Martina Rauch
Antun Galović
Zdravko Virag

ISSN 1333-1124
eISSN 1849-1391

OPTIMIZATION OF COMBINED BRAYTON-RANKINE CYCLE WITH RESPECT TO THE TOTAL THERMAL EFFICIENCY

Summary

In this paper combined Brayton–Rankine cycle was mathematically simulated. In the Matlab program package an adequate numerical procedure has been developed to determine the maximum thermal efficiency of the combined cycle limited by the temperature of exhaust gases at the entrance of the gas turbine and the temperature of the condensation of water vapour in the steam condenser. Furthermore, additional limitations were introduced: the exhaust gases temperatures at the exit of the gas turbine and dryness fraction at the exit of the steam turbine. Impact of adiabatic flame temperature, temperature difference of the working fluid (water) and the exhaust gases at the pinch point and dryness fraction on change of the overall thermal efficiency of the combined cycle was examined. It was concluded that adiabatic flame temperature has the most impact for the selected intervals of the observed values.

Key words: *Combined Brayton–Rankine cycle, maximum thermal efficiency, optimization*

1. Introduction

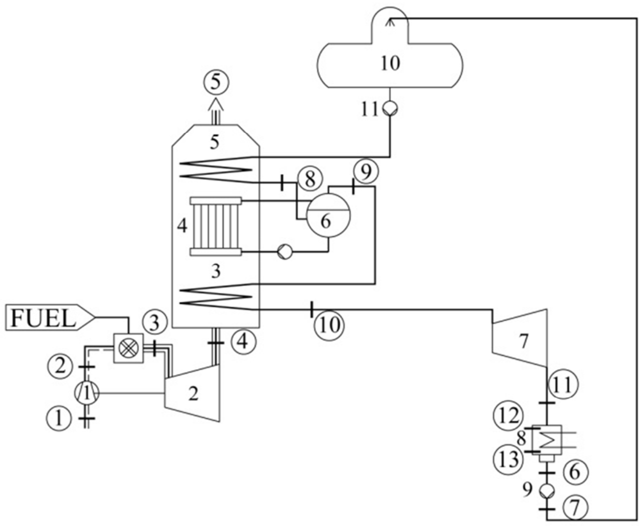
In the fossil fuel based electrical energy production the highest degree of conversion from chemical energy of the fuel into the electrical energy can be achieved by using the combined Brayton-Rankine cycle. From the engineering point of view it is logical to use the benefits of the very desirable characteristics of the steam cycle at high temperatures and utilize the waste heat of the exhaust gases at the gas turbine exit as a heat source in the heat recovery steam generator. This waste heat would usually stay unutilized in the classical gas turbine power plant. [1]

By examining the available literature, a lot of the papers dealing with energy and exergy efficiency of combined cycles are noted. Most researches of the combined cycle thermodynamic efficiency are based on improving the Rankine cycle, while a small number deals with the improvements of gas part of the cycle. There are a number of studies in the field of potential directions of gas turbines development. The turbine blade cooling technology is an object of analyzes presented in [2]. There are also concepts of inlet air cooling [3] and intercooling of the air in compressor [4]. On the other hand, some studies are dealing with steam turbine analysis. In [5] a new approach for a steam turbine simulation development using normalisation method is applied.

Optimization of combined cycle in point of energy and exergy analysis is usually carried out like this: a given input parameter in the optimization process, inter alia, is the temperature of the exhausted gases at the outlet of the gas turbine. That way we do not enter into optimization of Brayton cycle and do not include combination of both cycles in search of whole cycle optimum.

