COMPARISON OF SUPERCRITICAL CO2 GAS TURBINE CYCLE AND BRAYTON CO2 GAS TURBINE CYCLE FOR SOLAR THERMAL POWER PLANTS

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ABSTRACT

A supercritical CO_2 gas turbine cycle can achieve extremely high cycle thermal efficiency using compression work reduction near the critical point. However, the cycle must be provided with a bypass circuit, which presents some difficulty in the compressor design and operation, particularly in small capacity plants. However, in a typical Brayton cycle flow scheme without a bypass circuit, considerably higher thermal efficiency can be achieved than with other turbine cycles. In such cases, it will be useful to evaluate the two flow schemes of the CO_2 gas turbine. 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 a solar thermal power plant of 100 MW were compared for this study in terms of their design features. The solar power plant consists of a 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. Designs were conducted for flow schemes having an equal number of compressors. The values of cycle thermal efficiency were 48.9% for the supercritical CO_2 GT cycle and 45.3% for the Brayton CO_2 GT cycle. The former cycle exhibits a 3.6% advantage. However, compressor aerodynamic designs are more difficult for the former cycle than for the latter cycle, especially for the bypass compressor design. No distinct difference in turbine designs is

apparent between the cycles. With respect to recuperators, the recuperator weight for the latter cycle becomes half that for the former cycle.

INTRODUCTION

Research and development efforts by the Tokyo Institute of Technology and the Japan Atomic Research Institute have been devoted to the supercritical CO_2 gas turbine applied to nuclear fast reactors (Kato, 1999; Aritomi, 2011). In actuality, CO_2 shows no progressive reaction with Na. Moreover, the supercritical CO_2 gas turbine can achieve extremely high thermal efficiency at fast reactor temperatures. A 600 MW(t) plant with CO_2 turbine inlet pressure and temperature of 20 MPa and 800 K has been designed (Muto, 2006), justifying the assumed values of adiabatic efficiency of the compressor and the turbine in the cycle calculations. Then, design of turbomachinery for a larger capacity plant of 1,500 MWe (750 MWe for one loop) was conducted (Muto, 2010). This capacity is the target of the first commercial reactor in Japan. In this case, the design was assigned to a Japanese gas turbine maker. Designs of axial machines and centrifugal machines were compared. Results revealed that the axial type is superior to the centrifugal type for this capacity. The achievable values of adiabatic efficiency were, respectively, 88% and 92% for the compressor and the turbine. The supercritical CO_2 cycle necessitates a unique flow scheme in which the flow to the compressor is divided into two streams and a bypass compressor line is added. In addition, pressure of 20 MPa is needed. The value of cycle thermal efficiency is markedly reduced if the pressure is considerably lower than 20 MPa. At less than 13 MPa, the bypass flow effects disappear and the maximum achievable efficiency becomes equal for both the cycle with bypass flow and for the usual Brayton cycle. As the pressure decreases further, the cycle thermal efficiency continues to decrease gradually.

In fact, the CO_2 gas turbine is applicable to solar thermal power generation. In this application, the power level is lower, i.e., 10–30 MWe and the maximum temperature is 500–650°C the load is changed daily. Furthermore, the operation must be simple. Therefore, not only cycle thermal efficiency but also simplicity, easy operation and easy development are crucially important. The cycle thermal efficiency of the Brayton cycle CO_2 gas turbine is 4% lower than that of the supercritical CO_2 gas turbine, but it is 3% higher than that of the ideal (He) cycle gas turbine at the turbine inlet temperature of 650°C. The appropriate pressure is approximately 10 MPa. The component design becomes easier than that for the supercritical cycle of 20 MPa. In addition, the operation and control is simple and straightforward, except for the charge and discharge of CO_2. Therefore, this system is expected to be appropriate for solar power applications.

As described herein, the values of the cycle thermal efficiency are calculated and evaluated both for the supercritical CO_2 gas turbine and the Brayton cycle gas turbine for the solar thermal power heat source of 100 MW. Designs of components such as compressors, turbines, and recuperators are conducted and discussed for both cycles.

SOLAR THERMAL POWER GENERATION WITH A CO_2 GAS TURBINE

Many solar power generation systems exist, such as a concentrating solar power type, trough type, and dish type. Among these, an appropriate system applied to the CO_2 gas turbine cycle is the concentrating solar power type, which can achieve high temperatures. Existing concentrating solar power systems consist of a central tower heat receiver surrounded by numerous heliostats on the ground, a molten salt heat transport system, and a steam turbine. However, two problems persist. The first is the maximum allowable temperature of molten salt, which is 600°C. Consequently, the maximum CO_2 temperature is limited to about 550°C. At such temperatures, the cycle thermal efficiency of the supercritical CO_2 gas turbine cycle is only slightly higher than that of the steam turbine cycle. Therefore, the benefit of the supercritical CO_2 gas turbine cycle cannot be used to any great degree. A new transport material must be used. The second problem is enhanced radiation loss for the higher temperature. As the surface temperature of the receiver becomes higher, the radiation heat loss increases in proportion to the fourth power of an absolute temperature. Our solutions for these problems are the use of metal aluminum instead of the molten salt and the use of a beam-down solar energy-collecting system instead of a solar power tower collector. The new system is presented in Fig. 1.

