Parametric optimization of the semi-closed oxy-fuel combustion combined cycle

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Abstract. The report discloses thermodynamic studies of the semi-closed oxy-fuel combustion combined cycle. The computer simulation and parametric optimization approaches are described in details. The oxy-fuel cycle net efficiency in relationship to the carbon dioxide turbine exhaust pressure and the steam turbine inlet pressure is shown. The power production efficiency reduction is related to the turbine cooling losses is described. It is shown that the semi-closed oxy-fuel combustion cycle maximal net efficiency of 52.5% occurs at the initial temperature and pressure 1400°C and 60 bar at the carbon dioxide turbine exhaust pressure 0.5 bar and the steam turbine inlet pressure 90 bar. The cooling losses consideration leads to the net efficiency of 47.76% that is reached at the carbon dioxide turbine exhaust pressure 1 bar and the steam turbine inlet pressure 90 bar.

1. Introduction
During the last decades the atmosphere pollution became global and about a quarter of the man-produced carbon dioxide is emitted by the power production industry [1]. The large carbon dioxide emissions worsen the greenhouse effect which is one of the most crucial problems in industrialized countries. Now power plants successfully mitigate emissions of nitrogen and sulfur oxides [2-4]. On the other side the large amount of carbon dioxide produced by the organic fuel combustion is still a difficult problem [5].

One of the problems in Russia Power Sector strategy until 2035 is the development and introduction of measures for mitigation of harmful emissions by the heat and power production industry. This goal may be reached by the transition to environmentally friendly power production technology such as oxy-fuel cycles that burn organic fuels in pure oxygen and the carbon dioxide storage. The simplest oxy-fuel cycle is the SCOC-CC [6]. It is an oxy-fuel semi-closed gas turbine cycle. The carbon dioxide turbine exhaust gas produces steam in heat recovery steam generator for a steam turbine compartment (figure 1).

Gas fuel and high purity oxygen enter the combustor at the stoichiometric ratio. Flame of this mixture combustion may be up to 3500 °C so this temperature is limited with the third flow into the combustor, a gas mixture with a high CO₂ content. The formed gas has the 80% CO₂ content at 1300-1400 °C temperature. This gas enters the carbon dioxide gas turbine that drives the power generator. The gas expands in the turbine and enters the steam generator. The heat of turbine exhaust gas produces the overheated steam that feeds the power production steam turbine. The steam generator exhaust gas enters the condenser where a large part of the water content is condensed at the nearly atmospheric pressure. Then a part of the carbon dioxide content leaves from the cycle circuit and is further sequestered. The residual flow consisting mostly of carbon dioxide is recirculated to the compressor inlet.
Papers [7, 8] disclose detailed studies of the SCOC-CC cycle input parameters and their influence upon the power production efficiency. The research [9] show that at the carbon dioxide turbine inlet temperature of 1400 °C the optimal inlet pressure is 60 bar, the cycle net efficiency is 46% and the total coolant massflow is 21%.

The cycle is semi-closed so its minimal pressure not necessarily must be equal to the atmospheric one. An increase of the minimal pressure reduces the turbine and steam generator sizes but increases wall thicknesses of the power production equipment.

If the minimal pressure of semi-closed oxy-fuel cycle grows from 1 bar to 10 bar it changes the cycle efficiency only for 0.5% [9]. The important problem concerned to the pressure growth is the turbine blade cooling. The increased operating pressure reduces the volumetric flow and thus the cooled surface area. On the other side, the heat transfer coefficients on the both sides of cooling channels become higher and the related heat flow through the blade wall grows. This effect increases the temperature difference across the blade thickness and reduces the acceptable coolant temperature increase along the cooling channels. As the result, the same heat flux needs a larger coolant flow. The mentioned above effects show that the carbon dioxide turbine exit pressure should not be above 1 bar. The SCOC-CC efficiency data on the reduction of turbine exit pressure below the atmospheric pressure are absent yet.

This study goal is to improve the SCOC-CC thermal efficiency by the thermal flow parametric optimization that includes the following problems:
1. Development of the SCOC-CC computer simulation model to carry out thermodynamic studies,
2. Relation of the carbon dioxide turbine exit pressure with the SCOC-CC thermal efficiency,
3. Relation of the steam turbine inlet pressure and the SCOC-CC thermal efficiency,
4. Influence of the carbon dioxide turbine cooling losses upon the SCOC-CC thermal efficiency.

