




Article

# A Case Study of the Supercritical CO<sub>2</sub>-Brayton Cycle at a Natural Gas Compression Station

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**Abstract:** Heat losses caused by the operation of compressor units are a key problem in the energy efficiency improvement of the natural gas compression station operation. Currently, waste heat recovery technologies are expensive and have low efficiency. One of these technologies is organic Rankine cycle (ORC) which is often analyzed in scientific works. In this paper, the authors decided to investigate another technology that allows for the usage of the exhaust waste energy—the supercritical Brayton cycle with CO<sub>2</sub> (S-CO<sub>2</sub>). With a thermodynamic model development of S-CO<sub>2</sub>, the authors performed a case study of the potential S-CO<sub>2</sub> system at the gas compressor station with the reciprocating engines. By comparing the values of selected S-CO<sub>2</sub> efficiency indicators with ORC efficiency indicators at the same natural gas compression station, the authors tried to determine which technology would be better to use at the considered installation. Investigations on parameter change impacts on the system operation (e.g., turbine inlet pressure or exhaust gas cooling temperatures) allowed to determine the direction for further analysis of the S-CO<sub>2</sub> usage at the gas compressor station. When waste heat management is considered, priority should be given to its maximum recovery and cost-effectiveness.

**Keywords:** supercritical CO<sub>2</sub>-Brayton cycle; natural gas; compressor station; waste heat recovery; energy efficiency

## 1. Introduction

Due to the constant global increase in energy demand in various forms, humanity is more and more willingly looking for solutions that will reduce the energy consumption at the stage of production and transportation. In the case of natural gas transmission, it is assumed that the process that requires the consumption of the largest amount of energy is compression of natural gas—compressor stations may be responsible even for 50% of total cost of gas transmission [1]. Natural gas at compressor stations is most often compressed by means of combustion engines (reciprocating engines or gas turbines), which are powered by natural gas. From the point of view of managing the energy efficiency of the natural gas compression station, the heat losses caused by the operation of compressor units are a key problem. As a result of natural gas combustion, hot exhaust gases are emitted to the atmosphere, which may be treated as a potential source of energy. Currently, waste heat recovery technologies are expensive and have low efficiency. Therefore, it is important to (i) search for existing technologies used in other industries and which could be used at a gas compressor station; (ii) simulating various operating variants of these technologies, which will allow to optimize their performance indicators for operating conditions at the gas compressor station. One of the available options is waste heat recovery with the use of a system based on the organic Rankine cycle (ORC). This technology is often analyzed in scientific works e.g., [2–5]. For example, the possibilities of generating electrical

energy with the ORC systems were checked in Jarosław II natural gas compression station, localized in Jarosław, Podkarpacie Province [2]. Results have shown that the available temperature and the stream of exhaust gases confirm the sense of considering the use of ORC technology. On the assumption of pseudo-steady operation of the system, the average annual net electric power of about 32 kW could be obtained when feeding the ORC system with waste heat from a compressor engine. However, as Kim M.S. et al. presented in their work [6] the ORC has a few disadvantages: the working fluids in ORC (such as R113 and R123) are expensive and strong climate-changing gases [7]. Furthermore, during the heating process of working fluid, it changes the phase, which creates a so-called “pinching” effect in the heat exchanger and thus limits heat exchanger performance [8].

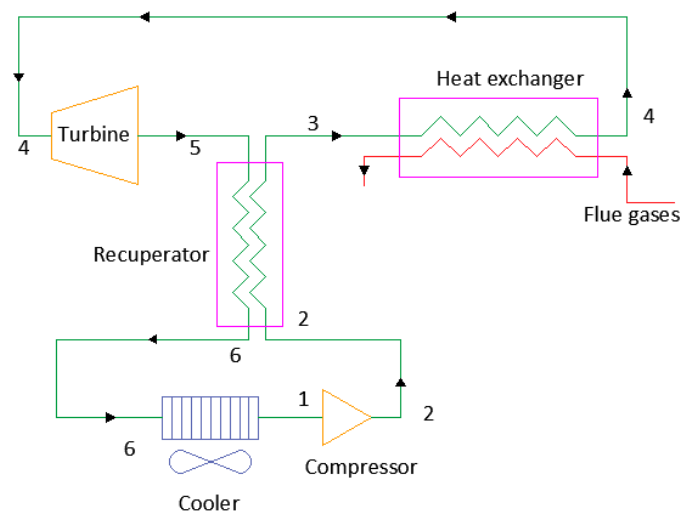
In this paper, the authors decided to investigate a case study of another technology that allows for the usage of the exhaust waste energy—the supercritical Brayton cycle with CO<sub>2</sub> (S-CO<sub>2</sub>). By analyzing different publications on S-CO<sub>2</sub>, it was found that this technology has a great potential to exceed organic Rankine cycles in certain conditions and applications [9,10]. One of the main reasons for the high potential is the relatively high cycle efficiency at moderate working medium temperatures at the turbine inlet (450–600 °C) [11]. This is particularly important in the context of waste heat temperature at gas compressor stations. Francis A. Di Bella (2011) in his work analyzed the development of a gas turbine engine exhaust waste heat recovery navy shipboard module [12]. Preliminary feasibility analysis has indicated a power improvement over the MT-30 gas turbine engine of from 20%, for the simple S-CO<sub>2</sub> waste heat recovery system. According to this publication, a typical Rankine cycle system using an organic-based refrigerant (i.e., R245fa or R113), working with the same heat source temperature, could achieve only approximately 15%–17% power improvement. Although, the supercritical Brayton cycle with CO<sub>2</sub> as a working medium is a solution to the disadvantages mentioned in the paragraph above. With the growing interest in increasing energy efficiency in the context of climate policy, high efficiency cycles are key to achieving cost-parity for non-renewable sources. The authors of this article did not find publications analyzing the use of S-CO<sub>2</sub> for waste heat recovery at a gas compressor station. The main directions where the usage of S-CO<sub>2</sub> systems is analyzed are the nuclear e.g., [13–15], geothermal e.g., [16], and solar industry e.g., [17–19]. Wright et al. [20] performed a thermo-economic analysis of four types of S-CO<sub>2</sub> systems used for waste heat recovery. They conclude that the “Cascade”, the “Dual Recuperated”, and the “Preheating” cycle maximize the net annual revenue produced by appropriately selecting the operating conditions of the S-CO<sub>2</sub> power system type, and by optimizing the amount of waste heat recovery that can be obtained. However, this study focused on the medium scale gas turbine (above 20 MWe). Sarkar and Bhattacharyya [21] carried out an optimization of compressor pressure ratio and intermediate pressure between HP and LP turbines leading to maximum thermal efficiency. They also calculated a 3.5% improvement in the efficiency when reheating is used. A new concept of the combined S-CO<sub>2</sub> recompression Brayton/organic Rankine cycle were studied by Akbari and Mahmoudi [22]. The rejected heat in the pre-cooler of SCRBC is at a temperature range suitable to run an ORC (organic Rankine cycle). These studies were carried out for the nuclear application, however, the idea of combining S-CO<sub>2</sub> Brayton cycle with ORC can be applicable at compression stations.

This manuscript focuses on presenting the case study of a Brayton cycle with S-CO<sub>2</sub> as a technology for waste heat recovery at the natural gas compressor station. The main purpose of the work is to determine the capabilities and operating parameters of the potential S-CO<sub>2</sub> system at the natural gas compressor station with the reciprocating engines. This paper presents also the impact of system operation parameters (e.g., turbine inlet pressure or exhaust gas cooling temperatures) changes on the energy efficiency indicators of this system. Investigating the impact of parameter changes on the system operation should allow determining the direction of further analysis of the S-CO<sub>2</sub> usage at the gas compressor station. The assumed waste heat parameters (e.g., stream, temperature, etc.) are real and have been determined for flue gas at the Jarosław II gas compressor station. This publication does not focus on issues related to the use of mechanical work generated by the turbine nor to technical aspects of the studied installation.

