NUMERICAL ANALYSIS OF REAL OPEN CYCLE GAS TURBINE

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Abstract: *The paper presents a thermodynamic analysis of gas turbine with real open cycle. Gas turbine operates in combined heat and power (CHP) system. Analysis is provided by using measured operating parameters of operating mediums (air and combustion gases) in all required operating points. Cumulative real turbine developed power amounts 78611.63 kW. In the whole gas turbine process, the highest losses occur in combustion chambers during the heat supply process and amounts 13689.24 kW. Turbine power losses are equal to 7976.22 kW, while the turbo-compressor power losses amounts 4774.24 kW. While taking into account all analyzed gas turbine components, the highest efficiency of 90.79% has turbine, followed by combustion chambers which efficiency is equal to 89.01%. Turbo-compressor efficiency amounts 88.59% and the whole gas turbine cycle has efficiency equal to 33.15%.*

KEYWORDS: GAS TURBINE, REAL OPEN CYCLE, NUMERICAL ANALYSIS, LOSSES, EFFICIENCY

1. Introduction

Gas turbines are today widely used in various power plants or cogeneration plants for simultaneous heat and power production [1], [2]. Investigation and analysis of many gas turbines and its processes are performed in a lot of scientific papers [3].

Energy and exergy analyses are usually used for the investigation of complete power plants or its components in order to obtain efficiencies and losses of the entire plant and each plant component [4], [5]. Such analyses are also very applicable for the gas turbines and all of its components [6].

This paper presents a thermodynamic analysis of gas turbine with real open cycle which operates in combined heat and power system. Real gas turbine cycles involve losses at each analyzed gas turbine component. Power distribution, distribution of delivered and released heat, losses and efficiencies of each constituent component and the entire gas turbine were calculated and discussed.

2. Real open cycle gas turbine process

Main scheme of a gas turbine with real open cycle is presented in Fig. 1, while the temperature-specific entropy diagram of this process is shown in Fig. 2.

Beginning of gas turbine operation from dead-state is ensured with starting electro-motor [7]. In the open gas turbine cycle turbo-compressor takes air from the atmosphere and increases its pressure. Air with increased pressure is then delivered to combustion chambers. In combustion chambers is injected a certain amount of high-quality fuel, fuel is mixed with air after which combustion occurred. Produced combustion gases leave combustion chambers and enter into the turbine. At the combustion chambers outlet (turbine inlet) produced combustion gases has the highest temperature (the highest gas turbine cycle temperature) - point 3, Fig. 1 and Fig. 2. Combustion gases expanded through the turbine and after expansion, they are released from the gas turbine cycle to the atmosphere. One part of produced turbine cumulative power (usually about 50%) is used for a turbo-compressor drive, while with the other part of cumulative produced power is driven any power consumer (usually electric generator).

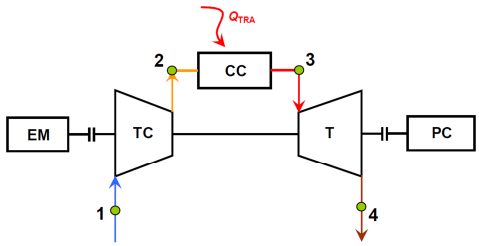

Fig. 1. *Main scheme of the gas turbine with real open cycle (EM = Electro-Motor; TC = Turbo-Compressor; CC = Combustion Chambers; T = Turbine; PC = Power Consumer)*

Real gas turbine cycle includes losses at each gas turbine component [8]. Turbo-compressor compresses air from the atmosphere by polytropic compression, which included compression losses if compared to ideal isentropic compression. Heat supply process in combustion chambers has several heat transfer losses and is characterized with a pressure drop. Real combustion gases expansion process on the turbine is polytropic which included expansion losses if compared to ideal isentropic expansion. Heat release from the real gas turbine process happens at the higher pressure in comparison with atmospheric pressure. For the analyzed gas turbine cycle, mentioned losses are included in measured operating parameters of air and combustion gases in all operating points.