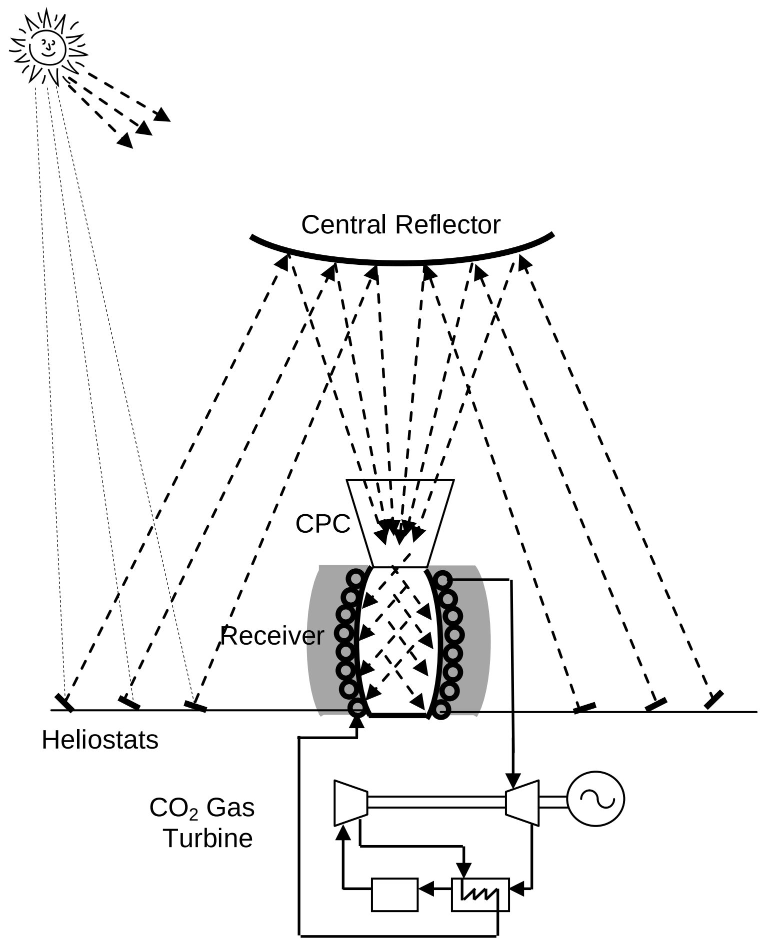

Fig. 1. Solar power generation system combined with aluminum and a CO_2 gas turbine.

Sun beams are reflected at the heliostats and are collected to the central reflector, from which they are reflected downward. The receiver comprises a compound parabolic concentrator (CPC), the receiver vessel, and a surrounding aluminum blanket, where vertical CO_2 heat transfer tubes are provided. The receiver vessel has a bottle shape. Although the receiver surface temperature becomes extremely high, such as 750°C and although much radiation energy is emitted, radiation energy loss is possible only through the top bottleneck small opening. Therefore, the radiation loss is confined and becomes less than that of a solar tower, which has a receiver exposed to the open atmosphere. The received heat energy is transferred via aluminum to CO_2. Aluminum is a good heat transfer material with thermal conductivity of 237 W/m/K. Actually, aluminum melts at 660°C; its melting heat is 397 kJ/kg, which is extremely high. In addition, its specific heat is 0.897 kJ/kg/K, which is extremely high. When aluminum is heated to 660°C in the morning, the solar energy is used to raise the CO_2 gas temperature and to melt the aluminum. After the sun sets, the melted aluminum is solidified and the heat of solidification is emitted while the melted aluminum exists. Subsequently, because of the high specific heat, a considerable amount of heat can remain available during the period when temperatures decrease.

As described in this paper, 100 MW net heat input to the receiver is assumed (Hasuike, 2006). This condition is achieved in a heliostat field of 400 m radius. Such a field would have 42,519 circular heliostats of 3.4 m diameter. The central reflector is placed at 114 m height. The aluminum is charged in a coaxial cylindrical vessel with ID16 m, OD28 m, and 9 m effective height. The aluminum, weighing 4,800 tons, has volume of 2,000 m^3. Within the aluminum vessel, 15,000 heat transfer tubes with OD34 mm are placed in 136 mm triangular pitch.

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

Two Flow Schemes for the CO_2 Gas Turbine Cycle

Two flow schemes are used for the CO_2 gas turbine cycle with supercritical pressure. One is a *supercritical CO_2 gas turbine cycle* presented in Fig. 2, where a bypass compressor circuit is provided and the turbine inlet pressure is extremely high, such as 20 MPa. Such high pressure is necessary to obtain the merit of compression work reduction near the CO_2 critical point (31.06°C, 7.38 MPa). However, a bypass circuit is necessary to ease the reduction of recuperator performance. In this flow scheme, the main compressor and the bypass compressor must operate in parallel. Then, the system and its operation become extremely complicated. The other has the original simple flow scheme of the typical closed cycle gas turbine depicted in Fig. 3. In this case, the benefits of compression work reduction cannot be obtained sufficiently. However, the greater the degree to which the pressure exceeds the critical pressure, the lower the amount of compression work becomes. Then the thermal cycle efficiency becomes slightly higher than that of the Brayton cycle with an ideal gas such as helium. To discriminate between the two flow schemes, the system depicted in Fig. 3 is designated as the *Brayton CO_2 gas turbine cycle* herein.

