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Optimal Design and Thermodynamic Analysis of Gas Turbine and Carbon Dioxide Combined Cycles

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ABSTRACT

Gas turbines deliver high temperature exhaust, which can be recovered for clean power generation. Carbon dioxide power cycles are viable choices for waste heat recovery from gas turbines. In this paper, several gas turbine and carbon dioxide combined cycles with different configurations are proposed. Simulation results show that a cascaded CO_2 cycle, which consists of a supercritical CO_2 cycle and a transcritical CO_2 cycle could efficient waste heat recovery from gas turbines. Results show that the s-CO_2 compressor inlet conditions significantly affect the cascaded CO_2 system power output. This cascaded CO_2 cycle was optimized with a genetic algorithm to enhance its thermodynamic performance. The optimal combined cycle efficiency could reach 51.50%. Compared with a GT-steam Rankine combined cycle, the proposed GT-cascaded CO_2 combined cycle has a 4.53% increase in generation efficiency.

INTRODUCTION

In power generation applications, gas turbines are often combined with steam Rankine cycles to harness exhaust waste heat. Gas turbine and carbon dioxide combined cycles may be viable options to further enhance energy efficiency. Kim *et al.* [1, 2] investigated the performance of gas turbine and supercritical CO_2 power combined cycles, and found that supercritical CO_2 cycle have the advantages of higher efficiency, mechanical simplicity, and compact size relative to steam Rankine cycles. Shu *et al.* [3, 4] studied transcritical CO_2 cycles for waste heat recovery from gas engines. Walnum *et al.* [5] proposed that transcritical CO_2 bottoming cycles could increase the energy efficiency of offshore gas turbines. Yoon *et al.* [6] investigated the off-design performance of transcritical CO_2 systems for micro gas turbine heat recovery. Cao *et al.* [7] proposed a cascaded CO_2 cycle for gas turbine exhaust recovery. Manjunath *et al.* [8] performed a further study of this system. A number of studies have been performed on combined cycle gas turbine and carbon dioxide power cycles. However, few studies have been done on the optimal design of carbon dioxide power cycles for gas turbine exhaust recovery. Therefore, an optimal design and thermodynamic analysis is performed in this paper on gas turbine and carbon dioxide combined cycles.

SYSTEM MODELING AND SIMULATION CONDITIONS

System Configuration

Four carbon dioxide power cycles with different configurations are proposed to recover exhaust heat from gas turbines, as shown in Figs. 1 and 2. Thermal oil is used as heat transfer medium between carbon dioxide and gas turbine exhaust, because heat flow rate of carbon dioxide is quite high while that of gas turbine exhaust is much lower. Liquefied natural gas (LNG) is used as heat sink to offer a low back pressure which may greatly increase the cycle performance .Fig. 1 (a) illustrates a transcritical CO_2 cycle (t-CO_2) recovering exhaust heat from a gas turbine. Fig. 1 (b) presents a supercritical CO_2 cycle (s-CO_2) using exhaust heat from a gas turbine, and providing lower temperature heat to a t-CO_2 cycle. This system configuration is termed "supercritical and transcritical CO_2 cycle" (s-t-CO_2). In Fig. 1 (c), the gas turbine exhaust is recovered by a recuperative supercritical CO_2 cycle first, and then by a transcritical CO_2 cycle, and can be termed: "re-supercritical and transcritical CO_2 cycle" (res-t-CO_2). As an extension of these system configurations, a cascaded CO_2 cycle is proposed with improved utilization of gas turbine exhaust. Fig. 2 shows this cascaded CO_2 cycle, which consists of a supercritical CO_2 cycle and a transcritical CO_2 cycle. The carbon dioxide in the t-CO_2 stage is first preheated by the exhaust gas and then heated by the turbine exhaust of s-CO_2. The gas turbine combined cycle with this configuration can be termed: "GT-cascaded CO_2 combined cycle".

