The 6th International Symposium -
Supercritical CO_2 Power Cycles
March 27–29, 2018, Pittsburgh, Pennsylvania

Partial Load Characteristics of the Supercritical CO_2 Gas Turbine System for the Solar Thermal Power System with the Na-Al-CO_2 Heat Exchanger

Yasushi Muto, Noriyuki Watanabe
Masanori Aritomi, Takao Ishizuka
Tokyo Institute of Technology

Objective of this study

To show the technical feasibility of the new solar power system for the partial load operation

Contents

1. The design of the new solar thermal power generation system provided with a Na cooled receiver, an Al heat storage heat exchanger and the supercritical CO_2 gas turbine.
2. Partial load calculation method
3. Results of partial load calculation.
4. Conclusions

New Solar Thermal Power Generation with a S-CO2 Gas Turbine

Incident Sunbeam
Thermal Energy = 125 MW

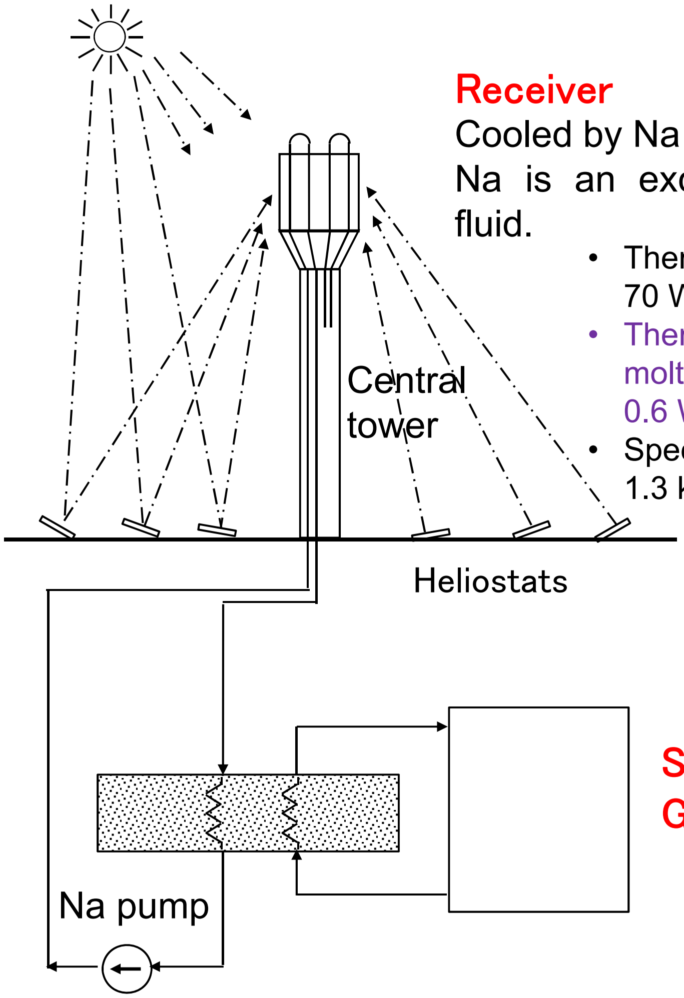
- Heliostat field diameter 800m
- Number of heliostats (φ3.4m) 42,519
- Central receiver height from the ground 100m

Due to the reference (Hasuike)