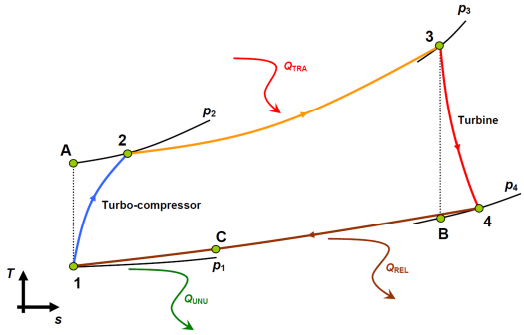

Fig. 2. *T-s diagram of the real open gas turbine cycle with all losses included*

3. Gas turbine with real open cycle numerical analysis – main equations

Most of the equations for the gas turbine with real open cycle analysis were found in [7] and [8]. For each operating point of real open gas turbine cycle, specific enthalpy of operating medium (air or combustion gases) can be calculated as:

$$h = c_p \cdot T \tag{1}$$

where c_p is the specific heat capacity of operating medium at constant pressure and T is current operating medium temperature. For real open cycle gas turbine, specific heat capacity at constant pressure (c_p) is a function of current operating medium temperature.

Specific heat capacity at constant pressure of air is calculated according to [9] by using an equation:

$$c_{p,air}(T) = 1.0484 - 0.0003837 \cdot T + \\ + \frac{9.45378}{10^7} \cdot T^2 - \frac{5.49031}{10^{10}} \cdot T^3 + \frac{7.92981}{10^{14}} \cdot T^4 \tag{2}$$

while for combustion gases (cg), specific heat capacity at constant pressure is also calculated according to [9] by using an equation:

$$c_{p,cg}(T) = 0.936087 + \frac{0.010749}{10^2} \cdot T + \\ + \frac{0.0172103}{10^5} \cdot T^2 - \frac{0.07247}{10^9} \cdot T^3 \quad (3)$$

In equations Eq. 2 and Eq. 3 temperature T must be inserted in (K) to obtain c_p of air or combustion gases in (kJ/kg·K). For the analyzed gas turbine are used measured temperatures in all operating points from Fig. 1 and Fig. 2, therefore specific enthalpies of operating medium can be easily calculated.

By using Fig. 1 and Fig. 2, the operating parameters of the real open cycle gas turbine are:

- Turbo-compressor real power:

$$P_{TC} = \dot{m}_{air} \cdot (h_2 - h_1) = \dot{m}_{air} \cdot (T_2 \cdot c_{p,2} - T_1 \cdot c_{p,1}) \quad (4)$$

Temperature of operating medium (air) after ideal (isentropic) compression is calculated by using an equation:

$$T_A = T_1 \cdot \left(\frac{p_2}{p_1} \right)^{\frac{\kappa_{air}-1}{\kappa_{air}}} \quad (5)$$

where κ_{air} is according to [10] equal to 1.4.

- Turbo-compressor ideal (isentropic) power:

$$P_{TC,IS} = \dot{m}_{air} \cdot (h_A - h_1) = \dot{m}_{air} \cdot (T_A \cdot c_{p,A} - T_1 \cdot c_{p,1}) \quad (6)$$

- Turbo-compressor power losses:

$$P_{TC,PL} = P_{TC} - P_{TC,IS} = \dot{m}_{air} \cdot (h_2 - h_A) \quad (7)$$

- Turbo-compressor efficiency:

$$\eta_{TC} = \frac{P_{TC,IS}}{P_{TC}} = \frac{h_A - h_1}{h_2 - h_1} = \frac{T_A \cdot c_{p,A} - T_1 \cdot c_{p,1}}{T_2 \cdot c_{p,2} - T_1 \cdot c_{p,1}} \quad (8)$$

- Combustion gases mass flow is the sum of air mass flow through a turbo-compressor and fuel mass flow (F) in combustion chambers:

$$\dot{m}_{cg} = \dot{m}_{air} + \dot{m}_F \quad (9)$$

- Turbine real cumulative power:

$$P_T = \dot{m}_{cg} \cdot (h_3 - h_4) = \dot{m}_{cg} \cdot (T_3 \cdot c_{p,3} - T_4 \cdot c_{p,4}) \quad (10)$$

Temperature of combustion gases after ideal (isentropic) expansion is calculated by using the following equation:

$$T_B = T_3 \cdot \left(\frac{p_4}{p_3} \right)^{\frac{\kappa_{cg}-1}{\kappa_{cg}}} \quad (11)$$

where κ_{cg} is equal to 1.3 according to [10] .