For the supercritical CO_2 gas turbine cycle, intercooling is provided. At 650°C solar power application, one intercooling has 1.7% thermal efficiency advantage compared with a non-intercooling cycle (Muto, 2013). The intercooling increases the number of compressors and necessitates the addition of an intercooler. However, it does not greatly complicate the system. For the Brayton CO_2 gas turbine cycle, two intercooling circuits are used. The cycle thermal efficiency of the cycle with two intercooling circuits is 2.6% higher than that of the cycle with one intercooling. Additionally, it will be reasonable to compare the supercritical cycle and Brayton cycle for the same number of compressors.

Figures 2 and 3 show the calculated optima of the temperature, pressure, heat capacity and flow rate.

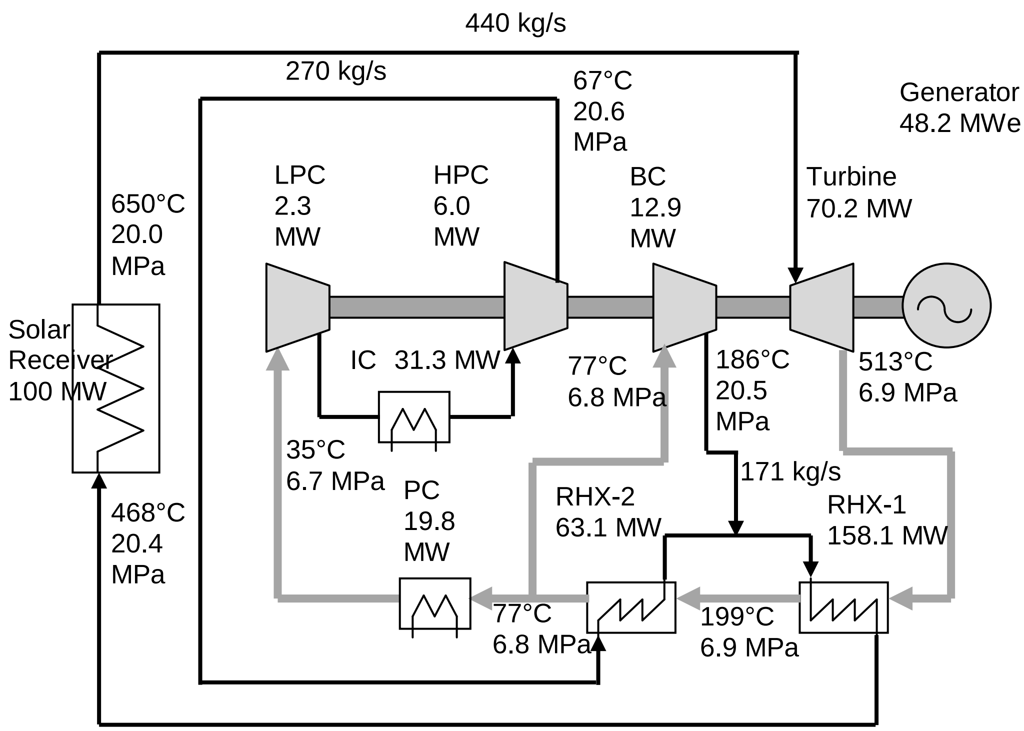

Fig. 2. Flow diagram of supercritical CO_2 gas turbine cycle with an intercooling.

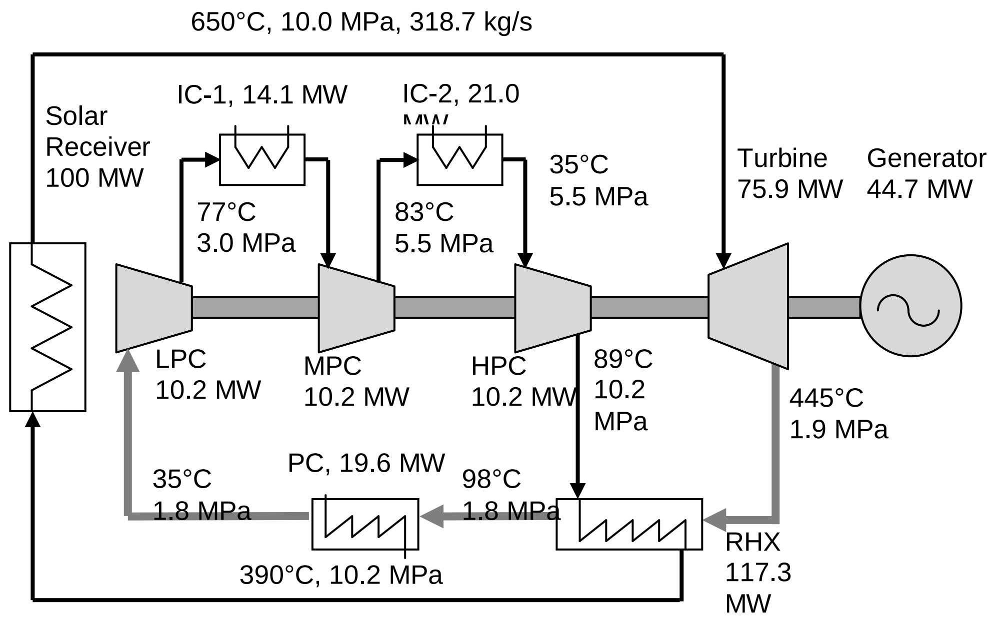

Fig. 3. Flow diagram of Brayton CO_2 gas turbine cycle with two intercoolings.

Cycle Thermal Efficiency

The cycle thermal efficiency was calculated under the following assumptions.

