Parametric Study of a Supercritical CO2 Power Cycle for Waste Heat Recovery with Variation in Cold Temperature and Heat Source Temperature

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Abstract:

A supercritical carbon dioxide (S-CO_2) power cycle is a promising development for waste heat recovery (WHR) because of its high efficiency despite its simplicity and compactness compared with a steam bottoming cycle. A simple recuperated S-CO_2 power cycle cannot fully utilize the waste heat because of the trade-off between heat recovery and thermal efficiency of the cycle. A split-cycle in which the working fluid is preheated by the recuperator and the heat source separately can be used to maximize the power output from a given waste heat source. In this study, the operating conditions of the split S-CO_2 power cycles for waste heat recovery from a gas turbine and an engine were studied to accommodate the temperature variation of the heat sink and the waste heat source. The results show that it is vital to increase the low pressure of the cycle along with a corresponding increase in the cooling temperature to keep the low-compression work near the critical point. The net power decreases by 6 to 9% for every 5 °C rise in the cooling temperature from 20 °C to 50 °C because of the decrease in heat recovery and thermal efficiency of the cycle. The effect of the heat-source temperature on the optimal low-pressure side is negligible, and the optimal high pressure of the cycle increases with a rise in the heat-source temperature. As the heat-source temperature increases in steps of 50 °C from 300 °C to 400 °C, the system efficiency increases by about 2% (absolute efficiency), and the net power significantly increases by 30 to 40%.

Keywords:

Supercritical CO_2 (S-CO_2) power cycle, Waste heat recovery, Exhaust gas, Cold temperature, Split-cycle

1. Introduction

A supercritical carbon dioxide (S-CO_2) power cycle has been studied for diverse applications, including nuclear, concentrated solar, fossil fuel, and waste heat recovery [1], and the S-CO_2 power cycle has been optimized for each application [2]. In waste heat recovery (WHR), the purpose of cycle optimization is not to maximize the thermal efficiency of the cycle, but to maximize the power output from the waste heat source. It is essential to incorporate the thermal efficiency of the cycle (cycle efficiency) and the utilization efficiency of the waste heat (heat recovery efficiency) to maximize the power output of the WHR cycle [3]. For the cycle efficiency, it is essential to minimize the temperature difference for the heat transfer (exergy loss). For the recovery efficiency, it is essential to reduce the outlet temperature of the waste heat source as much as possible by waste heat recovery. Therefore, the optimal system configuration for achieving the maximum power from the waste heat source is different from that for a high-temperature heat source (nuclear, concentrated solar, and combustor).

From our previous researches, a split-cycle in which the recuperator and the heat source separately preheat the working fluid was proposed as a promising WHR cycle from a gas turbine. The split-cycle was able to achieve higher efficiency at a lower upper pressure and a lower turbine inlet temperature, using a simpler system than the cascade cycle in which a low-temperature (LT) loop was added to the high-temperature (HT) loop of the simple recuperated S-CO_2 power cycle [4]. The optimization and performance of the S-CO_2 power cycle are significantly dependent on the cooling condition of the cycle because the cooling condition of CO_2 before the compressor is very close to the critical point (31.1 °C and 73.9 bar) where the properties of CO_2 change significantly [5]. However, the previous studies on S-CO_2 power cycle for gas turbine WHR [4] were limited to a fixed cooling CO_2 temperature of 20 °C.