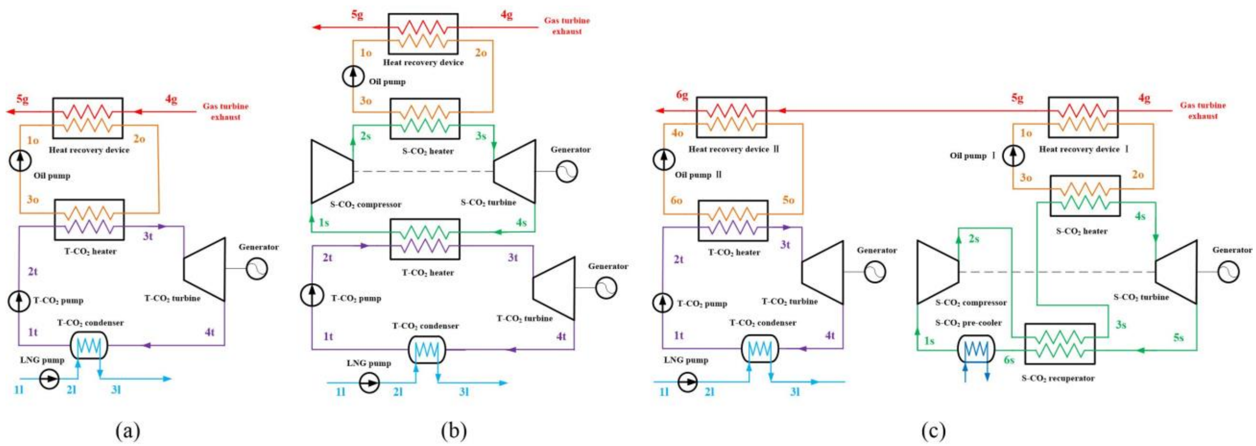


FIGURE 1. Schematic of three different carbon dioxide power cycles for exhaust heat recovery of gas turbines. (a) t-CO_2, (b) s-t-CO_2, and (c) res-t-CO_2.

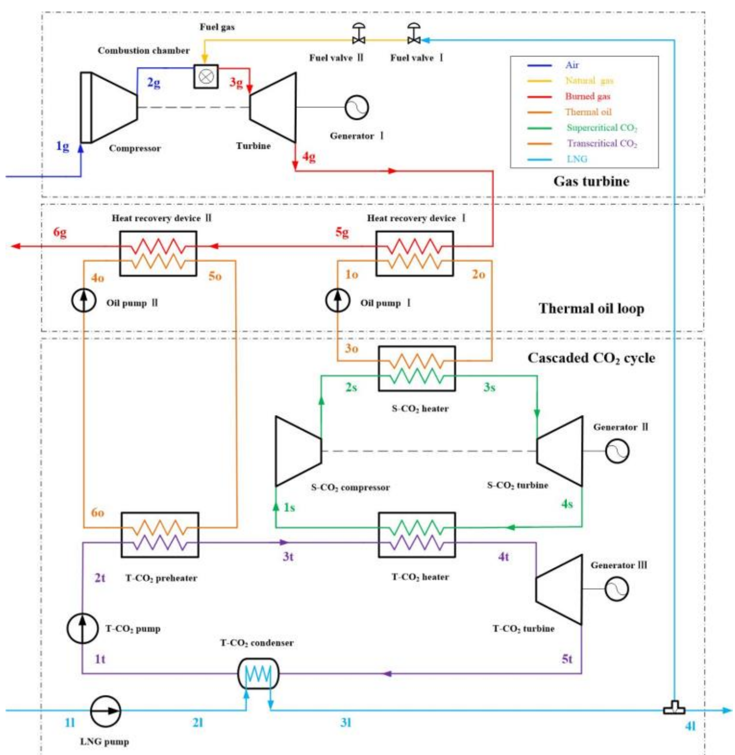


FIGURE. 2. Schematic of gas turbine and cascaded CO_2 combined cycle.

System Modeling

Thermodynamic models for carbon dioxide power cycles are formulate to predict system performances. A modular modeling approach is used to couple cascaded s-CO_2 and t-CO_2 models.

In an s-CO_2 system, the compressor and turbine are assumed coaxial, so its net power can be calculated by

$$P_s = (P_{s,turb} - P_{s,comp})\eta_e \tag{1}$$

where η_e is the generator efficiency, the turbine output power $P_{s,turb}$ and compressor power consumption $P_{s,comp}$ are calculated by their enthalpy changes.

For a t-CO_2 system, net power can be obtained by

$$P_t = P_{t,turb}\eta_{\mathrm{e}} - P_{t,pump} \tag{2}$$

where the turbine output power $P_{t,turb}$ and pump power consumption $P_{t,comp}$ are also calculated by their enthalpy changes.

