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Preliminary Conceptual Design of the Secondary Sodium Circuit-eliminated JSFR (Japan Sodium Fast Reactor) Adopting a Supercritical CO_2 Turbine System (2)
- Turbine System and Plant Size -

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JSFR Systems Development Planning Office
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Japan Atomic Energy Agency
日本原子力研究開発機構

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Preliminary Conceptual Design of the Secondary Sodium Circuit-eliminated JSFR (Japan Sodium Fast Reactor) Adopting a Supercritical CO_2 Turbine System (2) - Turbine System and Plant Size -

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Research and development of the supercritical CO_2 (S-CO_2) cycle turbine system is underway in various countries for further improvement of the safety and economy of sodium-cooled fast reactors. The Component Design and Balance-Of-Plant (CD&BOP) of the Generation IV International Nuclear Forum (Gen-IV) has addressed this study, and their analytical and experimental results have been discussed between the relevant countries.

JAEA, who is a member of the CD&BOP, has performed a design study of an S-CO_2 gas turbine system applied to the Japan Sodium-cooled Fast Reactor (JSFR). In this study, the S-CO_2 cycle turbine system was directly connected to the primary sodium system of the JSFR to eliminate the secondary sodium circuit, aiming for further economical improvement. This is because there is no risk of sodium-water reaction in the S-CO_2 cycle turbine system of SFRs.

This report describes the system configuration, heat/mass balance, and main components of the S-CO_2 turbine system, based on the JSFR specifications. The layout of components and piping in the reactor and turbine buildings were examined and the dimensions of the buildings were estimated.

The study has revealed that the reactor and turbine buildings could be reduced by 7% and 40%, respectively, in comparison with those in the existing JSFR design with the secondary sodium circuit employing the steam turbine. The cycle thermal was also calculated as 41.9-42.3%, which is nearly the same as that of the JSFR with the water/steam system.

Keywords : Secondary Sodium Circuit Elimination, JSFR, Supercritical CO_2 turbine System

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超臨界炭酸ガスタービンシステムを採用した 2 次系削除 JSFR（Japan Sodium Fast Reactor）の概念検討（2） －タービンシステムとプラントサイズ－

日本原子力研究開発機構
次世代原子力システム研究開発部門※
炉システム開発計画室

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ナトリウム冷却高速炉の安全性と経済性の更なる向上のために超臨界炭酸ガスサイクルタービンシステムの研究開発が世界各国で進められている。第 4 世代国際原子力システムフォーラム(Gen-IV)／機器設計及びバランス・オブ・プラント(CD&BOP)もこの研究テーマを取り扱い、これまで関係国間で議論されてきている。

原子力機構は CD&BOP のメンバーの一員でもあり、超臨界炭酸ガスタービンシステムの JSFR への適用した設計検討を行ってきた。本検討では、超臨界炭酸ガスタービンシステムを 1 次ナトリウム系に直結し、2 次系削除型 JSFR として、将来の更なる経済性向上を狙った。なぜなら、超臨界炭酸ガスタービンシステムは SFR においてはナトリウム／水反応がないからである。

本報告書は JSFR の仕様に基づき超臨界炭酸ガスタービンのシステム構成、ヒートマスバランス、主要機器の設計検討について述べる。原子炉建屋とタービン建屋内の機器・配管の配置を検討し、建屋の大きさを評価した。

従来の 2 次系を有する蒸気タービンを採用した JSFR と比較すると、原子炉建屋容積及びタービン建屋容積は、それぞれ 7%、40%削減することが明らかとなった。また、サイクル熱効率は 41.9～42.3%であり、水・蒸気システムの JSFR とほぼ同じであることわかった。

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1. Introduction

The sodium-cooled fast reactor (SFR) that is being developed by the Japan Atomic Energy Agency (JAEA) is a loop-type reactor. One of the advantages of the loop-type reactor is the possibility of eliminating the secondary sodium system. For the reason, a design study of the secondary sodium circuit-eliminated JFSR was conducted to evaluate its technical feasibility and the effect of reducing the volume as an option for JSFR systems for the future commercialization phase.

Important conditions for the elimination of the secondary sodium circuit are no risk of sodium-water reactions and greatly reduced impact on the primary sodium system and the reactor core in case of heat exchanger failure. To meet these conditions, a conceptual design study of the secondary sodium circuit-eliminated JSFR was conducted employing a supercritical CO_2 turbine system and a Na/CO_2 PCHE (printed circuit heat exchanger).

In this study, the 1500 MWe JSFR with the secondary sodium circuit and the steam turbine was used as the base reactor concept. A preliminary design study of the supercritical CO_2 gas cycle system and Na/CO_2 heat exchanger suitable for the reference JSFR was conducted. The other specifications, such as the reactor structure, 2-loop cooling system configuration, decay heat removal system, plant thermal power and the primary system heat balance, were set as the same as those of the reference JSFR.

This report describes the preliminary design of the supercritical CO_2 gas cycle (turbine) system for the JSFR. A reduction in the volume of the secondary sodium circuit-eliminated JSFR is also discussed in combination with a study on the Na/CO_2 heat exchanger.

2. Overview of secondary sodium circuit-eliminated JFSR

A supercritical CO_2 gas cycle and a PCHE-type Na/CO_2 heat exchanger were employed in the study of the secondary sodium circuit-eliminated JFSR. In addition, a SiC/SiC ceramic composite material was used for the Na/CO_2 heat exchanger (PCHE) to prevent a crack initiation and its growth. Although both the supercritical CO_2 gas turbine system and the PCHE are technologies under development, this study was conducted on the premise of the adoption of these technologies. The advantages of these technologies are described in (1) through (3) below;

2.1 Supercritical CO_2 gas turbine system

A configuration diagram of the supercritical CO_2 gas turbine system is shown in Fig. 2.1 with a conventional steam Rankin system. A high cycle thermal efficiency can be achieved because the compression of CO_2 gas in the supercritical region significantly reduces loss in the compressor. Also, the CO_2 gas turbine system is smaller and more simply configured compared with the steam turbine system, allowing for a reduction in the system dimension.

Applying the supercritical CO_2 gas cycle system to a sodium-cooled fast reactor requires Na/CO_2 heat exchangers. Even in the case that the heat exchanger fails and sodium-CO_2 reaction occurs, reaction products are Na_2CO_3 (solid) and CO; and there is no production of flammable gas (H_2) or strongly-alkaline corrosion product (NaOH) produced by the sodium-water reaction. Furthermore, the solid reaction product, Na_2CO_3, suppresses the progress of the sodium-CO_2 reaction. From these features, the sodium-CO_2 reaction is much milder than the sodium-water reaction in steam generators.

As described above, the safety of Na/CO_2 heat exchanger in the supercritical CO_2 gas turbine system can be dramatically improved compared with that of the conventional the steam turbine system.

2.2 PCHE (Printed Circuit Heat Exchanger)

A structural concept of the PCHE is shown in Fig. 2.2. This heat exchanger differs from the shell & tube heat exchanger that has been previously used in FBRs (Fig. 2.3). In a PCHE, the heating fluid (sodium) and the heated fluid (CO_2) flow in the reverse direction and their flow path are alternately arranged. The flow paths are partitioned by walls. Since the

PCHE has a large heat transfer area per unit volume, it is suitable for a gas with a low heat transfer coefficient.

The advantage of the PCHE is the capability of limiting the area affected by CO_2 leak resulting from a failure of the flow path because the flow paths are partitioned. In shell & tube heat exchangers, a reaction jet caused by one tube failure will affect many adjacent tubes, as illustrated in Fig. 2.3. However, in the PCHE, the influence of CO_2 leak is limited within one flow path, at least in the initial phase, as shown in Fig. 2.2.

Manufacturing a PCHE in a large-scale, however, is impractical. Accordingly, it is necessary to manufacture it as small-scale heat transfer modules and combine them in a heat exchanger vessel to form the flow paths.

2.3 SiC/SiC composite material for PCHE

The SiC/SiC ceramic composite material is based on silicon carbide ceramics using a silicon carbide fiber as reinforcement. Because of its pseudo-ductility, excellent high-temperature strength and low thermal expansion coefficient, this material is advantageous for a heat exchanger, in which hot-leg (sodium side) and cold-leg (CO_2 side) flow paths are incorporated together, from a structural integrity perspective.

Furthermore, it is possible to provide a crack arresting layer between the sodium and CO_2 flow paths in its manufacturing process. This would provide a mechanism to prevent through leaks.

This report describes the design of the supercritical CO_2 gas turbine system in (1) above. Other results are the estimated dimensions of the secondary sodium circuit-eliminated JSFR and the comparison of the building volume between plants with and without the secondary sodium circuit.

The conditions of the primary sodium system assumed for this study are the same as those of the reference 1500 MWe JSFR, as described below:

- Reactor thermal power: 3530 MWth
- Number of the primary sodium loops: 2
- Primary sodium flow rate: 3.24×10^7 kg/hr/loop
- Reactor outlet sodium temperature: 550°C
- Reactor inlet sodium temperature: 395°C