Study given in [6] is aimed at giving a thermodynamic comparison between the optimums of three configurations of heat recovery steam generator (HRSG) operating at exhaust gas temperature from 350 °C to 650 °C. The optimization results show that adding another pressure level allows achieving a higher pressure at the inlet of high pressure turbine, producing more steam quantities, destroying less exergy and finally producing more specific work independently of exhaust gas temperature. This work describes aforementioned optimization procedure including optimization of Rankine cycle with given parameter which is exhaust gases temperature at the exit of the gas turbine without Brayton cycle optimization procedure. In similar research study in [7] the influence of HRSG inlet gas temperature on the steam cycle efficiency is discussed. The result shows increasing HRSG inlet gas temperature until 650 °C leads to increase the thermal efficiency and exergy efficiency of the cycle and after that has less improvement and starts to decrease them. Exergy analysis of each part of HRSG shows that the high pressure-evaporator and high pressure-superheater have the most exergy destruction. Furthermore, in [8] a comprehensive thermodynamic analysis of three different combined cycle power plants based on the first and second law of thermodynamic is conducted. A double pressure and two triple pressure (with and without reheat) HRSG are modeled and the results are compared. The results show that the exhaust gas exergy and the exergy destruction due to heat transfer decrease with the increase in number of pressure levels of steam generation. Also, the increase of pressure levels of steam generation in HRSG increases the heat recovery from the exhaust gas and as a result, the energetic efficiency of the cycle increases. In [9] an advanced exergy analysis of a real combined cycle power plant with supplementary firing is done. A parametric study is presented discussing the sensitivity of various performance indicators to the gas turbine inlet temperature and compressor pressure ratio. It is observed that the thermal and exergy efficiencies increase when gas turbine inlet temperature and compressor pressure ratio rise. Results also show that combustion chamber concentrates most of the exergy destruction (more than 62%). Generally, to increase the power system efficiency, it is necessary to optimize the HRSG because it has a large impact on the overall performance of the combined cycle power plant. HRSG optimization is analyzed in several studies [10]. And interesting performance analysis of an operating power plant is given in [11]. The analysis is performed with actual operating data acquired from power plant control unit. Energy and exergy efficiencies of the each component of the power plant system are calculated and also parametric analysis is performed. After applying first law and second law of thermodynamics, energy and exergy efficiencies of the combined cycle power plant are found as 56% and 50.04% respectively. As well as in [9], it is found that combustion chamber has the most exergy destruction rate among the system components.

The simplest type of combined cycle that is analyzed in this paper is a basic single-pressure cycle, named so because the heat recovery steam generator generates steam for the steam turbine at only one pressure level as shown in Fig. 1. Optimization procedure includes both Brayton and Rankine cycle.



1. Compressor
2. Gas turbine
3. Superheater
4. Evaporator
5. Economizer
6. Drum
7. Steam turbine
8. Condenser
9. Condensate pump
10. Feedwater tank
11. Feedwater pump

Fig. 1 Flow diagram of a single-pressure cycle [1]

2. Mathematical model

The condition of the working fluid is given in 13 characteristic points of the process. The points are marked by an index (i = 1, 2, ...13) respecting the flow of the working fluids where the first point in the cycle, stage 1, is the entrance of the air into the compressor.

Figures 2 and 3 show the *T-s* diagram of Brayton and Rankine cycle with the associated points of the processes. At each point of the process two state variables are given and the rest of the variables are determined by the REFPROP program, which has been adapted to the Matlab user package. Optimization variable parameters used in definition of the mathematical model of combined cycle are compression ratio in Brayton cycle and pressure after the pump in Rankine cycle. The optimization of the combined process with the goal of acquiring the maximum thermal efficiency is carried out at given temperatures: the temperature of the exhaust gases at the entrance of the gas turbine and the temperature of the water vapor condensation in the steam stage of the cycle. Additional limitations were introduced: the exhaust gases temperatures at the exit of the gas turbine and dryness fraction at the exit of the steam turbine. Upon determining state variables of the working fluids at all stages of the cycle, the mass flow of the air and exhaust gases have also been determined, as well as the corresponding rate of the heat flow through heat exchanger, the compressor power, the gas and steam turbine power, the pump power, the thermal efficiency of the Rankine and Brayton cycle and the overall thermal efficiency of the combined cycle.