In these carbon dioxide power cycles, electric power is output by both the s-CO_2 generator and t-CO_2 generator. Therefore, the net power of carbon dioxide power cycle is expressed as

$$P_{CO2} = P_s + P_t \tag{3}$$

Therefore, the overall system thermal efficiency of gas turbine and carbon dioxide combined cycle is calculated by

$$\eta_{cc} = \frac{P_{gt} + P_{CO2}}{\dot{m}_{fuel}\eta_b LHV} \tag{4}$$

where $\dot{m}_{fuel}$ is the mass flow rate of fuel gas, *LHV* is the lower heating value of fuel gas, and η_b is the burning efficiency.

TABLE 1. Simulation conditions for GT-cascaded CO_2 combined cycle.

Item	Value	Item	Value
Taurus 60 GT exhaust temperature (K)	783	T-CO_2 condensing pressure (MPa)	0.8
Taurus 60 GT exhaust mass flow rate (kg/s)	21.77	S-CO_2 turbine isentropic efficiency (%)	85
Maximum temperature of thermal oil (K)	673	S-CO_2 compressor isentropic efficiency (%)	82
Effectiveness of heat recovery device (%)	85	T-CO_2 turbine isentropic efficiency (%)	85
S-CO_2 compressor inlet temperature (K)	305	T-CO_2 pump isentropic efficiency (%)	80
S-CO_2 compressor inlet pressure (MPa)	7.4	Generator efficiency (%)	98
S-CO_2 turbine inlet pressure (MPa)	20	S-CO_2 heater effectiveness (%)	95
T-CO_2 turbine inlet pressure (MPa)	20	T-CO_2 heater effectiveness (%)	95

Simulation Conditions

The Taurus 60 Gas Turbine was selected as the topping cycle and heat source for the carbon dioxide power cycles in this paper [9]. This is a 5.67 MW gas turbine with a thermal efficiency of 31.5%. Its exhaust parameters are presented in Table 1. In this paper, it was assumed that fuel gas was pure methane with LHV was 50.2 MJ kg^{-1} [10]. DOWTHERM A heat transfer fluid was selected as heat transfer medium, which can operate from 15 ℃ to 400 ℃ [11]. Plate-fin heat exchangers were chosen for heat transfer between exhaust gas and thermal oil while Printed circuit heat exchangers were selected for heat transfer between thermal oil and carbon dioxide. Pressure drops for these heat exchangers were assumed to be 2 %. The final exhaust temperature after heat recovery is greater than the water dew point ($\geq 350K$) [12]. The simulation conditions and assumptions for these carbon dioxide power cycles are summarized in Table 1.

RESULTS AND DISCUSSION

Thermodynamic Performance Comparison

Four carbon dioxide power cycles are proposed in this paper. The cycles are simulated using MATLAB [13]. Table 2 presents simulation results for these carbon dioxide power cycles. It shows that the cascaded CO_2 cycle has the highest net power of 3.318 MW, and its combined cycle net power and efficiency can reach 8.988 MW and 49.96%, respectively. The t-CO_2 has the second highest net power. This cycle can fully recover heat from exhaust gas and has a final exhaust temperature of 350 K. However, its turbine exhaust is cooled at the temperature of 351.7 K, which can be further utilized to generate power. Therefore, it is considered that the t-CO_2 can have better recovery of gas turbine exhaust while it can't utilize the recovered heat efficiently. The s-t-CO_2 also has an acceptable net power of 3.131 while its final exhaust temperature is 443.5 K. This means the exhaust heat from the gas turbine is not fully utilized. The res-t-CO_2 has the lowest net power. Although the recuperative configuration makes the supercritical CO_2 cycle output more effective, the low mass flow rate of t-CO_2 limits the t-CO_2 net power. In general, it is likely that the cascaded CO_2 cycle is the most efficient considered option for using turbine exhaust heat.

TABLE 2. Simulation results for carbon dioxide power cycles.

