

# Thermodynamic analysis of the double Brayton cycle with the use of oxy combustion and capture of CO<sub>2</sub>

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**Abstract** In this paper, thermodynamic analysis of a proposed innovative double Brayton cycle with the use of oxy combustion and capture of CO<sub>2</sub>, is presented. For that purpose, the computation flow mechanics (CFM) approach has been developed. The double Brayton cycle (DBC) consists of primary Brayton and secondary inverse Brayton cycle. Inversion means that the role of the compressor and the gas turbine is changed and firstly we have expansion before compression. Additionally, the working-fluid in the DBC with the use of oxy combustion and CO<sub>2</sub> capture contains a great amount of H<sub>2</sub>O and CO<sub>2</sub>, and the condensation process of steam (H<sub>2</sub>O) overlaps in negative pressure conditions. The analysis has been done for variants values of the compression ratio, which determines the lowest pressure in the double Brayton cycle.

**Keywords:** Inverse Brayton cycle, Brayton cycle, gas-steam unit, oxy combustion, CCS, thermodynamic analysis, numerical analysis, CFM

## Nomenclature

$h$  – specific enthalpy, kJ/kg  
 $l$  – specific work, kJ/kg

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$N$	–	power, kW
$\dot{m}$	–	mass flux rate, kg/s
$p$	–	pressure, Pa
$\dot{Q}_{chem}$	–	chemical energy flux, kW
$t$	–	temperature, °C
$T$	–	temperature, K
$W_d$	–	fuel calorific value, kJ/kg
$x$	–	vapor quality

### Greek symbols

$\beta$	–	coefficient of energy consumption (specific work for air separation), kWh/kg
$\Delta T$	–	temperature difference in the heat exchanger, K
$\eta$	–	efficiency, %
$\Pi$	–	compression/expansion ratio

### Subscripts

$a$	–	point of end regeneration
$amb$	–	ambient
$el$	–	electrical
$ex$	–	exhaust
$f$	–	fuel
$i$	–	internal
$ic$	–	internal compressor
$it$	–	internal turbine
$m$	–	mechanical
$o$	–	oxygen
$1s, 2s, \dots$	–	isentropic points of cycle
$1, 2, \dots$	–	real points of cycle
$IBC$	–	inverted Brayton cycle
$DBC$	–	double Brayton cycle
$BC$	–	Brayton cycle
$GT$	–	temperature on the turbine outlet
$TIT$	–	turbine inlet temperature

### Superscripts

$in$	–	point in inversed Brayton cycle
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## 1 Introduction

Along with the civilization development and technical progress, the worlds requirement on the electrical energy grows up. According to it, many problems with electricity appear [7]. At an increasing output of the electrical power the care about the environment it is needed to be taken. All of the

rules were determined by the EU Directive No. 2010/75/UE [10]. It ought to call attention to the fact that the alarming signals from the environment and meteorological effects have been noticed lately. However, the scientists do not live up to one theory about the causes of more often found climate disorders. One fact needs to be emphasized, the bigger and bigger combustion of the fossil fuels can affect the atmosphere, mainly due to ash dust and toxic oxides emissions. Economic growth of countries should be optimized with the sustainable energy policy to develop in environmental, economical and social order. And this is how we describe the sustainability, which main pillar is the even-tempered energy conversion [2,4].

The way to the clean and sustainable energy conversion is through the use of renewable sources of energy and also 'clean' carbon technology. However, it is worth to concentrate on the 'clean' gas technology, according to the large amount of shale gas in Poland. It would help to diversificate the energy sources [16,33]. It is worth to remind that the other sources of energy, such as geothermal source of energy, wind energy and nuclear energy, will not be able to have a big influence on energy market in Poland, which is the enhancement of the electrical energy output. In this situation, the main role may play the gas-steam power units, accommodated to the escape of carbon dioxide ( $\text{CO}_2$ ) [29]. The technologies, which base on high-efficiency power units with usage of oxy combustion and capture of  $\text{CO}_2$ , may stabilize and support the electroenergetical system in reduction of gas emissions according to EU policies. Such a system can be based on the double Brayton cycle with oxycombustion in wet combustor chamber and with water condensation combined with  $\text{CO}_2$  capture.

According to the fact that in this cycle working fluid is a mixture of steam and gas, it is an example of gas-steam turbine cycle which binds the advantages of both, gas and steam, systems. As it was mentioned, gas-steam turbine, operates at the higher temperature compared to steam turbines, which is about 1400 K (in some other cases, the temperature reaches the value of 1700 K) [1,12,17]. Inlet pressure above 4 MPa exceeds the levels of typical gas-turbines [17]. In the condenser the negative pressure is obtained [8]. In the USA the Clean Energy Systems and Siemens Corporation develop large scale oxy combustion power plant. This power plant combines the gas and steam turbine operating condition [1,12]. In the literature, this cycle, with oxy combustion and water injection is the so called water cycle [6,18,20], because of the 90% content of the steam. The rest 10% is  $\text{CO}_2$  from methane combustion [6,11]. More sophisticated

cycle, working on steam and allowing to reach higher efficiency, is the Graz cycle [14,22].