Turbine adiabatic efficiency	92%
Compressor adiabatic efficiency	88%
Pressure loss (ratios over the inlet pressure)	
Solar receiver	2.0%
Recuperator high temperature side	1.2%
Recuperator low temperature side	0.4%
Precooler	1.0%
Intercooler	0.8%
Recuperator average temperature effectiveness	91%

The solar receiver capacity is 100 MW thermal. The value has no relation with the cycle thermal efficiency in the cycle calculations but important for the components design.

The values of adiabatic efficiency are determined based on the conceptual design (Muto, 2010) of the gas turbine of the 1,500 MWe commercial power plant for the Na-cooled fast reactor.

Generally speaking, the following definitions of recuperator effectiveness are used.

$$\eta_{RHX,Hot} = \frac{T_{H,in} - T_{H,out}}{T_{H,in} - T_{C,in}}$$

$$\eta_{RHX,Cold} = \frac{T_{C,out} - T_{C,in}}{T_{H,in} - T_{C,in}},$$

In those equations,

$\eta_{RHX,Hot}$ = High-temperature side temperature effectiveness

$\eta_{RHX,Cold}$ = Low-temperature side temperature effectiveness

$T_{H,in}$ = High-temperature side inlet temperature

$T_{H,out}$ = High-temperature side outlet temperature

$T_{C,in} =$ Low-temperature side inlet temperature

$T_{C,out} =$ Low-temperature side outlet temperature

For a usual gas turbine cycle such as a helium turbine, the values of the high-temperature side and low-temperature side are almost equal. In the case of S-CO_2, however, these values differ considerably. Therefore, for these analyses, the next average value $\eta_{RHX,av}$ of the high-temperature and low-temperature side is adopted for the calculation base.

$$\eta_{RHX,av} = \frac{1}{2}\left(\eta_{RHX,Hot} + \eta_{RHX,Cold}\right)$$

The value of 91% is assumed, as determined, based on our previous experience, as the value at which a recuperator design of reasonable size is possible. Nevertheless, in some extreme cases, the value of high-temperature or low-temperature side temperature effectiveness can become extremely large. The other side temperature effectiveness becomes exceedingly small. In such cases, the hot leg or cold leg temperature difference can become extremely small. To exclude such cases, the minimum acceptable value of 5°C is assumed for the hot leg or the cold leg temperature difference.

The gas properties of CO_2 such as enthalpy and entropy are based on a computer dataset "PROPATH" (PROPATH Group, 1990).

Pressure dependences of cycle thermal efficiency and turbine pressure ratio for the supercritical CO_2 cycle are portrayed in Fig. 4. Pressure dependences of the bypass flow ratio and minimum recuperator end temperature difference are presented in Fig. 5. As the pressure increases, the cycle thermal efficiency increases, and the values of turbine pressure ratio become roughly the same, i.e., about 2.8–3.1. The values of the bypass flow ratio and the minimum end temperature difference change their slope suddenly at 16 MPa. For pressures less than 16 MPa, the desirable effect of the bypass flow decreases markedly. It disappears completely at less than about 7 MPa. Therefore, 20 MPa is selected as the optimum pressure because structural designs of the solar receiver and recuperators become extremely difficult.

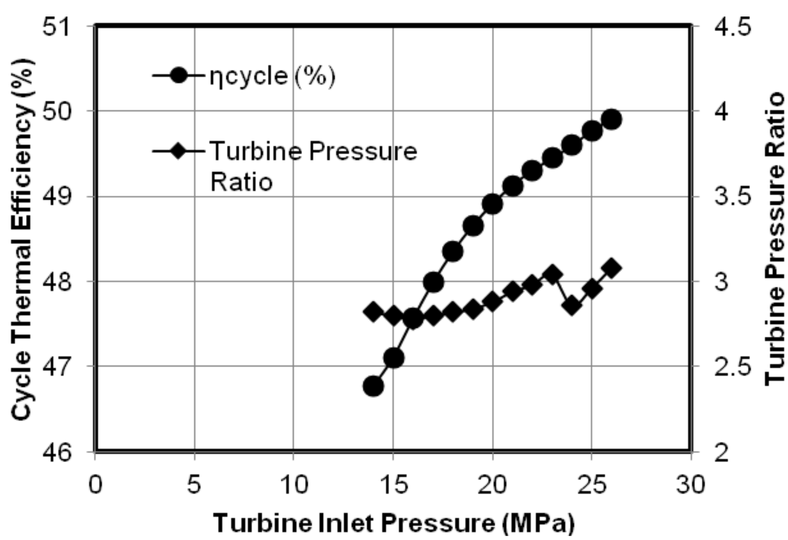

Fig. 4. Pressure dependences of cycle thermal efficiency and turbine pressure ratio for the 650°C supercritical CO_2 gas turbine cycle.