- Turbine ideal (isentropic) cumulative power:

$$P_{T,IS} = \dot{m}_{cg} \cdot (h_3 - h_B) = \dot{m}_{cg} \cdot (T_3 \cdot c_{p,3} - T_B \cdot c_{p,B}) \quad (12)$$

- Turbine power losses:

$$P_{T,PL} = P_{T,IS} - P_T = \dot{m}_{cg} \cdot (h_4 - h_B) = \dot{m}_{cg} \cdot (T_4 \cdot c_{p,4} - T_B \cdot c_{p,B}) \quad (13)$$

- Turbine efficiency:

$$\eta_T = \frac{P_T}{P_{T,IS}} = \frac{h_3 - h_4}{h_3 - h_B} = \frac{T_3 \cdot c_{p,3} - T_4 \cdot c_{p,4}}{T_3 \cdot c_{p,3} - T_B \cdot c_{p,B}} \quad (14)$$

- Real useful power (which can be used for power consumer drive):

$$P_{US} = P_T - P_{TC} \quad (15)$$

- Ideal (isentropic) useful power (which can be used for power consumer drive if the compression and expansion processes were ideal ones):

$$P_{US,IS} = P_{T,IS} - P_{TC,IS} \quad (16)$$

- Chemical energy delivered by fuel in the combustion chambers:

$$Q_{CHE} = \dot{m}_F \cdot LHV \quad (17)$$

where LHV is the fuel lower heating value.

- The amount of heat transferred in combustion chambers:

$$Q_{TRA} = \dot{m}_{cg} \cdot (h_3 - h_2) = \dot{m}_{cg} \cdot (T_3 \cdot c_{p,3} - T_2 \cdot c_{p,2}) \quad (18)$$

- Heat supply losses in the combustion chambers:

$$Q_{HSL} = Q_{CHE} - Q_{TRA} \quad (19)$$

- Heat supply (combustion chambers) efficiency:

$$\eta_{HS} = \frac{Q_{TRA}}{Q_{CHE}} = \frac{\dot{m}_{cg} \cdot (h_3 - h_2)}{\dot{m}_F \cdot LHV} = \frac{\dot{m}_{cg} \cdot (T_3 \cdot c_{p,3} - T_2 \cdot c_{p,2})}{\dot{m}_F \cdot LHV} \quad (20)$$

- The cumulative amount of heat released from the process:

$$Q_{REL} = \dot{m}_{cg} \cdot (h_4 - h_1) = \dot{m}_{cg} \cdot (T_4 \cdot c_{p,4} - T_1 \cdot c_{p,1}) \quad (21)$$

- Useful heat released from the process:

$$Q_{REL,US} = \dot{m}_{cg} \cdot (h_4 - h_C) = \dot{m}_{cg} \cdot (T_4 \cdot c_{p,4} - T_C \cdot c_{p,C}) \quad (22)$$

Useful heat released from the process is the amount of heat which can be used for additional heating purposes. Combustion gases which exit gas turbine can be used for additional heating if its temperature exceeds 160 °C (433.15 K). Temperature of combustion gases equal to 160 °C is represented with point C in Fig. 2. Intensive low-temperature corrosion is the main reason why combustion gases with temperature lower than 160 °C cannot be used for additional heating, so one part of heat will always be released from the process as unused heat.

- Unused released heat:

$$Q_{UNU} = \dot{m}_{cg} \cdot (h_C - h_1) = \dot{m}_{cg} \cdot (T_C \cdot c_{p,C} - T_1 \cdot c_{p,1}) \quad (23)$$

- Gas turbine process overall efficiency:

$$\eta_{GT} = \frac{P_{US}}{Q_{TRA}} = \frac{P_T - P_{TC}}{Q_{TRA}} \quad (24)$$

- Specific fuel consumption:

$$SFC = \frac{\dot{m}_F}{P_{US}} = \frac{\dot{m}_F}{P_T - P_{TC}} \quad (25)$$

4. Measured operating parameters of the real open cycle gas turbine

The operating parameters of real open cycle gas turbine process were found in [9], Table 1. The operating points of the gas turbine process in Table 1 are presented in accordance to Fig. 1 and Fig. 2. The real gas turbine process is characterized with real (polytropic) compression and expansion processes with pressure drops during heat transferring in combustion chambers and during heat releasing from the process. Analyzed real open cycle gas turbine operates in combined heat and power (CHP) system, Turkey.