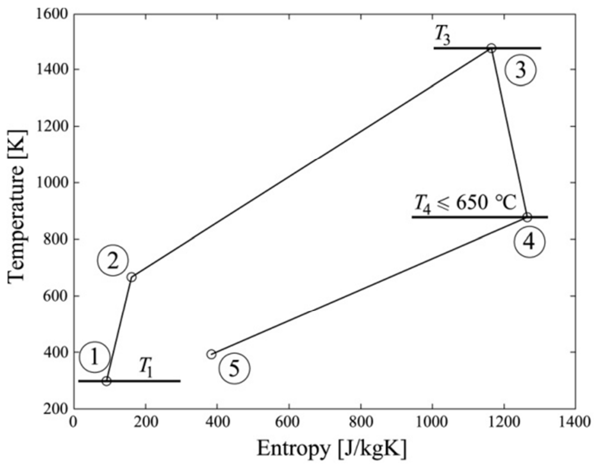


Fig. 2 T-s diagram of Brayton cycle

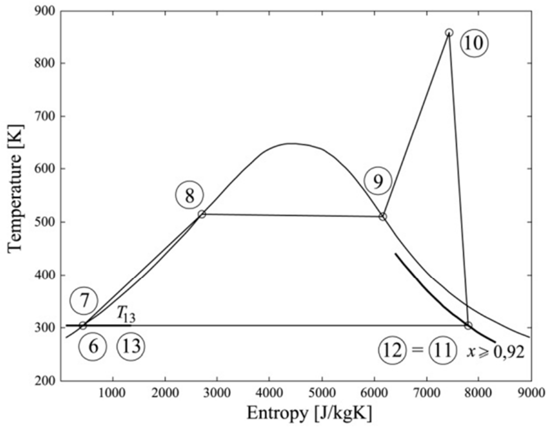


Fig. 3 T-s diagram of Rankine cycle

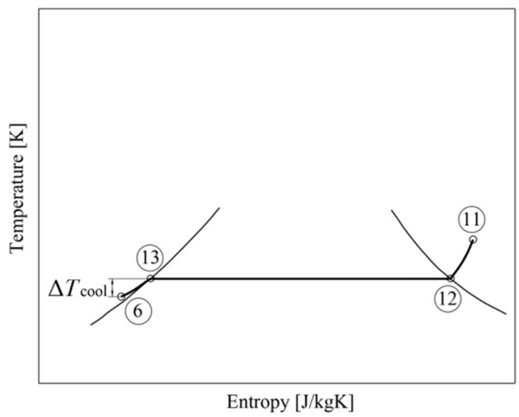


Fig. 4 Detail of process in condenser

Point 11 describes steam condition at the outlet of the steam turbine which can be superheated steam, saturated steam or wet steam. If superheated steam is at the exit of the steam turbine, state in diagram is marked with point 11 and that is the state at the entrance in condenser. Then point 12 is an auxiliary point in condenser and presents saturated steam condition, as shown in Fig. 4. Upon determining the maximum thermal efficiency of combined cycle, steam condition at the outlet of steam turbine is wet steam. Subsequently, point 12 loses its sense and point 11 and 12 show exact the same condition of wet steam, as shown in Fig. 3.

Point 13 presents saturated liquid condition within condenser, while point 6 presents exit of subcooled liquid from condenser. Cooling of the condensate is described with given ΔT_{cool}.