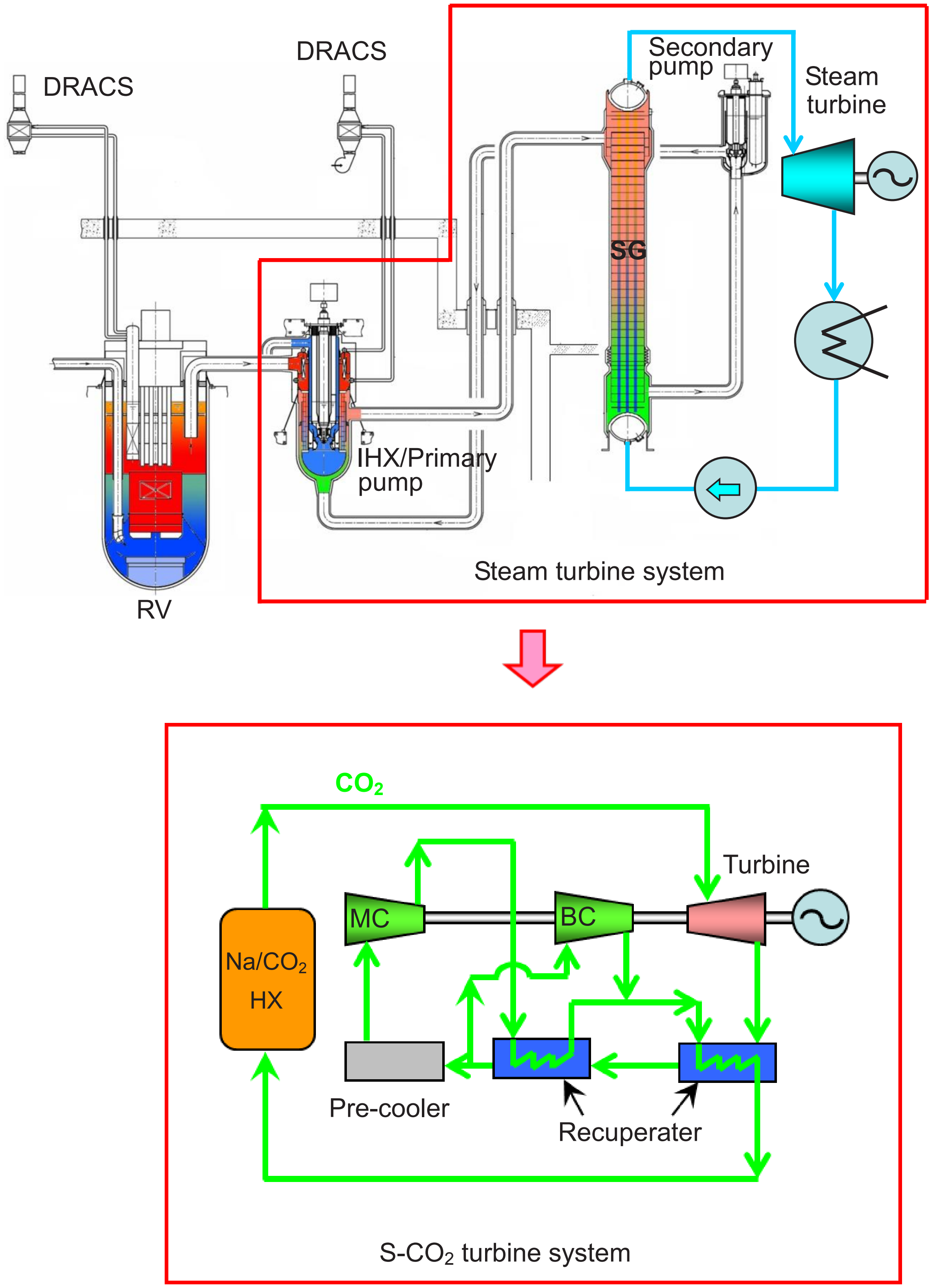


Fig. 2.1 Supercritical CO_2 turbine system for JSFR

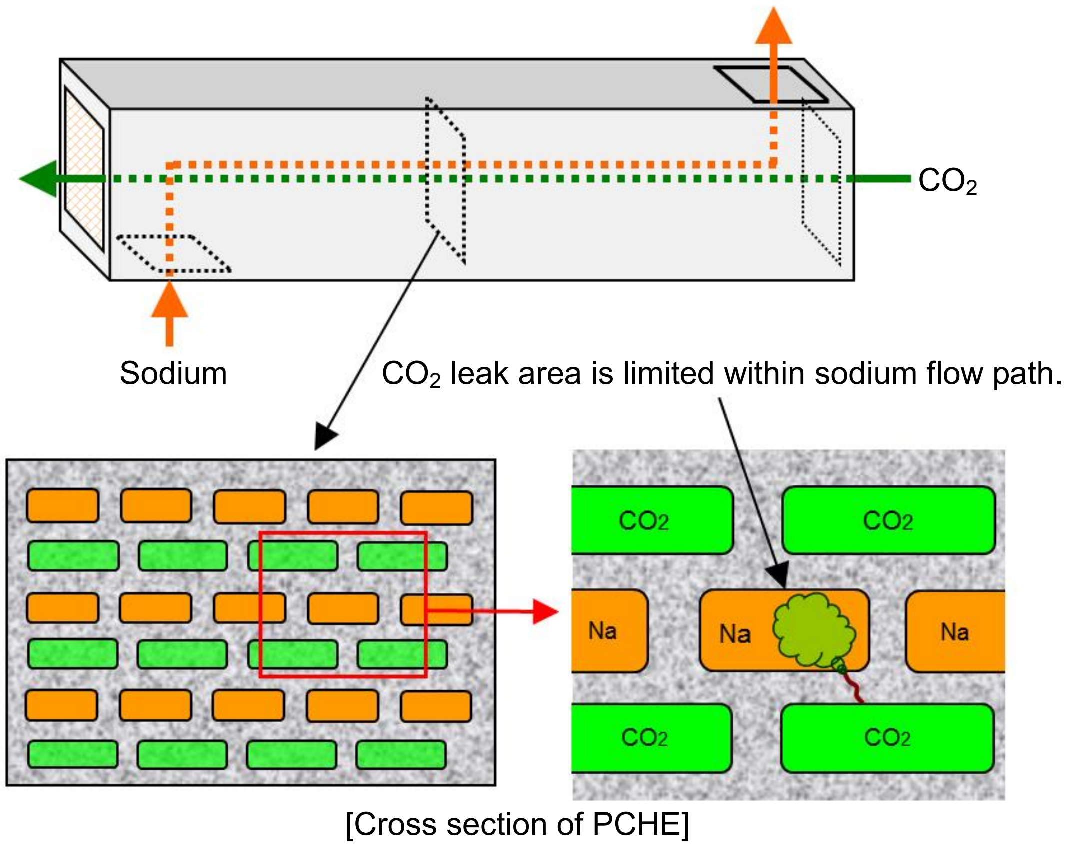


Fig. 2.2 Printed circuit heat exchanger (PCHE)

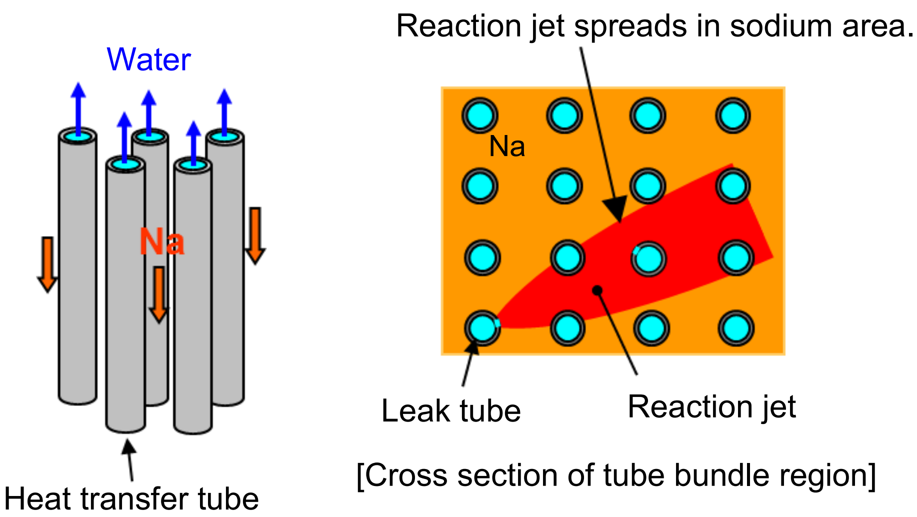


Fig. 2.3 Shell and tube type heat exchanger

3. Supercritical CO_2 turbine system

3.1 Compressor inlet CO_2 temperature

3.1.1 Basic concepts

A study on the compressor inlet CO_2 temperature that affects the cycle thermal efficiency was conducted in order to design the supercritical CO_2 turbine cycle.

The influencing factor that determines the compressor inlet CO_2 temperature is the inlet coolant (sea water) temperature at the pre-cooler installed upstream from the compressor.

In LWRs, the variation of the sea water temperature by season affects the condenser vacuum, varying the steam turbine efficiency. Accordingly, at constant reactor (LWR) power operation, a reduced sea water temperature improves the condenser vacuum, raises the steam turbine efficiency and increases its generating output.

Also in the supercritical CO_2 cycle, the design sea water and compressor inlet CO_2 temperatures needs to be determined by taking into account the effect of the sea water temperature change on the turbine power efficiency. Furthermore, when the compressor inlet temperature and pressure are equal to or lower than the critical point (7.382 MPa, 31.06°C), CO_2 liquefies, and prolonged liquefaction may lead to compressor failure. Therefore, determination of the compressor inlet CO_2 temperature must also take control/protection into consideration to avoid the critical point.

This report describes a semi-quantitative study on the compressor inlet CO_2 temperature, based on the above 2 points (turbine efficiency, control/protection of the compressor) and the effect on the volume of the heat exchanger.

3.1.2 Design sea water temperature

(1) Prerequisites

Although the site for the reference JSFR (with the secondary sodium circuit and steam turbine) has not yet been decided, the design condenser sea water temperature is assumed to be 23°C. The averaged values of the highest and lowest sea water temperatures at Japan's LWR sites are as follows:

Monthly-averaged highest sea water temperature	23.9°C≒24°C
Monthly-averaged sea water temperature	16.4°C≒16°C
Monthly-averaged lowest sea water temperature	10.4°C≒10°C
Sea water temperature with a 75% incidence	20°C

As described above, LWRs can be operated at a constant power level even when the sea water temperature varies, however, the turbine efficiency varies due to the change in condenser vacuum. Therefore, the effect of sea water temperature on the S-CO_2 turbine efficiency was examined based on the following conditions:

(a) Reactor operation at a constant power
(b) Reactor outlet/inlet temperatures 550°C / 395°C (ΔT:155°C)
(c) Na/CO_2 heat exchanger CO_2 outlet/inlet temperatures
527°C / 388°C (ΔT:139°C)
(d) Compressor inlet CO_2 temperature 35°C
(e) Sea water temperature

Monthly-averaged highest sea water temperature	24°C
Annual average sea water temperature	16°C (Ref.)
Monthly-averaged lowest sea water temperature	10°C

(f) The compressor inlet CO_2 temperature is assumed to be determined with sea water temperature as follows:

Sea water temperature	Compressor inlet CO_2 temperature
24°C	43°C
16°C	35°C
10°C	29°C

(2) In the case of a decreased sea water temperature (16°C --> 10°C)

A decrease in sea water temperature lowers the turbine system temperature and consequently the pressure, however, the turbine inlet pressure is maintained at a constant by the pressure control system. The compressor inlet CO_2 temperature decreases from 35°C to 29°C, and the compressor flow rate changes to (273+35)/(273+29)×100 = 102.0%, i.e., it increases by 2%. Since the Na/CO_2 heat exchanger (HX) inlet CO_2 temperature falls down with the decrease in the compressor inlet CO_2 temperature (6°C), it changes from 388°C to 382°C (=388-6). The Na/CO_2 HX outlet CO_2 temperature falls down with the decrease in the outlet/inlet temperature difference due to the increase in CO_2 flow rate and the decrease in the inlet CO_2 temperature. And then it changes from 527°C to 518°C (= 527°C - (139°C × 2% + 6°C)).

Even if the sea water temperature declines, the reactor outlet/inlet temperature difference remains unchanged (155°C). Because the reactor power and the sodium flow rate are maintained to be constant. Since the heat exchange capacity of the Na/CO_2 heat exchanger is not changed, the logarithmic mean temperature difference (LMTD) also stays constant. The reactor outlet sodium temperature in the case of decreased sea water temperature transitions to a new level that meets the above conditions.

The LMTD, ΔTm, of the Na/CO_2 heat exchanger during the rated operation is as follows:

$$\Delta Tm = \frac{(550-527)-(395-388)}{\ln\left(\dfrac{550-527}{395-388}\right)} = 13.45℃$$

Reactor Na outlet/inlet temperatures 550°C / 395°C [rated operation]

Na/CO_2 HX CO_2 outlet/inlet temperatures 527°C / 388°C [rated operation]

A new reactor inlet sodium temperature (Na/CO_2 HX sodium outlet temperature) is set so that the LMTD is maintained at the same value during the rated operation (13.45°C) even when the sea water temperature decreases to 10°C. From the previous calculation, the Na/CO_2 HX CO_2 -side temperatures are:

・Inlet CO_2 temperature:382°C [In the case of the decreased sea water temperature]

・Outlet CO_2 temperature:518°C [In the case of the decreased sea water temperature]

A new reactor outlet sodium temperature (Na/CO_2 HX sodium inlet temperature) is (T_1+155)°C and the LMTD is the same as before, and accordingly, the following equation holds:

$$\Delta Tm = \frac{(T_1+155-518)-(T_1-382)}{\ln\left(\dfrac{T_1+155-518}{T_1-382}\right)} = \frac{19}{\ln\left(\dfrac{T_1-363}{T_1-382}\right)} = 13.45℃$$

From the above, the reactor inlet/outlet sodium temperatures in the case of decreased sea water temperature have been obtained as follows:

- Reactor inlet sodium temperature: T_1 388°C
- Reactor outlet sodium temperature : 388+155=543°C

The operational conditions in the case of decreased sea water temperature to 10 °C are summarized below:

- Reactor power 100% (constant)
- Primary sodium flow rate 100% (constant)
- Reactor outlet/inlet temperature difference 155°C (constant)
- LMTD 13.45 °C (constant)
- Reactor outlet/inlet sodium temperatures 543°C /388°C
- Na/CO_2 HX CO_2 outlet/inlet temperatures 518°C /382°C
- CO_2 flow rate 102%

The sea water temperature decrease leads to the power decline of the supercritical CO_2 turbine due to the following reasons:

- Decrease in efficiency due to a (2%) deviation from the rated point of turbine and compressor
- Decrease in turbine power due to a decrease in the turbine inlet temperature (527°C --> 518°C)

To prevent the power decrease due to decreased sea water temperature, the coolant flow rate in the pre-cooler needs to be declined so that the compressor inlet CO_2 temperature is maintained at 35°C.

(3) In the case of an increased sea water temperature (16°C --> 24°C)

An increase in sea water temperature raises the turbine system temperature and pressure; however, the turbine inlet pressure is maintained at a constant by the pressure control system.

The compressor inlet CO_2 temperature increases from 35°C to 43°C on the basis of the above assumption, and the compressor outlet mass flow rate changes to (273+35)/(273+43)×100 = 97.5%, i.e., decreases by 2.5%. Since the Na/CO_2 HX inlet CO_2 temperature goes up with the increase in the compressor inlet CO_2 temperature (8°C), it changes from 388°C to 396°C (=388+8). The Na/CO_2 HX outlet CO_2 temperature changes from 527°C to

538°C by the increase in outlet/inlet temperature difference due to the CO_2 flow rate decrease and the inlet temperature increase (=527°C+(139°C×2.5%+8°C)).

The reactor power and the sodium flow rate are maintained at a constant regardless of sea water temperature increasing. The reactor outlet/inlet temperatures are, therefore, calculated under the assumption that the reactor outlet/inlet temperature difference is kept to be constant (155°C). In this case, the heat exchange capacity of the Na/CO_2 heat exchanger and the LMTD stay unchanged. The reactor outlet sodium temperature in the case of an increased sea water temperature transitions to a new level that meets the above conditions.