2. Methodology

2.1. Input data and method for the SCOC-CC simulation.

The AspenONE code makes a basis for the SCOC-CC parametric optimization computer model (figure 2). The heat flow chart includes the following elements: carbon dioxide compressor, oxy-fuel combustor, carbon dioxide gas turbine, heat recovery steam generator, cooler separator, multi-stage intercooled compressor, fuel and oxygen compressors, steam turbine, condenser, pumps and deaerator. Specific features of this chart are the absence of coolant bleeding for the carbon dioxide turbine and the straight flow build of heat recovery steam generator.

Calculation of thermo-physical parameters of the multi-component working fluid involved the NIST REFPROP database. The fuel was pure methane at 15°C temperature and 7bar pressure. The oxidizer was pure oxygen with the power consumption for its production of 900 KW/kg/s [10]. The carbon dioxide storage pressure and temperature are assumed as 100bar and – 28°C respectively [11]. Table 2.1 summarizes the heat flow chart simulation input data.
Calculation of the carbon dioxide compressors and turbines involved the isentropic expansion method with the following input data: turbo-machine pressure / expansion ratio, turbo-machine specific mechanical and internal efficiencies. The pump calculation input data were internal specific and mechanical efficiencies and its pressure ratio.

The combustor model was a stoichiometric reactor with the following input data: fuel, working fluid and oxygen massflows. The reaction equation for the combustion of methane in oxygen was as follows (1):

\[
CH_4 + 2O_2 \rightarrow CO_2 + 2H_2O
\] (1)

There are no available efficiency test data for the oxygen-fuel mixture combustion in the excessive carbon dioxide environment so the heat losses into atmosphere were not considered and the combustion efficiency was assumed as equal to 1.

Calculation of losses for cooling the carbon dioxide turbine blades is described in the paper [12]. The analysis assumed 7 turbine stages with 4 of them cooled [14]. The heat drop distribution along the stages is taken as uniform.

Figure 3 shows the location of cooled and non-cooled turbine compartments. The coolant flow taken downstream compressor enters the blades and vanes cooling channels where it is heated. Then the coolant is discharged into the flowpath through the film cooling holes and the trailing edge slots.

The turbine flowpath efficiency drops due to mixing the coolant and working fluid flows was evaluated by the method described in [13].
Table 1. Input data for the SCOC-CC heat flow chart simulation

| Parameter                                      | Unit    | Value  |
|------------------------------------------------|---------|--------|
| Carbon dioxide turbine inlet temperature      | °C      | 1400   |
| Carbon dioxide turbine inlet pressure         | bar     | 60     |
| Compressor inlet massflow                     | kg/s    | 100    |
| Fuel temperature                              | °C      | 15     |
| Fuel pressure                                 | bar     | 7      |
| Fuel low combustion heat                      | KJ/kg   | 50025  |
| Fuel pressure at supercharger exit            | Bar     | 60     |
| O₂ temperature at the air separator exit      | °C      | 30     |
| O₂ pressure at the air separator exit         | bar     | 10     |
| O₂ production power consumption               | KW/kg/s | 900    |
| Supercharger exit fuel pressure               | bar     | 60     |
| Steam turbine inlet temperature               | °C      | 560    |
| Steam turbine exit pressure                   | bar     | 0.05   |
| Condenser pressure                            | bar     | 0.05   |
| Condensate pump exit pressure                 | bar     | 1.44   |
| Condensate gas heater inlet temperature       | °C      | 55     |
| Condensate gas heater exit under-heating to the deaerator saturation point | °C | 14 |
| Deaerator pressure                            | bar     | 1.26   |
| Pinch point temperature in the heat recovery steam generator | °C | 10 |
| Flow temperature difference between the economizer exit and the drum saturation point | °C | 10 |
| CO₂ storage pressure                          | bar     | 100    |
| CO₂ storage temperature                       | °C      | 28     |
| Carbon dioxide turbine specific internal efficiency | % | 89 |
| Compressor specific internal efficiency        | %       | 91     |
| Power generator efficiency                    | %       | 99     |
| Steam turbine specific internal efficiency     | %       | 87     |
| Mechanical efficiency                         | %       | 99     |

Figure 3. Location of carbon dioxide turbine cooled and uncooled compartments
2.2. SCOC-CC cycle parametric optimization.
SCOC-CC optimization involved the enumerative scan method. In all considered cases the fixed parameters were the carbon dioxide initial temperature 1400°C and pressure 60 bar. The varying parameters were the carbon dioxide turbine exit temperature and pressure and the steam turbine inlet pressure. Figure 4 illustrates the 35 studied combinations of parameters. The humidity limit at the steam turbine exit corresponded to the recommendation [15] that is 13%.