2.1 Modeling of system's components

Compressor

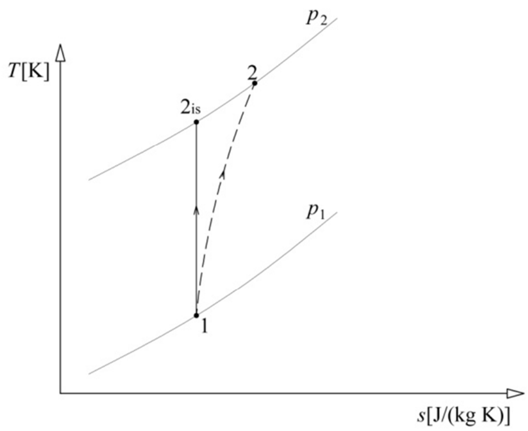


Fig. 5 Isentropic efficiency of compressor [12]

Pressure at the outlet of the compressor is determined by optimization variable parameter r_{p1} by the following expression:

$$p_2 = p_1 \cdot r_{\mathrm{p1}} \tag{1}$$

Optimization variable parameters used in definition of the mathematical model of combined cycle are compression ratio in Brayton cycle and pressure after the pump in Rankine cycle.

$$\eta_{\mathrm{comp}} = \frac{h_{2\_\mathrm{is}} - h_1}{h_2 - h_1} \tag{2}$$

Combustion chamber

With the total combustion of 1 kg/s of methane (CH_4) in combustion chamber exhaust gases are produced, consisting of carbon dioxide, oxygen, nitrogen and water vapour. The energy balance of the combustion chamber is based on the first law of thermodynamics which states [13]:

$$h_2 \cdot q_{\mathrm{m\_AIR}} + LHV = h_3 \cdot q_{\mathrm{m\_EXH.GAS.}} \tag{3}$$

where:

$q_{\mathrm{m\_AIR}}$ – the mass flow rate of air

$q_{\mathrm{m\_EXH.GAS.}} = q_{\mathrm{m\_AIR}} + q_{\mathrm{m\_F}}$ – the mass flow rate of exhaust gases

In order to determine enthalpy in point 3 it is necessary to know the mass flow rate of air and exhaust gases and composition of exhaust gases. As the given quantities are functions of the excess air factor, for which calculation requires composition of exhaust gases, the process is carried out iteratively, using Newton's method.

Gas turbine

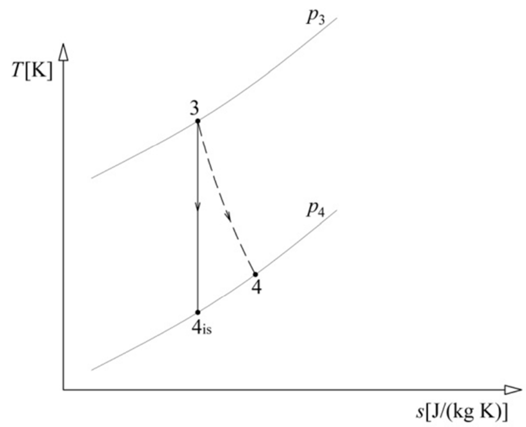


Fig. 6 Isentropic efficiency of gas turbine

$$\eta_{\mathrm{t\_gas}} = \frac{h_4 - h_3}{h_{4\mathrm{\_is}} - h_3} \tag{4}$$

Heat recovery steam generator

When calculating the heat recovery steam generator it is usually assumed that the pinch point is at the entrance of the evaporator (point 8 in this model). Under this assumption temperature, pressure and enthalpy of exhaust gases at the outlet of the economizer was determined by the following expressions:

$$T_{\mathrm{EXH.GAS\_8}} = T_8 + \Delta T_{\mathrm{pinch}} \tag{5}$$

$$p_{\mathrm{EXH.GAS\_8}} = 0{,}5 \cdot (p_1 + p_4) \tag{6}$$

$$h_{\mathrm{EXH.GAS\_8}} = h\,(T_{\mathrm{EXH.GAS\_8}}\,, p_{\mathrm{EXH.GAS\_8}}) \tag{7}$$