Other cycles utilizing the oxy combustion in traditional combustion chamber, which also provide huge perspectives are: SCO-CC (semi-closed oxy-fuel combustion combined cycle) [25], Matian [21,26], COOPERATE cycle [26,28], COOLENERG cycle [23] In turn, in these cycles, mentioned above, CO<sub>2</sub> predominates as a medium.

It is ought to emphasize that there are some other cycles, which are also worth to be mentioned. These are such systems as:

- ZEITMOP (zero emission ion transport membrane oxygen power) system with Ion Transport Membrane [27],
- AZEP (advanced zero emission power plant) – system with a mixed conductive membrane reactor [18,30],
- CLCC (chemical looping combustion cycle) [18],
- system with natural gas reforming [30],
- system with fuel cells in hybrid cycle (SOFC-GT) [19].

The main aim of this paper is to investigate the thermodynamic parameters of the double Brayton cycle with oxy combustion and with the capture of the CO<sub>2</sub>. This was carried by accessible numerical CFM codes type, by step-by-step modeling of separates apparatus.

## 2 Double Brayton cycle

The whole system consists of the first traditional Brayton cycle and the second inversed one. The ‘inversed Brayton cycle’ means that the order of the compressor and turbine is inversed, so that first there is the expansion of the working-fluid and than its compression [5]. Wet and hot exhaust gases at the atmospheric pressure (from the gas turbine or from the fuel cell) are able to generate extra turbine power, by expanding to the negative pressure (it is the expansion, which is very similar to the expansion in the steam turbine) [19]. The gas mixture, at that level, is still a high temperature fluid and it needs to be colled. It is all done by using a special regenerative heat exchanger (HE). After desuperheating, steam is envisaged to condensate (or just a part of it). The gases pressure is lower than the atmospheric one and that is why the exhaust needs to be compressed in the compressor (C). Compressed gases are directed to the second precooling heat exchanger and

second condenser, which dries them. Additionally, the battery of devices realizing the cycle, consists of the turbine (GT), the heat exchanger (HE) and the compressor (C) (Fig. 1). The cycle described above is the inversed Brayton cycle (IBC) indeed. Temperature-entropy diagram of the double Brayton cycle is presented in Fig. 2, where the characteristic points (1, 2, 2s, ..., 5<sup>in</sup>) are denoted.

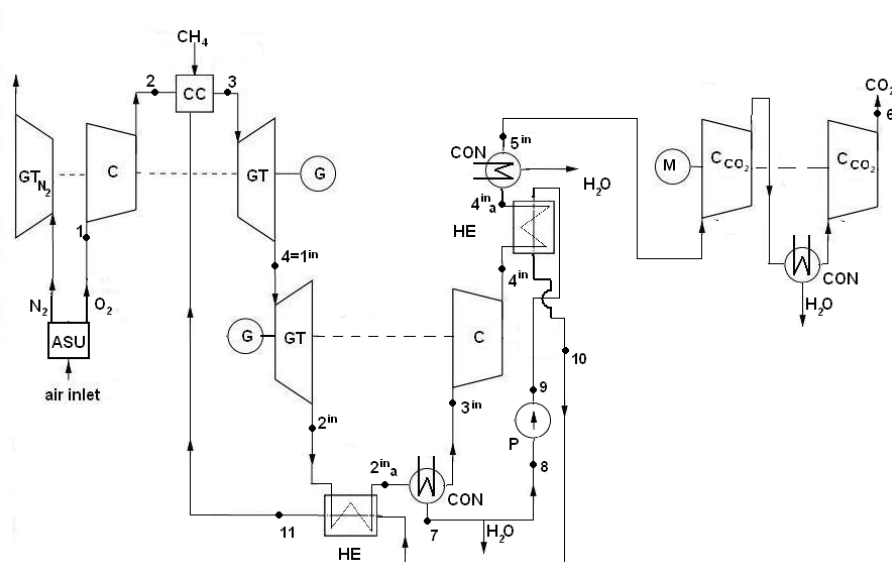


Figure 1. Schematic diagram of the double Brayton cycle with the use of oxy combustion and capture CO<sub>2</sub> (ASU – air separation unit, CC – combustion chamber, C – compressor, GT – gas turbine, HE – heat exchanger, G – electric generator, M – motor, CON – condenser, P – pump, GT<sub>N<sub>2</sub></sub> – additional gas turbine of N<sub>2</sub>, C<sub>CO<sub>2</sub></sub> – compressor of CO<sub>2</sub>).

As it was mentioned, in the condenser (CON), working fluid is separated into water and CO<sub>2</sub>. Next, the 'clean' CO<sub>2</sub> goes to the compressor (C), where it compresses and, after that, it is cooled and condensated. Liquid carbon may be sold or might be used as a fracturing fluid [33]. The main disadvantage of the whole system is the necessity of the air separating station (ASU), to supply combustion chamber in pure oxygen. Moreover, the 95%-oxy-combustion eliminates almost entirely the problem of the NO<sub>x</sub> emission. Additionally, the nitrogen turbine (GT<sub>N<sub>2</sub></sub>) might be used and would be fueled from the oxygen and nitrogen separating station.