## 2. Supercritical Brayton Cycle with CO<sub>2</sub> (S-CO<sub>2</sub>)

Cycles operating with carbon dioxide as a working fluid can be divided into two groups in terms of the operating temperature level. The first group is high-temperature cycles, which are usually designed for a heat source temperature above 300 °C. The second group is low-temperature cycles, designed mainly for waste heat recovery, where the temperature of the heat source is below 300 °C. The low temperature systems operate in a transcritical cycle, in which the parameters of the working medium are below the critical point in a certain area of circulation. This type of cycle achieves lower efficiency compared to high-temperature cycles, however, it can be profitable in cases when high heat source parameters are not available but the available quantities of heat are significant.

The technological scheme of the transcritical CO<sub>2</sub> cycle is basically identical to the ORC. The cycle consists of a turbine (expander), heat exchanger, recuperator (not necessary), a pump, and a condenser. In the case of a high-temperature cycle, like S-CO<sub>2</sub>, a compressor and cooler are used in place of the pump and the condenser. A simple schematic of S-CO<sub>2</sub> cycle is presented in Figure 1.



**Figure 1.** Schematic of a simple supercritical Brayton cycle with CO<sub>2</sub> (S-CO<sub>2</sub>) cycle.

The following processes take place as the medium flows through the successive parts of the system.

- Stage 1 to 2 is a compression stage. Cooled CO<sub>2</sub> is pressurized with a compressor. The process is treated as adiabatic because the heat losses are negligible. However, Heo J, Y et al. in their work [23] showed that for certain conditions of the system's operation it is worth considering the isothermal compression—assuming the isothermal compressor for S-CO<sub>2</sub> cycles it is possible to reduce the compression work by nearly 50%, compared to the adiabatic process in some range of inlet and outlet pressures. This is because the generated heat is rejected after each small stage compression, the working fluid becomes more liquid-like in density as the pressure is increased near the ambient temperature, requiring much less work. The main mechanism to improve the efficiency in the S-CO<sub>2</sub> cycle is to reduce the compression work by performing a compression process near the critical point of CO<sub>2</sub>. Due to the fact that the temperature range near the critical point corresponds to the ambient temperature (30.98 °C) it gives the possibility to set the compressor's work near the point where CO<sub>2</sub> behaves like liquid and the compression work decreases (approaching pumping work).
- Stages 2 to 3 and 5 to 6 are the process of heat exchange in the recuperator. The working medium, which has been compressed, goes to the recuperator where it flows on the cold side of the exchanger. On the hot side of the recuperator CO<sub>2</sub> flows, which has been previously expanded, but it still has a relatively high temperature. Heat transport takes place to the compressed CO<sub>2</sub>, increasing its temperature. This step is theoretically not necessary, but it significantly improves

the internal efficiency of the cycle. At this stage the temperature of the returning CO<sub>2</sub> is also decreasing, so that less heat will have to be taken away in the cooler.

- Stage 3 to 4 is a process of main heat exchange. Preheated and highly pressurized CO<sub>2</sub> is directed to heat exchanger, where energy is obtained from waste heat. It is important that working medium undergoes heating in one phase—differently than in ORC where working medium undergoes three stages: heating, evaporation, and overheating. Thanks to that, during the heating process there is no so-called “pinching” in the heat exchanger and thus heat exchange performance in S-CO<sub>2</sub> cycle is better than ORC.
- Stage 4 to 5 can be described with the decompression process. A CO<sub>2</sub> at high pressure and high temperature is directed to the turbine, where the pressure and temperature values drop, and part of the generated mechanical energy can be used for instance to drive electric generator. The heat loss is negligible, and the process can be treated as adiabatic.
- Stage 6 to 1 is a process during which the working medium gives off the heat to the cooling medium in the cooler.

Carbon dioxide was used as a working medium a relatively long time ago (the end of the 19th century). Since the application of systems for supercritical CO<sub>2</sub> was re-proposed at the beginning of the 21st century, intensified research and development works on these systems have begun. The compressor’s efficiency was the key issue for the work being carried out, because it works closest to the critical point, where the properties of the fluid differ most significantly between all other points of the cycle. Currently, modules for generating electricity using the supercritical CO<sub>2</sub> cycle are commercially available. An example can be the product developed by Echogen company and called Thermefficient. The aim of the article is to present S-CO<sub>2</sub> as an alternative to ORC in the utilization of waste heat at a gas compressor station. ORC has a number of advantages, but also disadvantages, which include:

- high investment costs,
- large size due to the low temperature level at the heat flow,
- large streams of working medium in circulation,
- high price of the working medium,
- properties of the working medium (often dangerous),
- low efficiency of the cycle.

S-CO<sub>2</sub> technology is a solution for some of the above-mentioned disadvantages. One of the key advantages of S-CO<sub>2</sub> technology is the compactness of the solution due to the high density of the working medium. Compactness is particularly visible when the size of this system is compared to technology based on steam Rankine cycle. For example, Ahn et al. [10] write that the S-CO<sub>2</sub> turbomachinery becomes 10 times smaller than that of the steam Rankine cycle. Furthermore, the overall system size can be reduced up to four times when compared to the steam Rankine cycle. Another advantage mentioned earlier is that the cycle can operate in single-phase without passing through the saturation dome because the critical point of CO<sub>2</sub> is below the heat sink condition. Apart the lack of appearance of so-called “pinching” in the heat exchanger, this feature also results in the no need for devices adapted to operate in two phases, which reduces the total costs of the system [13]. One more advantage is the relatively high thermal cycle efficiency at turbine inlet temperatures, which are moderate. This is mainly due to the smaller work needed to compress the medium thanks to the incompressible nature of the working fluid near the critical point. Thus, the use of a S-CO<sub>2</sub> allows to increase the efficiency of the circulation and is characterized by a low ratio of the working fluid volume to the obtained mechanical power. It is also important that CO<sub>2</sub> is a non-toxic, easily available, and relatively cheap working medium.

### 3. Thermodynamic Model of S-CO<sub>2</sub>

As mentioned in the introduction, this publication focuses on checking the applicability of the S-CO<sub>2</sub> system at the Jarosław II gas compressor station. The most important issue for this type of

calculation is to determine the amount of waste heat and its parameters. To make the calculations, it was decided to use the data achieved in a previous preformed analysis [2]. As the amount of waste heat and its parameters during the year did not differ significantly for particular compressor units, it was decided to use the average parameters of flue gases. Table 1 shows the exhaust gas average properties and mole fraction of each component.

**Table 1.** Exhaust gases average properties [2].

Component of Wet Waste Gases	Molar Participation (% mol)
Carbon dioxide	5.10
Oxygen	9.40
Nitrogen	73.55
Water vapor	11.95
<b>Stream parameters</b>	
$M_h = 3.55 \text{ kg/s}$	
$T_{IN} = 616.38 \text{ K}$	

To compare the performance of different cycle operation parameters under the same exhaust gas conditions, the working out of a thermodynamic model was preceded by the following assumptions:

- The maximum operating pressure is limited to 22.5 MPa (compressor outlet pressure) in order to avoid high capital costs and to create a realistic S-CO<sub>2</sub> power cycle model. The maximum temperature (turbine inlet temperature) is limited by the exhaust gas temperature.
- Constant temperature and pressure of working medium at the compressor inlet were assumed. The values of 35 °C (308.15 K) and 7.7 MPa were appropriately assumed to assure that the heat sink temperature is not very far from the ambient temperature while the fluid in the cycle maintains the supercritical state.
- Compression and expansion of the S-CO<sub>2</sub> is an adiabatic process.
- The temperature difference between CO<sub>2</sub> and flue gas at the heat exchanger inlet and outlet is always constant for the analyzed variant.
- The limit temperature to which exhaust gases can be cooled is 100 °C (373.15 K).
- Irreversibilities such as due to turbine and compressor isentropic efficiencies, finite temperature differences, pressure losses in the heat exchanger and recuperator were also considered so as to generate thermodynamic data for real cycles. For the sensitivity analysis of cycles, component efficiency and heat exchanger effectiveness are based on the literature review. For convenience, the pressure losses in each heat exchanger and recuperator are considered as 0.5% of the inlet pressure. Pressure losses in exhaust gas side are neglected in this paper because it has practically no effect on the energy efficiency of the system. However, it should be remembered that in reality the heat exchanger must be chosen in such a way that the pressure drops of the exhaust gases do not exceed the limit value given by the manufacturer of the compressor unit. In the case of the compressor station under consideration, the value for installed machines is 100 kPa.

