Fig. 2.

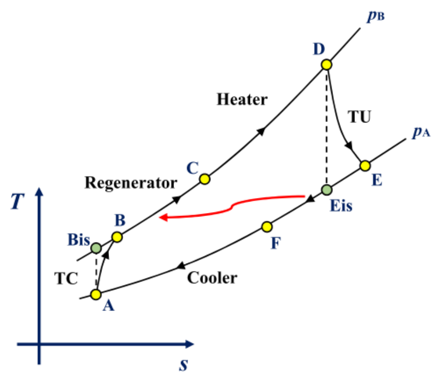

Fig. 2. *T-s diagram of the analyzed CO_2 closed-cycle gas turbine (general overview) along with operating points required for the analysis (in accordance with Fig. 1)*

3. Equations for the energy and exergy evaluation of the observed system

3.1. Overall energy and exergy equations

Energy analysis of any system or a control volume is related to the first law of thermodynamics [12-14]. The general energy balance equation valid for any system or a control volume (while disregarding potential and kinetic energies) [15, 16] can be defined as:

$$\dot{Q}_{\mathrm{IN}} + P_{\mathrm{IN}} + \sum \dot{En}_{\mathrm{IN}} = \dot{Q}_{\mathrm{OUT}} + P_{\mathrm{OUT}} + \sum \dot{En}_{\mathrm{OUT}}, \quad (1)$$

where P in (kW) is used or produced mechanical power and $\dot{Q}$ in (kW) is energy heat transfer. Index IN is related to the inlet (input), while index OUT is related to the outlet (output). $\dot{En}$ in (kW) is a total energy of operating medium flow [17] which is defined as:

$$\dot{En} = \dot{m} \cdot h, \quad (2)$$

where $\dot{m}$ in (kg/s) is operating medium mass flow rate and h in (kJ/kg) is operating medium specific enthalpy.

Exergy analysis of any system or a control volume is related to the second law of thermodynamics [18]. The general exergy balance equation valid for any system or a control volume is defined according to [19] as:

$$\dot{X}_{\mathrm{heat}} + P_{\mathrm{IN}} + \sum \dot{Ex}_{\mathrm{IN}} = P_{\mathrm{OUT}} + \sum \dot{Ex}_{\mathrm{OUT}} + \dot{Ex}_{\mathrm{D}}, \quad (3)$$

where $\dot{Ex}_{\mathrm{D}}$ is exergy destruction in (kW) and $\dot{X}_{\mathrm{heat}}$ in (kW) is the exergy transfer by heat at the temperature T, which can be defined by an equation [20]:

$$\dot{X}_{\mathrm{heat}} = \sum (1 - \frac{T_0}{T}) \cdot \dot{Q}. \quad (4)$$

In Eq. 4, T is a temperature in (K), and index 0 is related to the state of the ambient in which system or a control volume operates. A total exergy of operating medium flow $\dot{Ex}$ in (kW) can be calculated according to [21]:

$$\dot{Ex} = \dot{m} \cdot \varepsilon, \quad (5)$$

where ε is specific exergy of operating medium in (kJ/kg) [22, 23].

3.2. Equations and principles for the energy and exergy evaluation of the analyzed CO_2 closed-cycle gas turbine

Equations and principles for the energy and exergy evaluation of the observed CO_2 closed-cycle gas turbine are based on operating points presented in Fig. 1 and Fig. 2.

Energy analysis equations

Equations for ideal and real mechanical power of Turbocompressor (used mechanical power), Turbine (produced mechanical power) and Useful power (the difference between produced and used mechanical power) as well as transferred heat to CO_2 (in Regenerator and Heater) and transferred heat from CO_2 to cooling medium in Cooler are presented in Table 1.

Table 1. *Power and transferred heat for the observed process*

Component	Real (polytropic) mechanical power	Eq.	Ideal (isentropic) mechanical power	Eq.
Turbocompressor (TC)	$P_{\mathrm{TC,RE}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{B}} - h_{\mathrm{A}})$	(6)	$P_{\mathrm{TC,ID}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{Bis}} - h_{\mathrm{A}})$	(9)
Turbine (TU)	$P_{\mathrm{TU,RE}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{D}} - h_{\mathrm{E}})$	(7)	$P_{\mathrm{TU,ID}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{D}} - h_{\mathrm{Eis}})$	(10)
Useful	$P_{\mathrm{Useful,RE}} = P_{\mathrm{TU,RE}} - P_{\mathrm{TC,RE}}$	(8)	$P_{\mathrm{Useful,ID}} = P_{\mathrm{TU,ID}} - P_{\mathrm{TC,ID}}$	(11)