Technical realization of the IBC may cause such problems as the increase

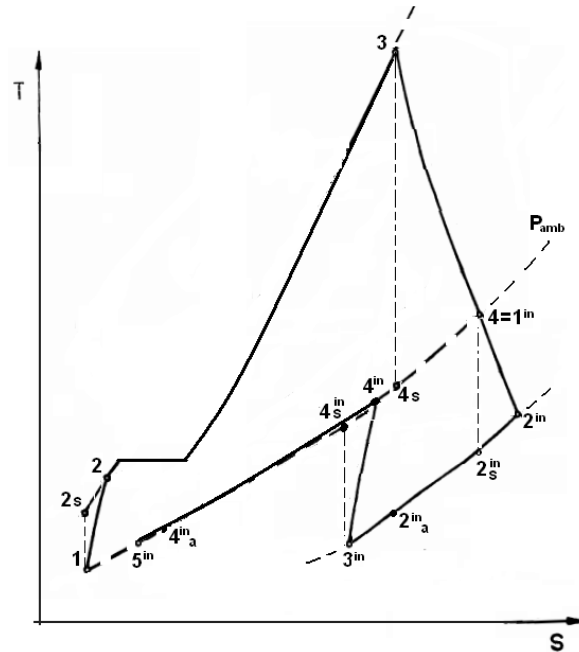


Figure 2. Temperature-entropy diagram of the double Brayton cycle.

of low pressure part size of gas turbine. The IBC turbine, in which the end of expansion runs in negative pressure conditions, what in turn causes that the end part needs to be as big as a low pressure steam turbine. The increase of the diameter is caused by the necessity of axial velocity reduction of the flowing gases through the blading system. Also, the nature of the exchanger operation needs more complex geometry to decrease the loss of movement [19].

### 3 Mathematical model in the CFM code

To analyse the double Brayton cycle with oxy combustion and capture of  $\text{CO}_2$ , the computational flow mechanics code (CFM) code was used [3,4,19,24]. Mathematical CFM models use mass, momentum and energy equation in the '0D' engineering form [3,24]. In this paragraph, computational procedures for each component of the turbo assembly are presented, namely: the compressor, combustion chamber, turbines, pumps and the heat exchangers. The output power and the efficiency calculations are also

presented in the paper.

The CFM code, relying to the internal efficiency of the compressor, also determines the power consumption. The input data include the internal efficiency of the compressor,  $\eta_{ic}$ , the mechanical efficiency of the compressor,  $\eta_{mc}$ , compression ratio and also the mass flow rate,  $\dot{m}$ . Having known the compressor inlet thermodynamic parameters, such as the pressure  $p_1$ , the temperature  $T_1$ , the pressure  $p_2$  can be calculated using the formula presented below [15,24]

$$p_2 = \Pi_{BC} p_1 . \quad (1)$$

For the inversed Brayton cycle there is [19]

$$p_{4^{in}} = \Pi_{IBC} p_{3^{in}} , \quad (2)$$

where  $\Pi_{IBC}$  is the compression ratio in inverse Brayton cycle and  $p_{3^{in}}$ ,  $p_{4^{in}}$  are the pressure at characteristic points ( $3^{in}$ ), ( $4^{in}$ ), respectively. Particular equations of CFM code, used in the inversed Brayton cycle, are similar to those in the traditional Brayton cycle. The total compression ratio is given as

$$\Pi = \Pi_{BC} \Pi_{IBC} . \quad (3)$$

Isentropic exponent,  $\kappa$ , of mixture has been predicted on the basis of thermodynamic tables for each medium and that is what makes the calculations shorter. The ideal compression process is described by the isentropic equation  $s = idem$  [15,24]

$$T_{2,s} = T_1 \left( \frac{p_2}{p_1} \right)^{\frac{\kappa-1}{\kappa}} . \quad (4)$$

This formula allows calculate the theoretical temperature of the end of the compressing process,  $T_{2s}$ . Having given the internal efficiency of the compressor,  $\eta_{ic}$ , it is possible to calculate the real temperature of the compressing process,  $T_2$ . The efficiency of the compressor is described by [15,24]

$$\eta_{ic} = \frac{l_{t1-2s}}{l_{t1-2}} = \frac{h_1 - h_{2s}}{h_1 - h_2} , \quad (5)$$

where  $l_{t1-2s}$  is the specific work of the compression process,  $l_{t1-2}$  is the real work of the compressor, and  $h_1$ ,  $h_2$ ,  $h_{2s}$  are the mixture enthalpy attributable to each point (1), (2), (2s), respectively. To calculate the compressor power and its enthalpy in characteristic points of the whole system, thermodynamic tables of H<sub>2</sub>O and CO<sub>2</sub> need to be used.