The model of S-CO<sub>2</sub> cycle was prepared with the use of CoolProp [24] and REFPROP [25] database. Table 2 shows the S-CO<sub>2</sub> assumptions and constraints.

For the purposes of this section, all subscripts represent the cycle stages, as in Figure 1. As already mentioned, compression stage is treated as adiabatic process because the heat losses are negligible. Due to this assumption, and information about pressure and temperature of CO<sub>2</sub> at the inlet to the compressor, as well as the desired value of pressure at the outlet from the compressor, it is possible to calculate the actual temperature of CO<sub>2</sub> after compression which is given by Equation (1):

$$T'_2 = \frac{T_2 - T_1}{\eta_c} + T_1 \quad (1)$$

where:

- $T'_2$ —actual temperature of CO<sub>2</sub> after compression (K),
- $T_2$ —ideal temperature of CO<sub>2</sub> after compression (K),
- $T_1$ —temperature of CO<sub>2</sub> at the inlet to the compressor (K),
- $\eta_c$ —isentropic efficiency of the compressor (-).

**Table 2.** Assumptions and constraints of S-CO<sub>2</sub> cycle.

Parameter	Value
Maximum operating pressure of the cycle	22.5 MPa
Maximum operating temperature of the cycle	Limited by the exhaust gases temperature (616.38 K)
Minimum operating pressure of the cycle	7.7 MPa
Minimum operating temperature of the cycle	308.15 K
Pressure drop during the heat exchange	0.5% of the inlet pressure
Isentropic efficiency of the compressor	0.85
Isentropic efficiency of the turbine	0.85
Efficiency of the heat exchanger	0.95

Knowing the temperature and pressure of CO<sub>2</sub> at the compressor outlet, it is possible to calculate the enthalpy of CO<sub>2</sub> in stage 2. As a result, the required compressor power can be calculated based on Equation (2):

$$W_{1-2} = m_{CO_2} \cdot (h_2 - h_1) \quad (2)$$

where:

- $W_{1-2}$ —required compressor power (W),
- $m_{CO_2}$ —mass stream of CO<sub>2</sub> (kg/s),
- $h_2$ —specific enthalpy of CO<sub>2</sub> at the compressor outlet (J/kg),
- $h_1$ —specific enthalpy of CO<sub>2</sub> at the compressor inlet (J/kg).

At this stage of the calculations, the CO<sub>2</sub> stream is unknown, but it is not required. In the next step, the model determines the state parameters (pressure and temperature) of the CO<sub>2</sub> in stage 3 of the cycle (at the inlet to the main heat exchanger). The pressure is determined based on the assumption of a 0.5% pressure drop in the recuperator (in relation to the pressure from stage 2). The temperature at the inlet to the heat exchanger depends on the assumed cooling temperature of the flue gas and should be lower by a few degrees from the mentioned cooling temperature of the exhaust gases. Based on the pressure and temperature in stage 3, it is possible to calculate the enthalpy and entropy ( $h_3, s_3$ ).

On the same principle, the model calculates the state parameters in stage 4, except that the CO<sub>2</sub> pressure at the heat exchanger outlet is calculated based on the assumption of a 0.5% pressure drop in relation to the pressure from stage 3. The temperature in stage 4 depends on the temperature of the exhaust gases at the heat exchanger inlet and should be a few degrees lower than the mentioned exhaust gases temperature. Based on the pressure and temperature in stage 4, it is possible to calculate the enthalpy and entropy ( $h_4, s_4$ ). Having calculated the enthalpy at stages 3 and 4, the following Equation (3) determines the CO<sub>2</sub> mass stream, which is dependent on the assumed exhaust gas cooling temperature.

$$m_{CO_2} = \frac{m_h \cdot (h_{IN} - h_{out}) \cdot \eta_h}{h_4 - h'_3} \quad (3)$$

where:

- $m_{CO_2}$ —mass stream of CO<sub>2</sub> (kg/s),
- $m_h$ —mass stream of waste heat (kg/s),
- $h_{IN}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side inlet (J/kg),
- $h_{OUT}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet (J/kg),
- $h_4$ —specific enthalpy of CO<sub>2</sub> at the heat exchanger cold side outlet (J/kg),
- $h'_3$ —specific enthalpy of CO<sub>2</sub> at heat exchanger cold side inlet (J/kg),

$\eta_h$ —efficiency of the heat exchange (-).

The next stage of the cycle is CO<sub>2</sub> passing through the turbine to generate mechanical energy (step 5). First, the model assumes a pressure to which the CO<sub>2</sub> will expand. Knowing the pressure at the compressor inlet (minimum operating pressure of the cycle—7.7 MPa) and knowing that CO<sub>2</sub> after passing through the turbine goes to the recuperator, it is necessary to take into account the pressure drop in the recuperator (no pressure drop was assumed on the cooler) and thus determine the pressure at the turbine outlet. The CO<sub>2</sub> temperature in stage 5 is calculated based at the assumption that the turbine expansion can be described by an adiabatic process which is given by Equation (4):

$$T'_5 = (T_5 - T_4) \cdot \eta_T + T_4 \quad (4)$$

where:

$T'_5$ —actual CO<sub>2</sub> temperature after expansion (K),

$T_5$ —ideal CO<sub>2</sub> temperature after expansion (K),

$T_4$ —CO<sub>2</sub> turbine inlet temperature (K),

$\eta_p$ —isentropic efficiency of the compressor (-).

Knowing the temperature and pressure of CO<sub>2</sub> at the turbine outlet, it is possible to calculate the CO<sub>2</sub> enthalpy at stage 5. Thanks to this, the power on the turbine shaft can be calculated based on Equation (5):

$$W_{4-5} = m_{CO_2} \cdot (h'_4 - h_5) \quad (5)$$

where:

$W_{4-5}$ —the power on the turbine shaft (W),

$m_{CO_2}$ —mass stream of CO<sub>2</sub> (kg/s),

$h'_4$ —specific enthalpy of CO<sub>2</sub> at the turbine inlet (J/kg),

$h_5$ —specific enthalpy of CO<sub>2</sub> at the turbine outlet (J/kg).

The net power of the system was assumed as the power difference on the turbine shaft and the required power of the compressor by Equation (6):

$$W_{net} = W_{4-5} - W_{1-2} \quad (6)$$

where:

$W_{net}$ —the net power of the system (W),

$W_{4-5}$ —the power on the turbine shaft (W),

$W_{1-2}$ —required compressor power (W).