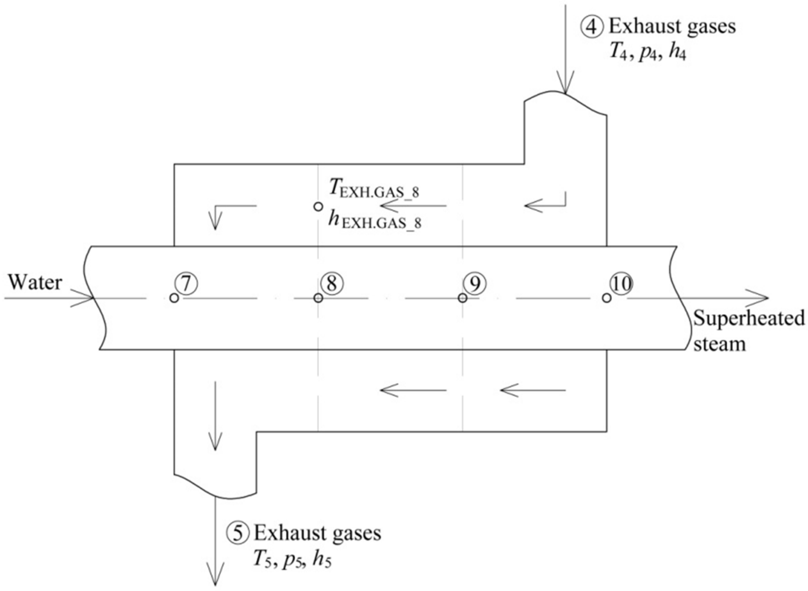


Fig. 7 Scheme of the counterflow heat recovery steam generator

From the energy preservation law [14] set for heat recovery steam generator for the sequence exhausted gases - superheater and evaporator, the mass flow rate of water that evaporates in the heat recovery steam generator is calculated by the following expression:

$$q_{\mathrm{m\_W}} = q_{\mathrm{m\_EXH.GAS}} \frac{h_4 - h_{\mathrm{EXH.GAS\_8}}}{h_{10} - h_8} \tag{8}$$

The enthalpy of the exhaust gases at the exit of the HRSG is calculated by following expression, which is obtained from the first law of thermodynamics applied to economizer - exhausted gases.

$$h_5 = h_{\mathrm{EXH.GAS\_8}} - \frac{q_{\mathrm{m\_W}}}{q_{\mathrm{m\_EXH.GAS}}} (h_8 - h_7) \tag{9}$$

<u>Steam turbine</u>

$$\eta_{\mathrm{t\_steam}} = \frac{h_{11} - h_{10}}{h_{11\_\mathrm{is}} - h_{10}} \tag{10}$$

This model gave restriction of the minimum dryness fraction at the exit of the steam turbine x_{11} at 92% with the aim of reducing erosion of blades caused by water droplets.

<u>Condenser</u>

Temperature of the working media at the outlet of the condenser is determined from the known temperature of saturated liquid T_{13} and temperature drop due to under cooling ΔT_{cool}.

$$T_6 = T_{13} - \Delta T_{\mathrm{cool}} \tag{11}$$

<u>Pump</u>

$$\eta_{\mathrm{p}} = \frac{h_{7\_\mathrm{is}} - h_6}{h_7 - h_6} \tag{12}$$

The pressure at the outlet of the pump p_7 is determined by optimization variable parameter r_{p2} by the following expression:

$$p_7 = p_6 \cdot r_{p2} \tag{13}$$

Thermal efficiency of the Brayton cycle is defined as:

$$\eta_{\mathrm{BR}} = \frac{P_{\mathrm{t\_gas}} - P_{\mathrm{comp}}}{\Phi_{\mathrm{add}}} \cdot 100, \tag{14}$$

where:

$P_{\mathrm{t\_gas}}$ – the gas turbine power

P_{comp} – the compressor power

Φ_{add} – the rate of the heat flow brought to the combustion chamber

Thermal efficiency of the Rankine cycle is defined as:

$$\eta_{\mathrm{RA}} = \frac{P_{\mathrm{t\_steam}} - P_{\mathrm{p}}}{\Phi_{\mathrm{h.exch}} + \Phi_{\mathrm{unutil}}} \cdot 100, \tag{15}$$

where:

