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COMPARISON OF SUPERCRITICAL CO_2 GAS TURBINE CYCLE AND BRAYTON CO_2 GAS TURBINE CYCLE FOR SOLAR THERMAL POWER PLANTS

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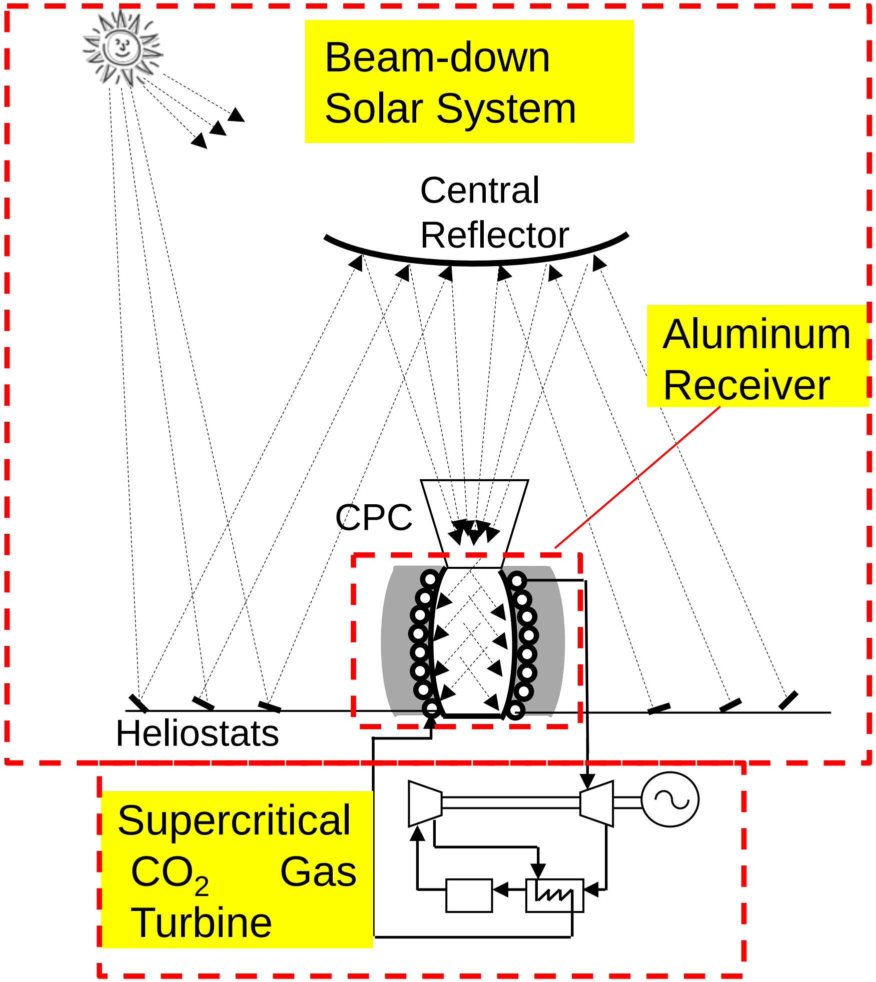
INTRODUCTION

- In Tokyo Institute of Technology, we have devoted our efforts for the development of supercritical CO_2 gas turbine (527℃, 750MWe) connected to the Na-cooled fast reactor.
- We are now devoting our efforts for the application of the supercritical CO_2 gas turbine (650℃, 100MWt) to the solar thermal power plants.

- Though the supercritical CO_2 gas turbine can achieve very high thermal efficiency, very high pressure (20MPa) and bypass flow circuit cause difficulties in the operation and gas turbine design.
- On the other hand, the thermal efficiency of a typical Brayton cycle flow circuit for CO_2 is some percentage lower, but still higher than those for the helium or nitrogen. In addition, there is no problem in the turbomachinery design and operation.