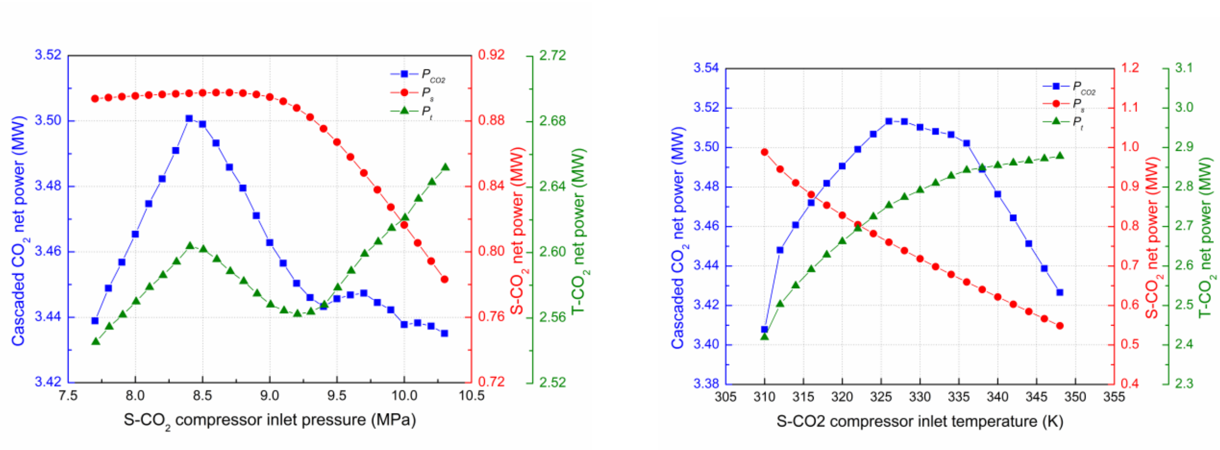
	t-CO_2	s-t-CO_2	res-t-CO_2	cascaded CO_2 cycle
P_s / MW	–	1.086	1.476	1.086
P_t / MW	3.141	2.045	1.249	2.232
P_{CO2} / MW	3.141	3.131	2.725	3.318
P_{cc} / MW	8.811	8.801	8.395	8.988
η_{cc} / %	48.97	48.92	46.67	49.96
T_{5g} / K	350.0	443.5	547.7	443.5
T_{6g} / K	–	–	350.0	350.0
m_s / kg·s^{-1}	–	19.85	27.76	19.85
m_t / kg·s^{-1}	10.15	12.67	9.17	21.75

Thermodynamic Analysis of Cascaded CO_2 Cycle

Design parameters have significant effect on the thermodynamic performance of carbon dioxide power cycles. In the cascaded CO_2 cycle, s-CO_2 compressor inlet conditions are assumed to be the main governing parameters. Therefore, simulations with different s-CO_2 compressor inlet pressures and temperatures are conducted.

Fig. 3 (a) illustrates the effect of s-CO_2 compressor inlet pressure on the cascaded CO_2 net power when the inlet temperature is 315 K. It can be seen the cascaded CO_2 net power reaches a maximum of 3.5 MW. The variations of s-CO_2 net power and t-CO_2 net power explain why the cascaded CO_2 net power has such trends. At low s-CO_2 compressor inlet pressure, the t-CO_2 net power increases nearly linearly with increasing s-CO_2 compressor inlet pressure. At higher inlet pressure, the reducing s-CO_2 net power offsets any gains from the t-CO_2 system.

Fig. 3 (b) shows the variation of cascaded CO_2 net power under different s-CO_2 compressor inlet temperatures when the inlet pressure is fixed at 8 MPa. The cascaded CO_2 net power reaches a maximum of 3.51 MW at 326 K. Moreover, this figure also indicates how this corresponds to an optimum point with opposite trends of power output for the two sub-cycles. At higher s-CO_2 compressor inlet temperatures, increasing compression work reduces the s-CO_2 net power. Also, the t-CO_2 turbine inlet temperature goes up with the increase of s-CO_2 compressor inlet temperature, which allows the t-CO_2 turbine to utilize higher grade heat and results in the increase of t-CO_2 net power.



(a) (b)

FIGURE 3. Effect of s-CO_2 compressor inlet conditions on net power from the cascaded CO_2 cycle. (a) S-CO_2 compressor inlet pressure, (b) s-CO_2 compressor inlet temperature

TABLE 3. GA optimization results for cascaded CO_2 net power.

Number	η_{cc} / %	P_{CO2} / MW	p_{1s} / MPa	T_{1s} / K
1	51.49	3.593	9.423	325.6
2	51.46	3.588	9.298	326.2
3	51.50	3.595	9.486	325.4
4	51.50	3.595	9.597	325.3
5	51.50	3.594	9.695	325.3
6	51.48	3.592	9.874	325.4
7	51.49	3.594	9.676	325.4
8	51.50	3.595	9.671	325.3
9	51.50	3.595	9.531	325.3
10	51.46	3.588	9.280	326.2

Genetic Algorithm Optimization

The above discussion indicates that there may be optimal operating conditions to maximize power output from the cascaded CO_2 cycle. The genetic algorithm (GA) optimization method is use to identify the optimal operating conditions. Population size, crossover probability and stop generation for the GA optimization are 20, 0.8 and 100, respectively. Constraint bounds of GA optimization for the compressor inlet pressure and temperature are 7.5 – 10.5 MPa and 310 – 350 K. Table 3 presents ten optimization results for the cascaded CO_2 net power. The overall system efficiency reaches a maximum of 51.50% when the cascaded CO_2 net power is maximum at 3.595 MW.