Component	Transferred heat	Eq.
Regenerator	$\dot{Q}_{\mathrm{Regenerator}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{C}} - h_{\mathrm{B}})$	(12)
Heater	$\dot{Q}_{\mathrm{Heater}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{D}} - h_{\mathrm{C}})$	(13)
Cooler	$\dot{Q}_{\mathrm{Cooler}} = \dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{F}} - h_{\mathrm{A}})$	(14)

The energy efficiency of the whole observed cycle (with the Regenerator) is:

$$\eta_{\mathrm{n,cycle,Reg.}} = \frac{P_{\mathrm{TU,RE}} - P_{\mathrm{TC,RE}}}{\dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{D}} - h_{\mathrm{C}})}. \quad (15)$$

The importance of Regenerator operation in the observed closed-cycle can be seen on the best possible way if the energy efficiency of the observed process (Eq. 15) is compared with the energy efficiency of the same process, but without Regenerator (in such process CO_2 after Turbocompressor will be delivered directly to Heater and after the expansion in the Turbine CO_2 will be delivered directly to Cooler). It should be noted that Regenerator operation influences heat transfer in the Heater and Cooler, it did not have any influence on the Turbocompressor and Turbine operation. The energy efficiency of the whole observed cycle (without the Regenerator) is:

$$\eta_{\mathrm{n,cycle,Without\ Reg.}} = \frac{P_{\mathrm{TU,RE}} - P_{\mathrm{TC,RE}}}{\dot{m}_{\mathrm{CO_2}} \cdot (h_{\mathrm{D}} - h_{\mathrm{B}})}. \quad (16)$$

Exergy analysis equations

Due to insufficient data of heating medium in Heater and of cooling medium in Cooler, exergy analysis is performed for three components of the observed process – Turbocompressor, Turbine and Regenerator. In Table 2 are presented final exergy analysis equations which define exergy destructions (exergy losses) and exergy efficiencies of each observed component.

Table 2. *Exergy analysis equations for the Turbocompressor, Turbine and Regenerator*

Component	Exergy destruction	Eq.
Turbocompressor (TC)	$\dot{Ex}_{\mathrm{D,TC}} = \dot{m}_{\mathrm{A}} \cdot \varepsilon_{\mathrm{A}} + P_{\mathrm{TC,RE}} - \dot{m}_{\mathrm{B}} \cdot \varepsilon_{\mathrm{B}}$	(17)
Turbine (TU)	$\dot{Ex}_{\mathrm{D,TU}} = \dot{m}_{\mathrm{D}} \cdot \varepsilon_{\mathrm{D}} - P_{\mathrm{TU,RE}} - \dot{m}_{\mathrm{E}} \cdot \varepsilon_{\mathrm{E}}$	(18)
Regenerator	$\dot{Ex}_{\mathrm{D,Reg.}} = \dot{m}_{\mathrm{E}} \cdot \varepsilon_{\mathrm{E}} + \dot{m}_{\mathrm{B}} \cdot \varepsilon_{\mathrm{B}} - \dot{m}_{\mathrm{C}} \cdot \varepsilon_{\mathrm{C}} - \dot{m}_{\mathrm{F}} \cdot \varepsilon_{\mathrm{F}}$	(19)

Component	Exergy efficiency	Eq.
Turbocompressor (TC)	$\eta_{\mathrm{x,TC}} = \frac{\dot{m}_{\mathrm{B}} \cdot \varepsilon_{\mathrm{B}} - \dot{m}_{\mathrm{A}} \cdot \varepsilon_{\mathrm{A}}}{P_{\mathrm{TC,RE}}}$	(20)
Turbine (TU)	$\eta_{\mathrm{x,TU}} = \frac{P_{\mathrm{TU,RE}}}{\dot{m}_{\mathrm{D}} \cdot \varepsilon_{\mathrm{D}} - \dot{m}_{\mathrm{E}} \cdot \varepsilon_{\mathrm{E}}}$	(21)
Regenerator	$\eta_{\mathrm{x,Reg.}} = \frac{\dot{m}_{\mathrm{C}} \cdot \varepsilon_{\mathrm{C}} - \dot{m}_{\mathrm{B}} \cdot \varepsilon_{\mathrm{B}}}{\dot{m}_{\mathrm{E}} \cdot \varepsilon_{\mathrm{E}} - \dot{m}_{\mathrm{F}} \cdot \varepsilon_{\mathrm{F}}}$	(22)

4. CO_2 operating parameters required for the analysis

Operating parameters of CO_2 (temperature, pressure and mass flow rate) in each operating point of the observed process from Fig. 1 and Fig. 2, are presented in Table 3. Specific enthalpies, specific entropies and specific exergies of CO_2 are calculated in each operating point by using NIST REFPROP 9.0 software [24].