- In nuclear power plants, their unit capacities are large (～1000MWe/unit) and their operations are simple. Then, the supercritical CO_2 GT cycle is preferred.
- In solar thermal plant, unit capacities are small (～20MWe/unit) and daily operation control is needed. Therefore, not only thermal efficiency but also simple and easy operations are important.
- In this paper, the supercritical CO_2 GT cycle and Brayton GT cycle are compared for the solar thermal power plant.

SOLAR THERMAL POWER GENERATION WITH A CO_2 GAS TURBINE



- Beam-down sun-shine collecting system to reduce radiation heat loss.
- Use of aluminum phase change at 660℃. Then, turbine inlet temperature is 650℃.
- Connected to supercritical CO_2 gas turbine

Aluminum

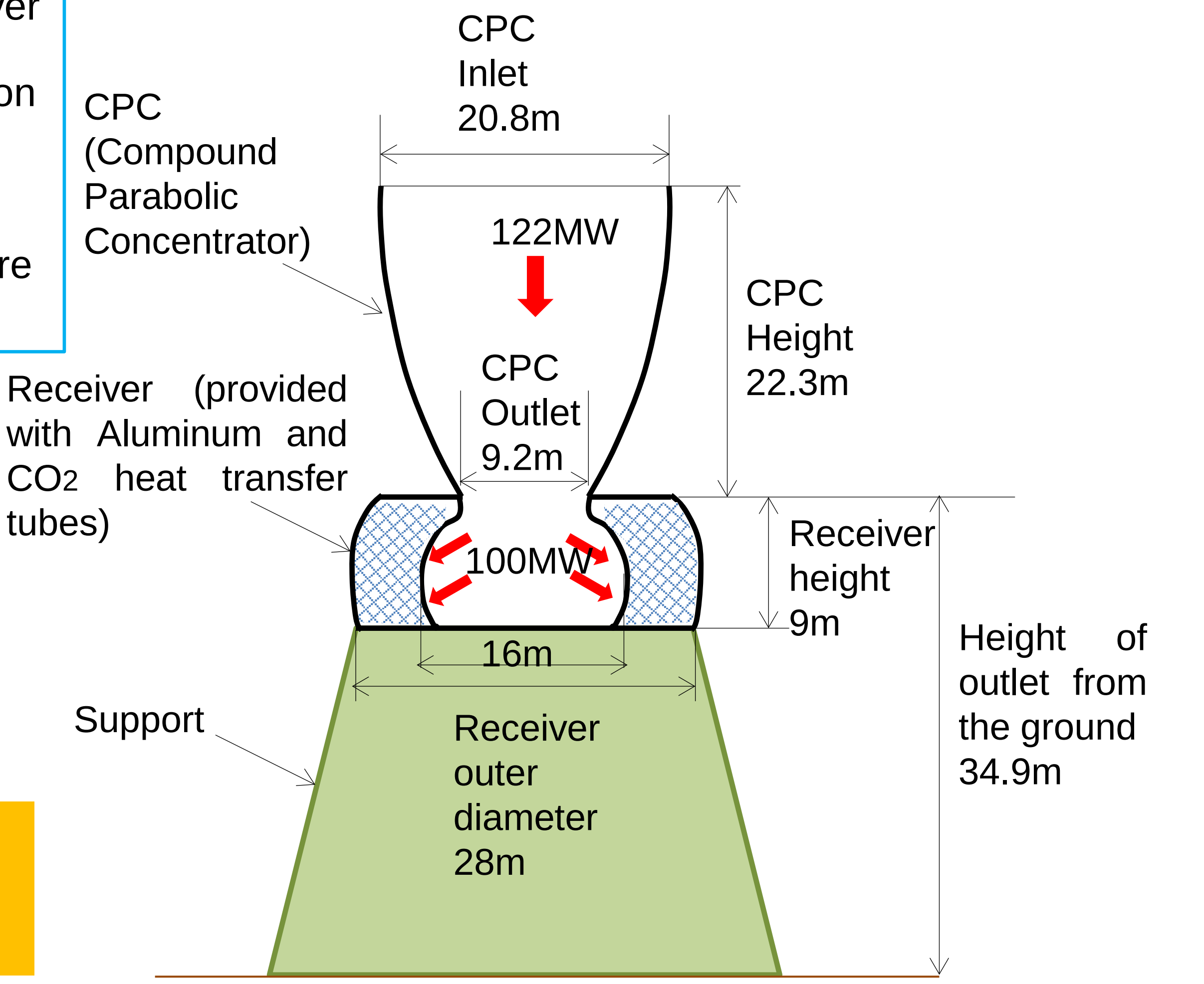
- Thermal conductivity 237 W/m/K
- Melting point 660℃
- Heat of fusion 397 kJ/kg
- Specific heat 0.897 kJ/kg.K