The LMTD at the rated power operation is 13.45°C. The reactor inlet sodium temperature (Na/CO_2 HX outlet sodium temperature) is set so that the LMTD is maintained at the same value (13.45°C) even when the sea water temperature increases.

- Inlet CO_2 temperature: 396°C
- Outlet CO_2 temperature: 538°C
- Reactor inlet sodium temperature
  (Na/CO_2 HX outlet sodium temperature) : T_1°C
- Reactor outlet sodium temperature
  (Na/CO_2 HX inlet sodium temperature) : T_1+155°C

$$\Delta Tm = \frac{(T_1 + 155 - 538) - (T_1 - 396)}{\ln\left(\dfrac{T_1 + 155 - 538}{T_1 - 396}\right)} = \frac{13}{\ln\left(\dfrac{T_1 - 383}{T_1 - 396}\right)} = 13.45$$

From the above, the reactor inlet/outlet sodium temperatures are obtained as follows:

- reactor inlet sodium temperature T_1: 404°C, and
- reactor outlet sodium temperature: 404 + 155 = 559°C.

Therefore, the operational conditions in the case of increased sea water temperature are summarized below:

- Reactor power 100% (constant)
- Primary sodium flow rate 100% (constant)

- Reactor outlet/inlet temperature difference 155°C (constant)
- LMTD 13.45 °C (constant)
- Reactor outlet/inlet sodium temperatures 559°C/404°C
- Outlet/inlet CO_2 temperatures 538°C / 396°C
- CO_2 flow rate 97.5%

The above operational conditions are beyond the reactor operation limit because the reactor outlet temperature (559°C) is 9°C higher than the rated operation level (550°C). Therefore, the reactor power requires to be lowered to maintain the reactor outlet temperature at 550°C. It is obtained by the following equations.

$$\frac{Q}{100} = \frac{550 - T_1}{550 - 395} \quad \cdots\cdots\cdots\cdots\cdots (1)$$

$$T_2 = 396 + \frac{139}{0.975} \times \frac{Q}{100} \quad \cdots\cdots\cdots\cdots\cdots (2)$$

$$\Delta Tm = \frac{(550 - T_2) - (T_1 - 396)}{\ln\left(\dfrac{550 - T_2}{T_1 - 396}\right)} = 13.45 \quad \cdots\cdots (3)$$

[Initial conditions]

Reactor inlet temperature: 395°C (the rated operation)
Reactor outlet temperature: 550°C (the rated operation, constant)
Reactor outlet/inlet temperature difference
:139°C (the rated operation)
CO_2 flow rate: 0.975 (relative value to that at 16°C)
Na/CO_2 HX inlet CO_2 temperature: 396°C (constant)

Q: decreased reactor power (%)
T_1: reactor inlet temperature (°C)
T_2: Na/CO_2 HX outlet CO_2 temperature (°C)

From equations (1)-(3), Q=94%, T_1=404°C and T_2=530°C.
The above operation conditions are summarized as follows:

Reactor power 94%
Primary coolant flow rate 100% (constant)

Reactor outlet/inlet temperatures	550°C/404°C
Na/CO_2 HX outlet/inlet CO_2 temperatures	530°C/396°C
CO_2 flow rate	97.5%

The causes for the decrease in supercritical CO_2 turbine power are as follows:

- Decrease in reactor power (100% --> 94%)
- Increase in compressor input power due to the compressor temperature rise (35°C --> 43°C)

The turbine power is significantly decreased due not only to the decrease in turbine system efficiency but also the decrease in reactor power attributed to the reactor operation limit.

(4) Design sea water temperature

The following 4 options were candidates for the design sea water temperature.

- Monthly-averaged lowest sea water temperature 10°C
- Monthly-averaged sea water temperature 16°C
- Sea water temperature with a 75% incidence 20°C
- Monthly-averaged highest sea water temperature 24°C

When the sea water temperature decreases below the design value, decreasing the coolant flow rate in the pre-cooler keeps the compressor inlet CO_2 temperature at 35°C to maintain the power at a constant.

On the other hand, in the case that the sea water temperature increases above the design value, the compressor inlet CO_2 temperature rises and the power falls down. To prevent a decrease in power, the design sea water temperature was determined to be 24°C, the highest value.

3.1.3 Compressor inlet CO_2 temperature and turbine pressure ratio

(1) Control to avoid a gas-liquid mixing region and protection method

i. Basic concept

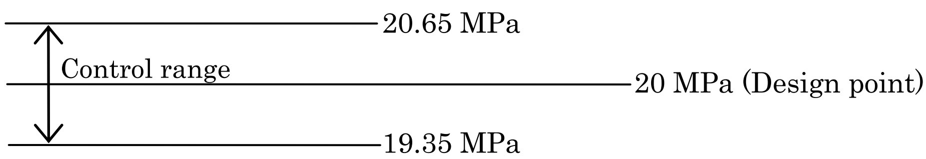
The operating condition of the compressor is based on the following concepts:

- To protect the compressor, measures shall be taken to avoid operation near the critical point (7.382 MPa, 31.06°C).
- Control shall be provided to maintain the CO_2 system pressure and the compressor inlet temperature at a constant in order to avoid immediate entry of gas-liquid mixture into the compressor even when either fails.
- The entry of gas-liquid mixture into the compressor for a short period of time is allowed because the entry of a small amount of gas-liquid mixture, such as mist, does not lead to immediate failure of the compressor.

ii. Turbine pressure control and protection

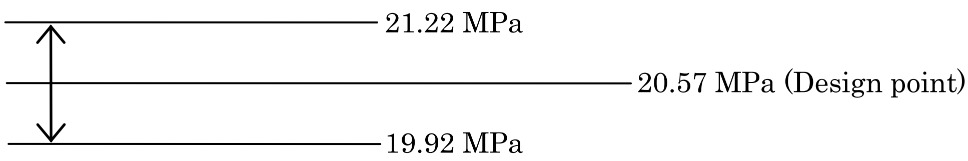
The turbine pressure shall be controlled by injection and extraction of CO_2 (on-off control) to maintain the turbine inlet pressure at a constant of approximately 20 MPa. The control width (including instrumentation error) of the pressure shall be 0.65 MPa taking into consideration of the accuracy of the control system and pressure gauge, etc.

[Turbine inlet]

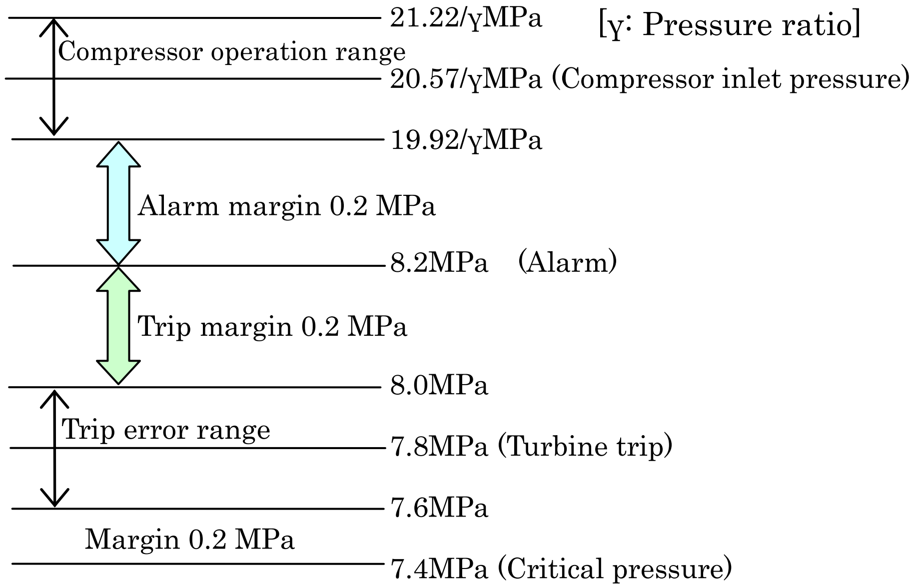


Since the compressor outlet pressure is 0.57 MPa higher than the turbine inlet pressure, its pressure control rage is below.

[Compressor outlet]



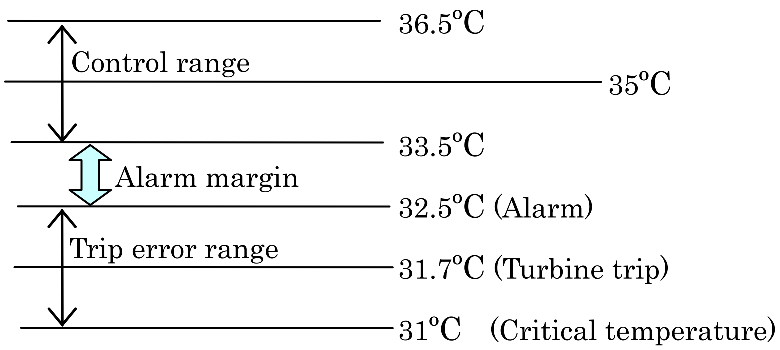
To protect the compressor, it is necessary to trip the turbine to prevent the compressor inlet pressure from decreasing to the critical pressure.



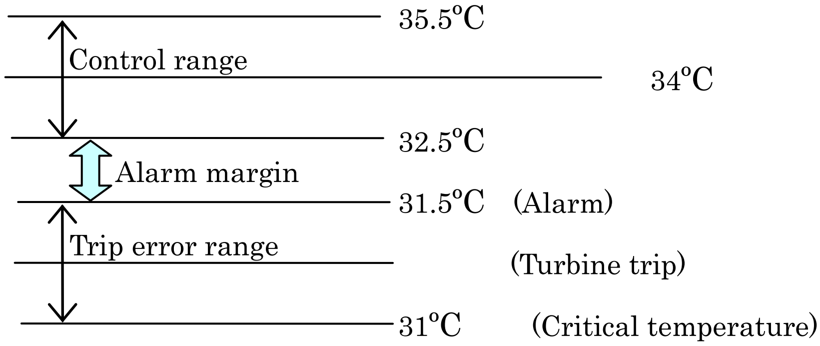
iii. Control of compressor inlet CO_2 temperature and protection

The compressor inlet CO_2 temperature is controlled by the coolant (sea water) flow rate in the pre-cooler. The alarm and turbine trip set values that are determined between the critical temperature (31°C) and operating temperature is described below. A study was conducted on setting the compressor inlet temperature at 35°C, 34°C and 33°C; however, protection using temperature signal is impractical in any case because the operating temperature is close to the compressor protection limit temperature.

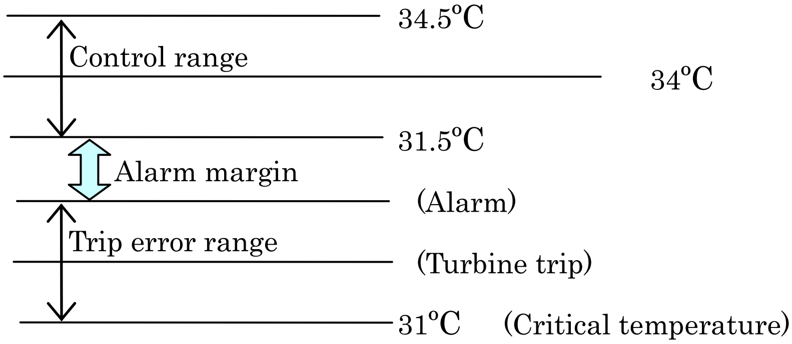
- At a compressor inlet temperature of 35°C



- At a compressor inlet temperature of 34°C



- At a compressor inlet temperature of 33°C



From the above, temperature signals is used only for alarms, and the compressor protection is provided using pressure signals that are responsive and allow for reliable trip.

As shown in the compressor protection diagram, taking the margin of 0.2 MPa between the lowest operational limit and the alarm value (8.2 MPa), the compressor's pressure ratio is calculated as follows:

$$19.92/\gamma - 0.2 = 8.2$$

$$\therefore \ \gamma = \frac{19.92}{8.4} = 2.37$$

The turbine pressure ratio corresponding to γ = 2.37 at a compressor inlet temperature of 35°C is 2.22.

(2) Compressor inlet CO_2 temperature

Based on the result of the last section, the compressor inlet CO_2 temperature is examined under the condition of a compressor pressure ratio of 2.37 (turbine pressure ratio of 2.22).

Possible compressor inlet CO_2 temperatures include 2 cases: 35°C and 34°C. It is not desirable to select temperatures of 33°C or below from the perspective of protecting the compressor. Temperatures of 36°C or above are also not adopted because of the significant reduction in the cycle thermal efficiency.

In the case of a closed-cycle gas turbine, it is desirable to maintain the compressor inlet CO_2 temperature at a constant regardless of sea water temperature change. Taking the monthly-averaged maximum sea water temperature of 24°C as the design temperature allows the compressor inlet CO_2 temperature to be a constant almost all year.