Compared with traditional GT-steam Rankine combined cycles, this GT-cascaded CO_2 combined cycle can have an increase of 4.53% for combined cycle efficiency [7]. This is a significant improvement for energy efficiency. In general, the simulation results indicate that the proposed cascaded CO_2 cycle is a good option for exhaust heat recovery from gas turbines.

CONCLUSIONS

In this paper, thermodynamic analyses were presented for four carbon dioxide power cycles. Four different carbon dioxide power cycles were studied, including t-CO_2, s-t-CO_2, res-t-CO_2 and cascaded CO_2 cycles. The cascaded CO_2 cycle has the highest net power and results in the highest combined cycle efficiency. The net power of the cascaded CO_2 cycle reaches a maximum with at a specific s-CO_2 compressor inlet pressure and temperature. Through GA optimization, the cascaded CO_2 cycle can reach a maximum net power of 3.595 MW. Moreover, the overall system efficiency of the GT-cascaded CO_2 combined cycle can achieve 51.50% which is 4.53% higher than typical GT-steam Rankine combined cycle. This suggests that the cascaded CO_2 cycle is a viable option for the exhaust heat recovery from gas turbines.

REFERENCES

[1] Kim, M. S., Ahn, Y., Kim, B., and Lee, J. I., 2016, "Study on the supercritical CO_2 power cycles for landfill gas firing gas turbine bottoming cycle," Energy, 111, pp. 893-909.
[2] Kim, Y. M., Sohn, J. L., and Yoon, E. S., 2017, "Supercritical CO_2 Rankine cycles for waste heat recovery from gas turbine," Energy, 118, pp. 893-905.
[3] Shu, G., Shi, L., Tian, H., Li, X., Huang, G., and Chang, L., 2016, "An improved CO_2-based transcritical Rankine cycle (CTRC) used for engine waste heat recovery," Applied Energy, 176, pp. 171-182.
[4] Shu, G., Shi, L., Tian, H., Deng, S., Li, X., and Chang, L., 2017, "Configurations selection maps of CO_2-based transcritical Rankine cycle (CTRC) for thermal energy management of engine waste heat," Applied Energy, 186, pp. 423-435.
[5] Walnum, H. T., Nekså, P., Nord, L. O., and Andresen, T., 2013, "Modelling and simulation of CO_2 (carbon dioxide) bottoming cycles for offshore oil and gas installations at design and off-design conditions," Energy, 59, pp. 513-520.
[6] Yoon, S. Y., Kim, M. J., Kim, I. S., and Kim, T. S., 2017, "Comparison of micro gas turbine heat recovery systems using ORC and trans-critical CO_2 cycle focusing on off-design performance," Energy Procedia, 129, pp. 987-994.
[7] Cao, Y., Ren, J., Sang, Y., and Dai, Y., 2017, "Thermodynamic analysis and optimization of a gas turbine and cascade CO_2 combined cycle," Energy Conversion and Management, 144, pp. 193-204.
[8] Manjunath, K., Sharma, O., Tyagi, S., and Kaushik, S., 2018, "Thermodynamic analysis of a supercritical/transcritical CO_2 based waste heat recovery cycle for shipboard power and cooling applications," Energy Conversion and Management, 155, pp. 262-275.
[9] "Solar Turbines. www.solarturbines.com."
[10] Cao, Y., Gao, Y., Zheng, Y., and Dai, Y., 2016, "Optimum design and thermodynamic analysis of a gas turbine and ORC combined cycle with recuperators," Energy Conversion and Management, 116, pp. 32-41.
[11] Dowtherm, A., 1997, "Heat transfer fluid," product technical data.
[12] Chacartegui, R., Sánchez, D., Muñoz, J., and Sánchez, T., 2009, "Alternative ORC bottoming cycles for combined cycle power plants," Applied Energy, 86(10), pp. 2162-2170.
[13] MATLAB [Computer software]. MathWorks, Natic, MA.

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