Beam-down Sunbeam Collecting System with Aluminum Receiver

- Net thermal input to receiver 100MW
- Aluminum weight 4,670 ton
- Number of heat transfer tubes (φ34mm/ 18mm) 14,688
- Maximum shell temperature 729℃

- Heliostat field diameter 800m
- Number of heliostats (φ3.4m) 42,519
- Central reflector height from the ground 114m
- Central reflector diameter 34.7m

Beam down sunbeam collecting system is based on the paper (Hasike, 2006)



SUPERCRITICAL CO_2 GAS TURBINE CYCLE FLOW SCHEMES AND THE PRESSURE DEPENDENCY OF CYCLE THERMAL EFFICIENCY

There are two flow schemes for the supercritical CO_2 gas turbine cycle.

- Flow scheme with bypass compressor circuit.

  ➡ Supercritical CO_2 gas turbine cycle

- Flow scheme of typical intercooled closed gas turbine cycle. Higher than critical pressure of 8.4MPa.

  ➡ Brayton CO_2 gas turbine cycle

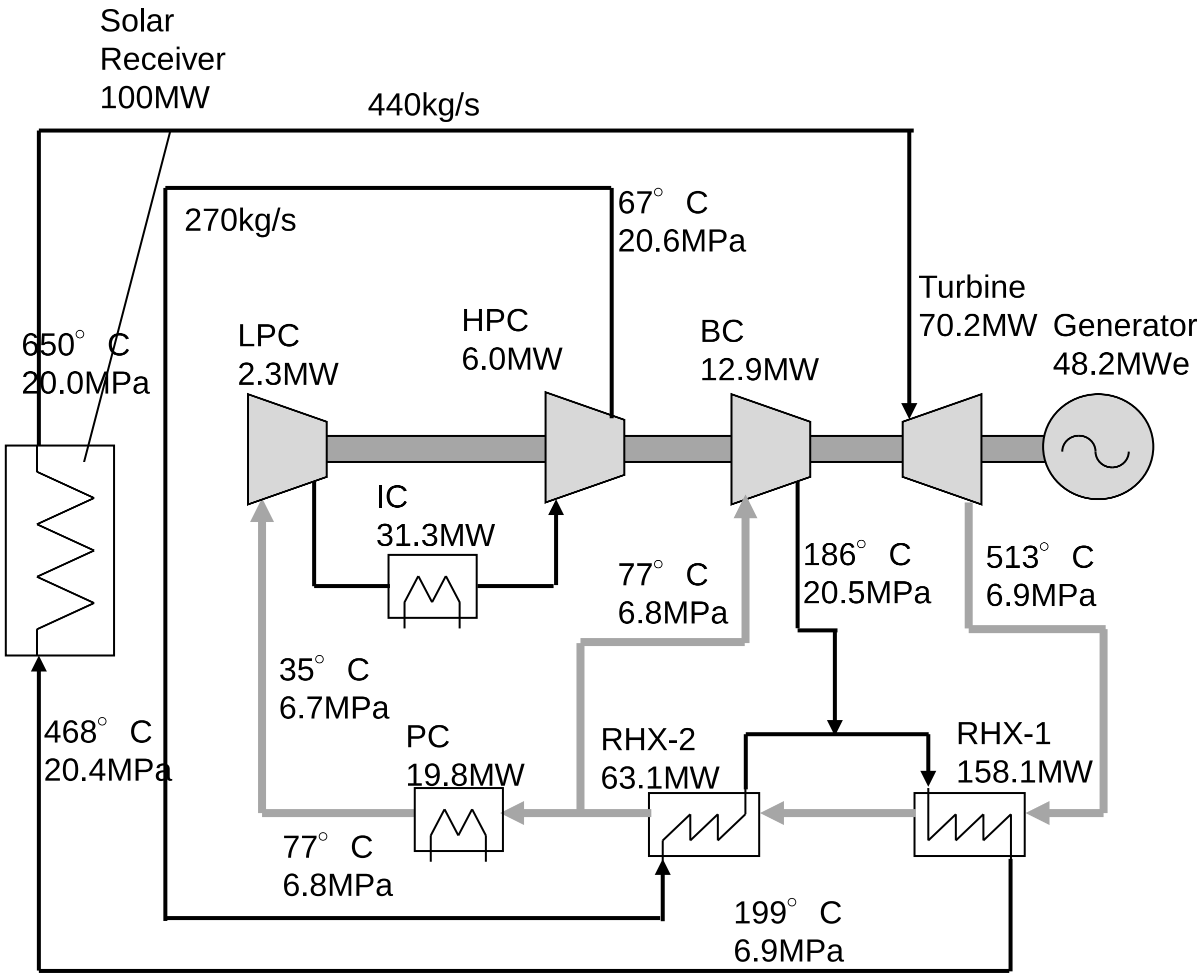
Supercritical CO2 GT Cycle

Assumptions

- Turbine adiabatic efficiency 92%
- Compressor adiabatic efficiency 88%
- Pressure loss (ratios over the inlet pressure)

① Solar receiver 2.0%
② Recuperator high temperature side 1.2%
③ Recuperator low temperature side 0.4%
④ Precooler 1.0%
⑤ Intercooler 0.8%

- Recuperator average temperature effectiveness 91%



Cycle Thermal Efficiency = 48.9%

Brayton CO_2 GT Cycle

Assumptions

- Turbine adiabatic efficiency 92%
- Compressor adiabatic efficiency 88%
- Pressure loss (ratios over the inlet pressure)