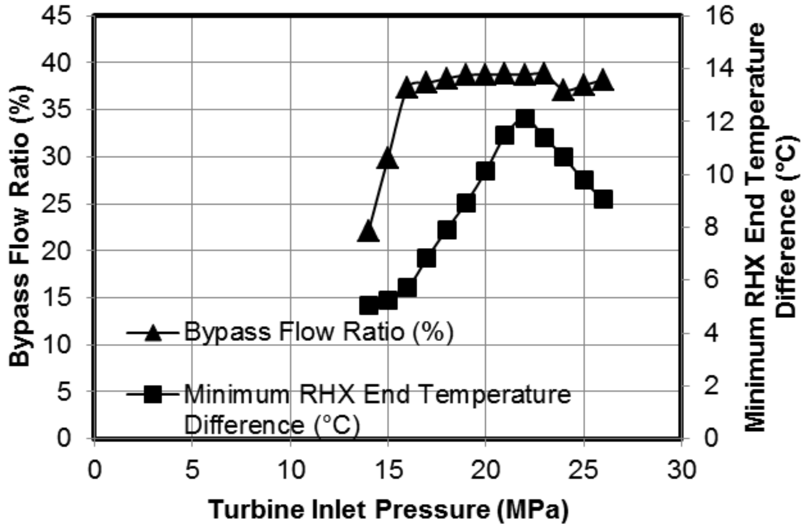

Fig. 5. Pressure dependences of bypass flow ratio and minimum recuperator end temperature difference for the 650°C supercritical CO_2 gas turbine cycle.

Pressure dependences of the cycle thermal efficiency and turbine pressure ratio for the Brayton CO_2 cycle are depicted in Fig. 6. Pressure dependences of the minimum recuperator end temperature difference are portrayed in Fig. 7. In the Brayton cycle, the cycle thermal efficiency increases linearly with the pressure increase. The turbine pressure ratios are about 5, which are considerably larger than those for the supercritical cycle. The values of minimum recuperator end temperature difference decrease linearly and become 5°C at 11 MPa. Then, 11 MPa is the maximum available pressure for this flow scheme. Actually, 9 MPa or 10 MPa is selected as the optimum pressure because the value of the recuperator end temperature difference greater than around 10°C is desired.

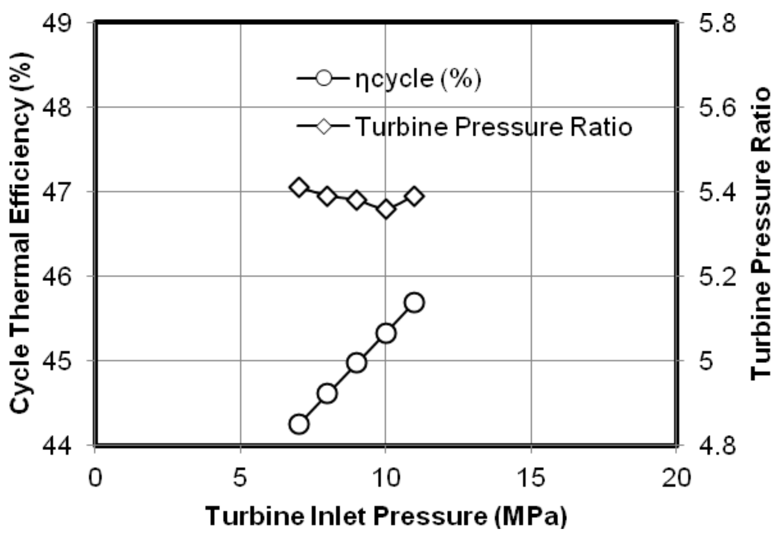

Fig. 6. Pressure dependences of cycle thermal efficiency and turbine pressure ratio for the 650°C Brayton CO_2 gas turbine cycle.

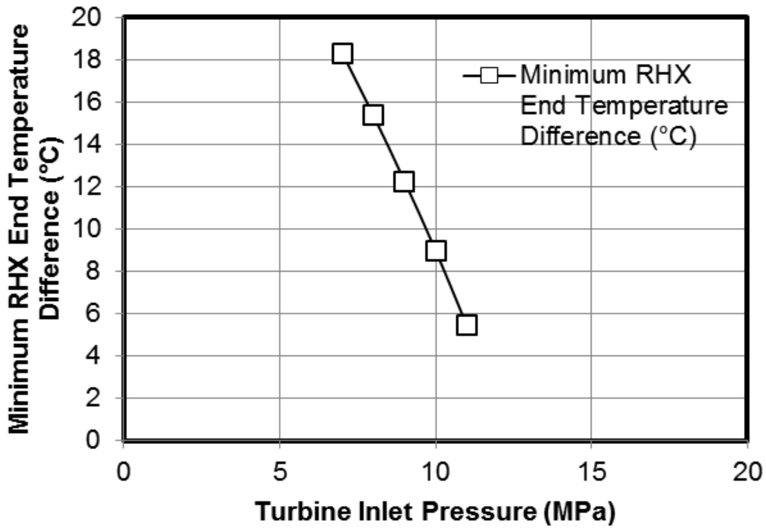

Fig. 7. Pressure dependences of minimum recuperator end temperature difference for the 650°C Brayton CO_2 gas turbine cycle.

Values of the cycle thermal efficiency for the two kinds of CO_2 gas turbine cycles are presented in Fig. 8, where the values for the helium turbine are also shown. The values of cycle thermal efficiency increase monotonically for both the supercritical GT cycle and Brayton GT cycle, although their numbers of intercooling differ. The reason is not clear. However, a probable reason is related to the values of recuperator temperature effectiveness. In the calculations above, the same value 91% is assumed for both the cycles. However, two recuperators are in the supercritical GT cycles. In this case, the total recuperator effectiveness of the supercritical GT cycle becomes higher than 91%.