[Prerequisite condition]

	Case 1	Case 2
Compressor inlet CO_2 temp. (°C)	35	34
Turbine pressure ratio (Compressor pressure ratio)	2.22 (2.37)	2.22 (2.37)
Design seawater temp. (°C) (Appearance ratio)	24 (100%)	24 (100%)

The heat transfer area of a heat exchanger and the cycle thermal efficiency are compared between the case 1 and 2. The comparison of the heat exchanger's heat transfer area is listed in the table below using the relative value to the value in the case 1 based on the heat exchange capacity and the logarithmic mean temperature difference.

[Calculation results]

	Case 1	Case 2
Compressor inlet CO_2 temp. (°C)	35	34
Pre-cooler (-)	1	1.11
Recuperator 1 (-)	1	1.01
Recuperator 2 (-)	1	1.01
Na／CO_2 HX (-)	1	0.95
Cycle thermal efficiency (%)	43.5	43.6

The table shows the decrease in compressor inlet CO_2 temperature from 35°C to 34°C improves the cycle thermal efficiency by only 0.1%. The comparison in the heat exchanger heat transfer areas indicates insignificant difference between these cases. Therefore, 35°C, which allows for a larger margin on the critical temperature (31.06°C), is chosen as the (design) compressor inlet CO_2 temperature to protect the compressor.

(3) Compressor and turbine conditions

Based on the study in the preceding section, the conditions for the compressor and turbine were determined as follows:

- Compressor inlet CO_2 temperature: 35°C
- Compressor pressure ratio: 2.37
- Turbine pressure ratio: 2.22
- Turbine inlet pressure: 20 MPa

A conceptual design of the system heat/mass balance and component/piping is described in the next section.

3.2 Conceptual design of turbine system components and piping

3.2.1 Turbine

A basic configuration of the supercritical CO_2 turbine was studied based on the followings:

(1) The required number of turbine stages was calculated on the basis of the inlet-outlet enthalpy difference obtained from the inlet/outlet conditions.

(2) The rotor diameter and the blade outer diameter were calculated from the CO_2 volumetric flow rate at the inlet/outlet.

The specifications and external view of the turbine are shown in Table 3.2-1 and Fig. 3.2-1, respectively. The turbine adopts a four-divided flow type consisting of 2 double-flow casing. The turbine has 6 stages, a mean diameter of 1,700 mm and a rotor blade height of 150 to 250 mm.

3.2.2 Compressor

A basic configuration of the main and bypass compressors was studied based on the followings.

(1) The required number of stages was calculated based on the inlet-outlet enthalpy rise obtained from the inlet/outlet conditions.
(2) The rotor diameter and the blade outer diameter were calculated from the CO_2 volumetric flow rate at the inlet/outlet.

The specifications of the main and bypass compressors are listed in Tables 3.2-2 and 3.2-3, respectively. The external view of the main and bypass compressors is shown in Fig. 3.2-1.

The main compressor has 9 stages, a 1st-stage blade outer diameter of 1,255 mm and a rotor outer diameter of 990 mm. The bypass compressor has 12 stages, a 1st-stage blade outer diameter of 1,745 mm and a rotor outer diameter of 1,470 mm.

3.2.3 Generator

With reference to the generator planned for the JSFR, the overall dimensions of the generator were determined as follows:

6.5m wide, 15.5 m long and 5.5 m high

The external view of the generator is shown in Fig. 3.2-1. The space for the generator system in which the turbine, main and bypass compressors, and generator are connected along one axis is approximately 7 m wide and 49 m long.

3.2.4 Recuperator 1

Recuperator 1 is the first heat exchanger to which the CO_2 from the turbine flows in. It is a plate-fin type in which cold-leg (high pressure side) and hot-leg (low pressure side) flow paths are alternatively layered. The specifications and structural concept are shown in Table 3.2-4 and Fig. 3.2-2.

Recuperator 1 is divided into 2 pressure vessels. Six heat exchanger modules are installed in each of the pressure vessels, and the dimensions of the heat transfer module are 0.9m wide, 1.8 m long and 24 m high. Each module is separately produced at a 2 m elevation and connected with a short pipe. The pressure vessel is filled with CO_2 under the condition at the low pressure-side outlet (8.86 MPa, 155.2°C), and its dimensions are an inner diameter of 7 m and a height of 31 m.

3.2.5 Recuperator 2

Recuperator 2 is a heat exchanger to which the CO_2 from the turbine flows in through recuperator 1. Recuperator 2 is also a plate-fin type in which cold-leg (high pressure side) and hot-leg (low pressure side) flow paths are alternatively layered, in the same way as recuperator 1. The specifications and structural concept are shown in Table 3.2-5 and Fig. 3.2-3.

Recuperator 2 is divided into 2 pressure vessels, in the same way as recuperator 1. Six heat exchanger modules are installed in each of the pressure vessels, and the dimensions of the heat transfer module are 0.9m wide, 1.9m long and 23m high. The dimensions of the pressure vessel are an inner diameter of 7m and a height of 30m.

3.2.6 Pre-cooler

The shell & tube type is employed for the pre-cooler. To reduce the pressure loss of CO_2, CO_2 and coolant (sea water) flow paths are arranged in the shell and in the tube, respectively. The heat transfer tube is a helically-coiled type, not a straight tube type. This is because a shell-bellows is not adopted in a straight tube type HX, due to the fact that the shell-bellows is subjected to high pressure.

The specifications and structural concept of the pre-cooler are shown in Table 3.2-6 and Fig. 3.2-4.

The pre-cooler was divided into 2 pressure vessels. The dimensions of heat transfer coiling region are an innermost coil diameter of 0.7m, an outermost coil diameter of 7.5m and a height of 7.4m. The dimensions of the pressure vessel are an inner diameter of 7.7 m and a height of 1.8m.

3.2.7 Main CO_2 piping

The main CO_2 piping from Na/CO_2 heat exchanger to turbine and from recuperator 1 to Na/CO_2 heat exchanger are comprised of 4 pipes (two CO_2 pipes are arranged for one Na/CO_2 heat exchanger vessel) with a large diameter to reduce pressure loss. The estimated pipe diameters and pressure losses are listed in Table 3.2-7.

3.2.8 System configuration

Two sets of the Na/CO_2 heat exchanger, recuperator 1, recuperator 2 and pre-cooler are arranged per one reactor. The main compressor, the bypass compressor, 2 double flow turbines and the generator are connected together on one axis. The turbine system configuration diagram is shown in Fig 3.2-5.

3.3 Heat/mass balance and cycle thermal efficiency

The pressure losses of the supercritical CO_2 system components/piping are listed in Table 3.2-8. The system heat/mass balance and state transition are shown in Figs. 3.2-6 and 3.2-7, respectively. The cycle thermal efficiency is 41.9%.

Table 3.2-1 S-CO_2 turbine specification

Prerequisite condition			
CO_2 flow rate		20,245	kg/s
Inlet	Pressure	20	MPa
	Temperature	527	℃
Outlet	Pressure ratio	8.97	MPa
	Temperature	428.9	℃
Pressure ratio		2.23	---
Adiabatic efficiency		93	%

Main specification			
Rotation speed		1,500	rpm
Number of flow division		4	
Stages		6 stages	× 4
Average diameter		1,700	mm
Rotor blade height		150 ～ 250	mm
Adiabatic efficiency		92.3	%
Outlet temp.		429.0	℃
Shaft power		2,132,000	kW
Nozzle diameter	Inlet	1,200	mm
	Outlet	1,300	mm

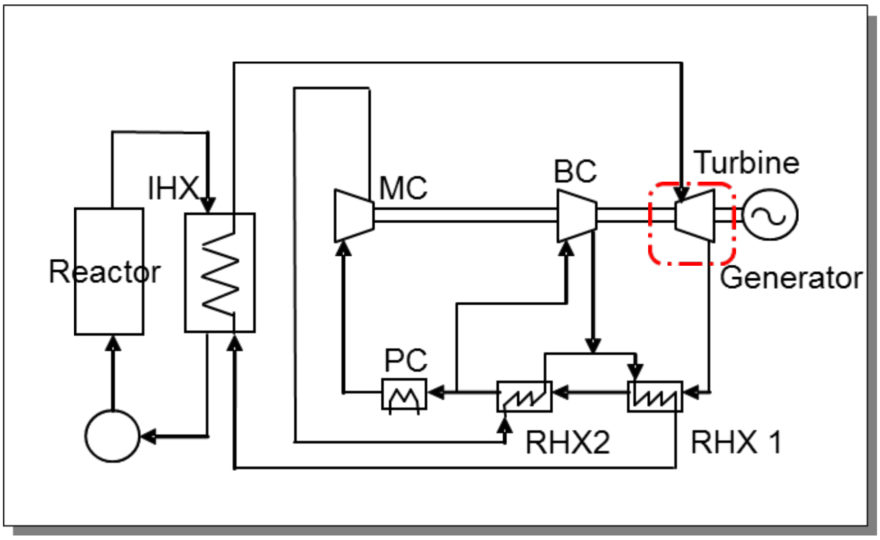


Table 3.2-2 S-CO_2 main compressor specification

Prerequisite condition			
CO_2 flow rate		12,256	kg/s
Inlet	Pressure	8.67	MPa
	Temperature	35.0	℃
Outlet	Pressure	20.6	MPa
	Temperature	60.5	℃
Pressure ratio		2.37	---
Adiabatic efficiency		88	%

Main specification			
Rotation speed		1,500	rpm
Blade Stages		9	stages
First stage Rotor blade outer diameter		1,255	mm
Rotor diameter		990	mm
Boss ratio		0.790	---
Adiabatic efficiency		88	%
First stage Rotor blade peripheral velocity		98.6	m/sec
Shaft power		244,100	kW
Nozzle diameter	Inlet	1,000	mm
	Outlet	1,000	mm

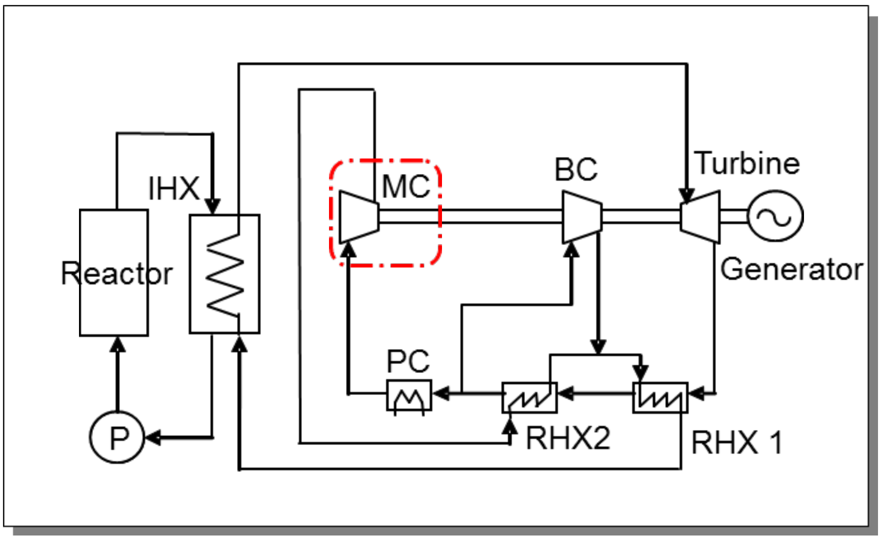


Table 3.2-3 S-CO_2 bypass compressor specification

Prerequisite condition			
CO_2 flow rate		7,988.8	kg/s
Inlet	Pressure	8.75	MPa
	Temperature	70.0	℃
Outlet	Pressure	20.5	MPa
	Temperature	147.7	℃
Pressure ratio		2.34	---
Adiabatic efficiency		88	%

Main specification			
Rotation speed		1,500	rpm
Blade Stages		12	stages
First stage Rotor blade outer diameter		1,745	mm
Rotor diameter		1,470	mm
Boss ratio		0.842	---
Adiabatic efficiency		88	%
First stage Rotor blade peripheral velocity		137.1	m/sec
Shaft power		394,200	kW
Nozzle diameter	Inlet	1,500	mm
	Outlet	1,200	mm