Na–Al–CO2 Heat Exchanger

Aluminum

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



Receiver

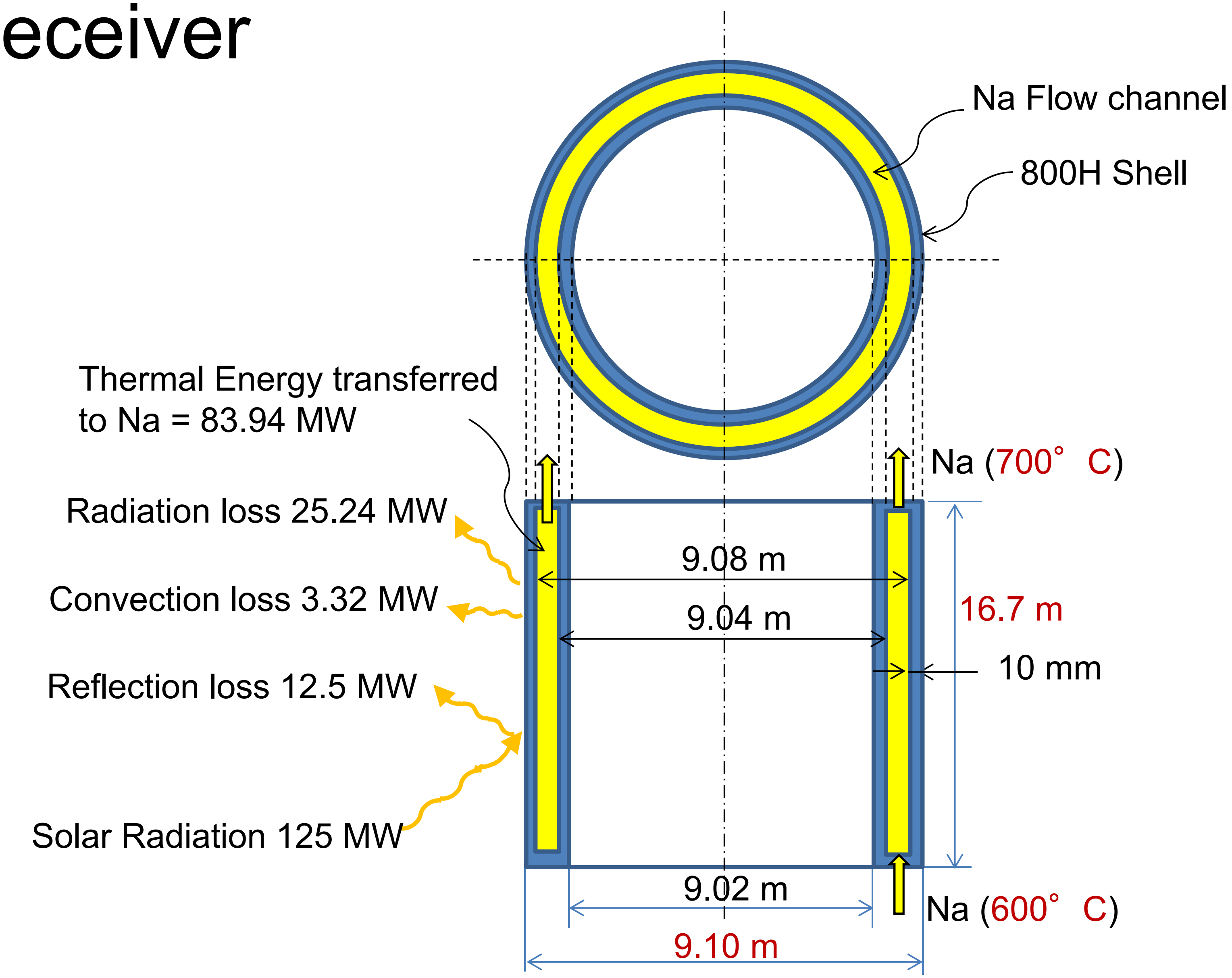
Cooled by Na

Na is an excellent heat transfer fluid.

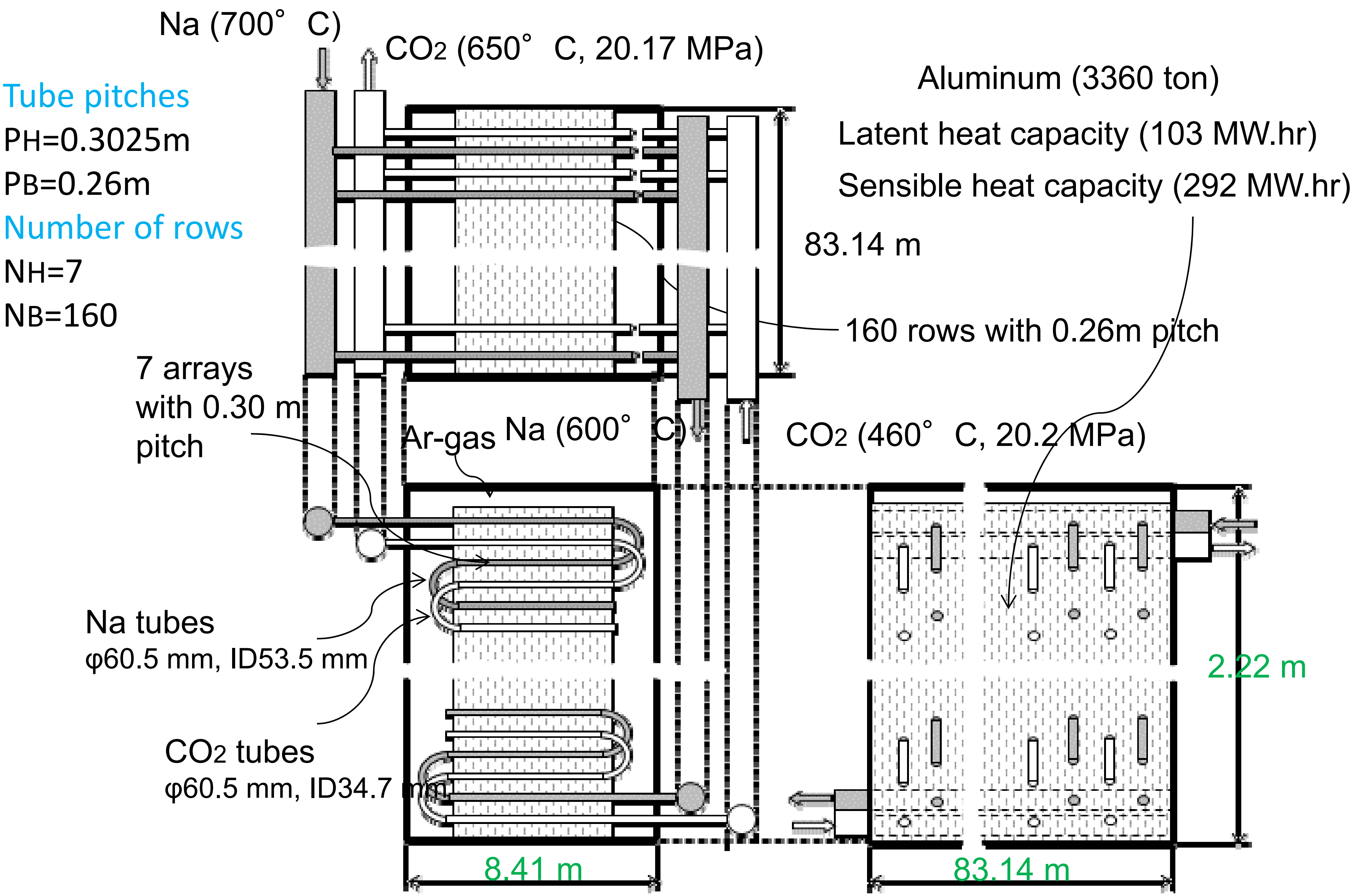
- Thermal conductivity 70 W/m/K
- Thermal conductivity of molten salt 0.6 W/m/K
- Specific heat 1.3 kJ/kg.K

Supercritical CO_2 Gas Turbine

Receiver


Na Flow channel
800H Shell
Thermal Energy transferred to Na = 83.94 MW
Radiation loss 25.24 MW
Convection loss 3.32 MW
Reflection loss 12.5 MW
Solar Radiation 125 MW
Na (700° C)
9.08 m
9.04 m
16.7 m
10 mm
9.02 m
Na (600° C)
9.10 m

Na-Al-CO_2 Heat Exchanger



Tube pitches

P_H=0.3025m

P_B=0.26m

Number of rows

N_H=7

N_B=160

Aluminum (3360 ton)

Latent heat capacity (103 MW.hr)

Sensible heat capacity (292 MW.hr)

Daily Power Demand and Aluminum Volume for the Heat Storage

- 50% of total solar thermal energy are needed to be stored (Reference Dunn).
- Demand assumed

Daytime	7:30 a.m. -	4:30 p.m.	100% of full power × 9 hr
Night	4:30 p.m. -	8:00 p.m.	90% of full power × 3.5 hr
	8:00 p.m. -	10:00 p.m.	80% of full power × 2 hr
	10:00 p.m. -	6:00 a.m.	38% of full power × 8 hr
	6:00 a.m. -	7:30 a.m.	80% of full power × 1.5 hr