The power,  $N_{C-BC}$ , required to drive the oxygen compressor, is given by the relation [15,24]

$$N_{C-BC} = \dot{m}_o \eta_{mc} (h_1 - h_2) , \quad (6)$$

where  $\dot{m}_o$  is the oxygen mass flow rate, and  $\eta_{mc}$  is the compressor mechanical efficiency [19]. The power,  $N_{C-IBC}$ , required to drive the compressor of exhaust gases

$$N_{C-IBC} = \dot{m}_{3in} \eta_{mc} (h_{3in} - h_{4in}) , \quad (7)$$

where  $\dot{m}_{3in}$  is the medium mass flow rate on the compressor inlet in the inversed Brayton cycle, and  $h_{3in}$ ,  $h_{4in}$  are the medium enthalpy determined at characteristic points ( $3^{in}$ ), ( $4^{in}$ ), respectively.

In CFM model of combustion chamber, an energetic balance including all energy fluxes, need to be taken in to account. The heat losses in the combustion chamber were determined using the efficiency of the combustion chamber,  $\eta_{CC}$ . The chemical energy contained in the fuel is defined as [15,24]

$$\dot{Q}_{chem} = \dot{m}_f W_d , \quad (8)$$

where  $\dot{m}_f$  is the fuel mass flow rate, and  $W_d$  is the the low calorific value of the fuel.

The heat balance of the combustion chamber may be written as [15,24]

$$\eta_{CC} (\dot{Q}_{chem} + \dot{m}_o h_2 + \dot{m}_f h_f + \dot{m}_{11} h_{11}) = \dot{m}_{ex} h_3 , \quad (9)$$

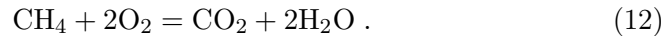
where  $\dot{m}_{ex}$  is the exhaust mass flow rate,  $\dot{m}_{11}$  is the heated (regenerated) water injected mass flow rate, and  $h_f$ ,  $h_3$ ,  $h_{11}$ – fuel, exhaust and heated water enthalpy, respectively.

Moreover, the equations given above Eqs. (10) and (11) were fulfilled following with mass balances [15]

$$\dot{m}_{ex} = \dot{m}_f + \dot{m}_o + \dot{m}_{11} , \quad (10)$$

$$\dot{m}_{2in} = \dot{m}_{3in} + \dot{m}_7 , \quad (11)$$

where  $\dot{m}_{2in} = \dot{m}_{ex}$  is the medium mass flow rate out the turbine outlet in the inversed Brayton cycle,  $\dot{m}_7$  is the condensates steam mass flow rate in the condenser. To simplify, fuel (gas) is assumed to undergo absolutely and entirely combustion. General the chemical reaction is shown below [24]





Oxygen, from a cryogenic air separation unit, is fed in a stoichiometric ratio with the fuel in the combustor. Based on the composition of the resulting exhaust gases, its temperature  $T_3 = T_{TIT}$  and enthalpy  $h_3$  were computed from the energy balance. The expansion process in the gas turbine is defined as [15,24]

$$\eta_{it} = \frac{l_{t3-4}}{l_{t3-4s}} = \frac{h_3 - h_4}{h_3 - h_{4s}}, \quad (13)$$

where  $l_{t3-4s}$  is the unitary isotropic expansion work  $l_{t3-4}$  is the real expansion work, and  $h_3, h_4, h_{4s}$  are the enthalpy of the medium ( $\text{CO}_2 + \text{H}_2\text{O}$ ) in points (3), (4), (4s), respectively; there is an analogy adopted for other points of the cycle.

The turbine power output of Brayton cycle [24]

$$N_{GT-BC} = \dot{m}_{ex}\eta_{mt}(h_3 - h_4), \quad (14)$$

where  $\eta_{mt}$  is the turbine mechanical efficiency, and turbine power output,  $N_{GT-IBC}$ , of inverted Brayton cycle formula is [19]

$$N_{GT-IBC} = \dot{m}_{ex}\eta_{mt}(h_{1in} - h_{2in}), \quad (15)$$

where  $h_{1in}, h_{2in}$  are the medium enthalpy determined at characteristic points  $(1^{in}), (2^{in})$ , respectively.

The power of the pump was calculated using formula

$$N_P = \dot{m}_8\eta_{mp}(h_8 - h_9), \quad (16)$$

where  $\dot{m}_8$  is the water injected mass flow rate,  $\eta_{mp}$  is the pump mechanical efficiency, and  $h_8, h_9$  are water enthalpy at characteristic points (8), (9).

Further, heat flux, exchange in the regenerative exchanger, with the assumed heat exchanger efficiency  $\eta_{he}$ , is expressed to be

$$\dot{m}_{ex}\eta_{he}(h_{2in} - h_{2a}^{in}) + \dot{m}_{4in}\eta_{he}(h_{4in} - h_{4a}^{in}) = \dot{m}_9(h_{11} - h_9), \quad (17)$$

where  $\dot{m}_{4in} = \dot{m}_{3in}$  is the medium mass flow rate out the compressor outlet in the inversed Brayton cycle;  $h_{2in}, h_{4in}, h_{2a}^{in}, h_{4a}^{in}$  are the medium enthalpy determined at characteristic points  $(2^{in}), (4^{in}), (2_a^{in}), (4_a^{in})$ ;  $\dot{m}_9 = \dot{m}_8$  is the water injected mass flow rate;  $h_9, h_{11}$  are the water enthalpy at characteristic points (9), (11).