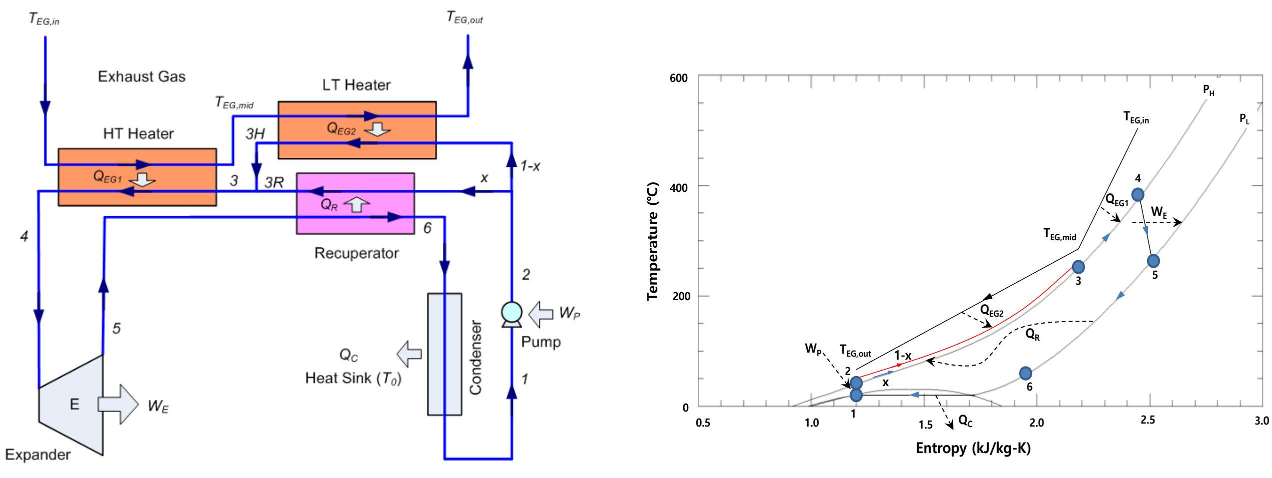
In this study, the operating conditions of the split S-CO_2 power cycle for a gas turbine WHR were investigated to accommodate the temperature variation of the heat sink. Furthermore, to apply the split S-CO_2 power cycle for WHR from an engine exhaust gas, the same parametric studies were done to accommodate the temperature variation of the waste heat source.

2. System analysis

2.1. System considered in investigation

First, as a waste heat source, an exhaust gas with a mass flow-rate of 69.8 kg/s at 538 °C (811 K) from a 25-MWe-class gas turbine was selected [6]. If the ambient temperature was 15 °C (288 K), the corresponding amount of waste heat was 40.9 MWth. Second, an exhaust gas with a mass flow rate of 51 kg/s at 300 °C (up to 400 °C) from a 30-MWe-class engine was selected as the waste heat source (waste heat of 15.7 MWth).

Fig. 1 and 2 show the configuration and temperature-entropy (T-s) diagram, respectively, of a split S-CO_2 power cycle for waste heat recovery (WHR) from a gas turbine [4]. A split S-CO_2 power cycle was used to recover the remaining waste heat from the simple recuperated S-CO_2 cycle and minimize the exergy loss in the recuperator. In the split S-CO_2 power cycle for the WHR, the remaining waste heat from the HT heater was used to heat the high-pressure side of CO_2 together with the recuperator, because the isobaric specific heat of the CO_2 on the high-pressure side is much higher than that on the low-pressure side [4]. The portions denoted by x and (1-x) after the pump are sent to the recuperator and the LT heater, respectively, and are preheated to the same temperature, and they merge before the HT heater. The pinch temperature (i.e., the minimum temperature difference required for heat transfer) was assumed to be 30 °C for the exhaust gas-to-CO_2 part and 10 °C for the internal recuperator.



a) (b)

Fig. 1. Split S-CO2 power cycle for WHR from the exhaust gas: a) schematic, b) T-s diagram [4].

2.2. Energy analysis

The following general assumptions were made for the purpose of analysis: the kinetic and potential energies and the heat and friction losses were assumed to be negligible, both isentropic efficiencies of the pump were assumed to be 80%, and the effectiveness of the recuperator was assumed to be 0.90. The properties of CO_2 were obtained from the Engineering Equation Solver software [7]. The equations for the different components of the simple S-CO_2 Rankine cycle shown in Figs. 1 and 2 are as follows:

For the pump (compressor):

$$\eta_P = \frac{h_{2,s} - h_1}{h_2 - h_1} \qquad (1)$$

$$\dot{W}_P^+ = \dot{m}_{CO2}(h_2 - h_1) \qquad (2)$$

For the turbine:

$$\eta_T = \frac{h_4 - h_5}{h_{4,s} - h_5} \qquad (3)$$

$$\dot{W}_E^- = \dot{m}_{CO2}(h_4 - h_5) \qquad (4)$$

The efficiency of the recuperator, ε_R, is expressed as follows:

$$\varepsilon_R = \frac{\dot{m}_{CO2}(h_5 - h_6)}{\dot{Q}_{\max}} = \frac{\dot{m}_{CO2}(h_3 - h_2)}{\dot{Q}_{\max}} \qquad (5)$$

The rate of maximum heat exchange, $\dot{Q}_{\max}$, is expressed as follows:

$$\dot{Q}_{\max} = \dot{m}_{CO2}(h_5 - h_6) \text{ assuming } T_6 = T_2 \qquad (6)$$

For the heater:

$$\dot{Q}_H^+ = \dot{m}_{EG}(h_{EG,in} - h_{EG,out}) = \dot{m}_{CO2}(h_4 - h_3) \qquad (7)$$

where $\dot{m}_{EG}$ is the mass flow rate of the exhaust gas and the subscripts *in* and *out* indicate the inlet and outlet states of the exhaust gas in the heater, respectively.