Table 3. *CO_2 operating parameters in each operating point*

O.P.*	Temperature (°C)	Pressure (MPa)	Specific enthalpy (kJ/kg)	Specific entropy (kJ/kg·K)	Specific exergy (kJ/kg)	Mass flow rate (kg/s)
A	33.00	7.5	381.36	1.5937	217.07	50
Bis	100.88	22.5	415.77	1.5937	251.48	
B	102.57	22.5	419.59	1.6039	252.27	
C	321.48	22.5	749.84	2.3123	371.29	
D	550.00	22.5	1033.40	2.7159	534.55	
Eis	406.45	7.5	875.59	2.7159	376.73	
E	420.04	7.5	891.37	2.7389	385.65	
F	130.23	7.5	561.13	2.1218	239.39	

* O.P. = Operating Point (in accordance with Fig. 1 and Fig. 2)

In Table 3 are presented CO_2 operating parameters of the Turbocompressor (TC) and Turbine (TU) for both real and ideal processes. It can be observed that in an ideal (isentropic) process specific entropies are the same for operating points A-Bis (TC) and for operating points D-Eis (TU).

In Table 3 are presented specific exergies of CO_2 in each operating point at the base ambient state. For this analysis, the base ambient state is defined by the ambient pressure of 0.1 MPa and the ambient temperature of 25 °C.

5. Results and discussion

5.1. Energy analysis results

Turbocompressor (TC) is a mechanical power consumer. In ideal (isentropic) compression process, TC will use lower mechanical power (1720.5 kW) than in real (polytropic) compression process (1911.67 kW), Fig. 3.

Gas turbine (TU) is a mechanical power producer. In ideal (isentropic) expansion process TU will develop higher power (7890.5 kW) in comparison to real (polytropic) expansion process (7101.45 kW). Regardless if the expansion process is real or ideal, CO_2 will expand through the TU between the same pressures (between p_B and p_A), Fig. 2. Ideal (isentropic) compression or expansion processes assume always the same CO_2 specific entropy.

Mechanical power produced by TU will firstly be used for the TC drive. The rest of produced mechanical power by TU will be used for any mechanical power consumer drive (Useful power). If the TU and TC processes are ideal (isentropic), produced Useful power will be equal to 6170 kW, while in the real (polytropic) TU and TC processes, produced Useful power is equal to 5189.78 kW, Fig. 3.

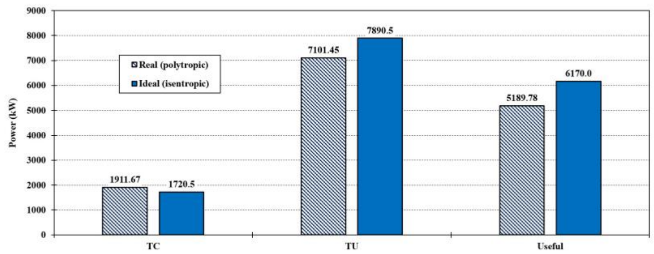

Fig. 3. *Real (polytropic) and ideal (isentropic) mechanical power of Turbocompressor (TC), Turbine (TU) and Useful power*

Transferred heat to CO_2 in Regenerator and Heater is equal to 16512.22 kW and 14178.12 kW, respectively (calculated by using Eq. 12 and Eq. 13, Table 1), Fig. 4. It is interesting and important to note that transferred heat to CO_2 in Regenerator is higher in comparison to Heater, what indicates significant influence of Regenerator on the delivered heat savings in Heater. Transferred heat from CO_2 to the cooling medium in Cooler (calculated by using Eq. 14, Table 1) is equal to 8988.33 kW.

Fig. 4. *Transferred heat to the cooling medium (in Cooler) and to operating medium (CO_2) in Regenerator and Heater*

Energy efficiency of the whole observed cycle with the included Regenerator is 36.6% (calculated by using Eq. 15). Obtained energy efficiency of the CO_2 closed-cycle gas turbine is acceptable and is in the range with energy efficiencies of other open-cycle gas turbines. However, Regenerator inclusion in the observed closed-cycle is essential, without Regenerator (or in the case of Regenerator malfunction) energy efficiency of the analyzed CO_2 closed-cycle will be unacceptably low and equal to 16.91% (calculated by Eq. 16).