Electric power of the generator terminals is defined upon the mechanical power of the individual components of the thermodynamic cycle, i.e., the

gas turbine,  $N_{GT}$ , compressor,  $N_C$ , the water pump,  $N_P$ , and the generator efficiency,  $\eta_g$ . Electrical power of the Brayton cycle,  $N_{el-BC}$ , is the difference between the devices generating and consuming the power [15]

$$N_{el-BC} = \eta_g(N_{GT-BC} - N_{C-BC} - N_P). \quad (18)$$

The electrical power of the inverse Brayton cycle,  $N_{el-IBC}$ , is given [19]

$$N_{el-IBC} = \eta_g(N_{GT-IBC} - N_{C-IBC}). \quad (19)$$

The electrical power of the double Brayton cycle  $N_{el-DBC}$  is determined by

$$N_{el-DBC} = N_{el-BC} + N_{el-IBC}. \quad (20)$$

The electrical power,  $N_{el}$ , of the whole system, including the requirement of the ASU station,  $N_{el-ASU}$ , and the electric power needed to drive the CO<sub>2</sub> capture system,  $N_{el-CO_2}$ , is described by

$$N_{el} = N_{el-DBC} - N_{el-ASU} - N_{el-CO_2}. \quad (21)$$

The electrical efficiency of the whole system,  $\eta_{el-netto}$ , is defined as a quotient of the electrical power generated by the block and chemical energy flux contained in the fuel [15,24]

$$\eta_{el-netto} = \frac{N_{el}}{\dot{Q}_{chem}} = \frac{N_{el}}{\dot{m}_f W_d}. \quad (22)$$

The efficiency of the Brayton cycle  $\eta_{el-BC}$  may also be given as a quotient of the electrical power  $N_{el-BC}$  and the chemical energy flux contained in the fuel  $\dot{Q}_{chem}$

$$\eta_{el-BC} = \frac{N_{el-BC}}{\dot{m}_f W_d}. \quad (23)$$

However, the double Brayton cycle efficiency is defined as a quotient of the electrical power  $N_{el-DBC}$  generated by the double Brayton cycle and fuel chemical energy flux  $\dot{Q}_{chem}$  contained in the fuel

$$\eta_{el-DBC} = \frac{N_{el-DBC}}{\dot{m}_f W_d}. \quad (24)$$

The efficiency of the inversed Brayton cycle  $\eta_{el-IBC}$  is described as a difference of the  $\eta_{el-DBC}$  and  $\eta_{el-BC}$  as

$$\eta_{el-IBC} = \eta_{el-DBC} - \eta_{el-BC}. \quad (25)$$

In turn, defining the specific work for air separation was corrected using the coefficient of energy consumption

$$\beta = \frac{N_{el-ASU}}{\dot{m}_o} . \quad (26)$$

The oxygen separating station was projected as a cryogenic station with double Linde column, which usage provides receiving oxygen, the purity of which varies in range 95–99.8% [9]. Moreover, the efficiencies of the elements system were all set up as

turbine: internal  $\eta_{it} = 88\%$ , mechanical  $\eta_{mt} = 99\%$ ,  
 compressor: internal  $\eta_{ic} = 87\%$ , mechanical  $\eta_{mc} = 99\%$ ,  
 pump: internal  $\eta_{iP} = 75\%$ , mechanical  $\eta_{mP} = 98\%$ ,  
 generator  $\eta_g = 97\%$ ,  
 combustion chamber  $\eta_{CC} = 99\%$ ,  
 heat exchanger  $\eta_{he} = 98\%$ .

## 4 Results of analysis

Calculations of the heat cycle had been done for the constant mass flow rate of: oxygen 51.8 kg/s, water 117.7 kg/s and fuel 12.83 kg/s on the combustion chamber inlet. Total exhaust mass flow rate is about 182.3 kg/s. The combustion chamber pressure was fixed to 4 MPa. Moreover, the temperature difference in the heat exchanger was also assumed to be 20 K. Additionally, the condensation temperature was assumed to be equal  $t_{3in} = 30^\circ\text{C}$ . During the thermodynamic analysis few efficiencies relations based on expansion/compression ratio parameter  $\Pi$  were evaluated  $\eta_{el-netto}$  including  $\eta_{el-BC}$ ,  $\eta_{el-IBC}$ ,  $\eta_{el-DBC}$ . The analysis results were presented in Fig. 3. The temperatures, on the turbine outlet,  $t_{GT}$ , and temperature inlet turbine,  $t_{TIT}$ , were also analyzed in inversed Brayton cycle (Fig. 4). The thermodynamic parameters of components of the medium in the characteristic points of the cycle were presented in Tab. 1.

