Cycle Thermal Efficiency of Supercritical CO₂ Gas Turbine Dependent on Recuperator Performance*

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Abstract

The supercritical CO₂ gas turbine cycle can achieve high cycle thermal efficiency by reducing compressor work near the critical point. The achievable cycle thermal efficiency value is strongly dependent on the recuperator performance. Values of the achievable cycle thermal efficiency are calculated by assuming a value of average recuperator temperature-effectiveness of 91%. The turbine inlet pressure is 20 MPa. The effects of turbine inlet temperature are examined. The calculations are conducted both for the non-intercooling cycle and the intercooling cycle. Results show that the non-intercooling cycle is preferred to the intercooling cycle up to 600°C because of its simplicity. However, the latter is preferred to the former at temperatures greater than 600°C attributable to its approximately 2% higher efficiency. For the typical temperature of 527°C and also 650°C, the diagrams of mass and heat balance are given. The maximum cycle thermal efficiencies are, respectively, 43.4% and 48.9% for 527°C and 650°C. The effects of pressure are examined and 20 MPa is justified as an optimal value. Finally, the effects of recuperator effectiveness on the cycle thermal efficiency are examined, which are revealed to be linear and the cycle thermal efficiency increases about 0.5% for a 1% increase of the recuperator effectiveness.

Key words: Carbon Dioxide, Gas Turbine, Cycle Thermal Efficiency, Medium Temperature, High Pressure, Recuperator

1. Introduction

The supercritical CO₂ (S-CO₂) gas turbine cycle is a kind of a closed gas turbine cycle (Brayton cycle). The flow diagram of the cycle is presented in Fig. 1, where variation of two kinds is shown: a nonintercooling cycle and an intercooling cycle. The cycle consists of a heat source, a turbine, two compressors, a precooler, an intercooler and two recuperators. The S-CO₂ cycle effectively uses the compressor characteristics that exist near the critical point (31.06°C, 7.38 MPa), where the compression work is reduced compared with that of the ideal gas compression because the gas properties approach liquid properties as a result of the exertion of molecular force. Thereby, the S-CO₂ cycle can achieve higher cycle thermal efficiency than the ideal gas Brayton cycle¹⁾. However, a shortcoming is that the recuperator performance is lower because of the difference of specific heat between the high-temperature side from the turbine to the precooler and the low-temperature side from the compressor to the heat source. To overcome this problem, a bypass method was

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proposed by Feher in 1967²⁾ and Angelino in 1968³⁾. In this cycle, the CO₂ gas at the inlet of precooler is divided into two streams; simultaneously, the recuperator is divided into two recuperator sections 1 and 2 (RHX1 and RHX2). One stream is to the main compressor (MC). The other is directed to the intermediate point of high-pressure side of the recuperator via a bypass compressor (BC) (Angelino designates it as a recompression compressor). Consequently, the values of heat capacity, which is the product of the mass flow rate and the specific heat, in the low pressure-high-temperature side and in the high pressure-low-temperature side can be matched. As in the usual Brayton cycle, intercooling in the compression process is also effective for additional reduction of compression work, where an intercooler is provided and the main compressor (MC) is divided into a low-pressure compressor (LPC) and a high-pressure compressor (HPC).

Although no research work was conducted after Angelino, in 2001, studies of the cycle were re-started in Tokyo Institute of Technology⁴⁾ in Japan and in MIT⁵⁾ in the USA. After a few years, because of inert characteristics of CO₂ compared to those of water and steam, great attention was devoted to the application of S-CO₂ to the power generation system of the liquid-metal-cooled fast reactor^{6), 7), 8)}; research programs were started with international collaboration⁹⁾. Some cycle characteristics of this cycle have been examined and reported to date. Nevertheless, the achievable cycle thermal efficiency has not been clarified sufficiently because of complicated performance that is dependent on temperature, pressure, the cycle scheme, and component performance.

As described in this paper, cycle characteristics are shown to be dependent on these factors, particularly for recuperator performance.

a) Non-intercooling cycle
 b) Intercooling cycle
 Fig. 1. Flow diagrams of non-intercooling and intercooling S-CO₂ gas turbine cycle.