For the condenser:

$$\dot{Q}_C^- = \dot{m}_{CO2}(h_6 - h_1) \qquad (8)$$

For the thermal efficiency of the cycle:

$$\eta_{cyc} = \frac{\dot{W}_E^- - \dot{W}_P^+}{\dot{Q}_H^+} \qquad (9)$$

The heat recovery efficiency of WHR from a waste heat source can be defined as follows [7]:

$$\eta_{HR} = \frac{\dot{Q}_H^+}{\dot{Q}_{H,\max}^+} = \frac{\dot{m}_w(h_{in} - h_{out})}{\dot{m}_w(h_{in} - h_0)} = \frac{h_{in} - h_{out}}{h_{in} - h_0} \qquad (10)$$

where $\dot{Q}_{H,\max}$ is the maximum allowable heating rate from the waste heat source; $\dot{m}_w$ is the mass flow rate of the waste heat source; h_{in} and h_{out} are the inlet and outlet specific enthalpies of the waste heat source, respectively; and the subscript 0 indicates that the properties are taken at the reference temperature and pressure (T_0, P_0) representing the dead state.

The thermal efficiency of the system for the WHR can be defined as the ratio of the net power to the maximum allowable heating rate from the waste heat source [7]. This is expressed as follows:

$$\eta_{sys} = \frac{\dot{W}_E^- - \dot{W}_P^+}{\dot{Q}_{H,\max}^+} = \eta_{HR}\eta_{tcyc} \qquad (11)$$

3. Split supercritical CO_2 Rankine cycle for waste heat recovery

3.1. Parametric study of the cycle

The parametric study of the WHR cycle has to be done for maximum system efficiency by incorporating the heat recovery (HR) efficiency together with the cycle efficiency to get the maximum power from the waste heat source. At a given cooling condition(temperature) of CO_2 and low- and high-pressure sides, the system efficiency was obtained for the rises in the turbine inlet temperature, when the mass flow rate of the working fluid and the split ratio x (the portion that flows to the recuperator) are adjusted to meet the pinch temperatures for the exhaust gas-to-CO_2 part (30 ℃) and the internal recuperator (10 ℃) [1]. With the rise in the turbine inlet temperature, as shown in Fig. 2, the cycle efficiency increases, but the HR efficiency decreases. Because of this trade-off relationship, the system efficiency increases and decreases and has a peak point in the middle range of the turbine inlet temperature. In this manner, the optimal turbine inlet temperature for maximum system efficiency can be obtained.

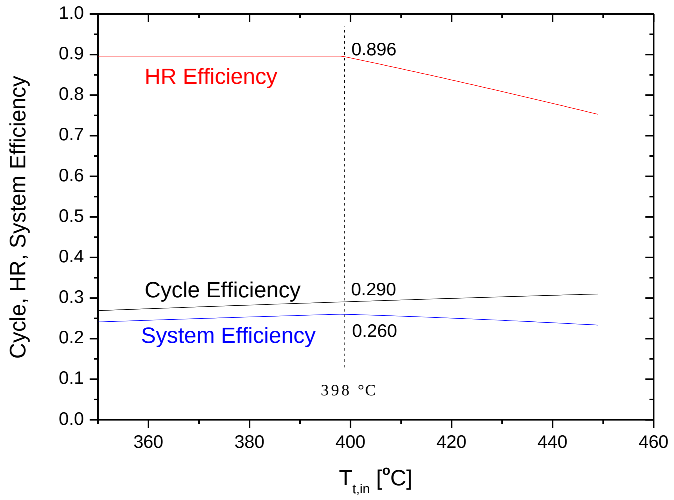
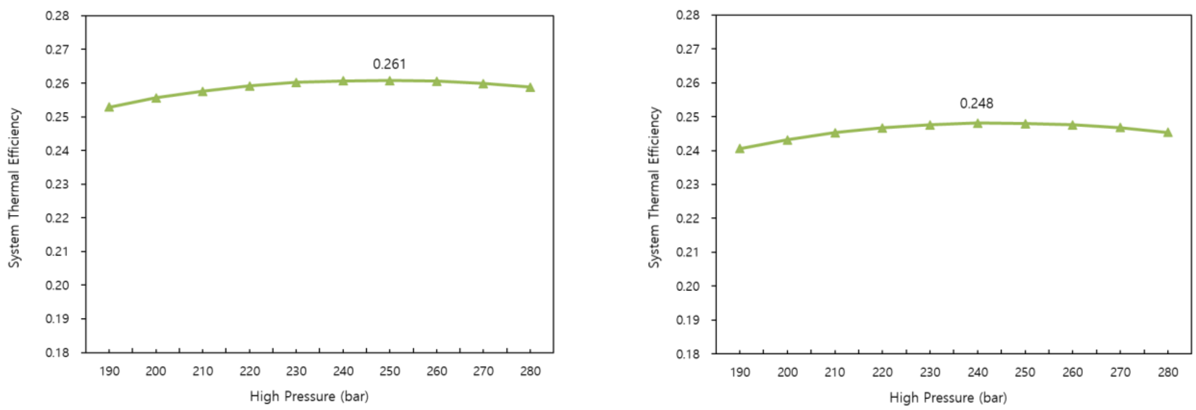