- Aluminum volume for the heat storage

Melting of 30% is assumed.
Aluminum vessel dimensions = W8.41m x H2.22 m x D83.1 m
weight = 3,360 ton
Heat storage capacity
Latent heat = 103 MW.hr
Sensible heat = 292 MW.hr for the temperature difference of 300° C

Supercritical CO_2 GT Cycle

Assumptions

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

① Heat source 1.0%

② Recuperator high temperature side 1.2%

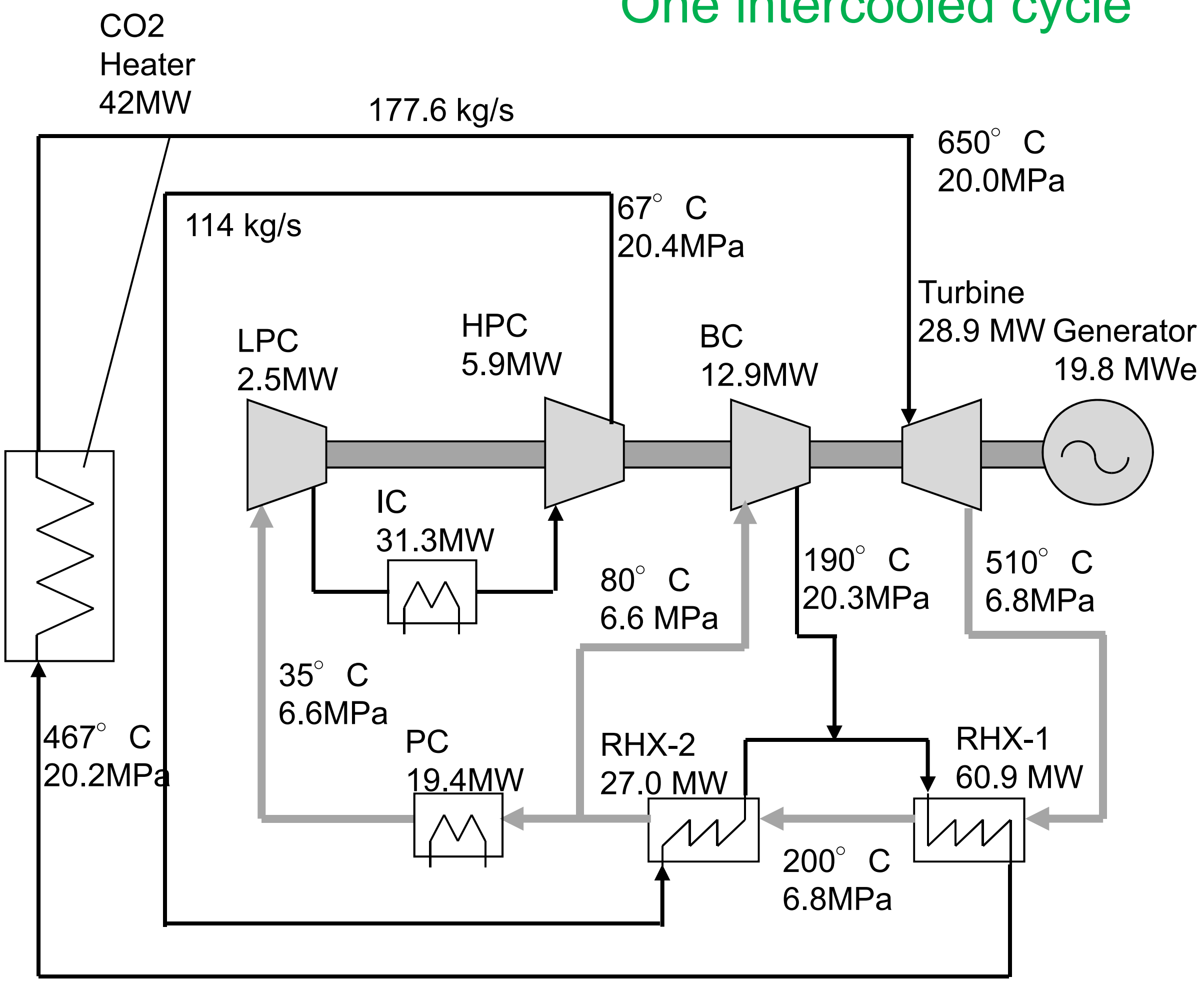
③ Recuperator low temperature side 0.4%

④ Precooler 1.0%

⑤ Intercooler 0.8%

- Recuperator average temperature effectiveness 89%