5.2. Exergy analysis results

Exergy analysis results at the base ambient state show that Turbocompressor (TC) has the lowest exergy destruction equal to 151.67 kW, while the Regenerator has the highest exergy destruction equal to 1362 kW, Fig. 5. In the exergy analysis of many systems [25, 26] is shown that, in general, exergy destruction and exergy efficiency of the components are reverse proportional. For the analyzed CO_2 closed-cycle gas turbine, Regenerator which has the highest exergy destruction of all observed components has also the lowest exergy efficiency (81.38%). However, reverse proportionality is not valid for the TC (which has the lowest exergy destruction, but not the highest exergy efficiency). The highest exergy efficiency of all observed components has Turbine (TU) equal to 95.39% (TC has a slightly lower exergy efficiency equal to 92.07%).

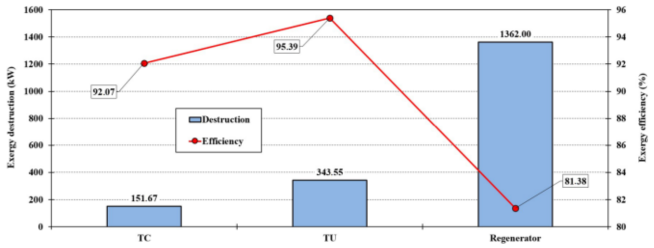

Fig. 5. *Exergy destruction and exergy efficiency of TC, TU and Regenerator (base ambient state)*

Regenerator influence on the observed CO_2 closed-cycle gas turbine is crucial. Therefore, for the Regenerator is investigated how the change in the ambient temperature influences its exergy destruction and exergy efficiency. Analyzed CO_2 closed-cycle gas turbine did not have any connection with the environment, but change in the ambient conditions (primarily the change in the ambient temperature) can have a notable influence on the exergy analysis parameters. During the ambient temperature change, the ambient pressure remains the same as at the base state (0.1 MPa).

Regenerator will achieve the highest exergy efficiency and the lowest exergy destruction (equal to 83.98% and 1270.5 kW) if the whole system operates at the lowest observed ambient temperature (in this analysis the lowest observed ambient temperature is 5 °C), Fig. 6. Increase in the ambient temperature will result with an increase in Regenerator exergy destruction and with a simultaneous decrease in Regenerator exergy efficiency.

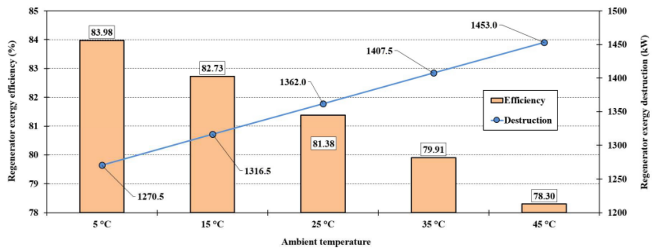

Fig. 6. *Exergy destruction and exergy efficiency of Regenerator during the change in the ambient temperature*

Further analysis of the observed CO_2 closed-cycle gas turbine will be based on finding optimal operating parameters in all operating points by using various artificial intelligence methods [27-30].

6. Conclusions

In the presented research is performed energy and exergy evaluation of CO_2 closed-cycle gas turbine. It is calculated main operating parameters of the whole observed cycle, as well as of each of its constituent components. For the Regenerator, as a crucial component of the analyzed process, is also performed variation in the ambient temperature. The main conclusions are:

- Useful real mechanical power of the observed process which will be used for mechanical power consumer drive is 5189.78 kW.
- Transferred heat to CO_2 in Regenerator is higher in comparison to Heater, what indicates significant influence of Regenerator on the delivered heat savings in Heater.
- Heat Regenerator is a crucial component of the observed CO_2 closed-cycle gas turbine. With Regenerator, overall cycle energy efficiency is equal to 36.6%, while without Regenerator (or if the Regenerator malfunction occurs) overall cycle energy efficiency will be equal to only 16.91%.
- From the exergy aspect, Turbocompressor (TC) has the lowest exergy destruction equal to 151.67 kW (when compared to Turbine and Regenerator), while the Turbine (TU) has the highest exergy efficiency equal to 95.39% of all three observed components.
- The ambient temperature variation shows that Regenerator will have the highest exergy efficiency and the lowest exergy destruction at the lowest observed ambient temperature of 5 °C (equal to 83.98% and 1270.5 kW, respectively).

7. Acknowledgment

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