As it was shown in the analysis, instead of the initial efficiency decrease in Brayton cycle, total efficiency of the block raised, as it was presented in Fig. 3. The Brayton cycle efficiency firstly decreased because of the temperature decrease and that is why the regeneration level was falling too in the heat exchanger. However, the efficiency of the inversed Brayton cycle has been rising until it has reached the value of  $\eta_{el-IBC} = 15.3\%$  for the expansion ratio equal  $\Pi = 560$ . It corresponds to the condensation pressure at

Table 1. Thermodynamic parameters of components of the medium in the characteristic points for the condenser pressure at the level of  $p_{3^{in}} = 7.8$  kPa.

Point	$t$	$p$	$x$	$\dot{m}$	Mole fraction				
					O <sub>2</sub>	H <sub>2</sub> O	N <sub>2</sub>	CO <sub>2</sub>	NO <sub>x</sub>
	[°C]	[kPa]	[-]	[kg/s]	[-]				
1	21	103.4	1.00	51.8	0.989	0.000	0.011	0.000	0.000
2	822	4050.0	1.00	51.8	0.989	0.000	0.011	0.000	0.000
3	1338	4000.0	1.00	182.3	0.000	0.908	0.002	0.089	0.000
4	641	101.3	1.00	182.3	0.000	0.908	0.002	0.089	0.000
2 <sup>in</sup>	302	7.8	1.00	182.3	0.000	0.908	0.002	0.089	0.000
2 <sup>in</sup> <sub>a</sub>	150	7.8	1.00	182.3	0.000	0.908	0.002	0.089	0.000
3 <sup>in</sup>	30	7.8	0.93	50.8	0.001	0.504	0.011	0.484	0.000
4 <sup>in</sup>	226	101.3	1.00	50.8	0.001	0.504	0.011	0.484	0.000
4 <sup>in</sup> <sub>a</sub>	60	101.3	0.60	50.8	0.001	0.504	0.011	0.484	0.000
5 <sup>in</sup>	30	101.3	0.99	36.4	0.003	0.042	0.021	0.934	0.000
6	155	8000.0	1.00	35.8	0.003	0.002	0.022	0.973	0.000
7	30	7.8	0.00	131.5	0.000	1.000	0.000	0.000	0.000
8	30	7.8	0.00	117.7	0.000	1.000	0.000	0.000	0.000
9	34	4000.0	0.00	117.7	0.000	1.000	0.000	0.000	0.000
10	129	4000.0	0.00	117.7	0.000	1.000	0.000	0.000	0.000
11	214	4000.0	0.00	117.7	0.000	1.000	0.000	0.000	0.000

the level of  $p^{3^{in}} = 7$  kPa. In turn, the optimal value of the efficiency of the whole cycle reached at the expansion ratio equal  $\Pi = 520$ , which conforms to the condenser pressure at the level of  $p^{3^{in}} = 7.8$  kPa. Additionally, the whole system efficiency zooms down for about 8.66% because of the oxygen production (6.38%) and CO<sub>2</sub> escapement (2.28%).

In the Fig. 4 the correlation of the temperature in the combustion chamber and on the turbines outlet was presented, for inversed Brayton cycle. As it is shown in the picture presented below, the temperature in the combustion chamber correlates to the regeneration level.

It ought to be added that CFM type numerical tool gives a possibility to model combined gas-steam turbine cycles, what has recently been demonstrated in articles [15,31,32].

It should be emphasized that the carbon dioxide capture from the exhaust containing steam and CO<sub>2</sub> is rather simple comparing to other pre- and post-combustion methods, being in use. The energy consumption amounts to  $\beta = 0.248$  kW/kgO<sub>2</sub>. Obtained value of the energy consumption corresponds to values  $\beta = 0.247$  and 0.250 given in [13,11].

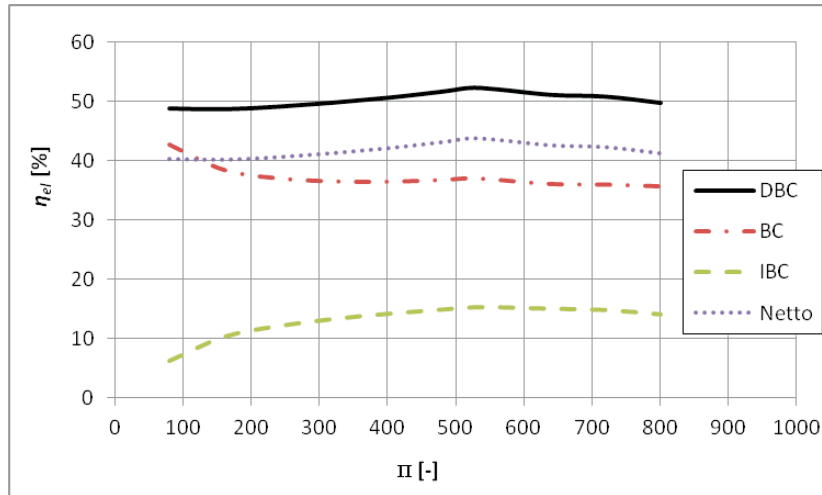


Figure 3. Dependency of electrical efficiency of the whole system  $\eta_{el-netto}$ , Brayton cycle efficiency  $\eta_{el-BC}$ ; inversed Brayton cycle efficiency  $\eta_{el-IBC}$  and double Brayton cycle efficiency  $\eta_{el-DBC}$  against expansion ratio  $\Pi$ .