① Solar receiver 2.0%
② Recuperator high temperature side 1.2%
③ Recuperator low temperature side 0.4%
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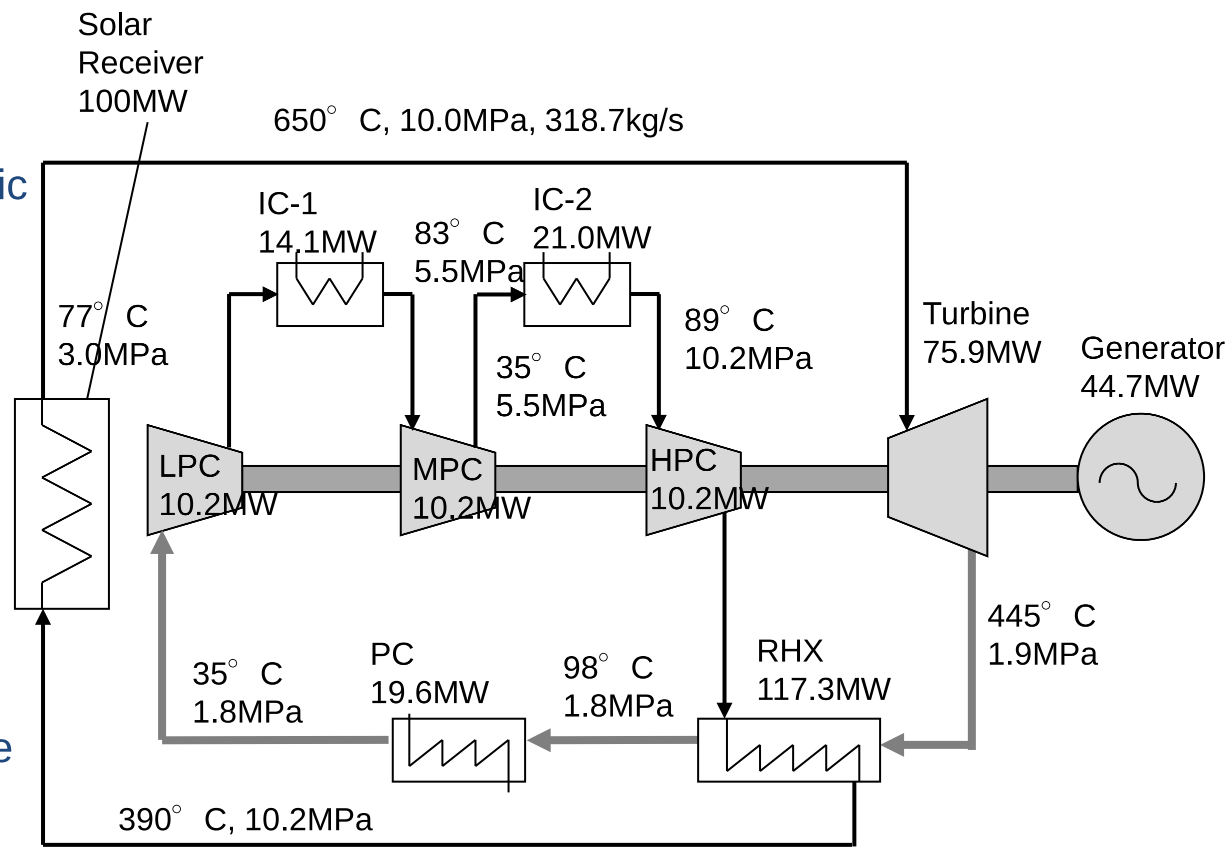
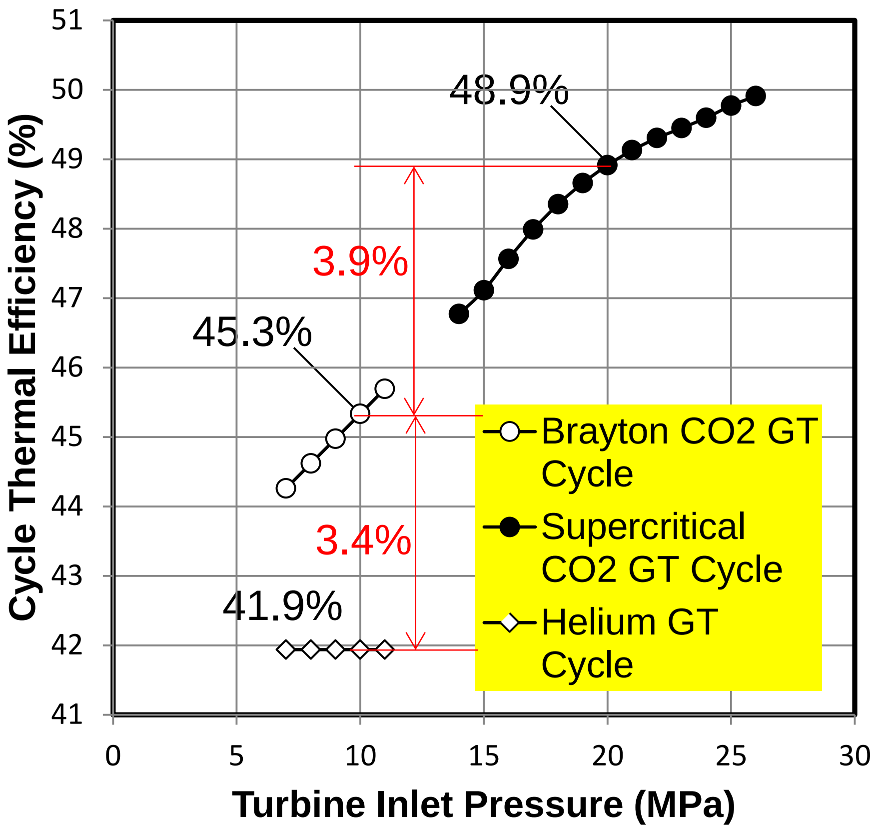