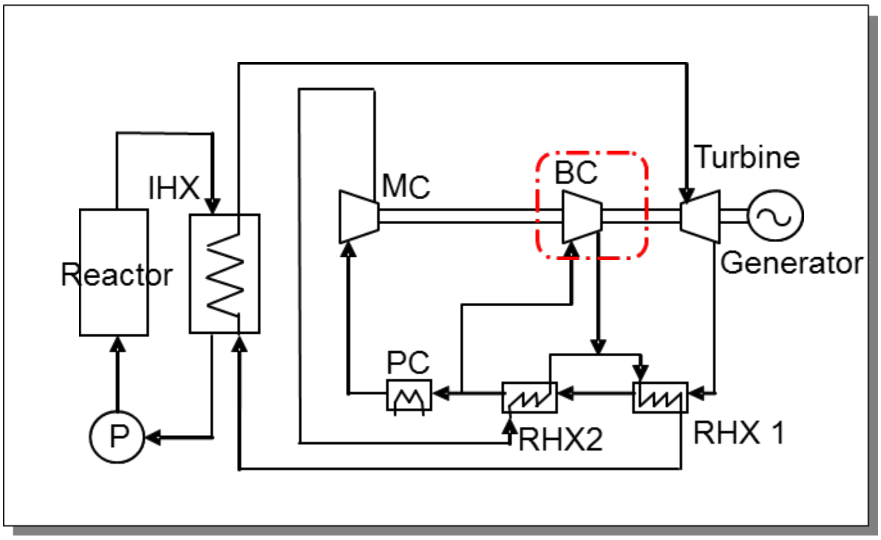


Table 3.2-4 Recuperater1 specification

Prerequisite condition		Low pressure side	High pressure side
CO_2 flow rate		20,245 kg/s	20,245 kg/s
Inlet	Pressure	8.97 MPa	20.5 MPa
	Temperature	428.9 ℃	147.7 ℃
Outlet	Temperature	155.2 ℃	385.8 ℃
Pressure loss		0.108 MPa	0.082 MPa
Heat exchanging capacity		6,418,052 kJ/s	6,418,052 kJ/s

Main specification		
Type		Plate-fin type
Unit number		2 unit
Module number per unit		6 Module
Size of heat transfer area (1 module)	Width	900 mm
	Length	1,800 mm
	Height	24,000 mm
Heat transfer area (1 unit)		154,600 m^2
Pressure loss	Low pressure side	0.100 MPa
	High pressure side	0.042 MPa
Size of recuperater vessel	Inner diameter	7,000 mm
	Height	31,000 mm

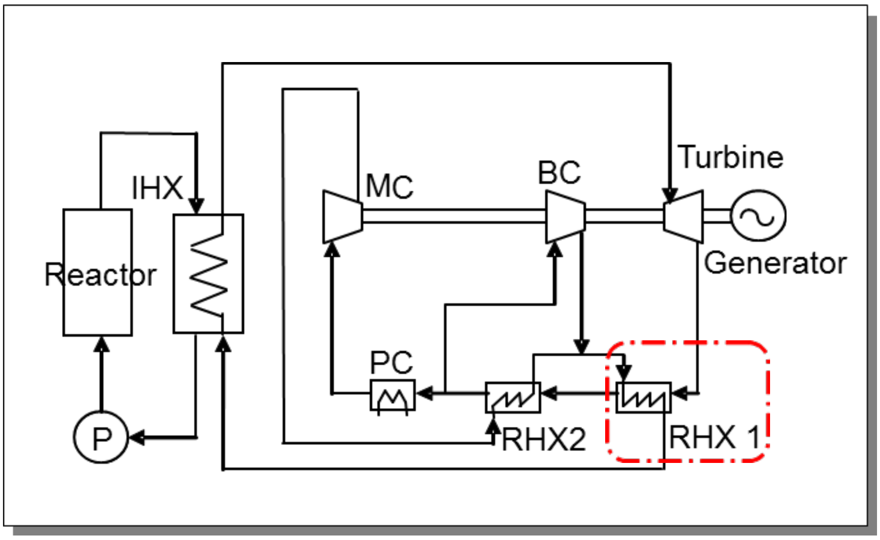


Table 3.2-5 Recuperater2 specification

Prerequisite condition		
	Low pressure side	High pressure side
CO_2 flow rate	20,245 kg/s	12,256 kg/s
Inlet Pressure	8.86 MPa	20.6 MPa
Temperature	155.2 ℃	60.5 ℃
Outlet Temperature	70 ℃	147.7 ℃
Pressure loss	0.106 MPa	0.082 MPa
Heat exchanging capacity	2,353,171 kJ/s	2,353,171 kJ/s

Main specification			
Type		Plate-fin type	
Unit number/Reactor		2	unit
Module number/unit		6	Module
Size of heat transfer area (1 module)	Width	900	mm
	Length	1,900	mm
	Height	23,000	mm
Heat transfer area (1 unit)		155,680	m^2
Pressure loss	Low pressure side	0.059	MPa
	High pressure side	0.012	MPa
Size of recuperater vessel	Inner diameter	7,000	mm
	Height	30,000	mm

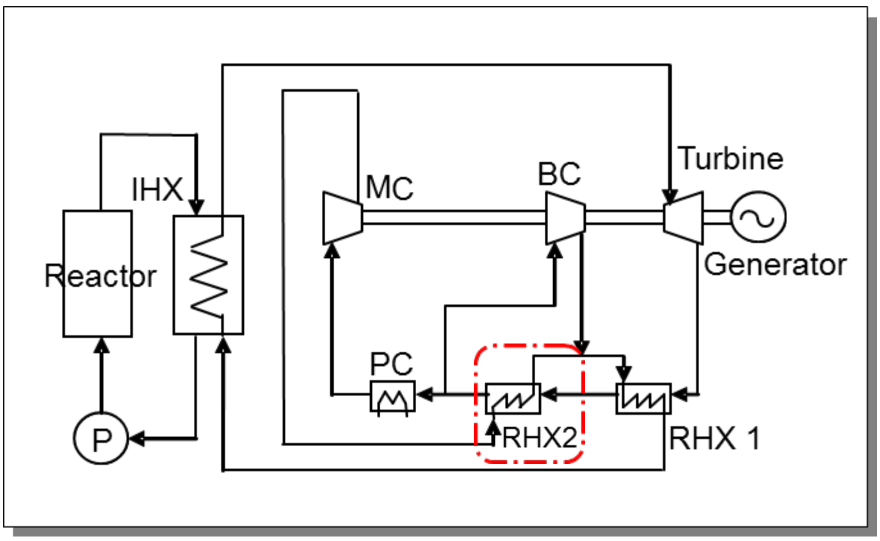


Table 3.2-6 Pre-cooler specification

Prerequisite condition				
		CO_2 side	Cooling water(Sea water)side	
CO_2 flow rate		12,256 kg/s	41,246 kg/s	
Inlet	Pressure	8.75 MPa	0.1 MPa	
	Temperature	70.0 ℃	24.0 ℃	
Outlet Temperature		35.0 ℃	36.0 ℃	
Pressure loss		0.088 MPa	— MPa	
Heat exchanging capacity		1,992,696 kJ/s	1,992,696 kJ/s	

Main specification			
Type		Shell & tube helical coil type	
Unit number/Reactor		2	unit
Tube number/Unit		24,345	tubes
Heat transfer coiling size	Inner coiling diameter	700	mm
	Outer coiling diameter	7,486	mm
	Height	7,400	mm
Heat transfer area/unit		20,224	m^2
Heat transfer tube size	Inner diameter	23.1	mm
	Thickness	2.85	mm
	Length	11.5	m
Pressure loss (CO_2 side)		0.025	MPa
Size of recuperater vessel	Inner diameter	7,700	mm
	Height	18,000	mm

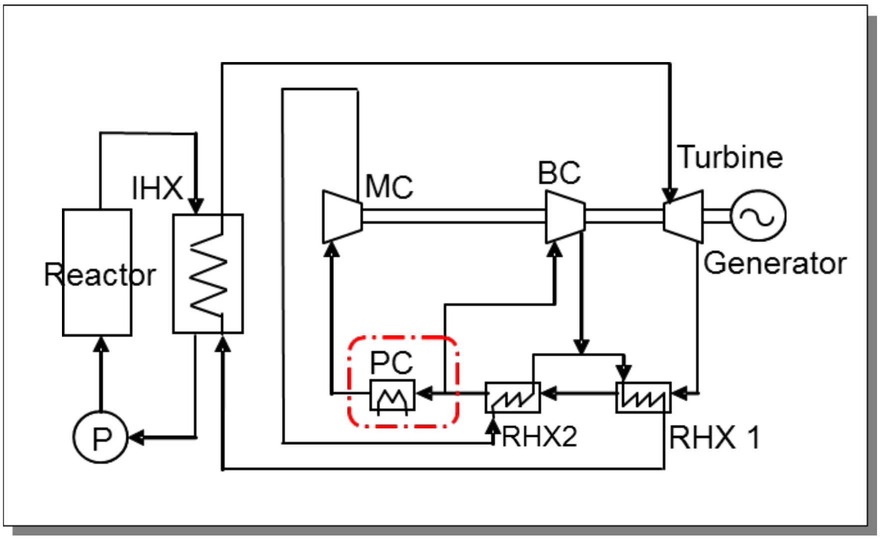


Table 3.2-7 Pressure loss of CO_2 main piping

CO_2 main piping		Piping inner diameter (m)	Pressure (MPa)	Tempe-rature (K)	Flow rate (m/s)	Pressure loss (MPa)
1	Na/CO_2 HX – Turbine	1.3×2→1.8	20	800	30/31	0.206
2	Turbine-RHX1(LP)	1.6×2→1.95	8.97	701.9	37/50	0.130
3	RHX1- RHX2(LP)	1.05×3→1.8	0.86	428.2	31/32	0.093
4	RHX2 - Pre-cooler	1.05×3→1.8 →1.6	8.86	343	31/32/25	0.114
5	Pre-cooler – Main compressor	1.6	8.67	308	4.8/9.6	0.013
6	Main compressor - RHX2(HP)	1.55	20.6	333.6	8.9/4.4	0.063
7	RHX2(HP) - RHX1(HP)	1.55	20.5	420.9	9.5/16	0.020
8	RHX1(HP)～ Na/CO_2 HX	1.7→1.35×2	20.4	658.8	27/22	0.166
9	RHX2 outlet - Bypass compressor	1.6×2→1.6	8.75	343	10/20	0.0086
10	Bypass compressor -RHX1 inlet	1.55→1.55×2	20.5	420.9	12/6.2	0.033

RHX1:Recuperater1　　LP:Low pressure
RHX2:Recuperater2　　HP:High pressure
PC:Pre-cooler

Table 3.2-8 Pressure loss of CO_2 system

Component and CO_2 piping	Pressure loss (MPa)
• Na/CO_2 HX • CO_2 piping between Na/CO_2 HX and turbine	0.611 0.206
• RHX1 (LP) • CO_2 piping between turbine and RHX1(LP)	0.1 0.13
• RHX2 (LP) • CO_2 piping between RHX1 and RHX2	0.059 0.093
• Pre-cooler • CO_2 piping between RHX2 and Pre-cooler • CO_2 piping between Pre-cooler and main compressor	0.025 0.114 0.013
• RHX2(HP) • CO_2 piping between main compressor and RHX2(HP) • CO_2 piping between RHX2 and RHX1(HP)	0.012 0.063 0.02
• RHX1(HP) • CO_2 piping between RHX1(HP) and Na/CO_2 HX	0.042 0.166