$P_{\mathrm{t\_steam}}$ – the steam turbine power

P_{p} – the pump power

$\Phi_{\mathrm{h.exch}}$ – the rate of the heat flow brought to the heat recovery steam generator

Φ_{unutil} – the unutilized rate of the heat flow

Thermal efficiency of the combined cycle is defined as:

$$\eta_{\mathrm{COMB}} = \frac{P_{\mathrm{COMB}}}{\Phi_{\mathrm{add}}} \cdot 100, \tag{16}$$

where:

$$P_{\mathrm{COMB}} = P_{\mathrm{t\_gas}} + P_{\mathrm{t\_steam}} - P_{\mathrm{comp}} - P_{\mathrm{p}} \tag{17}$$

3. Simulation results

The temperature T_3 was varied within the interval 900 ≤ T_3 ≤ 1400 °C, while the temperature T_4 was restricted to 650 °C. The entrance parameters which remained constant during the calculations are shown in Table 1.

Table 1 The parameter values that have remained constant during the calculation

T_1 = 298.15 K	Compressor inlet air temperature
p_1 = 1 bar	Compressor inlet pressure
LHV = 50.05 MJ/kg	Lower heating value of the fuel (methane)
η_{comp} = 0.88	Isentropic efficiency of compressor
$\eta_{\mathrm{t\_gas}}$ = 0.9	Isentropic efficiency of gas turbine
η_{p} = 0.6	Isentropic efficiency of pump

$\eta_{\mathrm{t\_steam}} = 0.92$	Isentropic efficiency of steam turbine
$x_1 = p_3 / p_2 = 0.95$	Pressure drop in combustion chamber
$x_2 = p_1 / p_4 = 0.95$	Pressure drop in heat exchanger on the exhaust gas side
$x_3 = p_9 / p_8 = 0.95$	Pressure drop in evaporator
$x_4 = p_{10} / p_9 = 1$	Pressure drop in superheater
$x_5 = p_8 / p_7 = 0.95$	Pressure drop in economizer
$T_{13} = 30$ °C	Condensing temperature
$\Delta T_{\mathrm{cool}} = T_{13} - T_6 = 1$ °C	Cooling down the condensate
$\Delta T_1 = T_4 - T_{10} = 20$ °C	The exhaust gases temperature difference at the turbine exit and the superheated steam at the superheater exit
$\Delta T_{\mathrm{pinch}} = 10$°C	The temperature difference of the working fluid (water) and the exhaust gases at the pinch point
$x = 0.92$	Dryness fraction at the exit of the steam turbine

Table 2 Optimization results

T_3 [°C]	p_2 [bar]	T_4 [°C]	p_7 [bar]	η_{BR} [%]	η_{RA} [%]	η_{COMB} [%]
900	5.1	575.4	28.9	24.26	27.5	45.64
1000	9.22	539.3	22.49	31.52	25.63	49.36
1100	10.08	593.3	32.77	32.99	27.75	53.15
1200	14.72	593	32.57	36.91	27.45	55.1
1300	19.08	611.2	36.8	39.49	29.98	57.1
1400	29.43	595.7	33.16	42.94	27.08	60.01

Table 2 summarizes the results in the form of the maximum achievable thermal efficiency as a function of T_3. Changing T_3 from 900 °C to 1400 °C, the thermal efficiency of the combined cycle rises from 45.64 % to 60.01 %. Pressure p_7 depends on temperature T_4, as temperature T_4 rises the pressure in steam generator will rise accordingly.

As expected, thermal efficiency of the Brayton cycle is rising with the increase of temperature T_3 because by increasing temperature higher value of enthalpy drop in the gas turbine is reached, which means more power output and rise of thermal efficiency of the Brayton cycle. With the increase of pressure p_7 thermal efficiency of the Rankine cycle also rises. If water is pressurized to a higher pressure, higher power output from the steam turbine is achieved.