The last stage of the model calculation is stage 6 of the cycle. The cooler has no pressure drop, so that the pressure at the compressor inlet is identical to the inlet pressure to the cooler. Using the CoolProp database, the model determines the CO<sub>2</sub> temperature at the inlet to the cooler based on the CO<sub>2</sub> pressure and enthalpy in stage 6. The CO<sub>2</sub> enthalpy value in stage 6 is calculated according to Equation (7):

$$h_6 = h'_5 - \frac{h_3 - h'_2}{\eta_h} \quad (7)$$

where:

$h_6$ —specific enthalpy of CO<sub>2</sub> at the recuperator hot side outlet (J/kg),

$h'_5$ —specific enthalpy of CO<sub>2</sub> at the recuperator hot side inlet (J/kg),

$h_3$ —specific enthalpy of CO<sub>2</sub> at the recuperator cold side outlet (J/kg),

$h'_2$ —specific enthalpy of CO<sub>2</sub> at the recuperator cold side inlet (J/kg),

$\eta_h$ —efficiency of the heat exchange (-).

For comparative purposes, it was decided to define parameters that would allow to evaluate the energy efficiency of the cycle for different operating conditions. The first parameter is the waste heat utilization rate which is given by Equation (8):



$$WHUR = \frac{W_{net}}{(h_{IN} - h_{AM})} \cdot 100\% \quad (8)$$

where:

- $WHUR$ —waste heat utilization rate (%),
- $h_{IN}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side inlet (J/kg),
- $h_{AM}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet, assuming exhaust gases cooling to 25 °C (298.15 K) (J/kg).

The next parameter is the percent of waste heat transferred to CO<sub>2</sub>, which is given by Equation (9):

$$PWHT = \frac{(h_{IN} - h_{OUT})}{(h_{IN} - h_{AM})} \cdot 100\% \quad (9)$$

where:

- $PWHT$ —percent of waste heat transferred to CO<sub>2</sub> (%),
- $h_{IN}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side inlet (J/kg),
- $h_{OUT}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet (J/kg),
- $h_{AM}$ —specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet, assuming exhaust gases cooling to 25 °C (298.15 K) (J/kg).

The last parameter allowing to evaluate the energy efficiency of the cycle is internal cycle efficiency, which is given by Equation (10):

$$CE = \frac{(h'_4 - h_5) - (h_2 - h_1)}{(h_4 - h_3)} \cdot 100\% \quad (10)$$

where:

- $CE$ —cycle efficiency (%),
- $h_5$ —specific enthalpy of CO<sub>2</sub> at the turbine outlet (J/kg),
- $h'_4$ —specific enthalpy of CO<sub>2</sub> at the turbine inlet (J/kg),
- $h_4$ —specific enthalpy of CO<sub>2</sub> at the heat exchanger cold side outlet (J/kg),
- $h'_3$ —specific enthalpy of CO<sub>2</sub> at the heat exchanger cold side inlet (J/kg),
- $h_2$ —specific enthalpy of CO<sub>2</sub> at the compressor outlet (J/kg),
- $h_1$ —specific enthalpy of CO<sub>2</sub> at the compressor inlet (J/kg).

#### 4. Cycle Operation Parameters and Calculation Results

As part of this research, the authors performed a series of simulations of the Brayton system with S-CO<sub>2</sub>. It was established that the lower heat source is the energy from the exhaust gas of reciprocating engines which drives compressors at the gas compressor station. During the simulation the following operating parameters of the system were changed: (i) turbine inlet pressure, (ii) degree of exhaust gas cooling, (iii) temperature difference of CO<sub>2</sub> and flue gas in the heat exchanger, (iv) temperature of CO<sub>2</sub> after passing the hot side of recuperator were changed.

Aggregated simulation results in the form of graphs and discussions are presented on the following pages. Additionally, exemplary system operation parameters are presented, which assumes the cooling of exhaust gases to a temperature of 200 °C and compression of CO<sub>2</sub> to a pressure of 20 MPa. It also has been established that the temperature difference between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger is 5 K. The simulated cycle for the above mentioned operating conditions is presented in Figures 2 and 3 in the form of temperature/entropy and pressure/enthalpy plots. Stages 1–6 correspond to the stages described in Sections 2 and 3.



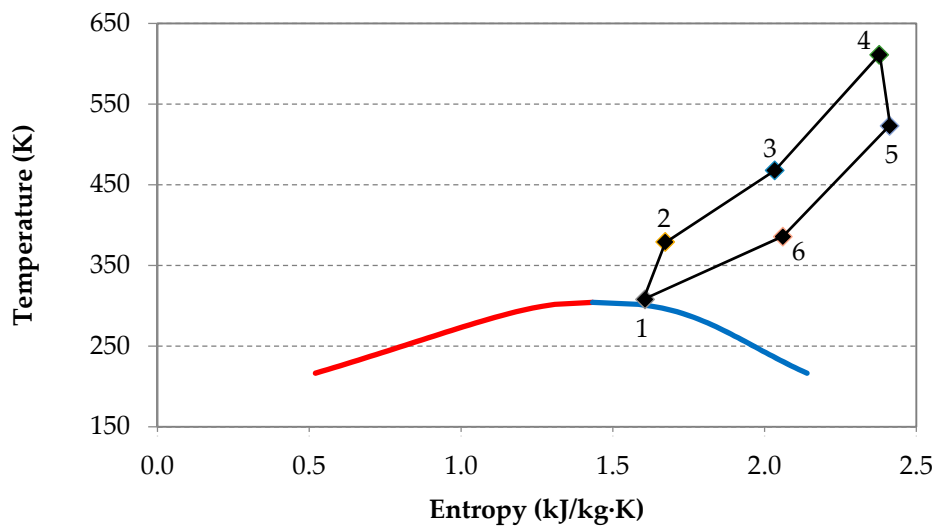


Figure 2. Temperature vs. entropy of S-CO<sub>2</sub> cycle.

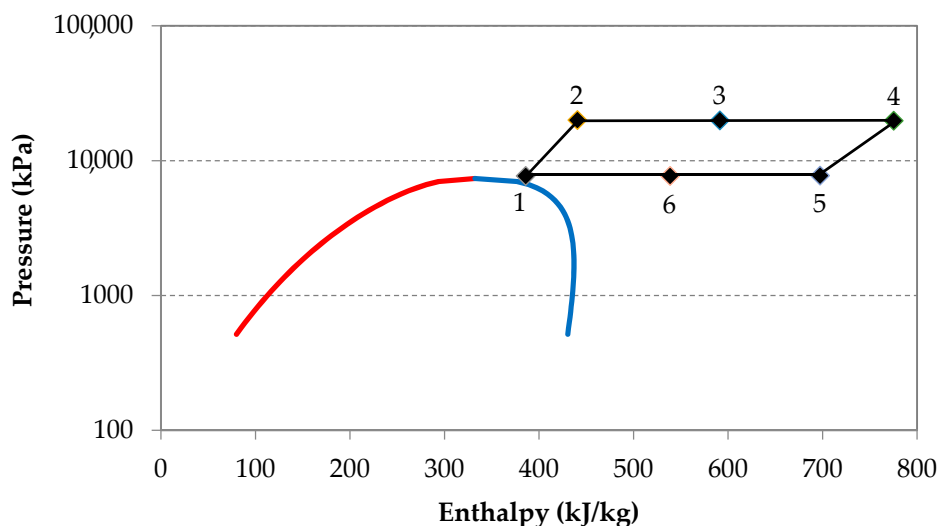


Figure 3. Pressure vs. enthalpy of S-CO<sub>2</sub> cycle.