RHX1:Recuperater1　　LP:Low pressure
RHX2:Recuperater2　　HP:High pressure

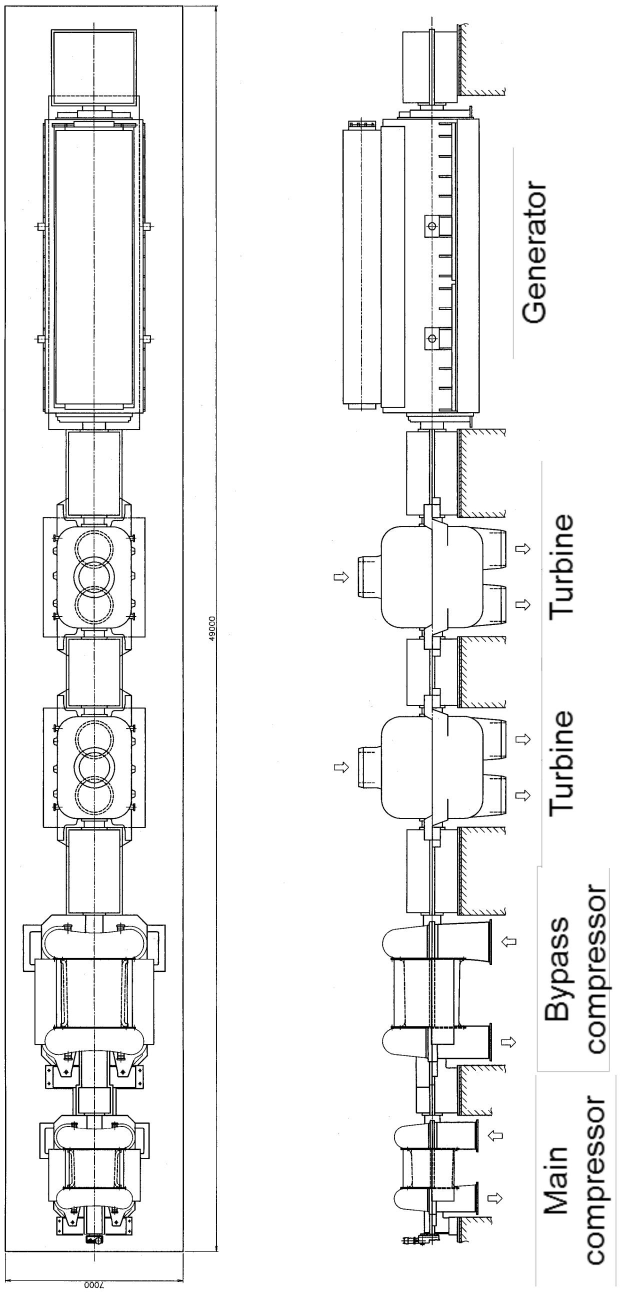


Fig. 3.2-1 S-CO_2 turbine system constitution

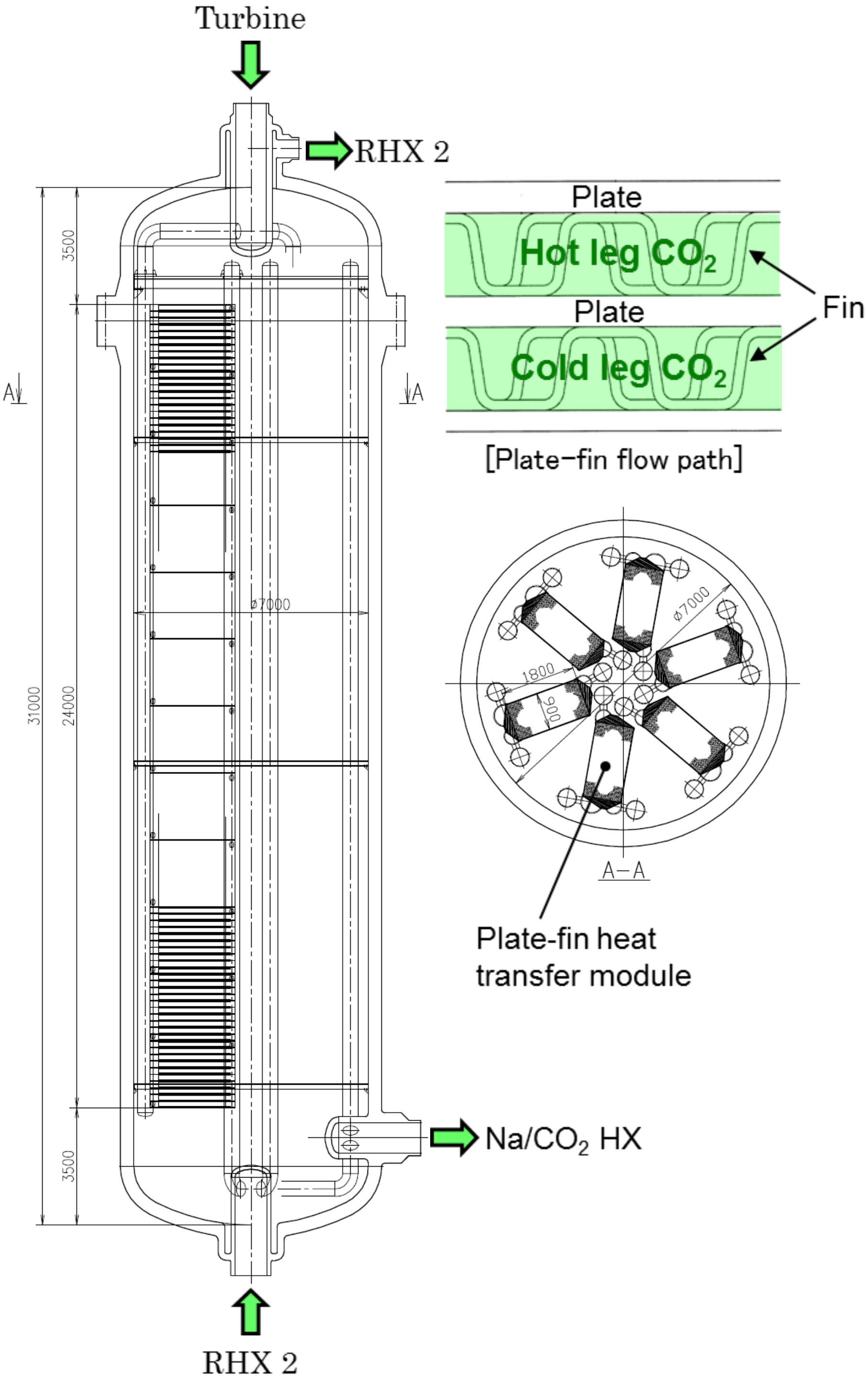


Fig. 3.2-2 Recuperater 1

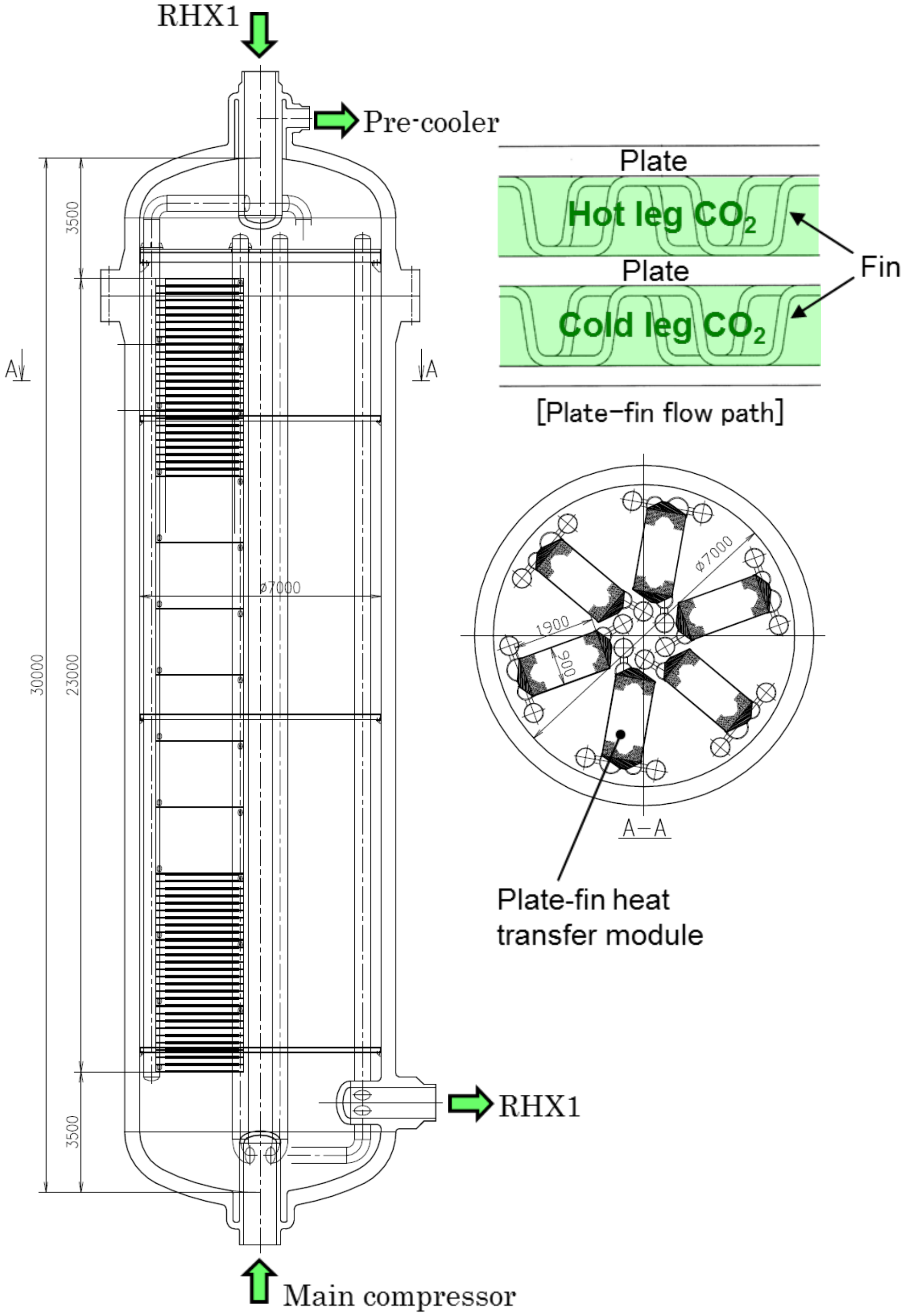


Fig. 3.2-3 Recuperater 2

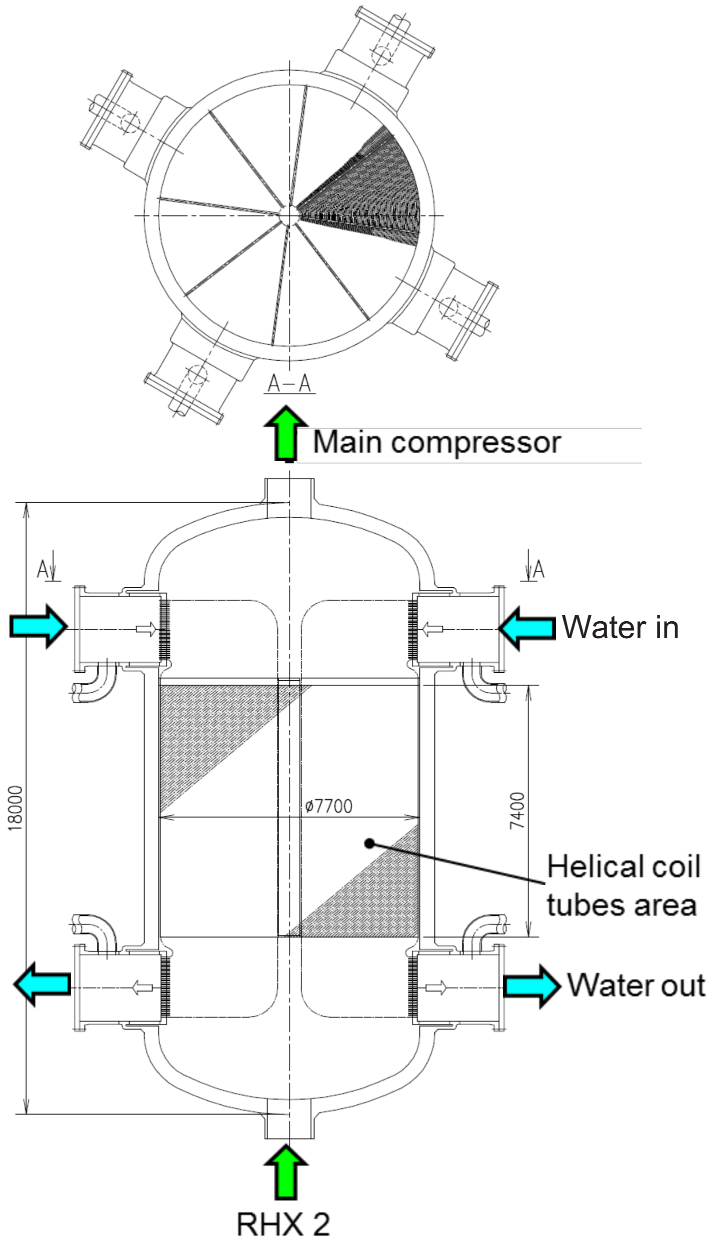


Fig. 3.2-4 Pre-cooler

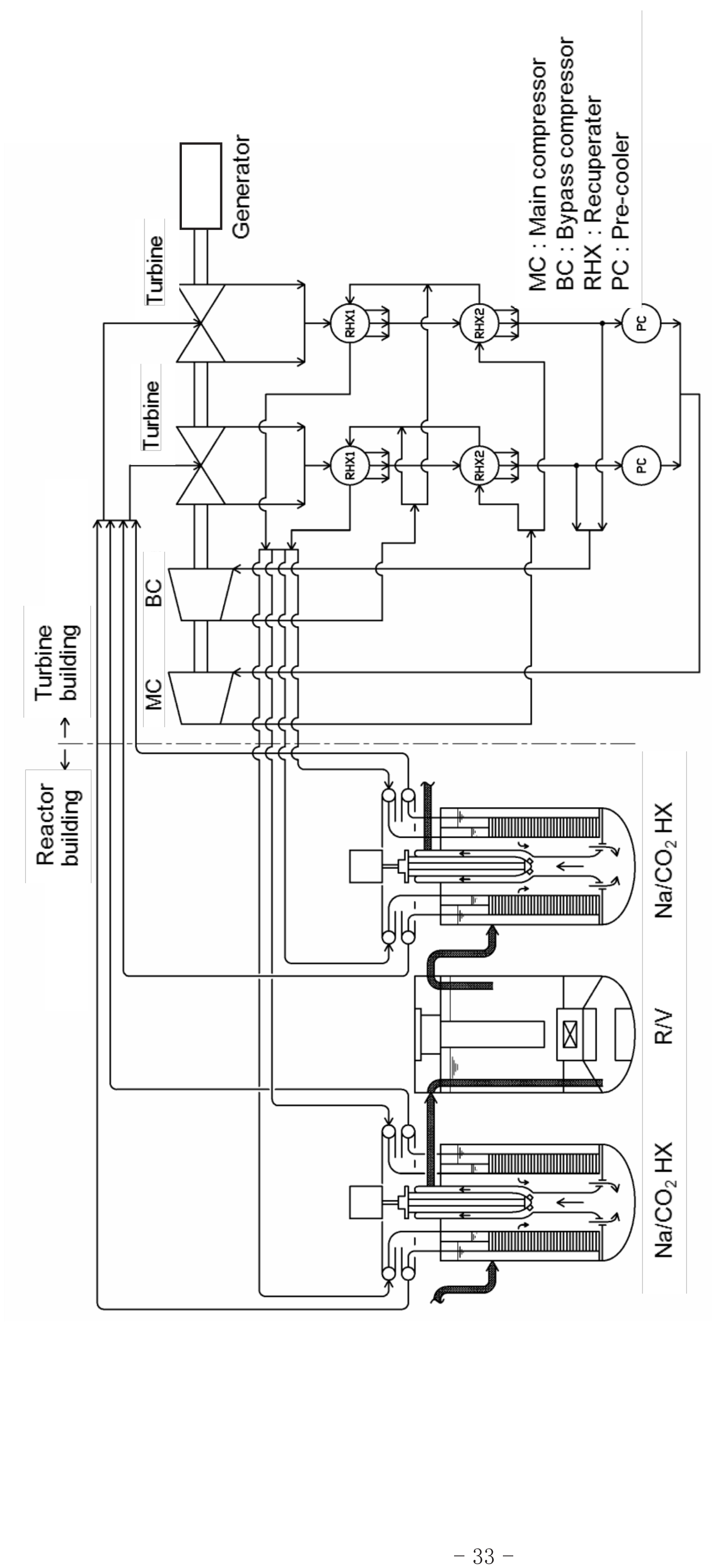


Fig. 3.2-5 JSFR S-CO$_2$ turbine system

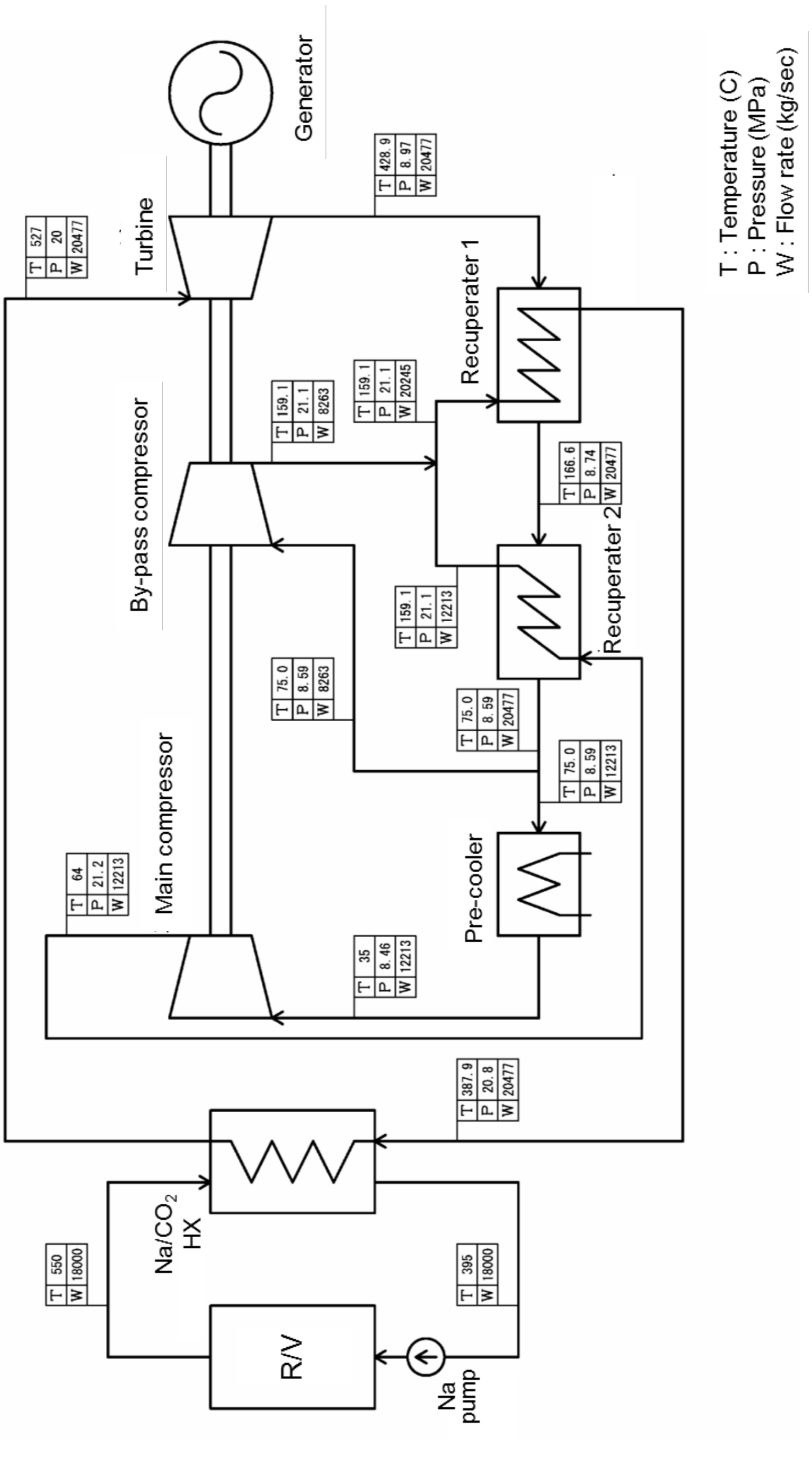


Fig. 3.2-6 JSFR S-CO_2 system heat mass balance

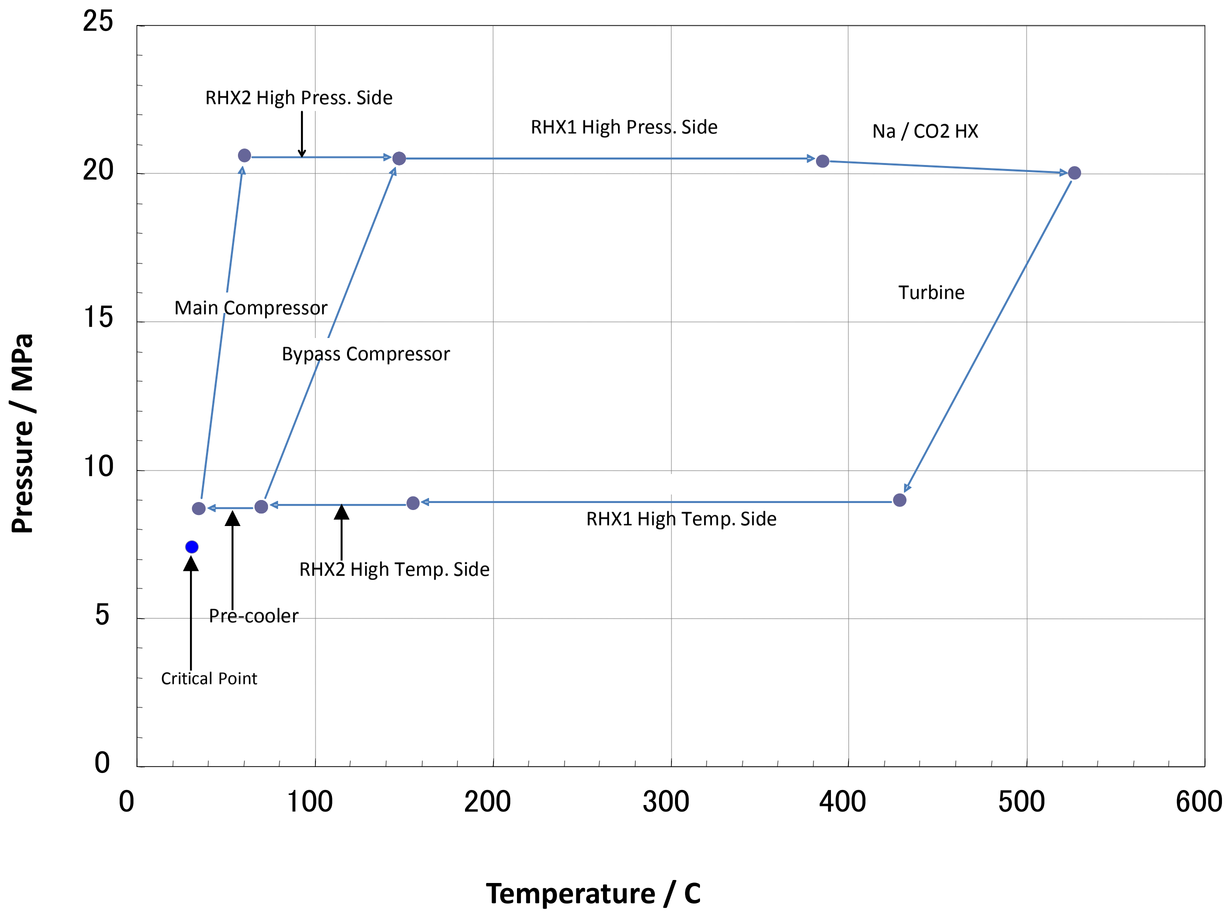


Fig. 3.2-7 JSFR S-CO_2 cycle

4. Plant dimension estimation

4.1 Turbine building layout

To clarify the dimension of the supercritical gas turbine building, the component/piping layout was studied based on the system configuration shown in Fig. 3.2-5 and the following conditions:

a. The generator unit consists of the generator (1 unit), turbine (2 units), main compressor (1 unit), and bypass compressor (1 unit). The cooling system equipment consists of 2 loops in which recuperator 1 and 2 (2 units for each) and pre-cooler (2 units) are installed.

b. Two sets of reactors are constructed as a twin plant in the same way as the reference JSFR.

c. Two sets of recuperators 1 (RHX1) and 2 (RHX2) and pre-cooler (PC) are arranged near the turbine to shorten the CO_2 main piping length.

d. The floor level on which the generator unit is installed (operating floor) is on the ground floor for easy installation of components.

e. The support skirt of the recuperators and pre-cooler are equipped at their center elevation, and these components are installed on the operating floor.