Cycle Thermal Efficiency = 45.3%

Cycle Thermal Efficiency

- Turbine inlet temperature = 650℃
- One intercooling for the supercritical CO_2 GT cycle.
- Two intercooling for the Brayton CO_2 GT and He GT cycles.
- Then, 3 compressors
- Recuperator effectiveness = 91% for CO_2 GT cycles
- It = 93% for He GT cycle (Recuperator effectiveness = 95%)



Compressor Designs

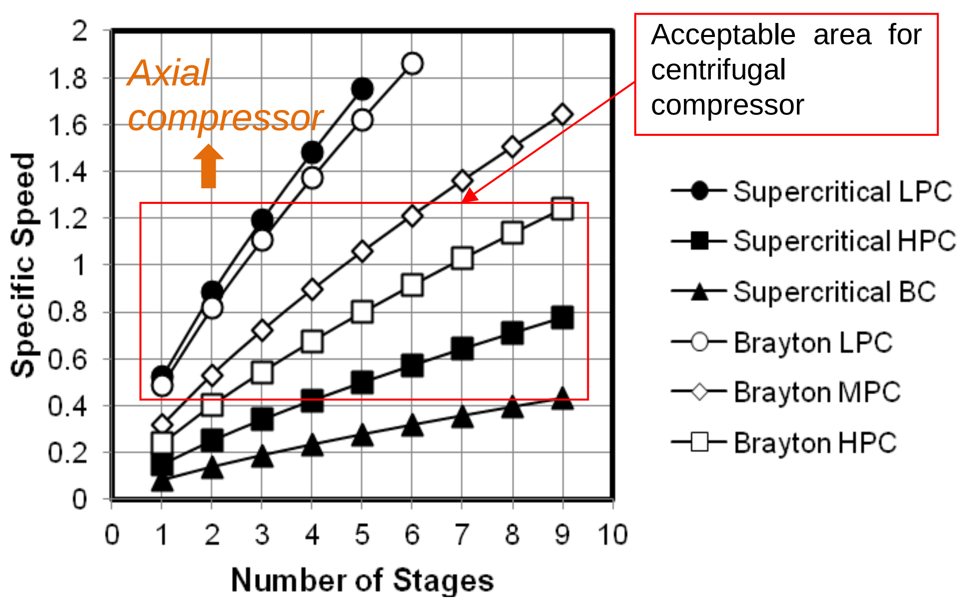
Design conditions of the CO_2 compressors for the 100MW solar thermal power plant (Rotational speed N = 3,600rpm)

Items	Supercritical CO_2 Cycle			Brayton CO_2 Cycle		
	LPC	HPC	BC	LPC	MPC	HPC
Temperature ℃	35	35	77.3	35	35	35
Inlet pressure MPa	6.71	8.26	6.78	1.83	2.97	5.46
Outlet pressure MPa	8.32	20.57	20.49	2.99	5.48	10.25
Mass flow rate kg/s	270	270	171	319	319	319

Specific Speed

$$Ns = \frac{m^{0.5}\rho^{0.25}N}{\Delta p^{0.75}}$$

- m=mass flow rate
- ρ=average density (kg/m3)
- N=rotational speed (radian/s)
- Δp=pressure rise (Pa)



- Design results for the supercritical CO_2 GT cycle compressors

The values of polytropic efficiency was predicted from the value of specific speed (Rogers)

Centrifugal compressors		**LPC**	**HPC**	**BC**
Number of stages		1	6	12
Impeller polytropic efficiency, %	First stage	91.6	91.5	91.9
	Last stage	-	89.4	87.4
Impeller outer diameter, m	First stage, m	0.571	0.319	0.458
	Last stage, m	-	0.431	0.529

Note: The efficiency doesn't include losses between stages. Therefore, the efficiency may reduce considerably, in particular for the bypass compressor.

- Design results for the Brayton CO_2 GT cycle compressors

Design method is based on that of Cohen.