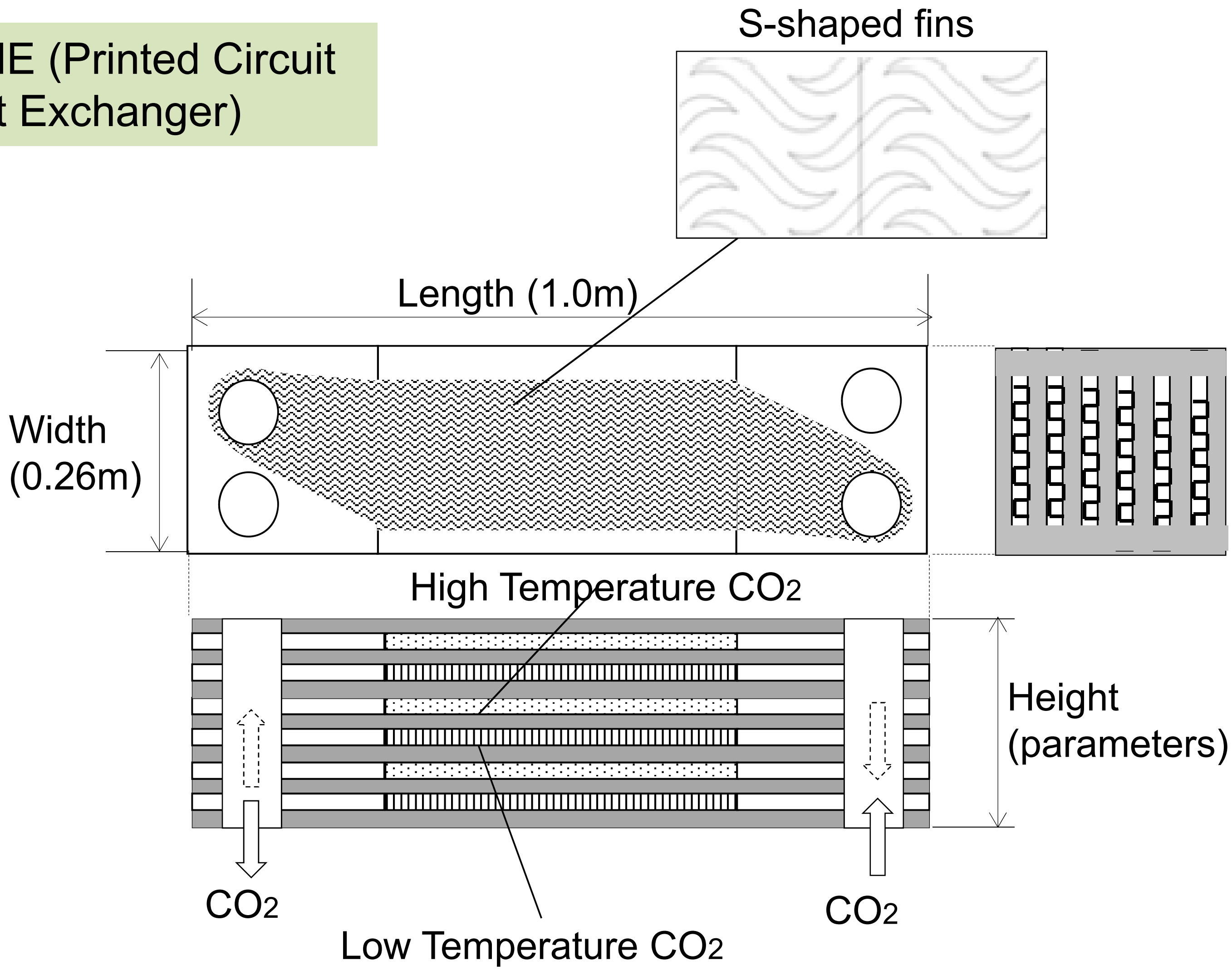
One intercooled cycle



Cycle Thermal Efficiency = 48.2%

Recuperator Designs

PCHE (Printed Circuit Heat Exchanger)



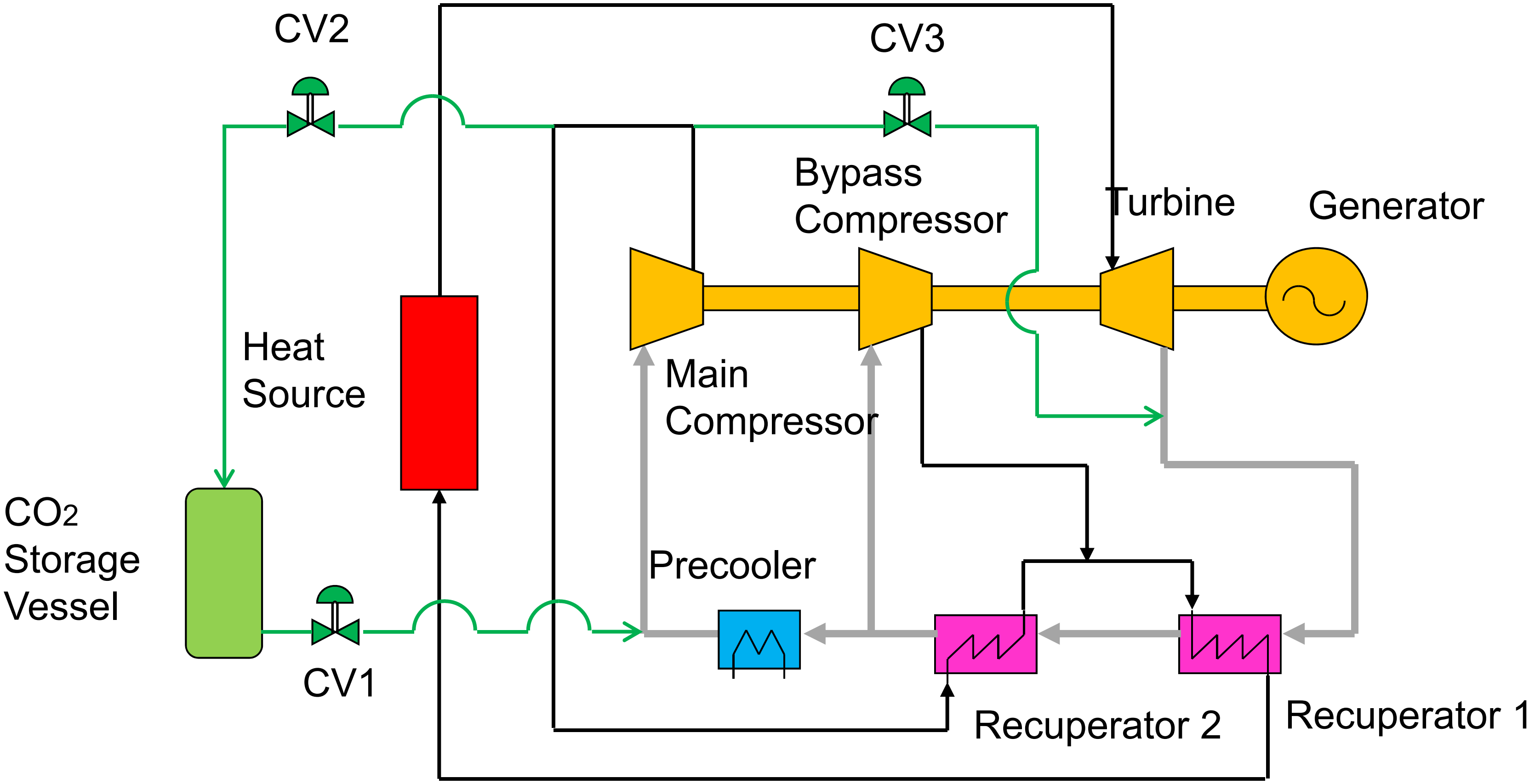
Design Conditions of the Recuperators

Items			**Supercritical CO_2 Gas Turbine**	
			RHX-1	RHX-2
Recuperator effectiveness		%	89	89
Number of modules			5	5
Heat load		MW/ modules	12.17	5.39
HT side	Flow rate	kg/s	35.52	35.52
	Inlet temperature	° C	509.9	210.7
	Inlet pressure	MPa	6.780	6.698
LT side	Flow rate	kg/s	35.52	22.84
	Inlet temperature	° C	190.3	69.0
	Inlet pressure	MPa	20.283	20.365