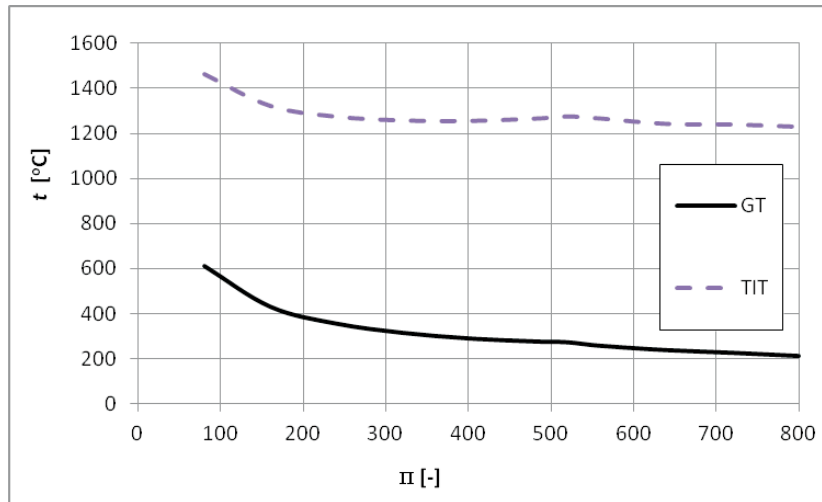


Figure 4. Dependency of outlet temperature of gas-steam turbine  $t_{GT} = t_{2in}$  and temperature inlet turbine  $t_{TIT} = t_3$  against expansion ratio  $\Pi$ .

## 5 Conclusion

The numerical analysis has shown that total energy output notably grows as the inversed Brayton cycle is being used. The highest netto efficiency

of double Brayton cycle with oxy combustion and capture CO<sub>2</sub>, at the level of  $\eta_{el-netto} = 43.67\%$ , was reached at the condenser pressure equal to  $p_{3in} = 7.8$  kPa. However, the efficiency of the double Brayton cycle is equal to  $\eta_{el-DBC} = 52.3\%$ . The decrease of the efficiency is caused by the oxygen producing (6.38%) and capture the CO<sub>2</sub> (2.28%). The indubitable advantage of the double Brayton cycle with oxy combustion and CO<sub>2</sub> capture is the lack of pollution emissions such as NO<sub>x</sub> and CO<sub>2</sub>.

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## References

- [1] ANDERSON R., MACADAM S., VITERI F., DAVIES D., DOWNS J., PALISZEWSKI A.: *Adapting gas turbines to zero emission oxy-fuel power plants*. ASME Paper GT2008-51377 (2008) 1–11.
- [2] BADUR J.: *Development of Energy Concept*. Wyd. IMP PAN, Gdańsk 2009 (in Polish).
- [3] BADUR J.: *Five lecture of contemporary fluid termomechanics*. Gdańsk 2005 (in Polish). [www.imp.gda.pl/fileadmin/doc/o2/z3/.../2005\\_piecwykladow.pdf](http://www.imp.gda.pl/fileadmin/doc/o2/z3/.../2005_piecwykladow.pdf).
- [4] BADUR J.: *Numerical modeling of sustainable combustion at gas turbine*. Wyd. IMP PAN Gdańsk 2003 (in Polish).
- [5] BADUR J., LEMAŃSKI M.: *Inverse Brayton cycle — high performance maner heat recovery from gas turbine*. Energetyka Ciepłna i Zawodowa **221**(2003), 46–48 (in Polish).
- [6] BOLLAND O., KVAMSDAL H.M., BODEN J.C.: *A thermodynamic comparison of oxy-fuel power cycles water-cycle, Graz-cycle, and Matiant-cycle*. Proc. of the Int. Conf. on Power Generation and Sustainable Development. Liège, Belgium, 2001.
- [7] CARAPELUCCI R., MILAZZO A.: *Repowering combined cycle power plants by a modified STIG configuration*. Energ Convers. Manage. **48**(2007), 1590–1600.
- [8] CHODKIEWICZ R., POROCHNICKI J., KACZAN B.: *Steam — gas condensing turbine system for power and heat generation*. ASME Paper 2001-GT-0097 (2001) 1–8.
- [9] CHOROWSKI M.: *Cryogenics. Basics and applications*. IPPU, Masta 2007.
- [10] *Directive 2010/75/Eu of the European Parliament and of the Council of 24 November 2010 on industrial emissions (integrated pollution prevention and control)*.
- [11] GOUL C., CAI R., HONG H.: *An advanced oxy-fuel power cycle with high efficiency*. Proc. IMechE Part A: J. Power Energ. **220**(2006) 315–324.
- [12] HOLLIS R., SKUTLEY P., ORTIZ C., VARKEY V., LEPAGE D., BROWN B., DAVIES D., HARRIS M.: *Oxy-fuel turbomachinery development for energy intensive industrial applications*. Proc. of ASME Turbo Expo 2012, GT2012-69988 (2012), 1–9.
- [13] HONG J., CHAUDHRY G., BRISSON J.G., FIELD R., GAZZINO M., GHONIEM A.: *Analysis of oxy-fuel combustion power cycle utilizing a pressurized coal combustor*. [web.mit.edu/mitei/docs/reports/hong-analysis.pdf](http://web.mit.edu/mitei/docs/reports/hong-analysis.pdf).