Axial Compressors	LPC	MPC	HPC
Number of stages	12	15	14
Hub-to-tip-ratio	0.62	0.78	0.88
Adiabatic efficiency, %	89.72	89.62	89.57
Inlet casing diameter, m	0.554	0.499	0.430
Blade height, mm	100-148	53-83	25-39
Axial blade length, m	1.75	1.21	0.56
Rotor blade stress, MPa	138	260	571

There seems no marked difference in the achievable efficiency values between both the cycles. However, extremely many number of stage is needed for the bypass compressor, which may reduce the efficiency markedly.

Turbine Designs

Design conditions of the CO_2 turbines for the 100MW solar thermal power plant

Turbines		Supercritical CO_2 GT	Brayton CO_2 GT
Inlet temperature	℃	650	650
Inlet pressure	MPa	20	10
Outlet pressure	MPa	6.94	1,866
Mass flow rate	kg/s	440.3	318.7
Rotational speed	rpm	3,600	3,600

Design Method

- Loss model: Craig & Cox
- Chord length: 30mm (Nozzles) 20mm (Blades)
- Tip clearance: 0.008
- Maximum allowable stress for blades: 400MPa (Mar-M47, 700℃)
- Parameters: Loading coefficient, Flow coefficient, Number of stages

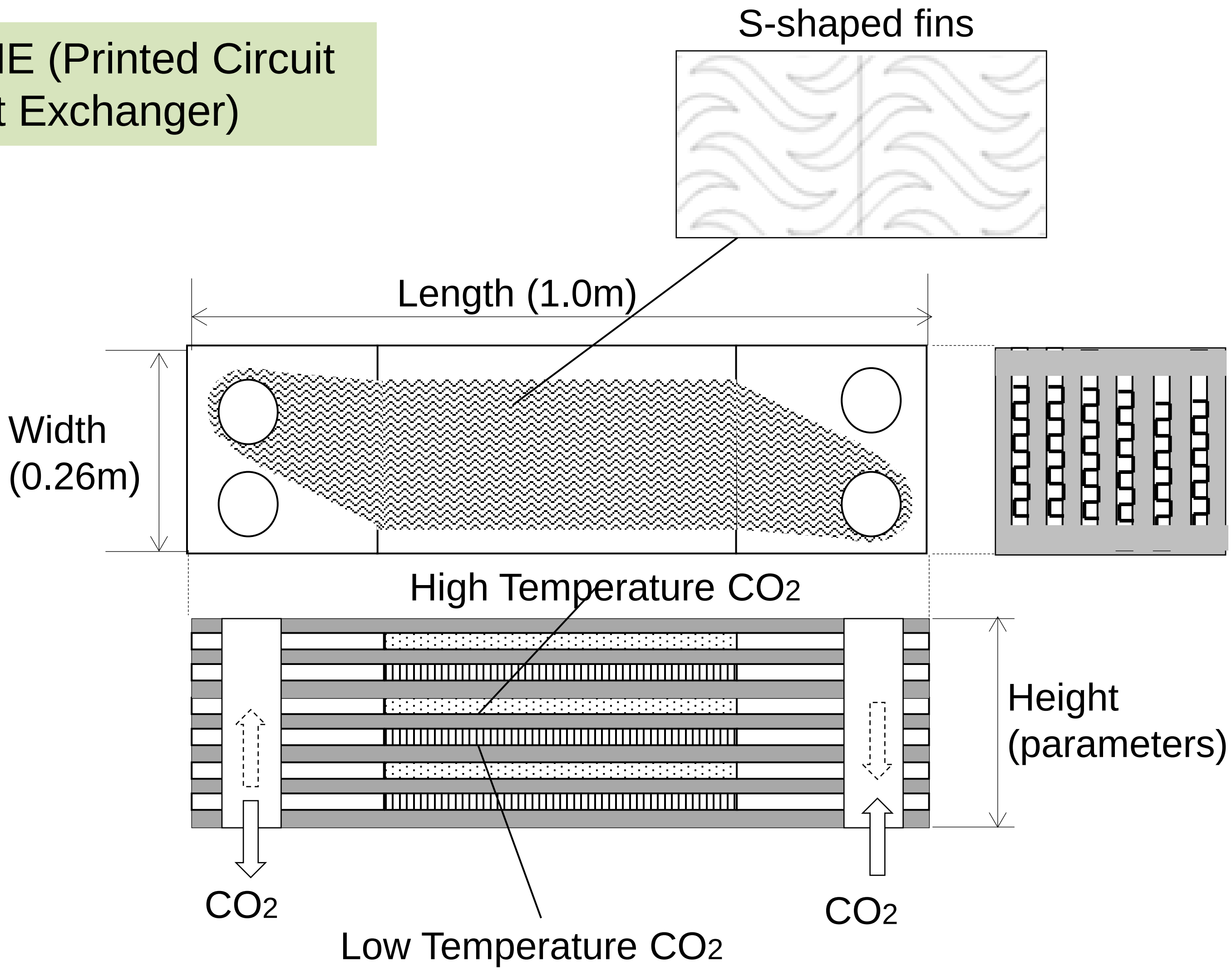
Design results of the CO_2 turbines for the 100MW solar thermal power plant