Results of the Recuperator Designs

Items		Supercritical CO_2 Gas Turbine	
		RHX-1	RHX-2
Width × Length	m/ module	0.26 × 1.0	0.26 × 1.0
Height	m/ module	4.24	3.65
Weight	ton/ module	7.90	6.80
Total weight	ton	39.5	34.0
Heat transfer capacity	MW	12.19	5.42
Pressure loss ratio (dP/Pinlet)	HT side %	0.74	0.27
	LT side %	0.58	0.08
Effective heat transfer area	HT side m^2	413.7	354.6
	LT side m^2	206.8	177.3
Overall heat transfer coefficient	$J/m^2/K/s$	909	1049

Partial Load Control
Inventory (pressure level) control



Partial Load Analysis Calculation Assumptions

1. The rotational speed is kept constant by the generator to meet the power grid frequency.
2. In this condition, the pressure ratio of both the turbine and the compressors should not be varied and should be kept in the rated values.
3. It is assumed that the bypass flow rate is kept constant.

Assumptions (Continued)

1. Pressure loss ratios of the components are assumed constant.
2. Inlet temperatures of both the low pressure compressor and the high pressure compressor are assumed to be kept constant by regulating the pre-cooler flow rate.
3. The inlet temperature of the RHX-1 high pressure side is calculated as the mixture of thermal energies of the RHX-2 high pressure side outlet gas and that of the bypass compressor outlet gas.

Fixed Input Data

1. Turbine pressure ratio (2.95), low pressure compressor pressure ratio (1.26)
2. Low pressure compressor inlet temperature (35° C), high pressure compressor inlet temperature (35° C)
3. Component pressure ratios, which are the same values with those for the design
4. Turbine adiabatic efficiency (92%), compressor adiabatic efficiency (88%), generator efficiency (98%)
5. Design turbine flow rate (166.72 kg/s)
6. Design bypass flow ratio (0.357)
7. Design turbine inlet temperature (650° C), Design turbine inlet pressure (20MPa)
8. Design turbine inlet CO_2 density (110 kg/m^3)
9. Recuperators dimensions

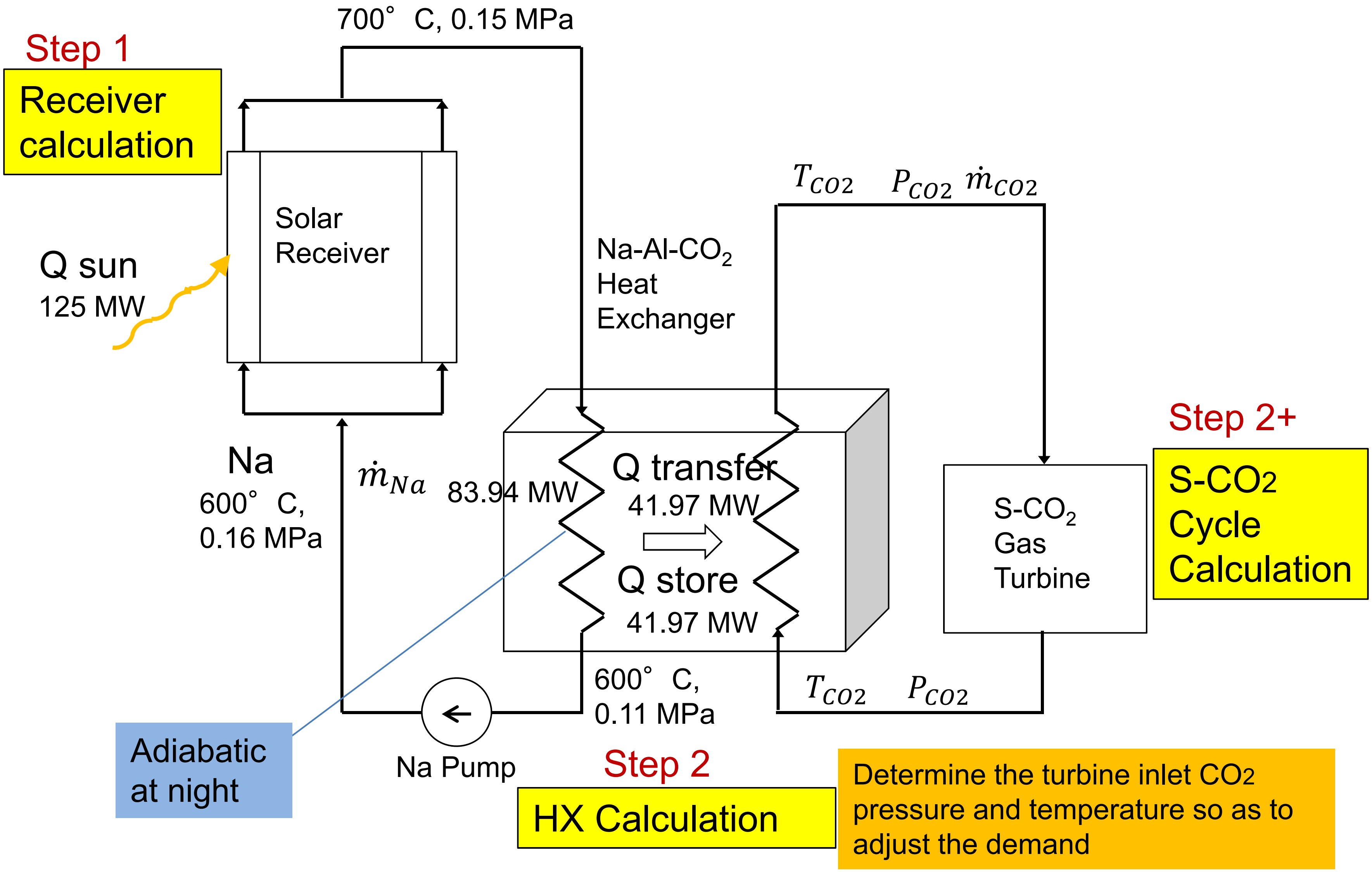
Input Data Varied

1. Turbine inlet pressure (MPa)
2. Turbine inlet temperature (°C)

To adjust the daily power demands.