Fig. 2. Heat recovery, cycle, and system efficiency of split S-CO2 power cycle for given turbine inlet temperature [11].

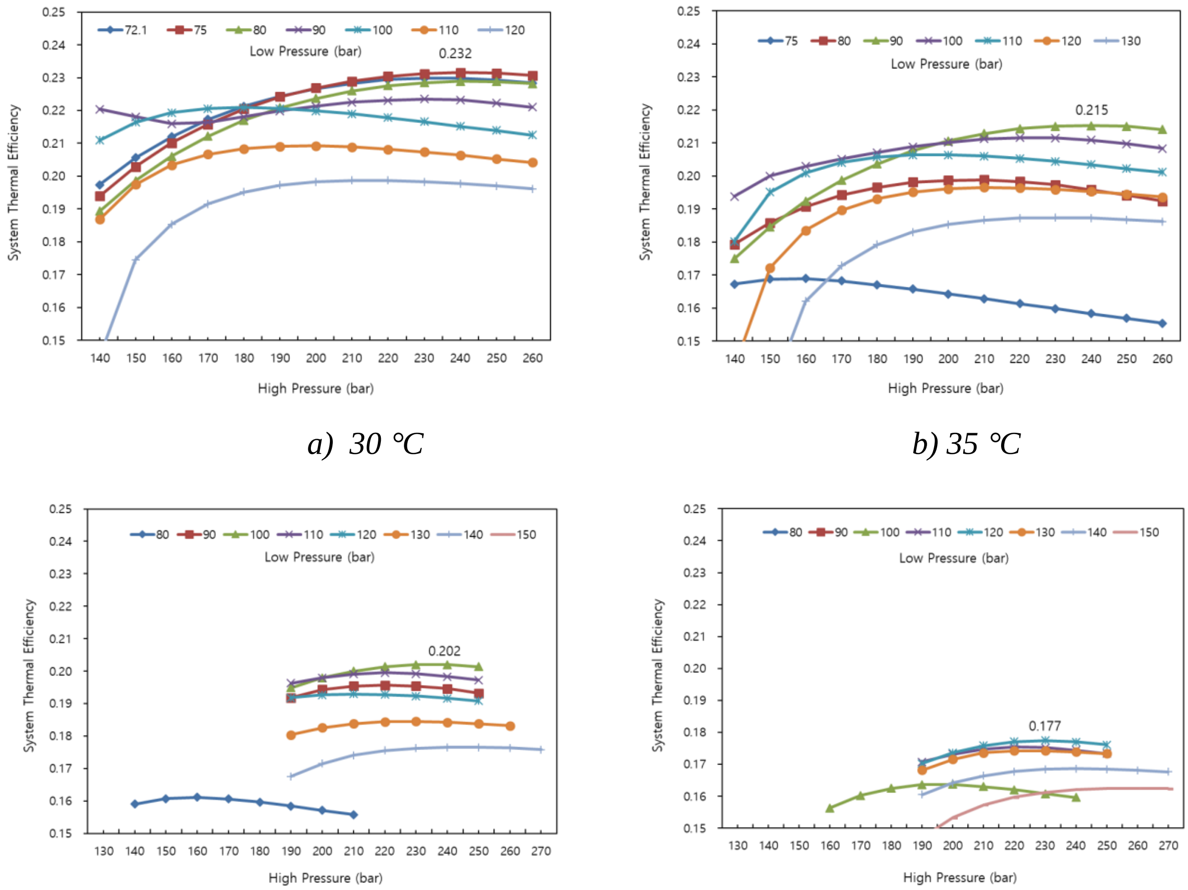
At a given cooling temperature of CO_2 below the critical temperature (in the case of the transcritical CO_2 power cycle), the low-pressure side can be easily optimized to be close to the saturation pressure. Therefore, the high-pressure side must be optimized together with the optimal turbine inlet temperature. Fig. 3 shows the system efficiency of the WHR CO_2 power cycle from the gas turbine, at cooling temperatures of 20 ℃ a) and 25 ℃ b) of CO_2, for the rises in the high-pressure side.