f. A crane is installed on the ceiling above the generator unit for maintenance work and transfer to a trailer. Because the recuperator and pre-cooler are lengthy components for which transfer cannot be performed indoors, a hatch is provided on the ceiling to allow for carrying in/out using an outside crane.

The layouts of the supercritical CO_2 turbine building based on the above considerations are shown in Figs. 4.1-1 through 4.1-4. Figures 4.1-1 and 4.1-2 show the floor plans, and Figs. 4.1-3 and 4.1-4 show the vertical cross-sectional views for the places where the turbines and heat exchangers are installed, respectively. The dimensions of the turbine building are 124 m × 67.5 m × 50 m in height. The volume of the turbine building is approx.

178,000 m^3. This is 40% smaller than the reference JSFR's 303,000 m^3. The supercritical CO_2 turbine building and the steam turbine building are compared in Fig. 4.1-5.

The turbine building volume of the supercritical CO_2 turbine system is significantly reduced compared with that of the steam turbine system because of the smaller turbine and simpler system configuration having no condenser and feed-water pump.

4.2 Reactor building layout

The reactor building layout in which the Na/CO_2 heat exchanger is arranged was developed based on the layout drawing of the reference JSFR. The layouts in the reactor building are shown in Figs. 4.2-1 through 4.2-8 (plane views of the respective floors and vertical cross-sectional views). In the figure, the primary sodium dump tank in which the sodium (100 m^3) of the ex-vessel fuel storage tank (EVST) can be received is also arranged.