2. Study of the cycle thermal efficiency

2.1 Calculation assumptions and calculation method

Cycle calculations are conducted both for non-intercooling and intercooling cycles. The heat source outlet temperature, i.e., turbine inlet temperature, is parametrically varied from 500°C to 800°C, every 50°C. The cycle lowest temperature (precooler outlet temperature and the main compressor inlet temperature) is 35°C. This value is appropriate both for seawater cooling and the air cooling considering that the temperature difference from the seawater in the neighborhood of Japan and that from the atmospheric temperature. The turbine inlet pressure is 20 MPa, which is a desirable pressure in this cycle. The effects of pressure are examined later. The values of the pressure loss ratio (pressure loss divided by inlet pressure) of the components including the connected piping are assumed as follows.

Heat source	2.0%
High-temperature side of the recuperator	1.2%
High pressure side of the recuperator	0.4%
Precooler	1.0%
Intercooler	0.8%
Adiabatic efficiencies of the turbomachine are as follows.	
Turbine	92%
Main and bypass compressor	88%

The values of adiabatic efficiency are determined based on the conceptual design¹⁰⁾ of the gas turbine of the 1,500MWe commercial power plant for the Na-cooled fast reactor. The plant capacity is too large to design cooling components and then, a cooling scheme of two loops was assumed. In this case, the power output of one loop is 750MWe. In the S-CO₂ gas turbine cycle, a volume flow rate of the turbine becomes very large compared with those of the compressors. Therefore, a double flow type machine was selected for the turbine. The results of the design are shown in Table 1. The adiabatic efficiency of the turbine is 92%. As for the compressors, those are 89% for the main compressor and 87% for the bypass compressor, respectively. Therefore, the mean value of 88% was assumed.

Table 1 Results of commercial S-CO₂ gas turbine conceptual design¹⁰⁾

Design items	Turbine	Main compressor	Bypass compressor
Type	Axial, double flow	Axial	Axial
Speed	1,500 rpm	1,500 rpm	1,500 rpm
Mass flow rate	4964 kg/s × 2	5849 kg/s	4034 kg/s
Inlet temperature	527°C	35°C	78°C
Inlet pressure	20 MPa	8.26 MPa	8.34 MPa
Outlet pressure	8.55 MPa	20.57 MPa	20.49 MPa
Number of stages	6 / flow	14	16
Power	1,072 MW	130 MW	232 MW
Adiabatic efficiency	92 %	89 %	87 %

At the point at which the outlet stream from the bypass compressor merges with the outlet stream from the high-pressure side of recuperator 2, the pressure values of both streams are adjusted automatically. Although the temperature values of both streams do not necessarily coincide, identical temperature was assumed to be achieved in the design point.

The assumed recuperator performance is controversial and has a large effect on the cycle thermal efficiency. In many cases, the temperature effectiveness is used, which is defined as follows.

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

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

In those equations,

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

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

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

$T_{H,out}$ = 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 identical. In the case of S-CO₂, however, these values differ considerably. Therefore, in this paper, the next average value $\eta_{RHX,av}$ of the high-temperature and low-temperature side is adopted for the calculation base.

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

The value of 91% is assumed, as determined from 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, and 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 turbine pressure ratio, i.e., the turbine inlet pressure versus turbine outlet pressure is varied parametrically.

Under the assumptions described above, the cycle calculation can be executed for the non-intercooling cycle. For the intercooling cycle, one more conditions, i.e., a value of outlet pressure of the low-pressure compressor (LPC) must be given. In gas turbine cycle of the ideal gas, the optimal value can be determined as the geometrical average between the low-pressure compressor inlet pressure and the high-pressure compressor outlet pressure. However, in the case of S-CO₂, the optimum value becomes different from case to case. Therefore, calculations were conducted by varying the LPC outlet pressure in each case. The condition achieving the maximum cycle thermal efficiency was determined.

Heat source capacity has no relation with the cycle thermal efficiency in the assumptions described above.

The gas properties of CO₂ such as enthalpy and entropy are based on a computer dataset: "PROPATH"¹¹⁾.

2.2 Characteristics of the non-intercooling cycle

Figure 2 shows the cycle thermal efficiency dependent on the turbine inlet temperature and turbine pressure ratio.