3.1 The analysis of the thermal efficiency of the combined cycle

There are multiple parameters that influence the value of thermal efficiency of the combined power plant. By determining the most influential ones we can define which part of the power plant is the most efficient to invest in. In the given analysis three parameters were selected that affect the thermal efficiency of the combined cycle. They are $\Delta T_{\mathrm{pinch}}$, T_3 and dryness fraction x.

The influence of each parameter generally denoted by P is defined by the following equation:

$$\{\eta - \eta_{\mathrm{REF}}\}_{\%} = \sum 100(P_{\max} - P_{\mathrm{REF}}) \frac{\partial \eta}{\partial P} \tilde{P} \tag{18}$$

where the dimensionless values defined as

$$\tilde{P} = \frac{P - P_{\mathrm{REF}}}{P_{\max} - P_{\mathrm{REF}}}, \tag{19}$$

where

P_{REF} - the selected reference value of a considered parameter

$P_{\max}$ - the maximum allowable value of a considered parameter

It is clear that when P changes from P_{REF} to $P_{\max}$, $\tilde{P}$ changes from zero to one.

The following values were chosen for the reference state: $\Delta T_{\mathrm{pinch}}$ = 10 °C, T_3 = 1200 °C, x = 0.92. The value of reference thermal efficiency η_{REF} = 55.9 % was obtained using the values of the parameters listed in Table 1 and at reference values of $\Delta T_{\mathrm{pinch}}$, T_3 and x. Partial derivations in Eq.(18) were calculated numerically according to:

$$\frac{\partial \eta}{\partial P} = \frac{\eta(P = P_{\max}) - \eta(P = P_{\min})}{P_{\max} - P_{\min}}, \tag{20}$$

where following values were used for maximum and minimum values of the parameters: $\Delta T_{\mathrm{pinch\_max}}$ = 15 °C, $\Delta T_{\mathrm{pinch\_min}}$ = 5 °C, $T_{3\_\max}$ = 1400 °C, $T_{3\_\min}$ = 1000 °C; and $x_{\max}$ = 0.96, $x_{\min}$ = 0.88.

By calculating the coefficients in equation (18) the following expression is obtained:

$$\{\eta - \eta_{\mathrm{REF}}\}_{\%} = -0.289\,\widetilde{\Delta T_{\mathrm{pinch}}} + 4.842\,\widetilde{T_3} - 1.85\,\tilde{x} \tag{21}$$

By analyzing the coefficients for the selected range of considered parameters in Eq. (21) it is clear that T_3 has the most influence on the thermal efficiency of the combined cycle. For example, increasing of T_3 from 1200 °C to 1400 °C increases thermal efficiency for 4.842 % and vice versa. The second important parameter is x. By decreasing x from 0.92 to 0.88, the thermal efficiency increases for 1.85 %. $\Delta T_{\mathrm{pinch}}$ has the least effect on thermal efficiency since the possibility for decrease from its reference value is limited. Eq.(21) can also be used as a quick estimation of the thermal efficiency of combined process in vicinity of the reference point. For example, for $\widetilde{\Delta T_{\mathrm{pinch}}}$ = 0.5 ($\Delta T_{\mathrm{pinch}} = 12.5$ °C), $\widetilde{T_3}$ = -0.5 ($T_3 = 1100$ °C) and $\tilde{x}$= 0.5 (x = 0.94) optimization procedure results in $\{\eta - \eta_{\mathrm{REF}}\}_{\%}$= -3.4 %, while Eq.(21) gives $\{\eta - \eta_{\mathrm{REF}}\}_{\%}$= -3.5 % which is fairly close to the exact value.