![Figure 4. Calculation parameters combinations.](image)

3. Parametric optimization
The parametric studies results in figure 5 show that a 1 bar increase of the steam turbine inlet pressure increases the SCOC-CC average net efficiency of 0.016%. Together with this the acceptable humidity at steam turbine exit (solid line in the picture) occurs at the initial pressures below 90 bar. When the turbine inlet pressure is above 90 bar the humidity is above 13% (dash line). The maximal efficiency of 52.5% occurs at the carbon dioxide turbine exit pressure of 0.5 bar and the steam turbine inlet pressure 90 bar.

![Figure 5. Influence of the steam turbine inlet pressure upon the SCOC-CC cycle efficiency at different pressures in the carbon dioxide turbine exit.](image)

The SCOC-CC net efficiency increase due to the steam turbine pressure increase is caused by the additional power production by steam turbine that is larger than the internal power consumption increase. For example an increase of the steam turbine inlet pressure of 1 bar produces the steam turbine power increase of 0.03 MW (figure 6) and a 0.005 MW increase of the feed pump power consumption (figure 7).
Figure 6. Influence of the steam turbine inlet pressure upon its electric power production at different pressures in the carbon dioxide turbine exit

Figure 7. Influence of steam turbine inlet pressure upon the feed pump electric power at different pressures in the carbon dioxide turbine exit

If the carbon dioxide turbine exit pressure grows for 0.1 bar above its optimal level of 0.5 bar at the steam turbine inlet pressure 90 bar, the SCOC-CC net efficiency reduces for 0.1%. On the other side the turbine exit pressure reduction pushes the efficiency down for 0.7% (figure 8). An increase of the counter-pressure above its optimal value for 0.1 bar reduces the turbine power output for 0.8% caused by the reduction of available heat drop together with the reduction of compressor power consumption of 6.6%. When the turbine exit pressure drops below its optimal value for 0.1 bar the power production increases for 0.1%, but the compressors power consumption grows for 8%

Thus the optimization besides the cooling losses shows that a transition of the carbon dioxide turbine exhaust pressure from 1 bar to 0.5 bar corresponds to the net efficiency increase of 0.25%. In this analysis the condenser-separator air suction was not accounted for but this effect will make the net efficiency increase smaller.

Figure 8. Influence of gas turbine exit pressure upon the SCOC-CC net efficiency at the steam turbine inlet pressure 90 bar.
Together with the turbine cooling losses the maximal cycle net efficiency of 47.76% is obtained at the turbine exit pressure 1 bar. When the exit pressure drops down to 0.5 bar the efficiency drops down to 47.67%. This effect is caused by the coolant injection into the flowpath and the exhaust gas temperature reduction from 627°C to 538°C (here the turbine exit pressure is 0.5 bar) and the steam cycle initial temperature reduction. The steam flow became 28.4% smaller and the turbine inlet temperature and pressure dropped from 560°C to 538°C, pressure dropped from 90 bar to 70 bar to ensure that is needed for the acceptable humidity downstream the turbine last stage.

At the carbon dioxide turbine exhaust pressure 1 bar and the cooling losses included, the steam cycle initial parameters do not change and the exhaust gas temperature reduction from 713°C to 699°C only reduces the steam flow for 30.0%. Thus the optimal pressure change is related to the steam cycle power production. The cooling losses consideration shows the steam turbine power reduction for 30.6% and 32.3% at the exhaust pressure values 1 bar and 0.5 bar respectively.

4. Conclusions
1. The developed computer simulation model based on the AspenONE code describes a semi-closed oxygen combustion cycle with a single pressure heat recovery steam generator.
2. The parametric studies have shown that the SCOC-CC cycle net efficiency with a single pressure heat recovery steam generator at the gas turbine inlet temperature 1400°C reaches its maximal value at the carbon dioxide turbine exhaust pressure of 0.5 bar and the steam turbine inlet pressure of 90 bar.
3. The increase of the steam turbine inlet pressure for 1 bar in the 60 – 130 bar range produces the net efficiency increase of 0.16% which is caused by an increase of the turbine available heat drop.
4. Deviation of the carbon dioxide turbine exit pressure from its optimal value of 0.5 bar for 0.1 bar causes the cycle net efficiency reduction of 0.1 – 0.7%. This is related to an increase of the internal power consumption and reduction of the turbine power output.
5. In the case of cooling losses included the carbon dioxide turbine exhaust pressure change from 1 bar to 0.5 bar causes the net efficiency reduction of 0.08%. This effect is caused by bleeding of some fluid pressurized in the compressor into the cooling system by-passing the combustor and this together with the smaller counter-pressure causes lower exhaust gas temperature. As the result it requires lower steam cycle inlet parameters and a remarkable reduction of the steam cycle power output.

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