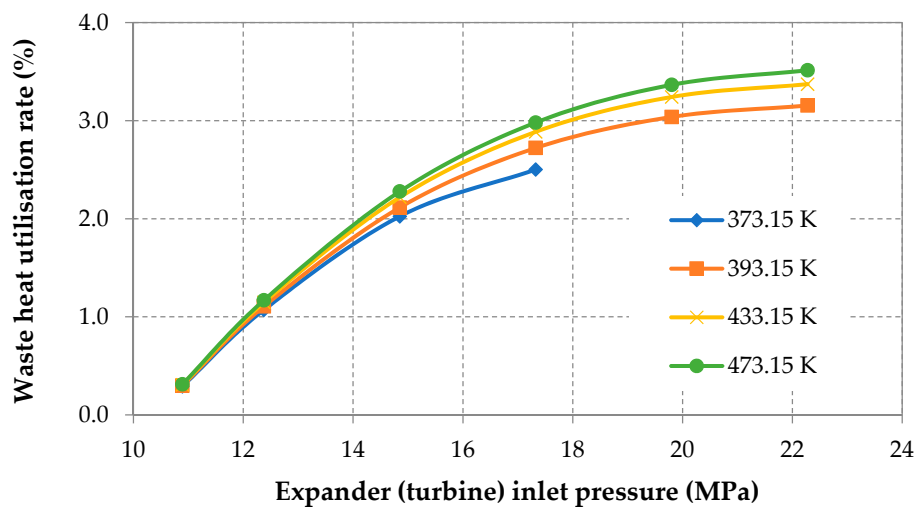
Additional conditions of the described above exemplary cycle are as follows: mass stream of flue gases: 3.55 kg/s, temperature of flue gases at heat exchanger inlet: 616.38 K, temperature of flue gases at heat exchanger outlet: 473.15 K, mass stream of CO<sub>2</sub>: 3.04 kg/s, the net power system  $W_{net}$ : 40.86 kW, waste heat utilization ratio  $WHUR$ : 3.37%, waste heat transferred to CO<sub>2</sub>  $PWHT$ : 47.29%, and cycle efficiency  $CE$ : 12.73%. Calculated temperature, pressure, enthalpy, and entropy values at successive stages are presented in Table 3.

Table 3. Parameters describing S-CO<sub>2</sub> cycle presented in Figures 2 and 3.

Stage	Temperature (K)	Pressure (kPa)	Enthalpy (kJ/kg)	Entropy (kJ/(kg·K))
1	308.15	7700.00	385.83	1.61
2	379.38	20,000.00	440.41	1.67
3	468.15	19,900.00	591.15	2.03
4	611.38	19,800.50	775.09	2.38
5	523.27	7738.50	697.10	2.41
6	385.83	7700.00	538.43	2.06

#### 4.1. Investigation on Turbine Inlet Pressure Change

It is well known that the efficiency of a Brayton cycle is highly dependent on the turbine inlet temperature and pressure ratio [6,26,27]. In this work, it was decided to investigate how the change in the turbine inlet pressure will affect the cycle performance, assuming constant pressure at the turbine outlet (7.74 MPa). Different turbine inlet pressures from 11 to 22.5 MPa were assumed. For each of the assumed pressures, the energy efficiency indicators were determined for different values of exhaust gas temperatures at the heat exchanger outlet, ranging from 100 (373.15 K) to 200 °C (473.15 K). In addition, it was assumed that the temperature difference between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger is always constant and is 5 K. Figure 4 presents the relationship between turbine inlet temperature and waste heat utilization rate.

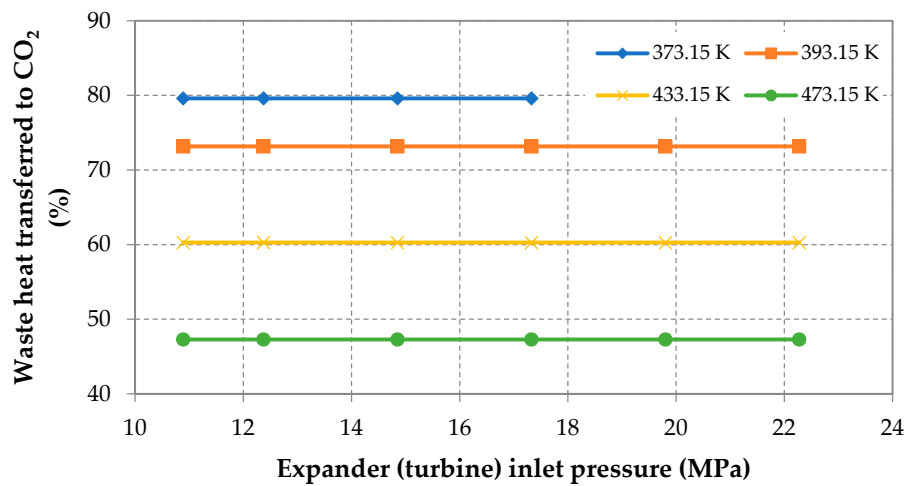


**Figure 4.** Relationship between turbine inlet temperature and waste heat utilization rate for the different exhaust gas temperature at the heat exchanger outlet.

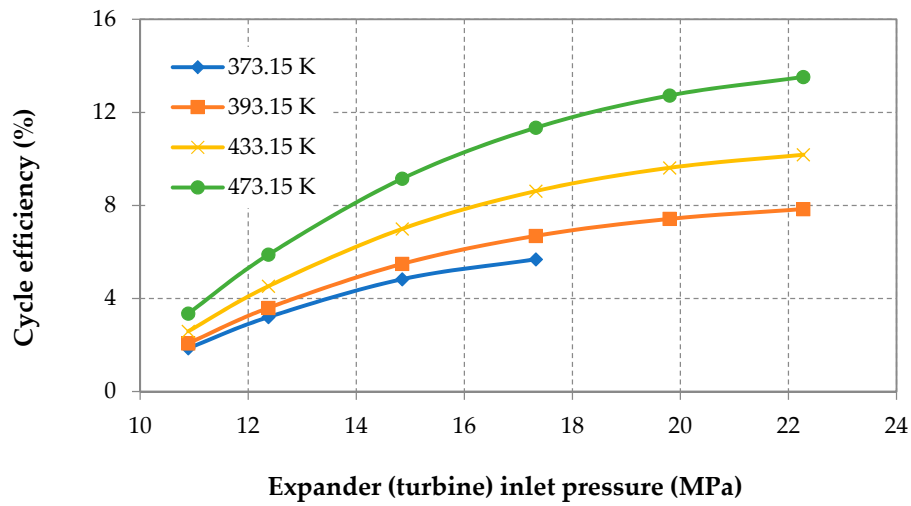
The plot presented in Figure 4 shows the dependence according to which the waste heat utilization rate grows with the exhaust gases temperature at the heat exchanger outlet and with the turbine inlet pressure. Waste heat utilization rate (Equation (7)) is a quotient in which the denominator has a constant value. This means that the higher the *WHUR* value, the larger the counter, which refers to the net power of the system. The higher the turbine inlet pressure, the greater the difference between the power on the turbine shaft and the required compressor power. This is due to the compression of CO<sub>2</sub> near the critical point. The *WHUR* growth is related to the exhaust gas temperature increase at the heat exchanger outlet, which results from the use of a recuperator. At the assumed flue gases temperature at the exchanger outlet—100 °C (373.15 K)—the calculation results are shown for the maximum turbine inlet pressure of 17 MPa. This is due to the fact that, for higher pressures, the temperature of CO<sub>2</sub> after compression was greater than the assumed exhaust gas cooling temperatures. This applies to the plots in Figures 4–7. In Figure 5 the relationship between turbine inlet temperature and percent of waste heat transferred to CO<sub>2</sub> is shown.

The analysis assumes that waste heat transferred to CO<sub>2</sub> is constant for a given exhaust gas temperature at the heat exchanger outlet. *PWHT* is described by Equation (8), where all variables are independent of the inlet pressure to the turbine. Figure 5 presents examples of *PWHT* values for the described above assumptions.