a) 20 ℃ *b) 25 ℃*

Fig. 3. System efficiency of the S-CO_2 power cycle for the high-pressure side at a given cooling temperature: a) 20 ℃, b) 25 ℃

At a given cooling temperature of CO_2 close to and over the critical temperature (in the case of supercritical CO_2 power cycle), both the low- and high-pressure sides must be optimized together with the optimal turbine inlet temperature. Fig. 4 shows the system efficiency of the WHR CO_2 power cycle from the gas turbine, at cooling temperatures of 30 ℃ a), 35 ℃ b), 40 ℃ c), and 50 ℃ d) of CO_2, respectively, for the rises in the high-pressure side.



a) 30 ℃ *b) 35 ℃*

c) 40 ℃ *d) 50 ℃*

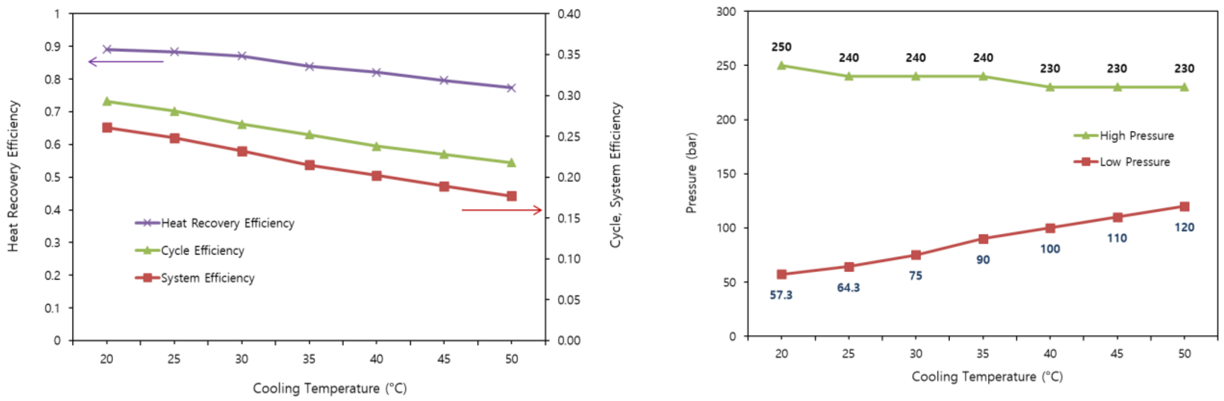
Fig. 4. System efficiency of the S-CO_2 power cycle for the high-pressure side at a given cooling temperature: a) 30 ℃, b) 35 ℃, c) 40 ℃, d) 50 ℃

3.3. Effects of the cooling temperature

From the previous optimization of the operating condition, including both the low- and high-pressure sides, and the turbine inlet temperature, for maximum power from the waste heat source, the maximum system efficiency at a given cooling temperature of CO_2 can be obtained as shown in Fig. 5 a). The cycle efficiency decreases along with the heat recovery efficiency with an increase in the cooling temperature of CO_2. Therefore, the system efficiency decreases by 4 to 7%, with each increase of 5 ℃ in the cooling temperature of CO_2.