Turbines		Supercritical CO_2 GT	Brayton CO_2 GT
Number of stages		7	4
Loading coefficient		1.3	1.3
Flow coefficient		0.45	0.35
Average peripheral velocity	m/s	132	214
Average mean diameter	m	0.702	1.135
Adiabatic efficiency	%	92.7	92.4
Blade stress	MPa	360	334

There is no marked difference between both the cycles.

Recuperator Designs

PCHE (Printed Circuit Heat Exchanger)



Design Conditions of the Recuperators

Items			Supercritical CO_2 Gas Turbine		Brayton CO_2 Gas Turbine
			RHX-1	RHX-2	RHX
Recuperator effectiveness		%	91	91	91
Number of modules			12	12	12
Heat load		MW/modules	13.173	5.261	9.775
HT side	Flow rate	kg/s	36.689	36.689	26.560
	Inlet temperature	℃	512.82	199.42	444.64
	Inlet pressure	MPa	6.944	6.861	1.866
LT side	Flow rate	kg/s	36.689	22.478	26.560
	Inlet temperature	℃	185.74	67.17	89.14
	Inlet pressure	MPa	20.490	20.572	10.245

Results of the Recuperator Designs

Items		Supercritical CO_2 Gas Turbine		Brayton CO_2 Gas Turbine
		RHX-1	RHX-2	RHX
Width × Length m/module		0.26 × 1.0	0.26 × 1.0	0.26 × 1.0
Height m/module		6.31	4.24	4.54
Weight ton/module		11.76	7.90	8.46
Total weight ton		141	95	102
Heat transfer capacity MW		11.755	5.261	9.777
Pressure loss ratio (dP/Pinlet)	HT side %	0.196	0.247	2.29
	LT side %	0.075	0.031	0.226

The total weight of recuperators for the supercritical CO_2 gas turbine cycle becomes twice that for the Brayton CO_2 gas turbine cycle.

CONCLUSIONS

Applications of two CO_2 GT cycles, i.e., “20 MPa supercritical CO_2 GT cycle” and “10 MPa Brayton CO_2 GT cycle” to the solar thermal power plant of 100 MW thermal have been compared in terms of their design features. The solar power plant consists of the beam-down sun-beam collecting system, sun-energy receiver provided with aluminum heat transfer and storage blankets and the CO_2 gas turbine with 650° C turbine inlet temperature. The designs were conducted for the flow schemes with the same number of compressors.

The following conclusions were obtained.

1. The values of the cycle thermal efficiencies are 48.9% for the supercritical CO_2 GT cycle and 45.3% for the Brayton CO_2 GT cycle. Therefore, the former cycle shows a 3.6% advantage.
2. Compressor aerodynamic designs are more difficult for the former cycle than for the latter cycle, especially in the bypass compressor design.
3. No distinct difference exists in the turbine designs between both the cycles.
4. With respect to recuperators, the recuperator weight for the CO_2 GT cycle becomes twice of the Brayton CO_2 GT cycle.

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