As shown in the curves for 500–600°C, the value of cycle thermal efficiency first increases and then reaches the maximum. Then it decreases with increased turbine pressure

Fig. 2. Cycle thermal efficiency dependent on the temperature and pressure ratio.

ratio. At 650–800°C, the area including the maximum was excluded because of the limit for the recuperator end temperature difference of less than 5°C. Regarding the curve appearance, the efficiency reduction in the small pressure ratio area is attributable to the effects of pressure loss. Efficiency reduction in the higher pressure ratio area results from the difference between turbine work and compressor work, which becomes smaller because the turbine outlet temperature gradually approaches the compressor outlet temperature. Although this is a general characteristic of the closed gas turbine cycle, in cases of S-CO₂ gas turbine, the curve shape is unique. The curve decreases more sharply immediately after the maximum point than the usual gas turbine cycle. This phenomenon is related to a feature: the gas turbine cycle working area is extremely close to the saturation line. As an example, at the turbine inlet temperature of 500°C, the diagram of pressure ratio 2.32 which

Fig. 3. Temperature-entropy diagrams for the turbine pressure ratios 2.32 and 2.50 in the non-intercooling cycle.

corresponds to the maximum cycle thermal efficiency and that of pressure ratio 2.50 are depicted in Fig. 3. In that calculation, the high pressure value of the cycle is given initially. Then the low pressure value is determined by the pressure ratio. When the pressure ratio is as small as 2.32, the compressor locates at the liquid-like supercritical region of the left-hand of the pseudo-critical line immediately above the liquid-side saturation line. When the pressure ratio increases slightly to 2.50, then the compressor inlet pressure is reduced. However,

the compressor inlet position cannot enter the gas-liquid mixture area. Therefore, it translates largely to the gas-like supercritical region of the right-hand side of the pseudo-critical line. Consequently, the temperature rise (and the enthalpy rise) of the compressor increases to a great degree. Therefore, the value of the cycle thermal efficiency decreases sharply.

The values of the maximum cycle thermal efficiency and the corresponding turbine pressure ratio versus turbine inlet temperature are presented in Fig. 4. The curves change their slope at 600°C because the maximum point is excluded, exceeding this temperature as presented in Fig. 2.

The values of the recuperator temperature effectiveness for the high-temperature side and for the low-temperature side are presented in Fig. 5. As the figure shows in the RHX1, the values of high-temperature side-temperature effectiveness are always greater than those of the low-temperature side-temperature effectiveness. On the other hand, in the RHX2, those values

Fig. 4. Maximum cycle thermal efficiency and relevant turbine pressure ratio dependent on the turbine inlet temperature for the non-intercooling cycle.

Fig. 5. Recuperator temperature effectiveness dependent on turbine-inlet temperature for the high and low temperature sides.

cross over and reversed at 550°C. The values of the end temperature difference are depicted in Fig. 6. The temperature difference at the bottom decreases with the temperature increase. It reaches the limit of 5°C at temperatures higher than 650°C. The value of temperature difference at the middle end lessens with the temperature decrease and becomes less than 5°C for temperatures lower than 500°C. Therefore, for the area less than 500°C, reduction of the value of the average recuperator temperature effectiveness of 91% is necessary.

Figure 7 shows the mass and heat balance for the non-intercooling cycle at 527°C. This temperature is appropriate for the application to the Na-cooled fast reactor. The cycle thermal efficiency of 43.3% can be achieved, which is slightly higher than the value of the steam turbine cycle. The capacities of components are those for the 1,000 MWe power (for 100% generator efficiency). In this cycle, the capacity of the recuperator is very large. It is roughly twice that of the heat source thermal power.

Fig. 6. Hot and cold leg temperature difference dependent on turbine inlet temperature for non-intercooling cycle.

Fig. 7. Mass and heat balance of 527°C non-intercooling S-CO₂ gas turbine cycle.
Cycle thermal efficiency = 43.4%

2.3 Characteristics of the intercooling cycle

The cycle thermal efficiency for the intercooling cycle is shown as dependent on the turbine inlet temperature and the turbine pressure ratio in Fig. 8. Compared with the characteristics for the non-intercooling cycle, two remarkable differences are apparent. First, the effect of turbine pressure ratio on the cycle thermal efficiency is smaller and second, the

maximum cycle thermal efficiency point translates to a larger pressure ratio with the temperature increase. The values of the cycle thermal efficiency are almost independent of the turbine pressure ratio for the broad area of 700–800°C.

To explain such dull dependence of the cycle thermal efficiency on turbine pressure ratio in the intercooling cycle, the temperature–entropy diagram for the pressure ratio of 2.40, where the maximum cycle thermal efficiency is achieved, and that for the lower pressure ratio of 2.50 is shown in Fig. 9 for 500°C.