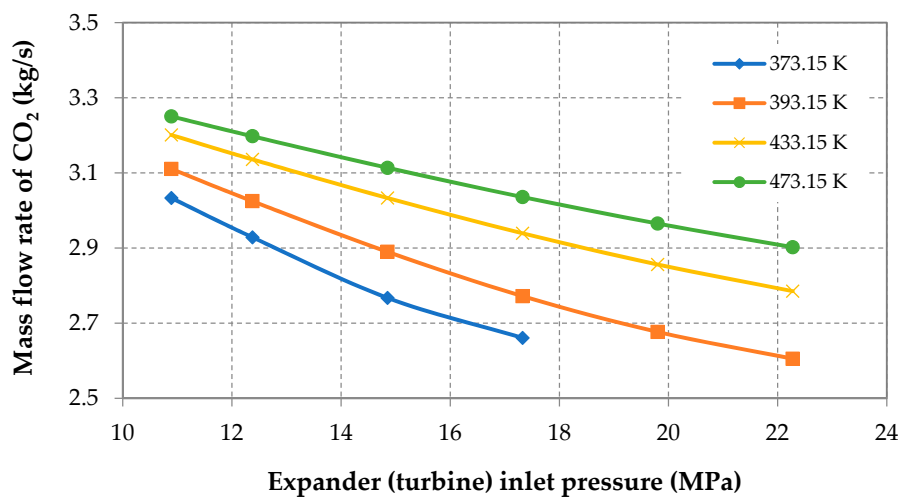
The last energy efficiency indicator is cycle efficiency. The relationship between turbine inlet temperature and cycle efficiency is presented in Figure 6.



**Figure 5.** Relationship between turbine inlet temperature and percent of waste heat transferred to CO<sub>2</sub> for the different exhaust gas temperatures at the heat exchanger outlet.



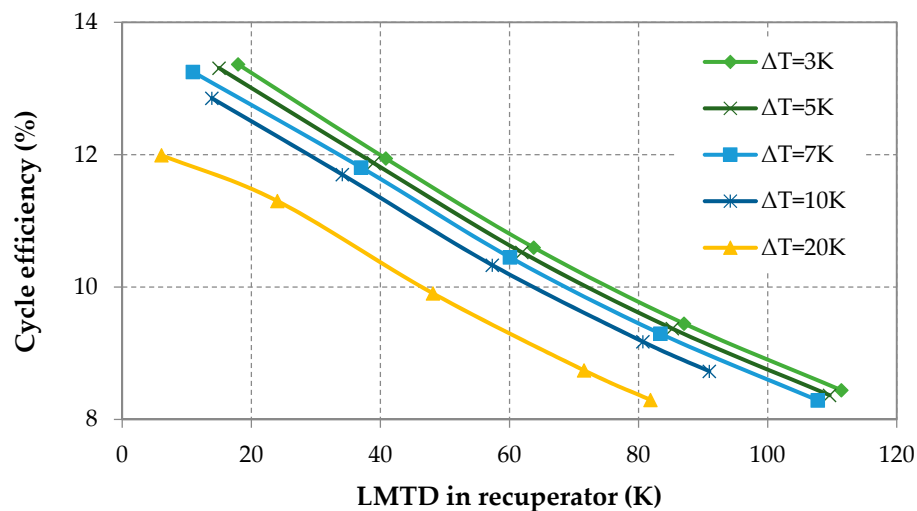
**Figure 6.** Relationship between turbine inlet temperature and percent of waste heat transferred to CO<sub>2</sub> for the different exhaust gas temperatures at the heat exchanger outlet.



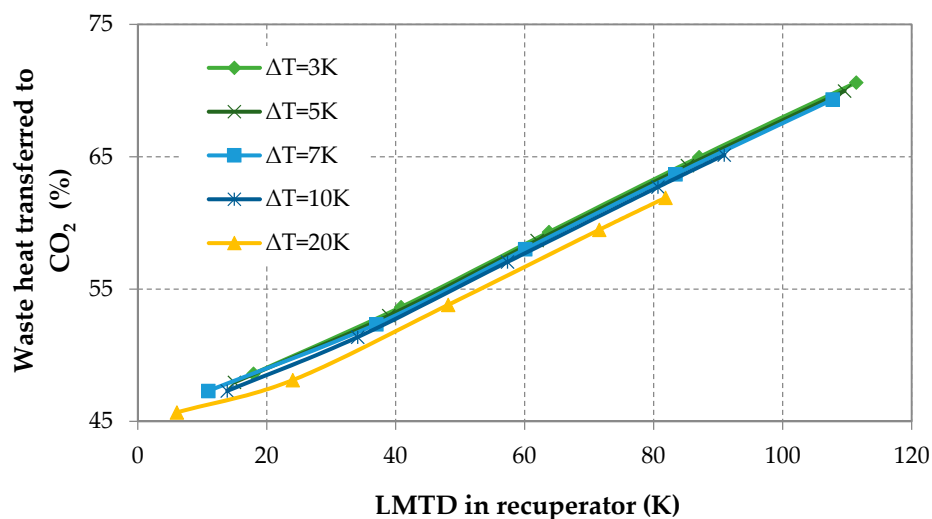
**Figure 7.** Relationship between turbine inlet pressure and mass flow rate of CO<sub>2</sub> for the different exhaust gas temperatures at the heat exchanger outlet.

Figure 6 shows that the higher the exhaust gases temperature at the heat exchanger outlet and the higher the turbine inlet pressure, the greater the cycle efficiency. Cycle efficiency refers to the internal efficiency of the system and is independent of the total waste heat amount, which is available in the exhaust gases (with assumption of exhaust gases cooling to 25 °C). The increase in the CE value with the increase of the turbine inlet pressure results from the fact that the higher the turbine inlet pressure, the greater the difference between the power on the turbine shaft and the required compressor power. This is due to the compression of CO<sub>2</sub> near the critical point. The increase in CE value with the growth of exhaust gases temperature at the heat exchanger outlet (similar to WHUR), results from the use of a recuperator, which improves the efficiency of the cycle (Figures 8–10).

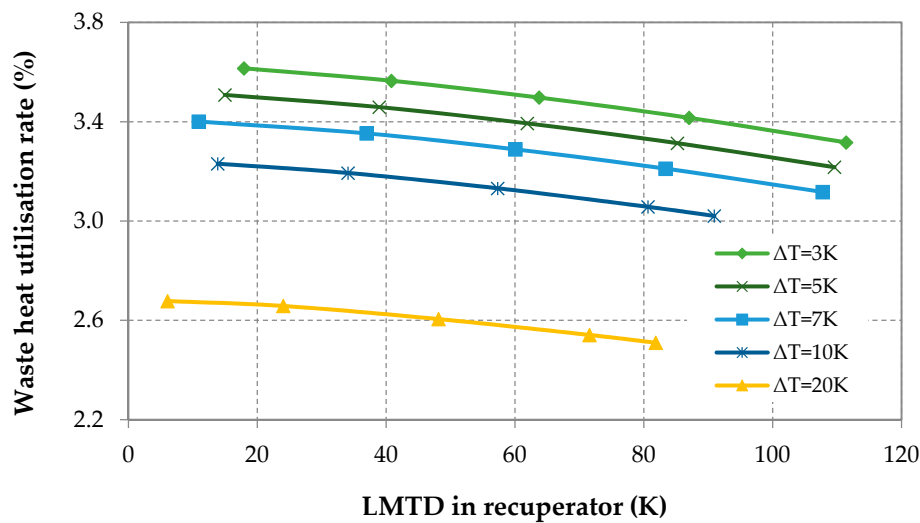
As the mass flow rate of CO<sub>2</sub> is also an important parameter, in Figure 7 the relationship between turbine inlet pressure and mass flow rate of CO<sub>2</sub> is presented.



**Figure 8.** Relationship between the *LMTD* and cycle efficiency for varied temperature differences between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger.



**Figure 9.** Relationship between the *LMTD* and percent of waste heat transferred to CO<sub>2</sub> for varied temperature differences between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger.



**Figure 10.** Relationship between the *LMTD* and waste heat utilization rate for varied temperature differences between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger.

