Optimization of combined Brayton-Rankine cycle with respect to the total thermal efficiency

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ABSTRACT

In this paper combined Brayton–Rankine cycle is 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 vapor 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 is examined. It is concluded that adiabatic flame temperature has the most impact for the selected intervals of the observed values.

KEY WORDS

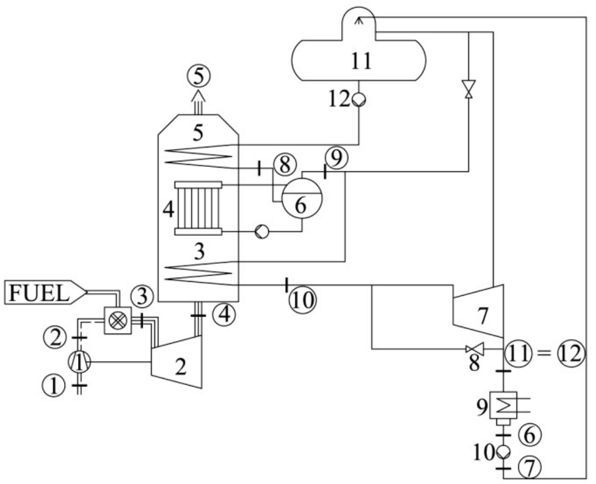
combined Brayton–Rankine cycle, maximum thermal efficiency, optimization

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.

Single-pressure combined cycle

The simplest type of combined cycle that is analyzed in this paper is a basic single-pressure cycle, so called because the heat recovery steam generator generates steam for the steam turbine at only one pressure level as shown in Figure 1.



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

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

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.

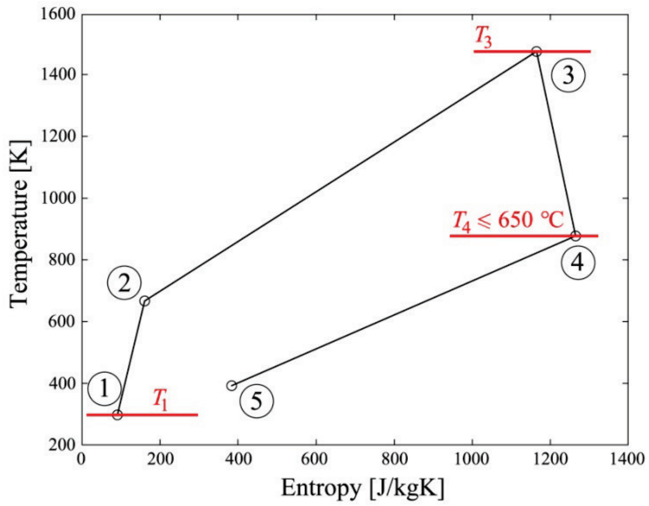


Figure 2. T-s diagram of Brayton cycle

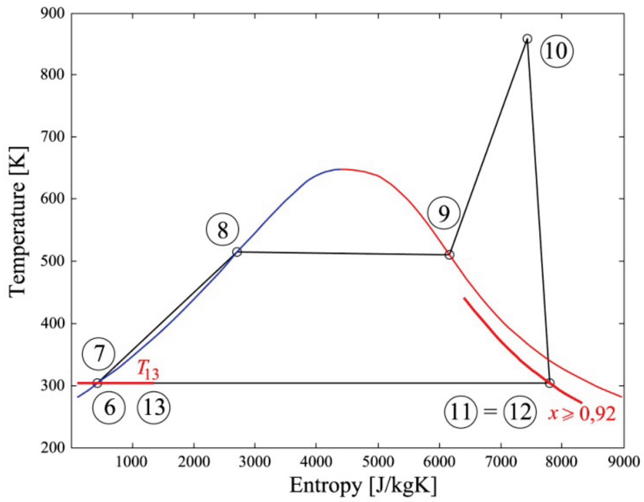


Figure 3. T-s diagram of Rankine cycle

Thermal efficiency of the Brayton cycle is defined as [2]:

$$\eta_{\mathrm{BR}} = \frac{P_{\mathrm{t\_gas}} - P_{\mathrm{comp}}}{\Phi_{\mathrm{add}}} \cdot 100, \tag{1}$$

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 [3]:

$$\eta_{\mathrm{RA}} = \frac{P_{\mathrm{t\_steam}} - P_{\mathrm{p}}}{\Phi_{\mathrm{h.exch}} + \Phi_{\mathrm{unutil}}} \cdot 100, \tag{2}$$

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:

$$\eta_{\mathrm{COMB}} = \frac{P_{\mathrm{COMB}}}{\Phi_{\mathrm{add}}} \cdot 100, \tag{3}$$

where:

$$P_{\mathrm{COMB}} = P_{\mathrm{t\_gas}} + P_{\mathrm{t\_steam}} - P_{\mathrm{comp}} - P_{\mathrm{p}} \tag{4}$$

SIMULATION RESULTS

The temperature T_3 was varied within the interval $900 \leq T_3 \leq 1400$ °C, while the temperature T_4 was restricted to maximal 650 °C. The entrance parameters which remained constant during the calculation 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)
$\eta_{\mathrm{comp}} = 0{,}88$	Isentropic efficiency of compressor
$\eta_{\mathrm{t\_gas}} = 0{,}9$	Isentropic efficiency of gas turbine
$\eta_{\mathrm{p}} = 0{,}6$	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.

THE ANALISYS 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 it most efficient to invest in. In the given analysis three parameters were selected that affect the thermal efficiency of the combined cycle. They are ΔT_{pinch}, T_3 and dryness fraction x.

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

$$\{\eta - \eta_{\text{REF}}\}_{\%} = \sum 100(P_{\max} - P_{\text{REF}}) \frac{\partial \eta}{\partial P} \tilde{P}, \tag{5}$$

where the dimensionless value is defined as

$$\tilde{P} = \frac{P - P_{\text{REF}}}{P_{\max} - P_{\text{REF}}}, \tag{6}$$

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: $\Delta T_{\text{pinch}} = 10$ °C, $T_3 = 1200$ °C, $x = 0{,}92$. The value of reference thermal efficiency $\eta_{\text{REF}} = 55{,}9$ % was obtained using the values of the parameters listed in Table 1 and at reference values of ΔT_{pinch}, T_3 and x. Partial derivations in Eq.(5) were calculated numerically according to:

$$\frac{\partial \eta}{\partial P} = \frac{\eta(P = P_{\max}) - \eta(P = P_{\min})}{P_{\max} - P_{\min}}, \tag{7}$$

where following values were used for maximum and minimum values of the parameters: $\Delta T_{\text{pinch\_max}} = 15$ °C, $\Delta T_{\text{pinch\_min}} = 5$ °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 (5) the following expression is obtained:

$$\{\eta - \eta_{\mathrm{REF}}\}_{\%} = -0{,}289\ \widetilde{\Delta T_{\mathrm{pinch}}} + 4{,}842\ \widetilde{T_3} - 1{,}85\ \tilde{x} \tag{8}$$

By analyzing the coefficients for the selected range of considered parameters in Eq. (8) 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.(8) can be also 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.(8) gives $\{\eta - \eta_{\mathrm{REF}}\}_{\%}$= -3,5 % which is fairly close to the exact value.

CONCLUSION

The paper defines the mathematical model of 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 the reference point with the change of three chosen parameters $\Delta T_{\mathrm{pinch}}$, 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 $\Delta T_{\mathrm{pinch}}$ has the least effect.

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