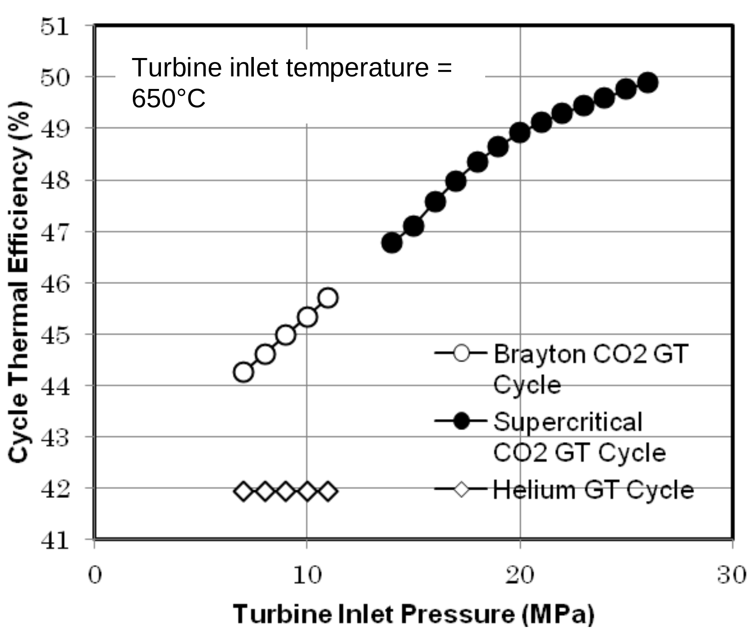

Fig. 8. Pressure dependences of cycle thermal efficiency between the supercritical CO_2 gas turbine cycle and the Brayton CO_2 gas turbine cycle at 650°C.

The cycle thermal efficiency for the 10 MPa Brayton GT cycle is 45.7%. This value is not so high, but it is still higher than the value of helium GT turbine cycle of 42%. This results from the fact that the compressor work reduction is still used effectively in the CO_2 Brayton GT cycle.

DESIGN CHARACTERISTICS OF SUPERCRITICAL CO_2 GAS TURBINE CYCLE COMPONENT DESIGNS

Compressor Designs

For the application of a supercritical CO_2 gas turbine for the solar thermal power plant, the compressor design is the most critical issue because of the design difficulty resulting from its small capacity. Their design conditions are presented in Table 1. Aerodynamic features for the compressors can be roughly evaluated using a non-dimensional parameter: *specific speed.* The specific speed is defined by the following equation for a centrifugal compressor (Cumpsty, 2004).

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

Therein, $m =$ mass flow rate (kg/s)

$\rho =$ average density (kg/m^3)

$N =$ rotational speed (radian/s)

$\Delta p =$ pressure rise (Pa)

Table 1 Design conditions of the CO_2 compressors for the 100 MW solar thermal power plant

Items	Supercritical CO_2 Cycle			Brayton CO_2 Cycle		
	LPC	HPC	BC	LPC	MPC	HPC
Inlet temperature °C	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	269.7	269.7	170.5	318.7	318.7	318.7

The specific speeds for the conditions above depend on the number of compressor stages shown in Fig. 9, where the rotational speed was assumed as 3,600 rpm.

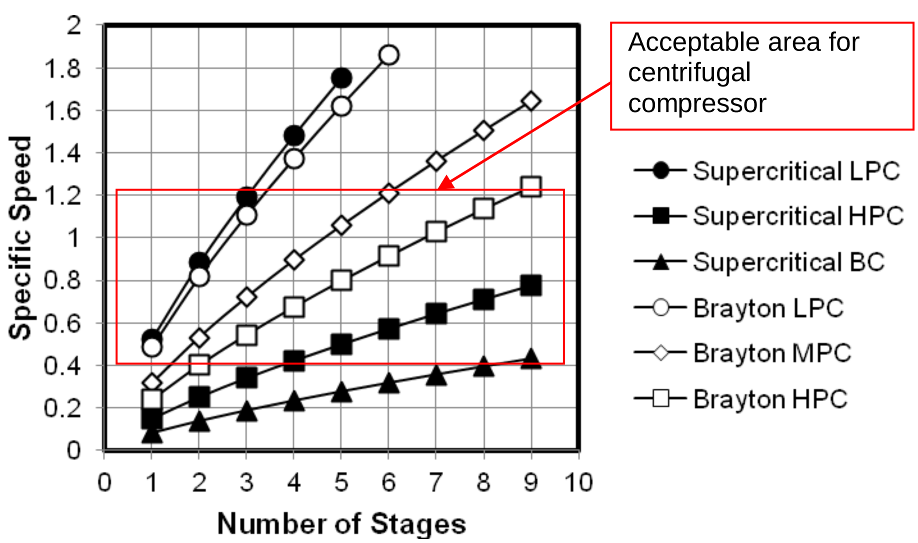

Fig. 9. Compressor specific speed dependent on the number of stages.

The design of a small compressor becomes easier for the selection of higher speed. However, this requires a mechanical speed reduction gear or frequency converter. The former is noisy, incurring mechanical loss. The latter is extremely expensive. Therefore, the same speed as the generator speed is desirable and is assumed in the analyses presented herein. An acceptable range of the specific speed is 0.4–1.2 for the centrifugal type. This figure reveals that aerodynamic designs are more difficult for the supercritical cycle. Particularly it is possible only for a number of stages greater than eight for the bypass compressor. Therefore, a centrifugal compressor is used for the supercritical CO_2 GT cycle. For the Brayton cycle, all compressors are located in the acceptable area for the few stages. Because the adiabatic efficiency is generally higher for the axial compressor than for the centrifugal compressor, the axial compressor is selected for the Brayton cycle compressors.