In this case, the inlet pressure of the LPC decreases considerably with the increase of the turbine pressure ratio. Then, the LPC, of which the capacity is small, translates to the gas-like regime. However, the position of HPC locates still at the liquid-like regime of the left-hand side of the pseudo-critical line. Therefore, the temperature rise (and enthalpy rise) at the HPC is still small. The sharp reduction of the thermal efficiency at the higher pressure ratio immediately after the thermal efficiency maximum does not occur, which differs from the behavior observed in the case of the non-intercooling cycle.

Fig. 8. Cycle thermal efficiency dependent on temperature and pressure ratio.

Fig. 9. Temperature–entropy diagrams for the turbine pressure ratio of 2.40 and 2.50 in the intercooling cycle.

The value of the maximum cycle thermal efficiency and the relevant turbine pressure ratio versus temperature is presented in Fig. 10. The values of the pressure ratio and also the cycle thermal efficiency increase almost linearly. For every increase of 100°C of turbine inlet temperature, the maximum cycle thermal efficiency increases 4.5% at 500°C and 3% at 700°C, respectively.

Values of the high-temperature-side and the low-temperature-side temperature effectiveness for recuperators are presented in Fig. 11. In RHX1, the value of the high-temperature side becomes greater than that of the low-temperature side, irrespective of temperature. However, for RHX2, the curves cross and reverse at 600°C and the value of the low-temperature side is lower than that of the high-temperature side for the low turbine-inlet-temperature area. The value of the high-temperature-side temperature effectiveness at 500°C is very large, i.e., 98% in RHX1.

The temperature difference between recuperator ends is presented in Fig. 12. The temperature difference at the top ends is very large, but the temperature difference at the middle and the bottom are small. The temperature difference at the middle

Fig. 10. Maximum cycle thermal efficiency and relevant turbine pressure ratio.

Fig. 11. Recuperator temperature effectiveness dependent on turbine inlet temperature at high- and low-temperature sides.

Fig. 12. Hot and cold leg temperature differences of RHX1 and RHX2.

gradually decreases along with the temperature decrease. Then, at 500°C it becomes 5°C of the limit. This is a similar inclination with the non-intercooling cycle.

The typical mass and heat balance of 650°C intercooling cycle is presented in Fig. 13. Very high cycle thermal efficiency of 48.9% can be achieved.

Fig. 13. Mass and heat balance of 650°C intercooling supercritical CO₂ gas turbine system.
Cycle Thermal Efficiency = 48.9%

2.4 Comparison of the cycle thermal efficiency between non-intercooling cycle and intercooling cycle

In this section, both the data of the maximum cycle thermal efficiency and a heat source inlet temperature are presented for comparison in Fig. 14 and Fig. 15, respectively, for the non-intercooling and the intercooling under the conditions of 20 MPa and average recuperator effectiveness of 91%. Although the difference between the non-intercooling and the intercooling is extremely small up to 600°C, a considerable difference exists and then magnifies at higher temperatures. For example, a difference of 2.4% prevails at 700°C. To achieve identical efficiency in the Non-IC cycle, a turbine inlet temperature of

Fig. 14. Maximum cycle thermal efficiency versus turbine inlet temperature.

800°C is necessary. Therefore, this difference is probably quite large. From this figure, the following can be said. At 527°C for the application to fast reactor, because of its simple scheme, the non-intercooling cycle is superior to the intercooling cycle. However, at 650°C for the application to solar thermal power generation, the intercooling cycle is superior to the non-intercooling cycle, even if the intercooling cycle engenders a more complicated

system. Regarding the heat source inlet temperature, they are 487°C and 468°C for the non-intercooling and the intercooling cycles, respectively, and their difference is about 20°C at 650°C. Although this value is apparently small, it is important for the strength design because the strength of structural material has strong temperature dependence at temperatures higher than 500°C. In addition, the temperature increases of the heat source, although the values are not shown in the figure, are, respectively, 163°C and 183°C for the non-intercooling cycle and the intercooling cycle: the difference is about 10%. This temperature difference reflects the difference of the mass flow rate. Then the mass flow rate for the non-intercooling cycle becomes 10% greater than that for the intercooling cycle. The difference of the mass flow rate influences the turbomachinery design.

Fig. 15. Heat source inlet CO₂ temperature at the maximum cycle thermal efficiency versus turbine inlet temperature.