Dependent on them, CO_2 density changes and then, CO_2 mass flow rate change.

Partial Load Analysis Procedure



Step 1

Receiver calculation

700° C, 0.15 MPa

Solar Receiver

Q sun
125 MW

Na
600° C,
0.16 MPa

$\dot{m}_{Na}$ 83.94 MW

Na-Al-CO$_2$
Heat
Exchanger

T_{CO2} P_{CO2} $\dot{m}_{CO2}$

Q transfer
41.97 MW

Q store
41.97 MW

600° C,
0.11 MPa

T_{CO2} P_{CO2}

Na Pump

S-CO$_2$
Gas
Turbine

Step 2+

S-CO2 Cycle Calculation

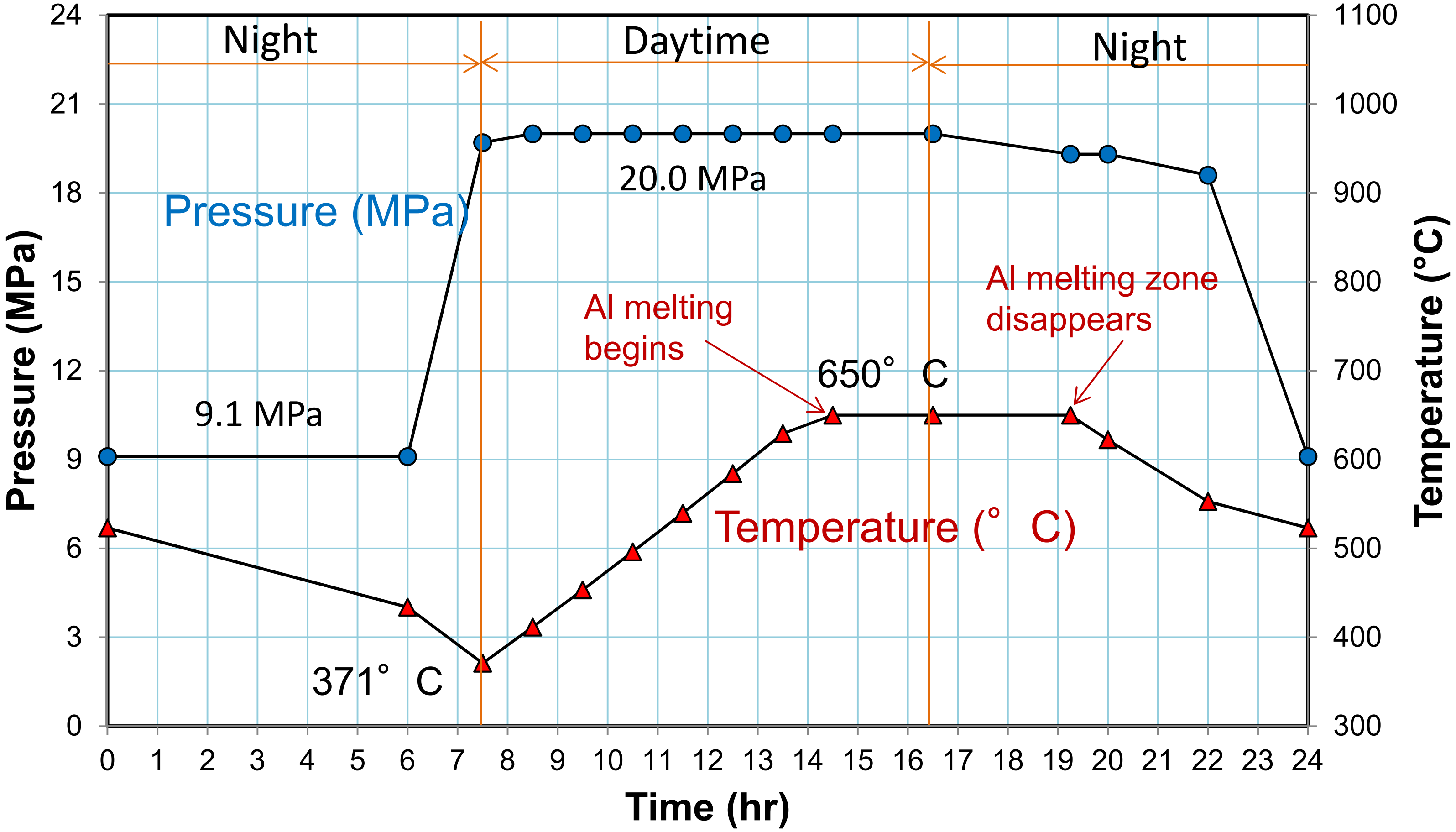
Adiabatic at night

Step 2

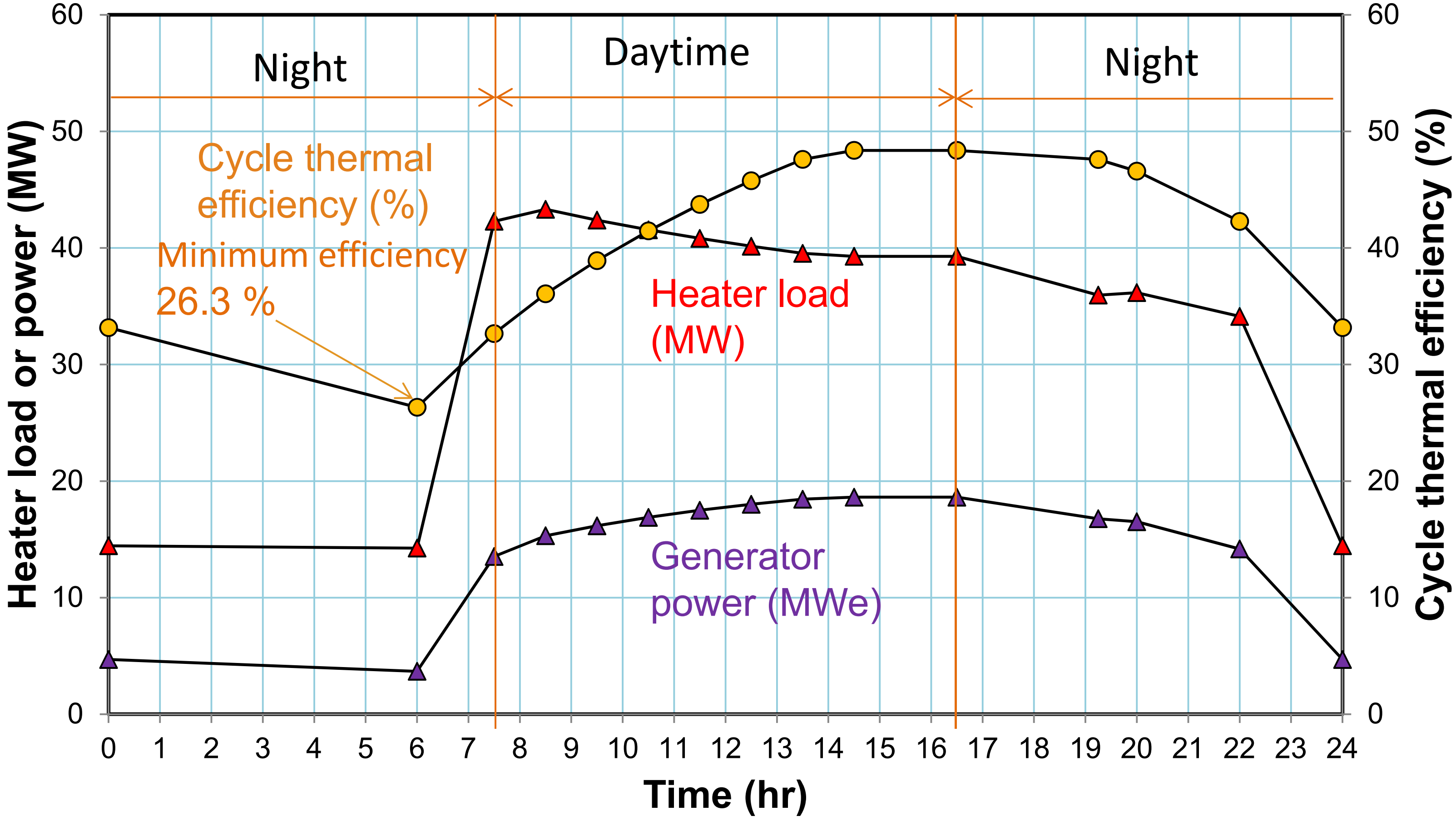
HX Calculation

Determine the turbine inlet CO2 pressure and temperature so as to adjust the demand

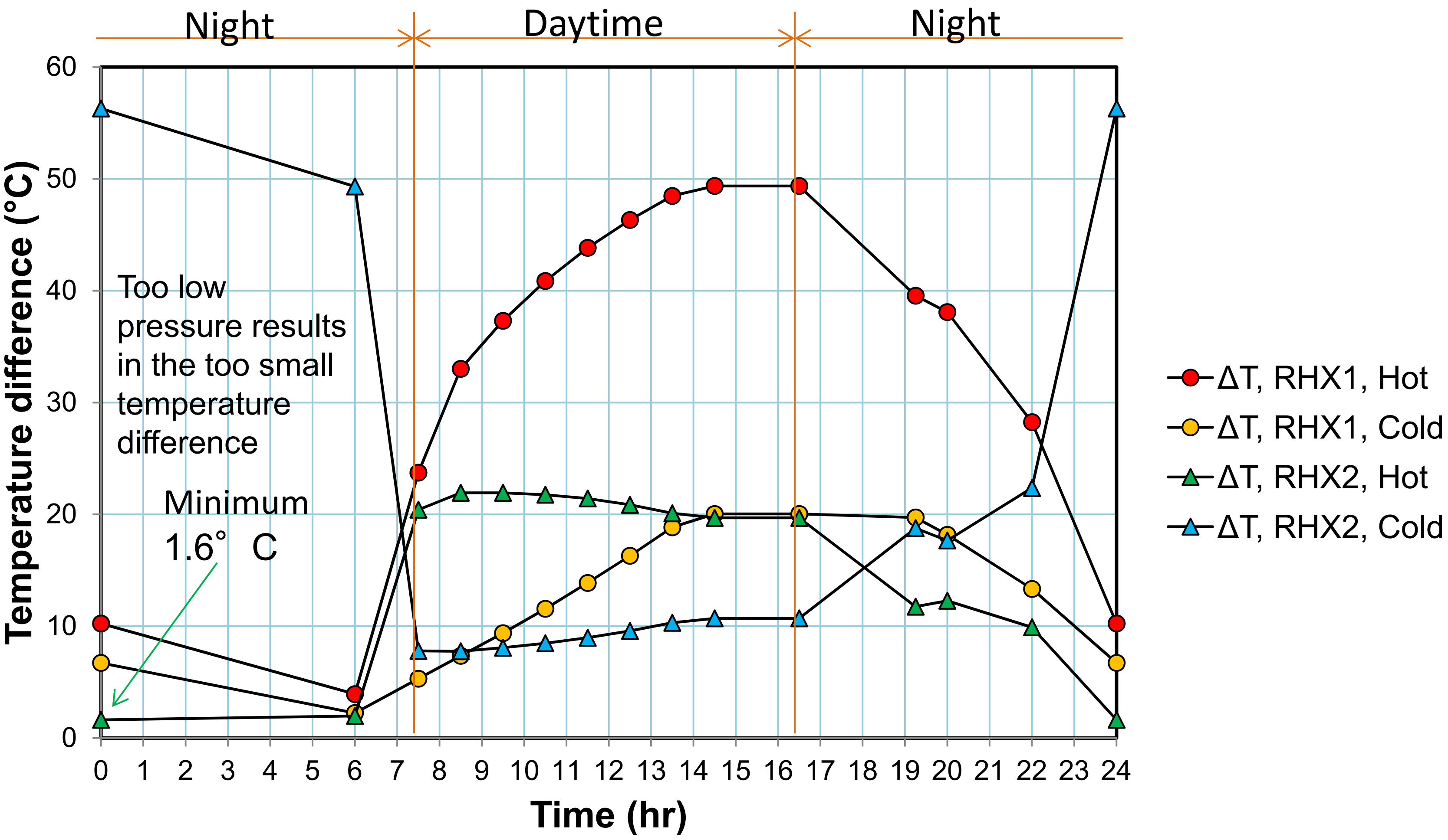
Results of Partial Load Analyses
Turbine inlet pressure and temp. (Input data)