Table 1. *Operating parameters of real open cycle gas turbine [9]*

Operating point*	Temperature (K)	Pressure (bar)
1	298.15	1.0133
2	615.15	10.5030
3	1287.57	9.9779
4	827.15	1.1171
Air mass flow	119.97 kg/s	
Used fuel	Natural gas	
Fuel lower heating value (*LHV*)	44661 kJ/kg	
Fuel mass flow	2.79 kg/s	
Combustion gases mass flow**	122.76 kg/s	

* According to Fig. 1 and Fig. 2
** According to Eq. 9

5. Numerical analysis results of real open cycle gas turbine

After real (polytropic) compression on the turbo-compressor and after real (polytropic) expansion on the turbine, temperatures of operating medium (air or combustion gases) are higher in comparison with ideal (isentropic) processes, Fig. 3. The air temperature after real compression is 33.58 K higher, while combustion gases temperature after real expansion is 50.33 K higher when compared to air and combustion gases temperature after isentropic compression and expansion.

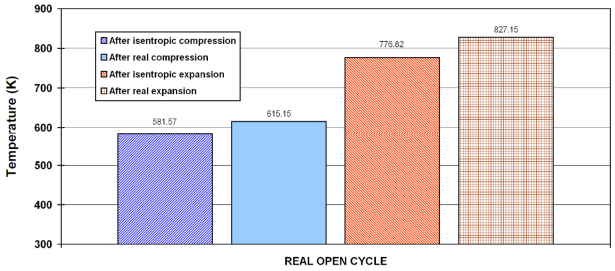

Fig. 3. *The change in real and isentropic temperatures after compression and expansion for the analyzed gas turbine*

A turbo-compressor is power consumer therefore in the real compression process it will use more power (due to losses) than in the ideal process. Real compression process will use 4772.24 kW more power in comparison with the ideal (isentropic) one, Fig. 4.

The turbine is a power producer - in the real expansion process turbine will produce 7976.23 kW less power (due to losses) than in the ideal (isentropic) process.

Real useful power produced by analyzed gas turbine amounts 36768.51 kW and will be used for any power consumer drive. Ideal (isentropic) useful power is calculated according to Eq. 16 and it assumes ideal compression and expansion processes. Isentropic useful power is equal to 49518.98 kW and represents the maximum theoretical power which can be in the ideal situation produced by analyzed gas turbine and delivered to power consumer, Fig. 4.

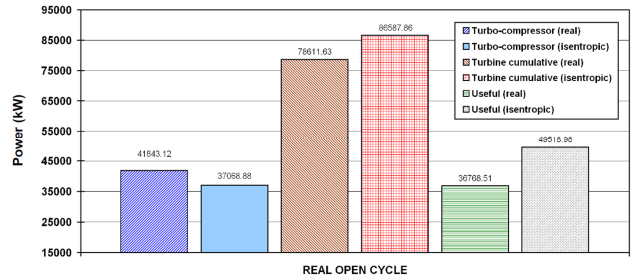

Fig. 4. *The change in real and isentropic power of gas turbine components and the entire gas turbine*

The most expensive element in the analyzed real open cycle gas turbine is used fuel. Chemical energy brought to combustion chambers by fuel represents the highest heat amount delivered in the gas turbine process. Due to several losses in the gas turbine combustion chambers - heat transferred from fuel to operating medium (combustion gases) will be lower in comparison with fuel chemical energy for 13689.24 kW, Fig. 5.

Cumulative heat released from the analyzed gas turbine process amounts 75119.53 kW. This heat amount will be divided on useful released heat (released combustion gases with temperature higher than 160 °C) and unused released heat (released combustion gases with temperature lower than 160 °C). In the analyzed gas turbine process useful released heat, which can be used for any heat consumer drive (or more of heat consumer's drive) amounts 58216.59 kW.

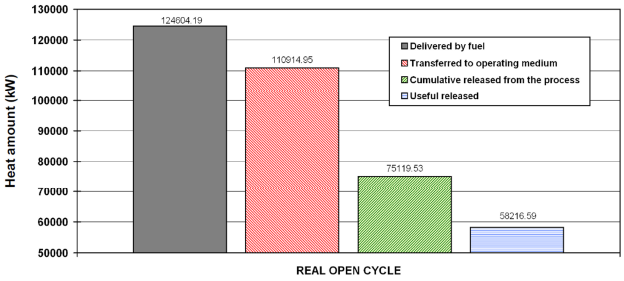

Fig. 5. *Change in heat amounts of the analyzed gas turbine*

Turbo-compressor power losses which amounts 4774.24 kW are represented as a difference between real and ideal (isentropic) power required for turbo-compressor drive. Turbine power losses amounts 7976.22 kW and those turbine losses were calculated as a difference between the ideal (isentropic) and real turbine developed power, Fig. 6.