The CO<sub>2</sub> mass flow was not the assumed value. CO<sub>2</sub> mass flow was calculated as the required flow so that the given amount of waste heat could be transferred to the S-CO<sub>2</sub> system. In the pressure and temperature range assumed for the system operation, the CO<sub>2</sub> enthalpy value (for a constant temperature) decreases with increasing pressure. In the case of the discussed system, this is especially noticeable after passing CO<sub>2</sub> through a recuperator (cold side). Assuming a constant heat flux to be transferred to the system, the smaller enthalpy at the main heat exchanger inlet causes that the CO<sub>2</sub> velocity in the exchanger must be lower, and hence also the mass stream. When it comes to the increase in the mass stream of CO<sub>2</sub> with the growth of exhaust gases temperature at the heat exchanger outlet, this is due to the fact that less energy from the outside is required for transfer to the system (the merit of the recuperator, which raises the temperature of CO<sub>2</sub> at the heat exchanger inlet). The less energy from the outside that must be transferred to the system, the higher the CO<sub>2</sub> velocity in the exchanger that can be achieved, and hence the mass stream. Figures 4–7 show that if the temperature of exhaust gases at heat exchanger outlet is higher, the cycle efficiency is better.

#### 4.2. Investigation on Temperature Difference between the Hot and Cold Side of the Recuperator

To investigate the reasons for the cycle efficiency growth related to the increase of temperature of exhaust gases at heat exchanger outlet, it was decided to additionally examine the influence of temperature difference between the hot and cold sides of the recuperator. For this purpose, the authors defined logarithmic mean temperature difference of the recuperator given by Equation (11):

$$LMTD = \frac{(T_2'' - T_3) - (T_6 - T_2'')}{LN\left(\frac{T_5'' - T_3}{T_6 - T_2''}\right)} \quad (11)$$

where:

*LMTD*—logarithmic mean temperature difference of the recuperator (K),

$T_2''$ —temperature of CO<sub>2</sub> at the recuperator cold side inlet (K),

$T_3$ —temperature of CO<sub>2</sub> at the recuperator cold side outlet (K),

$T_5''$ —temperature of CO<sub>2</sub> at the recuperator hot side inlet (K),

$T_6$ —temperature of CO<sub>2</sub> at the recuperator hot side outlet (K).

Calculations of the energy efficiency indicators were carried out for different *LMTD* values, for a constant turbine inlet pressure of 22.5 MPa. In addition, the same calculations were repeated for

varied temperature differences between CO<sub>2</sub> and the exhaust gases at the inlet and outlet of the heat exchanger, ranging from 3 to 20 K. The relationship between the *LMTD* and cycle efficiency is presented in Figure 8.

Figure 8 shows that the internal efficiency of the cycle increases if the difference in temperature of CO<sub>2</sub> at the entry to and exit from the recuperator (between hot and cold side) is smaller. The smaller temperature difference between CO<sub>2</sub> streams flowing countercurrent means that more heat of returning CO<sub>2</sub> is transferred to CO<sub>2</sub> after compression. Otherwise, this energy would have to be “taken” from the system at the cooler so the energy would be “wasted”. The increase in *CE* with the decrease in the *LMTD* results from the fact that the smaller the *LMTD*, the more energy is transferred from the hot side of recuperator to the cold side of recuperator. By this way, more internal energy in the system is re-used and thus the efficiency of the cycle is increasing. The relationship between the *LMTD* and percent of waste heat transferred to CO<sub>2</sub> is presented in Figure 9.

In Figure 9 a very clear relationship is presented, where, with the increase in *LMTD* in recuperator, the percent of waste heat transferred to CO<sub>2</sub> also increases. This results from the fact that with larger values of *LMTD* in recuperator, more energy is “wasted” and “taken” from the system at the cooler. To compensate this, a correspondingly greater amount of waste heat from the flue gases must be supplied to the system.

In Figure 10 the relationship between the *LMTD* and waste heat utilization rate is presented.

Waste heat utilization rate is an energy efficiency indicator, which combines *CE* and *PWHT*. The value of *WHUR* increases both with the growth in *CE* and *PWHT*. The data presented in Figure 10 shows that the *WHUR* value increases slightly with the decrease of *LMTD* in recuperator.

Results presented in Figures 8–10, show that for the adopted operating conditions, the increase in internal efficiency has a greater impact on the *WHUR* than decrease in percent of waste heat transferred to CO<sub>2</sub> (*PWHT*). The increase of *WHUR* is much more significant when the temperature difference between CO<sub>2</sub> and flue gases in the heat exchanger decreases. This is due to the fact that both *CE* and *PWHT* grow with the temperature difference drop. By overlapping these growths, the increase in *WHUR* is evident.

## 5. Results and Discussion

As mentioned in the introduction, no other publications were found that presented an analysis of S-CO<sub>2</sub> cycle application for the waste heat recovery at a natural gas compressor station. Detailed comparison of cycle efficiency indicators values between its applications in various industries is pointless, because these are different operating parameters (power of devices, waste heat temperature, etc.). For example, Kim et al. [6] studied the supercritical CO<sub>2</sub> power cycle for landfill gas fired gas turbine bottoming cycle. They concluded that for the simple recuperated cycle the net produced work is 2.18 MWe with 29.98% cycle net efficiency. For comparison, in this paper maximum net power system was 47 kW with cycle efficiency 14%. However, it should be noticed that Kim et al. assumed exhaust temperature 519.69 °C and mass flow rate of the flue gases of 21.3 kg/s. In this work it was, respectively, 343.23 °C and 3.55 kg/s. Similarly, the maximum system operating pressure was taken as 27.6 MPa (22.5 MPa in this paper).

Taking this into account, it was decided to compare the values of selected S-CO<sub>2</sub> efficiency indicators with ORC efficiency indicators, analyzed for the same gas compressor station. Kowalski et al. [2] analyzed the application of ORC system at Jarosław II natural gas compression station. The ORC system was proposed for the production of eclectic energy. They conclude that the average net electric power of about 32 kWe could be obtained when ORC system was fed with waste heat from a compressor engine. The obtained exergy efficiency of the cycle was about 7.06%. In another paper, Kostowski et al. [3] analyzed the methods of waste heat recovery at gas compressor station. They performed a case study for the Jarosław II gas compressor station and compared different technologies including ORC. The ORC output power was estimated in a simplified black-box approach based on

nominal parameters and performance curves given by the manufacturer (Heat Recovery Solutions). The average net electrical output was assumed as 140 kWe.

The above discrepancy does not allow to clearly state which technology would be better to use at the Jarosław II gas compressor station. The high value of net electrical power in for a Heat Recovery Solutions product is caused (probably) due to the advanced optimization of the entire ORC system. At first glance, the advantage of the ORC system seems to be indisputable, but in this paper the analysis of the S-CO<sub>2</sub> cycle was limited to the simple recuperated cycle and to conditions which occur at natural gas compressor stations.

The cycle efficiency and heat recovery coefficient must be carefully considered in the context of waste heat recovery system planning. These factors are directly related to the recovery work from the bottoming cycles. The net electrical efficiency of a heat recovery system is described as the product of cycle efficiency and the recovery rate of waste heat. In this case study, the defined waste heat utilization rate (*WHUR*) depends on both the cycle efficiency (*CE*) and the percent of waste heat transferred to CO<sub>2</sub> (*PWHT*). The value of *WHUR* increases both with the growth in *CE* and *PWHT*. Thus, when optimizing the operational conditions of the waste heat recovery system, the aim is not only to maximize cycle efficiency (as for example like in fired plants). During operation optimization of the waste heat recovery system, it is necessary to take into account both cycle efficiency and percent of waste heat transferred to working fluid. It results from the fact that a significant increase in the *CE* value does not have such a large impact on the *WHUR* value. Each decrease in *CE* is compensated (not 100%) by an increase in the *PWHT* value. This dependence is very important, because the internal efficiency of the system can be increased by using, e.g., (i) more recuperators, (ii) stream splitting between several heat exchangers, etc. However, the increase in *WHUR* will not be proportional to the growth of *CE*. Implementation of additional devices in the system will increase its investment costs. In this case, when the management of waste heat is considered, the priority should be its maximum recovery (*WHUR*) with taking into account economic aspects. It may turn out that, instead of a complicated system, it will be possible to create a simpler system that will use more waste heat (*PWHT*) to increase the CO<sub>2</sub> mass stream in the system. Increase in *WHUR* and *PWHT* is associated with heat exchange processes in exchangers and recuperators. That indicates one of the potential directions of further investigations on various heat exchanger solutions that will cooperate with S-CO<sub>2</sub> in a selected temperature range.