Supercritical CO_2 Cycle Compressors

Designs of centrifugal compressors were conducted based only on the value of the specific speed described above. A plot of the efficiency against specific speed was presented by Rogers (1980). In the equation, average values of the inlet density and the outlet density were used. To obtain these values, the compressor outlet temperature values are needed. They were estimated under the assumption of 85% adiabatic efficiency. The pressure increase from the inlet to the outlet was divided so that the value of pressure ratio becomes identical for each compressor stage.

The results of estimated impeller polytropic efficiency for the first and last stages are presented in Table 2. In addition, values of the impeller outer diameter are shown for the assumption of meridional speed of 30 m/s.

Table 2. Results of centrifugal compressor design for the supercritical CO_2 GT cycle

Items		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	First stage m	0.571	0.319	0.458
	Last stage m	-	0.431	0.529

Brayton Cycle Compressors

Design calculations for the axial compressor are conducted using the following procedure.

1. Assume the compressor outlet temperature for the sizing.
2. Ascertain the values of stage outlet pressure and temperature to achieve an equal isentropic enthalpy rise per stage.
3. Values of axial velocity (50 m/s was chosen) and nozzle inlet angle for each stage are given as input data. From these, the annulus flow area is determined for each stage.
4. Hub-to- tip ratio at the exit is given as input data. Values of mean peripheral diameter and peripheral speed are determined, which are assumed as constant.
5. All dimensions are determined by the annulus flow area and the mean peripheral diameter.
6. Velocity triangles of rotor blades are determined, where a value of tangential velocity change is calculated using the value of effective work, which is obtained using a stage enthalpy rise divided by the work-done-factor.
7. Stage reactions are calculated.
8. Loss is calculated for each stage.
9. Adiabatic efficiency is calculated. The calculations are repeated until the value converges.

In the method presented above, the values of the work-done-factor and loss calculation are based on the methods described by Cohen (1996).

The values of the axial gas velocity, solidity and the aspect ratio were assumed respectively as 50 m/s, 1.2 and 2.0. Under these assumptions, design calculations were conducted by varying the values of the hub-to-tip ratio and the number of compressor stages. The results are shown in Table 3. The values of adiabatic efficiency are higher than 89%, which exceeds the 88% assumed in the cycle calculation. Although the adiabatic efficiency values were not calculated for the centrifugal compressors of the supercritical CO_2 GT cycle, they are at most less than 88%. Therefore, the Brayton CO_2 GT cycle has an advantage in terms of the compressor design.

Table 3. Results of axial compressor design for the Brayton CO_2 GT cycle

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

Turbine Designs

The working conditions of the CO_2 turbines are remote from the critical point. Therefore, the existing design method for the air turbine is viable. One-dimensional aerodynamic design of an axial turbine has been conducted based on the loss model presented by Craig and Cox (1971).

The rotational speed was assumed as 3,600 rpm, as described in the compressor design. Chord lengths were assumed as 30 mm, 20 mm, and 20 mm, respectively, for nozzles, blades, and spaces between the nozzle and blade. The supercritical CO_2 turbine for the 100 MWt plant is rather smaller than commercial industrial turbines. Then the values of the aspect ratio tend to be small, sometimes less than 1.0, which means a considerable reduction of efficiency. To increase the efficiency, a small chord length was assumed, although the minimum producing limit remains unknown. Tip clearance was assumed as 0.008, although the producing accuracy for such a small blade size is also unknown.

Under these assumptions, aerodynamic calculations were conducted. Values of the adiabatic efficiency and stress were evaluated by varying the three parameter values of the number of stages, loading coefficient ψ , and flow coefficient ϕ , the definitions of which are the following.

$$\psi = \frac{\Delta h}{{u_m}^2}, \quad \phi = \frac{C_X}{u_m}$$

Therein, C_X = axial velocity (m/s),

Δh = Enthalpy drop (J/kg),

u_m = Mean peripheral velocity (m/s).

Regarding both the values of loading coefficient and flow coefficient, optimum values exist. As the number of stages increases, the mean rotor diameter decreases. Then, the blade heights increase, which causes higher adiabatic efficiency and simultaneously increases the blade stress. The maximum allowable stress is assumed as 400 MPa, which is the creep rupture strength of Mar-M247 for 10^5 h at 700°C. Therefore, the design point was determined as a condition achieving the maximum adiabatic efficiency for these three parameters within the blade stress limit.

The turbine design conditions are shown in Table 4.

Table 4. Design conditions of the CO2 turbines for the 100 MW solar thermal power plant

Turbines	Supercritical CO_2 GT	Brayton CO_2 GT
Inlet temperature °C	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

The design results are presented in Table 5.

Results show that no distinct difference exists between the supercritical CO_2 turbine and the Brayton cycle CO_2 turbine in terms of their performance. They can each achieve 92% adiabatic efficiency assumed in the cycle calculations.

Table 5. Design results of the CO_2 turbines for the 100 MW solar thermal power plant

Turbines	Supercritical CO2 GT	Brayton CO2 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

Recuperator Designs

In the closed cycle gas turbine systems, the recuperator capacity generally becomes extremely large. In addition, the gas pressure becomes extremely high, particularly for the supercritical CO_2 cycle. Therefore, a compact heat exchanger with both high heat transfer ability and high pressure resistance is needed. A Printed Circuit Heat Exchanger (PCHE) is such a device. The PCHE was developed by an English company, Heatric. Recently, Tokyo Institute of Technology has developed a new flow channel named the "S-shaped-fin" (Kato, 2005, Tsuzuki, 2009), with pressure loss of 1/6 of the PCHE made by Heatric for the same heat transfer ability. This type of heat exchanger is used in the present design studies.