- [14] JERICHA H., SANZ W., WOISETSCHLÄGER J, FESHARAKI M.: *CO<sub>2</sub> — Retention Capability of CH<sub>4</sub>/O<sub>2</sub> — Fired Graz Cycle*. CIMAC Conf. Paper, Interlaken, Switzerland 1995.
- [15] JESIONEK K., CHRZCZONOWSKI A., ZIÓLKOWSKI P., BADUR J.: *Power enhancement of the Brayton cycle by steam utilization*. Arch. Thermodyn. **33**(2012), 3, 39–50.
- [16] KAPROŃ H., WASILEWSKI A.: *Natural gas fuel XXI century*. Wyd. KAPRINT, Lublin 2012 (in Polish).
- [17] KOLEV N., SCHABER K., KOLEV D.: *A new type of a gas — steam turbine cycle with increased efficiency*. Appl. Therm. Eng. **21**(2001), 391–405.
- [18] KVAMSDAL H.M., JORDAL K, BOLLAND O.: *A quantitative comparison of gas turbine cycles with CO<sub>2</sub> capture*. Energy **32**(2007), 10–24.
- [19] LEMAŃSKI M.: *Analyses of thermodynamic cycles with fuel cells and gas-steam turbine*. PhD thesis, IF-FM PAS, Gdańsk 2007 (in Polish).
- [20] LIU C.Y., CHEN G., SIPOCZ N., ASSADI M., BAI X.S.: *Characteristic of oxy-fuel combustion in gas turbine*. Apl. Energ. **89**(2012), 387–394.
- [21] MATHIEU PH., NIHART R.: *Sensitivity analysis of the MATIANT cycle*. Energ. Convers. Manage. **40**(1999), 15, 1687–1700.
- [22] SANZ W., HUSTAD CARL-W., JERICHA H.: *First generation Graz cycle power plant for near-term deployment*. Proc. of ASME Turbo Expo 2011, GT2011-45135 (2011) 1-11.
- [23] STAICOVICI M.: *Further research zero CO<sub>2</sub> emission power production: the ‘COOLENERG’ process*. Energy **27**(2002), 831–844.
- [24] TOPOLSKI J.: *Combustion diagnosis in combined gas-steam cycle*. PhD thesis, IF-FM PAS, Gdańsk, 2002 (in Polish).
- [25] YANG H.J., KANG D.W., AHN J.H., KIM T.S.: *Evaluation of design performance of the semi-closed oxy-fuel combustion combined cycle*. Proc. of ASME Turbo Expo 2012, GT2012-69141 (2012) 1–12.
- [26] YANTOVSKY E., GÓRSKI J., SHOKOTOV M.: *Zero Emissions Power Cycles*. Taylor & Francis Group, 2009.
- [27] YANTOVSKY E., GÓRSKI J., SMYTH B, ELSHOF J.: *Zero-emission fuel-fired power plants with ion transport membrane*. Energy **29**(2004), 2077–2088.
- [28] YANTOVSKI E., ZVAGOLSKY K., GAVRILENKO V.: *The COOPERATE- demo power cycle*. Energy Convers. Manage **36**(1995), 861–864.
- [29] ZAPOROWSKI B.: *Perspectives of development of gas power sources in Polish electro energetic*. Polityka Energetyczna, **12**(2009), Z. 2/2, (in Polish).
- [30] ZHANG N., LIOR N.: *Two novel oxy-fuel power cycles integrated with natural gas reforming and CO<sub>2</sub> capture*. Energy **33**(2008), 340–351.
- [31] ZIÓLKOWSKI P., LEMAŃSKI M., BADUR J., NASTALEK L.: *Power augmentation of PGE Gorzow’s gas turbine by steam injection — thermodynamic overview*. Rynek Energii **98**(2012), 161–167.

- [32] ZIÓLKOWSKI P., LEMAŃSKI M., BADUR J., ZAKRZEWSKI W.: *Increase efficiency gas turbine by use the Szewalski idea*. Rynek Energii, **100**(2012), 63–70 (in Polish).
- [33] ZIÓLKOWSKI P., ZAKRZEWSKI W., SŁAWIŃSKI D., BADUR J.: *Clean gas technology – opportunity for Pomerania*. Rynek Energii **104**(2013), 79–85 (in Polish).