Internal efficiency increase can bring other benefits. Waste heat after passing through a heat exchanger, can be reused in another process. An analysis of operational parameters (*WHUR*, *PWHT*, *CE*) of the Bryton S-CO<sub>2</sub> cycle at the natural gas compressor station, showed that the temperature of the flue gases after passing the heat exchanger is high enough to reuse it (flue gases) as a heat source in an ORC system (there are organic fluids which operate at 100–200 K). That indicates additional direction for further research on Brayton S-CO<sub>2</sub> cycle which operates in a cascade with the ORC system.

## 6. Conclusions

The aim of this case study was to estimate the possibility of application of the Brayton cycle with supercritical CO<sub>2</sub> to utilize the waste heat from the exhaust gases at the natural gas compressor station. To achieve this, certain boundary conditions were assumed (e.g., waste heat parameters). The research was conducted in terms of impact of changes in the operating conditions of the S-CO<sub>2</sub> system on defined energy efficiency indicators (temperature, pressure). The research demonstrated that energy efficiency indicators depend on the adopted operating conditions.

Other research groups results present better efficiency indicators of the Bryton S-CO<sub>2</sub> cycle in comparison to the ORC cycle (in other branches of industry). Analysis of selected efficiency indicators values for S-CO<sub>2</sub> and ORC (analyzed for the same natural gas compressor station) did not show direct advantages of S-CO<sub>2</sub> cycle over ORC. This is due to the fact that ORC technology for the natural gas compressor station applications is relatively developed and thus the entire system is well optimized. In subsequent scientific studies, the authors focus on optimization of the Brayton S-CO<sub>2</sub> cycle in terms of net electricity produced, which will allow to compare results with industrially available ORC



technologies. However, it should be taken into consideration that the Brayton S-CO<sub>2</sub> system has a number of technical advantages over the ORC system. The most important advantages are (i) lower investment costs, (ii) smaller streams of working medium in circulation, (iii) lower price of the working medium, (iv) properties of the working medium (ORC fluids are often hazardous). Therefore, it may turn out that, despite the slightly lower efficiency of the Brayton cycle, it will be more attractive for the investor.

The case study demonstrated the possibility of application of the S-CO<sub>2</sub> Brayton cycle for waste heat parameters (temperature, stream) at a natural gas compressor station. It was proved that the increase in waste heat utilization rate (*WHUR*) is not proportional to the increase in cycle efficiency (*CE*). This makes it necessary to perform an individual case study, taking into account economic aspects. Additionally, disproportion indicates that it is better to create a simpler S-CO<sub>2</sub> installation (i.e., less recuperators), which will use more waste heat (*PWHT*), thus increasing the mass flow of CO<sub>2</sub> in the installation instead of complex system design (which is common practice). The study identified new potential directions for further research on application of S-CO<sub>2</sub> at natural gas compressor stations, e.g., (i) focus on different types of heat exchangers and study of the S-CO<sub>2</sub> heat flow in a specified temperature range; (ii) analysis of the Brayton S-CO<sub>2</sub> cycle which operates in a cascade with ORC.

The presented case study did not focus on issues related to the use of mechanical work generated by the turbine. However, it should be pointed out that this an issue is as important as maximizing indicators values of the (i) cycle efficiency (*CE*) and (ii) the percent of waste heat transferred to CO<sub>2</sub> (*PWHT*). Currently, the most commonly considered solution is the use of generated work to produce electricity (due to flexibility of electricity management system). Nowadays, heat recovery technologies are expensive, which means that there are not many such installations at natural gas compressor stations in the world. Regardless of that, applicability of results of this study may be beneficial when considering other scenarios of Brayton S-CO<sub>2</sub> use, e.g., a Brayton S-CO<sub>2</sub> system which operates in a cascade with the ORC system. Another application of the results of this study may be beneficial to underground gas storage magazines where the S-CO<sub>2</sub> Brayton Cycle also could be used.

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## Abbreviations

### Symbols

$h_1$	specific enthalpy of CO <sub>2</sub> at the compressor inlet (J/kg)
$h_2$	specific enthalpy of CO <sub>2</sub> at the compressor outlet (J/kg)
$h'_2$	specific enthalpy of CO <sub>2</sub> at the recuperator cold side inlet (J/kg)
$h_3$	specific enthalpy of CO <sub>2</sub> at the recuperator cold side outlet (J/kg)
$h'_3$	specific enthalpy of CO <sub>2</sub> at the heat exchanger cold side inlet (J/kg)
$h_4$	specific enthalpy of CO <sub>2</sub> at the heat exchanger cold side outlet (J/kg)
$h'_4$	specific enthalpy of CO <sub>2</sub> at the turbine inlet (J/kg)
$h_5$	specific enthalpy of CO <sub>2</sub> at the turbine outlet (J/kg)
$h'_5$	specific enthalpy of CO <sub>2</sub> at the recuperator hot side inlet (J/kg)
$h_6$	specific enthalpy of CO <sub>2</sub> at the recuperator hot side outlet (J/kg)
$h_{IN}$	specific enthalpy of waste heat (flue gases) at heat exchanger hot side inlet (J/kg)
$h_{OUT}$	specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet (J/kg)
$h_{AM}$	specific enthalpy of waste heat (flue gases) at heat exchanger hot side outlet, assuming exhaust gases cooling to 25 °C (298.15 K) (J/kg)
$m_{CO_2}$	mass stream of CO <sub>2</sub> (kg/s)
$m_h$	mass stream of waste heat (kg/s)

**Symbols**

$T_1$	temperature of CO <sub>2</sub> at the inlet to the compressor (K)
$T_2$	temperature of CO <sub>2</sub> at the recuperator cold side inlet (K)
$T'_2$	actual temperature of CO <sub>2</sub> after compression (K)
$T''_2$	temperature of CO <sub>2</sub> at the recuperator cold side inlet (K)
$T_3$	temperature of CO <sub>2</sub> at the recuperator cold side outlet (K)
$T_4$	CO <sub>2</sub> turbine inlet temperature (K)
$T_5$	ideal CO <sub>2</sub> temperature after expansion (K)
$T'_5$	actual CO <sub>2</sub> temperature after expansion (K)
$T''_5$	temperature of CO <sub>2</sub> at the recuperator hot side inlet (K)
$T_6$	temperature of CO <sub>2</sub> at the recuperator hot side outlet (K)
$W_{1-2}$	required compressor power (W)
$W_{4-5}$	the power on the turbine shaft (W)
$W_{net}$	the net power of the system (W)

**Greek symbols**

$\eta_c$	isentropic efficiency of the compressor (-)
$\eta_h$	efficiency of the heat exchange (-)
$\eta_p$	isentropic efficiency of the compressor (-)

**Abbreviations**

BC	Brayton cycle
CE	cycle efficiency (%)
LMTD	logarithmic mean temperature difference of the recuperator (K)
ORC	organic Rankine's cycle
PWHT	percent of waste heat transferred to CO <sub>2</sub> (%)
S-CO <sub>2</sub>	supercritical CO <sub>2</sub> Brayton cycles
WHUR	waste heat utilization rate (%)

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