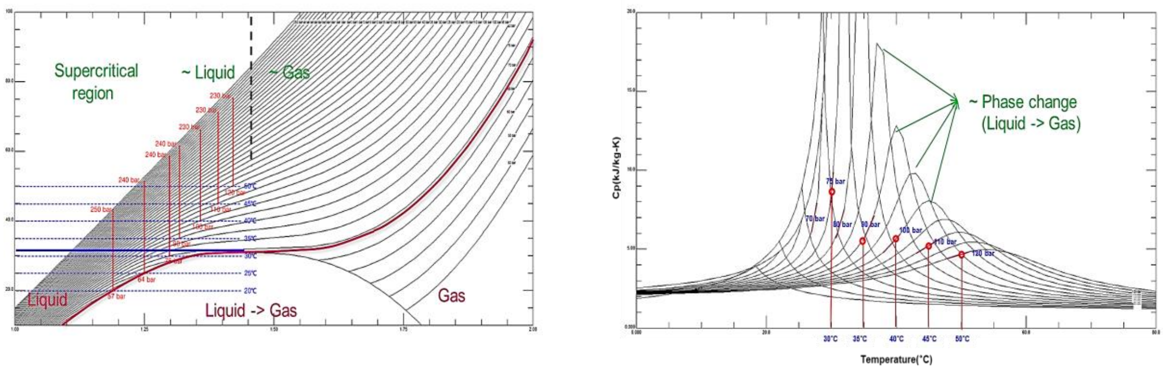
The optimal high- and low-pressure sides for the maximum system efficiency at a given cooling temperature of CO_2 are shown in Fig. 5 b). The effect of the cooling temperature on the optimal high-pressure side is insignificant because the optimal high pressure of the cycle is in the range of 230 bar to 250 bar and the maximum system efficiency of the split-cycle has a very flat curve over the wide range of high-pressure side, as shown in Fig. 3, and Fig. 4. However, the optimal low pressure of the cycle must be increased with a rise in the cooling temperature of CO_2, as shown in Fig. 5 b), because the effect of the low-pressure side on the maximum system efficiency of the split-cycle is significant, as shown in Fig. 3, and Fig. 4.

The compression processes from the optimal low-pressure side to the optimal high-pressure side with a rise in the cooling temperature are located in the T-s diagram, as shown in Fig. 6 a). The optimal low-pressure states before compression are located in the diagram of constant pressure-specific heat (c_p) over the range of the cooling temperature of CO_2. The low pressure of the cycle must be increased to maintain a liquid-like state of S-CO_2 with an increase in the cooling temperature of CO_2. In the supercritical region, there is no distinct phase change period. However, the left-hand side of the pseudo critical line vertical to the critical point is close to a liquid-like state, and the right-hand side is close to a gaseous-like state. In Fig. 6 b), the peak points of constant pressure-specific heat (c_p) correspond to the pseudo critical line (pseudo critical temperature and pressure). The optimal low-pressure state before compression at a given cooling temperature of CO_2, as shown in Fig. 6 b), must be located well before the pseudo critical point in order to reduce the compression work. Therefore, as shown in Fig. 7 a), the optimal low-pressure of the cycle before compression in the supercritical region is higher than the pseudo critical pressure at a given cooling temperature of CO_2. The net power (W_{net}) decreases by 6 to 9% with every 5 ℃ rise in the cooling temperature of CO_2 from 20 ℃ to 50 ℃, as shown in Fig. 7 b), because the compression work (W_c) increases, and the expansion work (W_e) decreases.



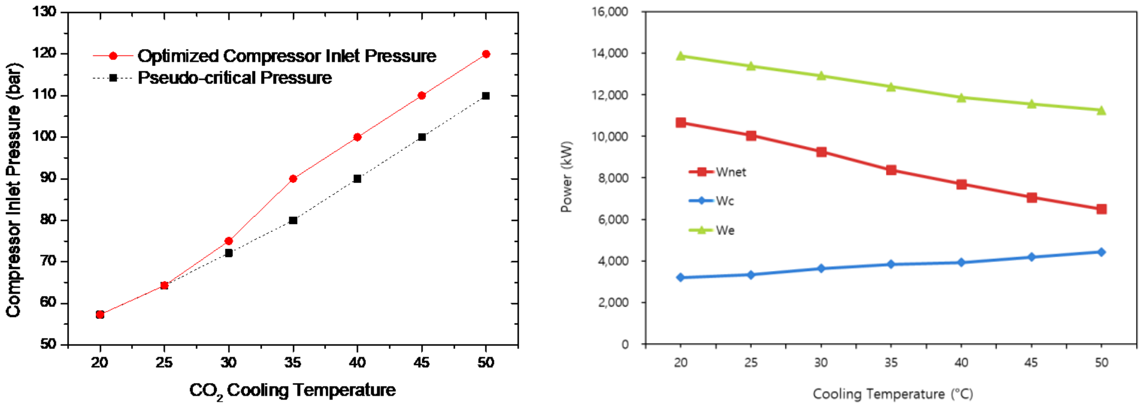
a) *b)*

Fig. 5. Optimal operating condition and performance of the S-CO_2 power cycle for a given cooling temperature: a) HR, cycle, system efficiency, b) low- and high-pressure sides



a) *b)*

Fig. 6. Optimal compression process of the S-CO_2 power cycle for a given cooling temperature: a) T-s diagram, b) constant specific heat (c_p) versus temperature and low-pressure sides before compression