4. Conclusion

The paper defines the mathematical model of a basic single-pressure combined Brayton-Rankine cycle. An appropriate procedure has been developed to determine the maximum thermal efficiency of the combined cycle limited by the temperature of exhaust gases at the entrance of the gas turbine and the temperature of the condensation of water vapor in the steam condenser. Additional limitations were introduced: the exhaust gases temperatures at the exit of the gas turbine and dryness fraction at the exit of the steam turbine. The maximum thermal efficiency of the combined cycle is achieved at the highest adiabatic flame temperature of 1400 °C and it is 60.01 %. For other given values of temperature T_3 which are 900 °C, 1000 °C, 1100 °C, 1200 °C and 1300 °C it is possible to achieve the maximum thermal efficiency within the 45.64 to 57.1 % interval.

The paper also gives the equation which approximates the change of the maximum thermal efficiency in vicinity of reference point with the change of three chosen parameters

ΔT_{pinch}, T_3 and dryness fraction x. From this equation it is visible that the change of T_3 has the most influence on the thermal efficiency, while the change of ΔT_{pinch} has the least effect.

REFERENCES

[1] Kehlhofer R., Hannemann F., Stirnimann F., Rukes B.: Combined-cycle gas steam turbine power plants, 3rd edition, PennWell, Tulsa, Oklahoma, 1999.

[2] Sanjay, Singh O., Prasad B.N.: Influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle, Applied Thermal Engineering Vol. 28 (2008) 2315-2326

[3] Mohopatra A.K., Sanjay: Thermodynamic assessment of impact of inlet air cooling techniques on gas turbine and combined cycle performance, Energy Vol. 68 (2014) 191-203

[4] Sanjay, Prasad B.N.: Energy and exergy analysis of intercooled combustion turbine based combined cycle power plant, Energy Vol. 59 (2013) 277-284

[5] Tot N., Jovic F., Slavek N.: Simulation model and analysis of the steam turbine subsystem, International Journal of Simulation Modelling Vol.2, No. 1-2 (2003) 43-49

[6] Nadir M., Ghenaietb A.: Thermodynamic optimization of several (heat recovery steam generator) HRSG configurations for a range of exhaust gas temperatures, Energy Vol. 86 (2015) 685–695

[7] Ganjeh Kavir A. et al.: Exergoenvironmental optimization of Heat Recovery Steam Generators in combined cycle power plant through energy and exergy analysis, Energy Conversion and Management Vol. 67 (2013) 27–33

[8] Mansouri M.T. et al.: Exergetic and economic evaluation of the effect of HRSG configurations on the performance of combined cycle power plants, Energy Conversion and Management Vol. 58 (2012) 47–58

[9] Boyaghchi F.A., Molaie H.: Sensitivity analysis of exergy destruction in a real combined cycle power plant based on advanced exergy method, Energy Conversion and Management Vol. 99 (2015) 374–386

[10] Xiang W., Yingying: Performance Improvement of Combined CyclePower Plant Based on the Optimization of the Bottom Cycle and Heat, Journal of Thermal Science Vol.16, No.1 84-89

[11] Ersayina E., Ozgenerb L.: Performance analysis of combined cycle power plants: A case study, Renewable and Sustainable Energy Reviews Vol. 43 (2015) 832–842

[12] Cengel Y. A., Boles M. A.: Thermodynamics, An engineering approach, sixth edition (SI units), McGraw-Hill, New York, 2007.

[13] Galović A.: Termodinamika II, Faculty of mechanical engineering and naval architecture, University of Zagreb, 2010.

[14] Bošnjaković, F.: Nauka o toplini, part I, 4th edition, Tehnička knjiga, Zagreb, 1970.

Submitted: 30.9.2015

Accepted: 20.4.2016

Martina Rauch*
Antun Galović*
Zdravko Virag**
Department of Thermodynamics, Thermal and Process Engineering *
Department of Energy, Power Engineering and Environment**
Faculty of Mechanical Engineering and Naval Architecture,
University of Zagreb
Ivana Lučića 5, 10 000 Zagreb