Figure 10 presents a schematic view and total dimensions for the module of PCHE with the flow channel of S-shaped-fins. The design conditions for the recuperators are shown in Table 6. The PCHE unit size is limited by production technology related to diffusion bonding. The maximum available space of the electric furnace is around 1 m.

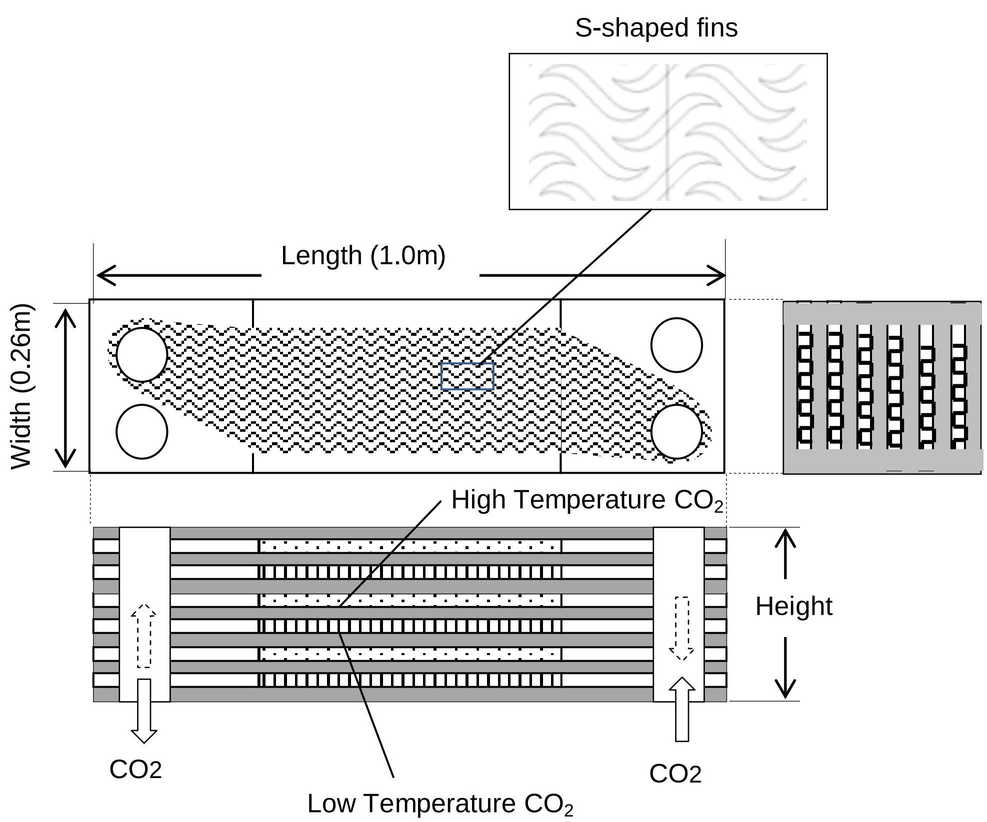

Fig. 10. Structure and dimensions of the recuperator module.

Calculations were conducted based on the following equations.
$\mathrm{Nu} = 0.167 R_e^{0.635} P_r^{0.354}$ for heat transfer (Tsuzuki, 2010).
For pressure drop (Muto, 2007).

$$f = \frac{d_H}{2\rho U^2} \cdot \frac{dP}{dx}$$

$$f = 0.11\mathrm{Re}^{-0.17}$$

Table 6. Design conditions of the recuperators

Items		Supercritical CO2 gas turbine		Brayton CO_2 GT
		RHX-1	RHX-2	RHX
Recuperator effectiveness %		91	91	91
Number of modules		12	12	12
Heat load MW/module		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 °C	185.74	67.17	89.14
	Inlet pressure MPa	20.490	20.572	10.245

Results of the design are presented in Table 7. Although the values of the recuperator effectiveness are equal, the necessary sizes differ depending on their heat loads, temperature profiles, and terminal temperature differences. Results show that the total weight of the recuperator modules for the supercritical CO_2 gas turbine is 236 tons, which is more than twice the total weight (102 ton) of the recuperator modules for the Brayton CO_2 gas turbine. Therefore, the Brayton CO_2 GT cycle is superior to the supercritical CO_2 GT cycle with respect to the recuperator.

Table 7. Results of the recuperator designs

Items		Supercritical CO_2 gas turbine		Brayton CO_2 GT
		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)	High T. side %	0.196	0.247	2.29
	Low T. side %	0.075	0.031	0.226

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 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. However, compressor aerodynamic designs are more difficult for the former cycle than for the latter cycle, especially in the bypass compressor design. No distinct difference exists in the turbine designs between both the cycles. With respect to recuperators, the recuperator weight for the latter cycle becomes half that of the former cycle.

NOMENCLATURE

BC = bypass compressor
CPC = compound parabolic concentrator
GT = gas turbine
HPC = high-pressure compressor
IC = intercooler
LPC = low-pressure compressor
MPC = medium pressure compressor
Ns = specific speed
PCHE = printed circuit heat exchanger
PC = precooler
RHX = recuperative heat exchanger
ψ= loading coefficient
φ= flow coefficient

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