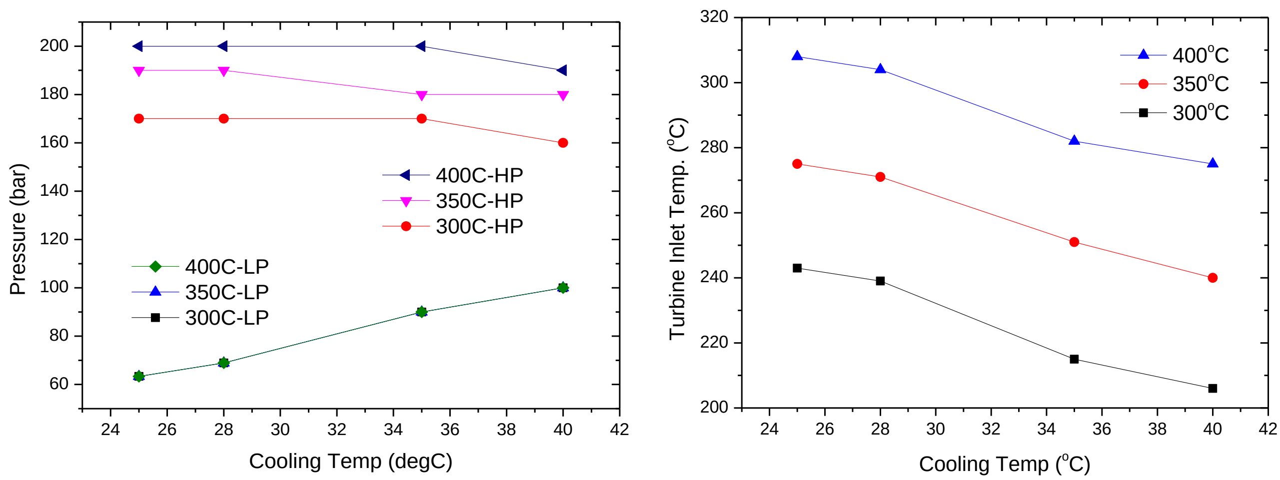
a) *b)*

Fig. 7. Optimal compression process of the S-CO_2 power cycle for a given cooling temperature: a) optimal low-pressure side, b) compression, expansion, and net work

3.4. Effects of the heat source temperature

The exhaust gas with a mass flow rate of 51 kg/s from 300 ℃ to 400 ℃ from a 30-MWe-class engine was selected as the waste heat source to investigate the effects of the heat source temperature on the WHR split S-CO_2 cycle.

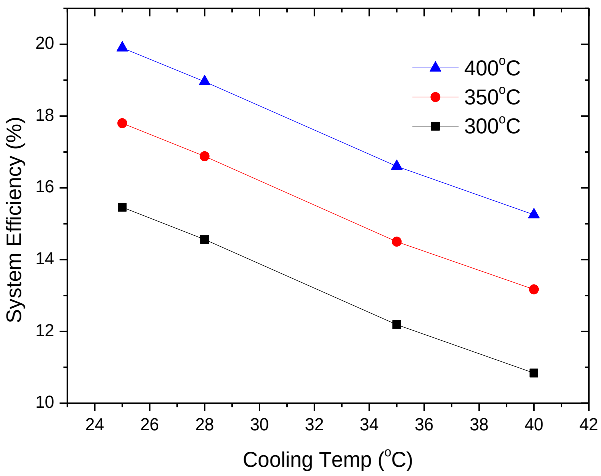
Similar to the previous parametric study of the cycle, the optimal low- and high-pressure sides can be obtained at a given exhaust gas temperature from 300 ℃ to 400 ℃, as shown in Fig. 8 a). The effect of the heat source temperature on the optimal low-pressure side is negligible, and the optimal low-pressure side is dependent on the cooling temperature of CO_2. However, the optimal high-pressure of the cycle increases with a rise in the heat source temperature and is much lower than the previous case with a higher temperature of the exhaust gas from the gas turbine (230 bar to 250 bar). The optimal turbine inlet temperature increases with a rise in the heat source temperature and decreases with a rise in the cooling temperature of CO_2, as shown in Fig. 8 b).