3. Effects of pressure

In this section, the turbine inlet pressure dependence of the cycle thermal efficiency is examined for the two typical heat source outlet temperature (= turbine inlet temperature); 527°C and 650°C.

Figure 16 depicts the case for 527°C, where the values of the cycle thermal efficiency denote the maximum obtained by varying a turbine pressure ratio and a LPC pressure ratio. It can be confirmed that in the case of the non-intercooling cycle, the cycle thermal efficiency increases first up to around 18 MPa and saturates at 23 MPa for the compression work for the constant pressure ratio. For the intercooling cycle, saturation is achieved at 22 MPa. Observing the low-pressure area less than 18 MPa, the cycle thermal efficiency of the non-intercooling cycle is considerably lower than that of the intercooling cycle because a compressor inlet point cannot enter the gas–liquid mixture regime and translates to the gas-like regime. In addition, a

Fig. 16. Effect of pressure on the cycle thermal efficiency for 527°C.

recuperator end temperature difference becomes smaller and ultimately less than 5°C with the pressure-ratio reduction. For pressure of 18 MPa to 23 MPa, the difference becomes small, although the value of the intercooling cycle is higher than that of the non-intercooling cycle. In previous sections, the value of the turbine inlet pressure was assumed as 20 MPa, which can be expected to be an appropriate value if one considers that the design becomes difficult at higher pressures and also that the efficiency difference between 20 MPa and the higher pressure is as small as 0.5%.

Figure 17 shows the case for 650°C . Compared with that of 527°C , the curves of the cycle thermal efficiency do not saturate. They continue to increase up to 26 MPa, although the general inclination is similar. In addition, the low cycle thermal efficiency regime of the non-intercooling cycle at the left-hand side is magnified up to the higher pressure.

Fig. 17 Effect of pressure on the cycle thermal efficiency for 650°C .

4. Effects of recuperator effectiveness

In the preceding sections, a constant value of recuperator effectiveness was assumed: 91%. For the analyses described in this section, the effects of variable recuperator effectiveness are examined for the constant turbine pressure ratio and the LPC pressure ratio (IC cycle), which are the values for the maximum cycle thermal efficiency for recuperator effectiveness of 91%. The turbine inlet temperatures are typically 527°C and 650°C .

Fig. 18. Effects of recuperator effectiveness on the cycle thermal efficiency at 527°C and 650°C .

The results are presented in Fig. 18. The slopes are linear and quite similar for the

non-intercooling cycle and the intercooling cycles. The slope value is 0.5%/%. Therefore, 1% improvement of recuperator effectiveness engenders 0.5% improvement of cycle thermal efficiency.

5. Conclusions

Cycle thermal efficiency for the S-CO₂ gas turbine cycle was examined for both the non-intercooling and the intercooling cycles with great attention to recuperator performance. Values of the achievable cycle thermal efficiency were calculated by assuming the value of average recuperator temperature effectiveness of 91%. The turbine inlet pressure is 20 MPa. The turbine pressure ratio and turbine inlet temperature varied parametrically. In addition, effects of pressure and recuperator effectiveness were examined. The following conclusions were obtained.

- (1) In the non-intercooling cycle, the maximum efficiency occurs in the area of small turbine pressure ratio. After the maximum, the cycle thermal efficiency is reduced sharply for temperatures below 650°C because a compressor inlet point cannot enter the gas-liquid mixture regime and translates to the gas side. In the intercooling cycle, the effect of pressure ratio on the cycle thermal efficiency is very small.
- (2) The values of the cycle thermal efficiency for the intercooling cycle are equivalent to those of the non-intercooling cycle up to 600°C. Exceeding 650°C, the former becomes considerably higher than the latter by 2%. Therefore, the simple non-intercooling cycle is expected to be appropriate below 600°C. An intercooling cycle with higher efficiency will be appropriate above 650°C.
- (3) The value of the maximum achievable cycle thermal efficiency was clarified for each turbine inlet temperature. In typical cases, it becomes 43.4% for non-intercooling at 527°C and 48.9% for the intercooling cycle at 650°C. Mass and heat balances for these cases are given.
- (4) The value of cycle thermal efficiency increases with increasing pressure at 527°C. The value is saturated at 22 MPa. At 650°C, the value continues to increase beyond 22 MPa. In consideration of the difficulty of strength design accompanied with higher pressure, 20 MPa will be reasonable.
- (5) The cycle thermal efficiency increases about 0.5% for a 1% increase of the recuperator effectiveness.

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