Heat supply losses are a difference between fuel chemical energy and heat amount transferred to operating medium (combustion gases) in combustion chambers - Eq. 19. Unused released heat from the analyzed gas turbine with real open cycle amounts 16902.94 kW and is calculated by using Eq. 23.

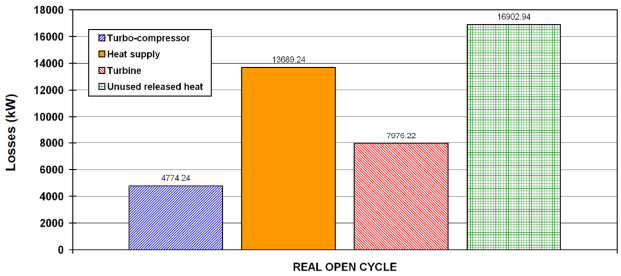

Fig. 6. *Losses of the analyzed gas turbine components*

For the analyzed gas turbine with real open cycle, the highest efficiency has a turbine (90.79%), Fig. 7. The heat supply process has efficiency equal to 89.01%, while the lowest efficiency of analyzed gas turbine components has a turbo-compressor and its efficiency is equal to 88.59%. Real open cycle gas turbine analyzed in this study has an overall efficiency equal to 33.15%, what is usual and expected efficiency for gas turbines used in power plants of any type.

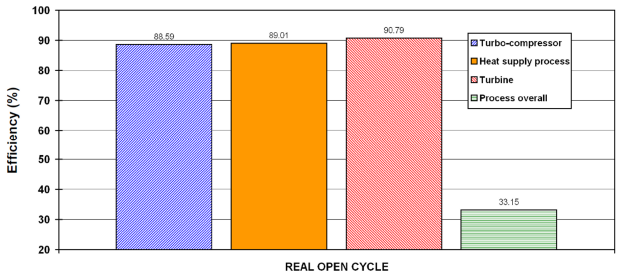

Fig. 7. *Efficiencies of gas turbine components and the entire gas turbine*

In the presented analysis are obtained some additional important operating data of the analyzed real open cycle gas turbine. By using natural gas as a fuel for combustion in combustion chambers, specific fuel consumption, calculated according to Eq. 25, is equal to 273.17 g/kWh. Turbo-compressor power share in the cumulative power developed by the turbine is equal to 53.23%, which leads to a conclusion that a majority of developed cumulative turbine power is used for turbo-compressor drive; only 46.77% of cumulative developed power is used for power consumer driving. Share of unused heat in cumulative released heat is equal to 22.50% therefore the conclusion is that the majority of cumulative released heat from the analyzed gas turbine can be used for the drive of other heat consumers. In the real analyzed gas turbine process turbo-compressor uses 12.88% more power, while the turbine produces 10.15% lower power when compared to ideal (isentropic) compression and expansion processes. Real produced gas turbine useful power which will be used for power consumer drive will be 34.68% higher if the compression and expansion processes are the ideal (isentropic) ones.

6. Conclusions

This paper presents a thermodynamic analysis of gas turbine with real open cycle, which means that all the losses at every gas turbine component were taken into account during analysis. Analyzed gas turbine takes the air from the atmosphere and after expansion process combustion gases were released from the process directly to the atmosphere - two-way atmosphere connection defines gas turbines with open cycle. The main conclusions of the presented analysis are:

- After real compression and expansion processes operating medium temperatures were significantly higher in comparison with the operating medium temperatures after ideal (isentropic) compression or expansion.
- Turbo-compressor power losses amounts 4774.24 kW and its efficiency is equal to 88.59%.
- Turbine power losses amounts 7976.22 kW and its efficiency is equal to 90.79%.
- Heat supply process in combustion chambers resulted with heat losses equal to 13689.24 kW, while the heat supply efficiency (combustion chambers efficiency) is equal to 89.01%.
- Cumulative heat released from the gas turbine process amounts 75119.53 kW. This cumulative heat amount is divided in two parts - first part is useful heat, which can be used for any heat consumer drive and amounts 58216.59 kW, while the second part is unused heat which cannot be used in heat consumers and must be released to the atmosphere. Unused heat amounts 16902.94 kW.
- The whole analyzed gas turbine cycle has an overall efficiency equal to 33.15%.

7. Acknowledgment

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8. References

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