Heater load, Generator power and cycle thermal efficiency



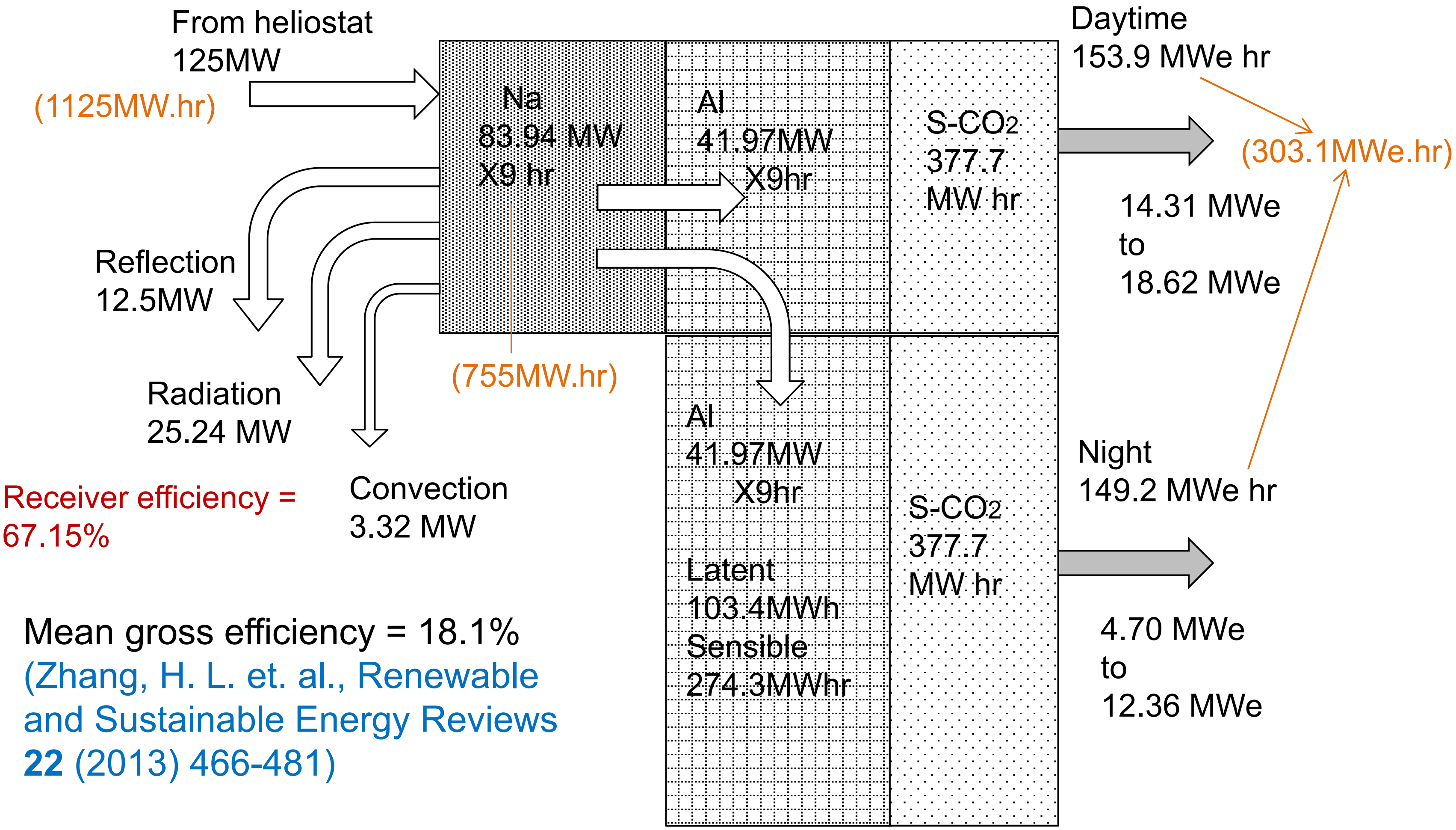
End temperature differences of recuperator 1 and 2



Too low pressure results in the too small temperature difference

Minimum 1.6° C

Energy Balance



From heliostat
125MW
(1125MW.hr)

Na
83.94 MW
X9 hr

(755MW.hr)

Al
41.97MW
X9hr

S-CO_2
377.7
MW hr

Al
41.97MW
X9hr

Latent
103.4MWh
Sensible
274.3MWhr

S-CO_2
377.7
MW hr

Reflection
12.5MW

Radiation
25.24 MW

Convection
3.32 MW

Receiver efficiency =
67.15%

Daytime
153.9 MWe hr

(303.1MWe.hr)

14.31 MWe
to
18.62 MWe

Night
149.2 MWe hr

4.70 MWe
to
12.36 MWe

Mean gross efficiency = 18.1%
(Zhang, H. L. et. al., Renewable and Sustainable Energy Reviews **22** (2013) 466-481)

- *Average thermal efficiency = 303.1MWe.hr/1125MW.hr = 26.9%*
- *Average power generating efficiency = 303.1 Mwe.hr/ 755 MW.hr = 40.2%*

Conclusions

1. 50% solar thermal energy should be stored in aluminum for the night power demand. The aluminum inventory becomes 3,000 tons, where the latent and sensible heat storages are 103.4 MW.hr and 274.9 MW.hr, respectively. The aluminum vessel dimensions are 8.41 m width, 2.22 m height and 83.14 m depth.

2. From the sunset (4:30p.m.) to 7:15 p.m., the power is supplied by the latent heat. In the midnight, the turbine inlet pressure falls to the minimum of 9.1 MPa. Minimum temperature difference of the recuperater becomes 1.6°C. The turbine inlet temperature becomes the minimum of 371°C at 7:30 a.m. early in the morning.

3. Total 303.1MWe.hr are produced. The average thermal efficiency and the average cycle efficiency are 26.9% and 40.12%, respectively.

4. The partial operation of this system is technically feasible though more comprehensive studies including turbomachinery are needed.