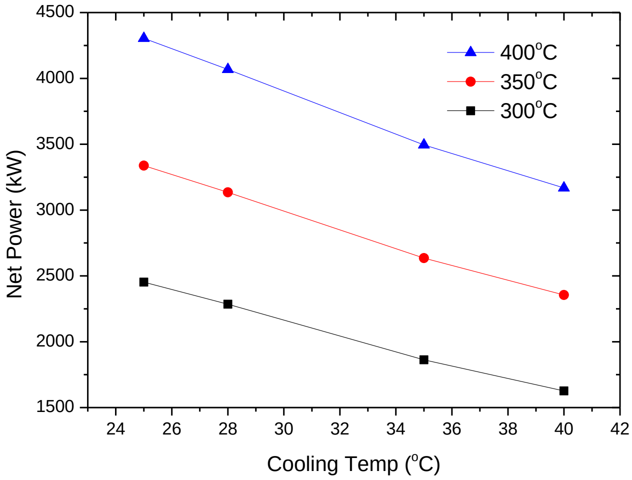
a) *b)*

Fig. 8. Optimal operating condition of the S-CO_2 power cycle for a given exhaust gas temperature from the engine: a) low-pressure (LP) and high-pressure (HP) sides, b) turbine inlet temperature

Fig. 9 shows the maximum system efficiency and the net power of the WHR split S-CO_2 cycle with a rise in the heat source temperature. As the heat source temperature increases in steps of 50 ℃ from 300 ℃ to 400 ℃, the system efficiency increases by about 2% (absolute efficiency), and the net power significantly increases by 30 to 40%.



a)



b)

Fig. 9. Performance of the S-CO_2 power cycle for a given exhaust gas temperature from the engine: a) system efficiency, b) net power

4. Conclusions

From our previous researches, a split S-CO_2 cycle was proposed as a promising WHR cycle from a gas turbine. In the S-CO_2 power cycle, the effects of the cooling condition close to the critical point on the optimization of the operating condition and the performance of the cycle are significant. In this study, the operating condition of the split S-CO_2 power cycle for WHR from a gas turbine was optimized to accommodate the temperature variation of the heat sink and the performance of the cycle was analyzed. Furthermore, to apply the split S-CO_2 power cycle for WHR from an engine exhaust gas, the operating condition of the cycle was optimized to accommodate the temperature variation of the waste heat source and the performance of the cycle was analyzed.

With an increase in the cooling temperature of CO_2, the low pressure of the cycle before compression must be increased to maintain a liquid-like state of S-CO_2 in the supercritical region to reduce the compression work, maintaining the liquid-like state before compression well before the pseudo critical point (peak point of the specific heat). The net power of the cycle decreases by 6 to 9% with every 5 ℃ rise in the cooling temperature of CO_2 from 20 ℃ to 50 ℃ because of the decrease in heat recovery and thermal efficiency of the cycle.

In the same manner, the operating condition of the split S-CO_2 cycle was optimized for WHR from an engine exhaust gas from 300 ℃ to 400 ℃ and the optimal low- and high-pressure sides were obtained at a given exhaust gas temperature. The effect of the heat source temperature on the optimal low-pressure side is negligible and the optimal low-pressure side is dependent on the cooling temperature of CO_2. However, the optimal high-pressure of the cycle increases with a rise in the heat source temperature and is much lower than in the previous case with a higher temperature of the exhaust gas from a gas turbine. As the heat source temperature increases by steps of 50 ℃ from 300 ℃ to 400 ℃, the system efficiency increases by about 2% (absolute efficiency) and the net power significantly increases by 30 to 40%.

Nomenclature

c_p isobaric specific heat, J/(kg K)

h specific enthalpy, kJ/kg

$\dot{m}$ mass flow rate, kg/s

P pressure, kPa

$\dot{Q}$ rate of heat, kW

T temperature, ℃

$\dot{W}$ rate of work, kW

Greek symbols

ε heat exchanger effectiveness

η efficiency

Subscripts and superscripts

0 atmospheric (environmental) state

C condenser

cyc cycle

e expander

EG exhaust gas

H heater

HR heat recovery

i state point

in inlet

max maximum

net net output

out outlet

P pump

R recuperator

s isentropic

sys system

T turbine

$+$ input

$-$ output

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