The volume of the spaces for installing the primary and secondary cooling systems of the reference JSFR is 92,230 m^3. On the other hand, the volume of the spaces for installing the primary cooling system of the secondary sodium system-eliminated plant employing the Na/CO_2 heat exchangers was calculated to be 86,079 m^3. It is reduced by 7%. The comparison of the reactor building size is shown in Fig.4.2-9.

Although the secondary sodium circuit-eliminated JSFR with the S-CO_2 system has no steam generators and secondary pumps, the volume of Na/CO_2 heat exchanger is approximately 3.6 times that of the Na/Na intermediate heat exchanger. The comparison of components (reactor and heat exchanger) and building volume listed in Table 4.2-1 shows that the volume of the building for the primary system components increases from 56,000 m^3 (reference JSFR) to 69,600 m^3 (secondary sodium circuit-eliminated JSFR). For this reason, the reduction rate in the reactor building volume was smaller than that in the turbine building volume.

4.3 3D view

The arrangements of the main components such as the heat exchangers, and piping are illustrated in Figs. 4.3-1~4.3-6 as 3D view to make it easy to understand their layout.

Table 4.2-1 Comparison of primary and secondary component building size

	Primary component building (R/V, HX, pump)	Secondary component building (SG, pump)
Reference JSFR (Secondary sodium circuit + Steam Rankin cycle)	55,959m^3	27,940m^3
Secondary circuit elimination JSFR ($S\text{-}CO_2$ cycle)	69,597m^3	—

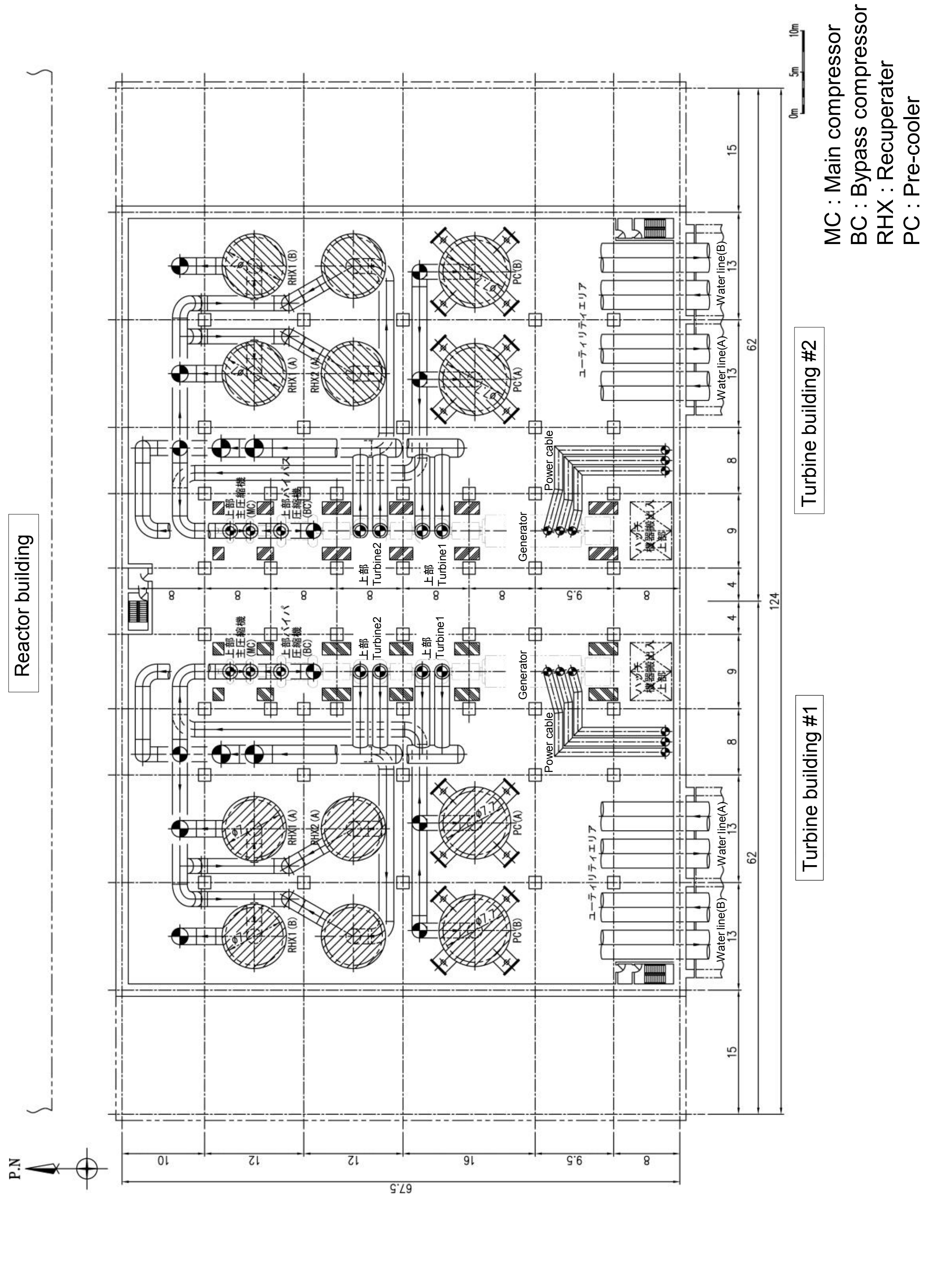


Fig. 4.1-1 S-CO_2 turbine system layout (Lower area)

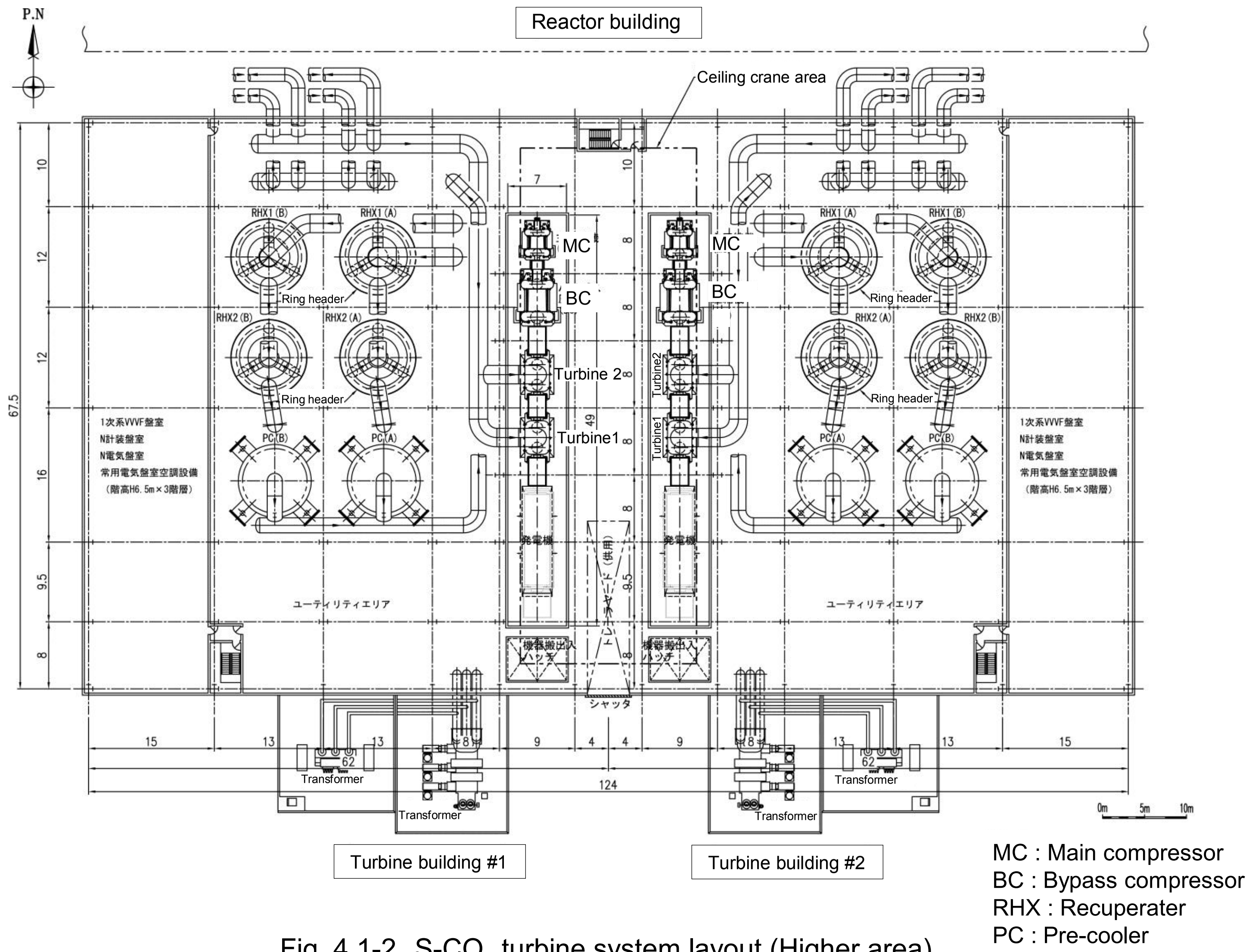
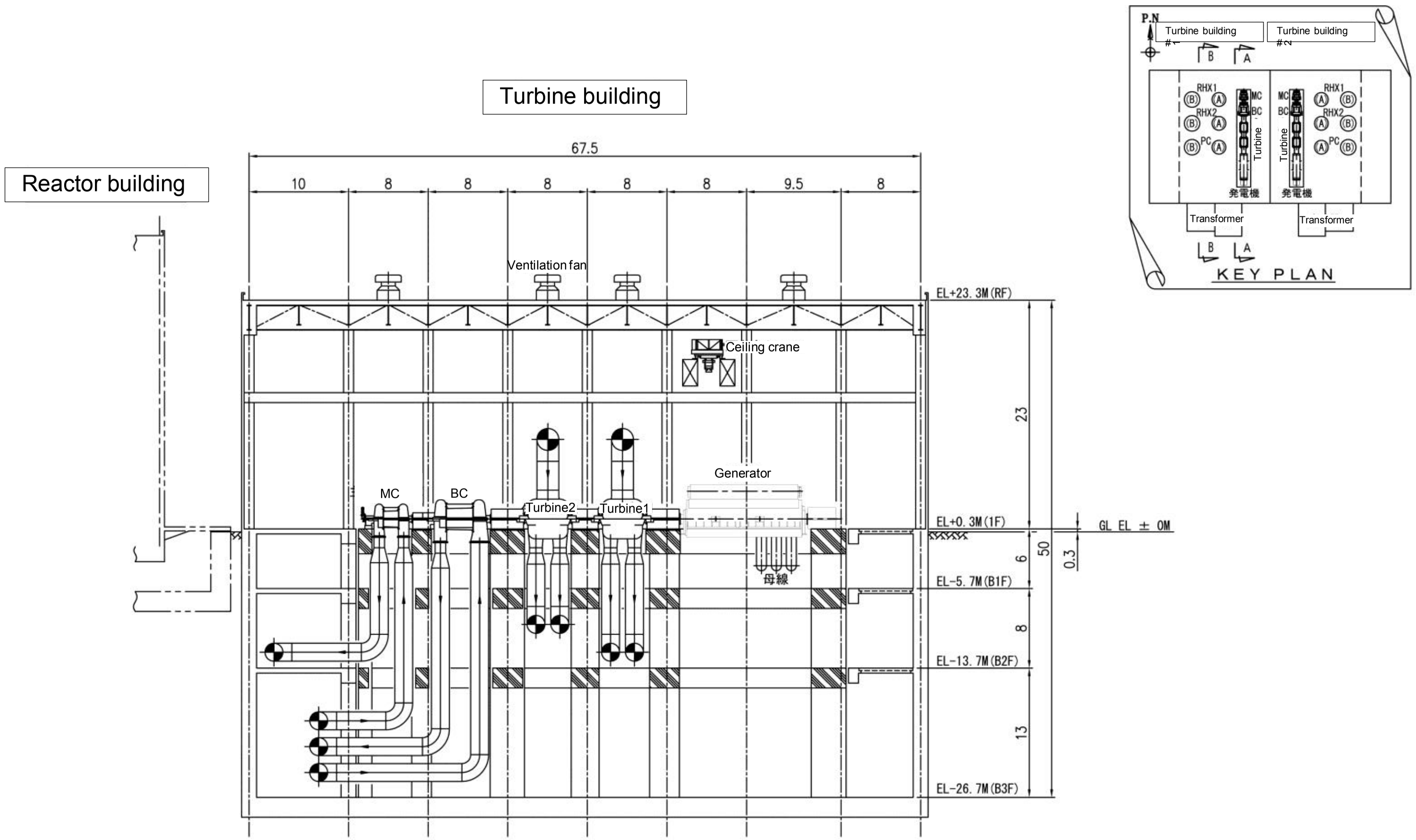


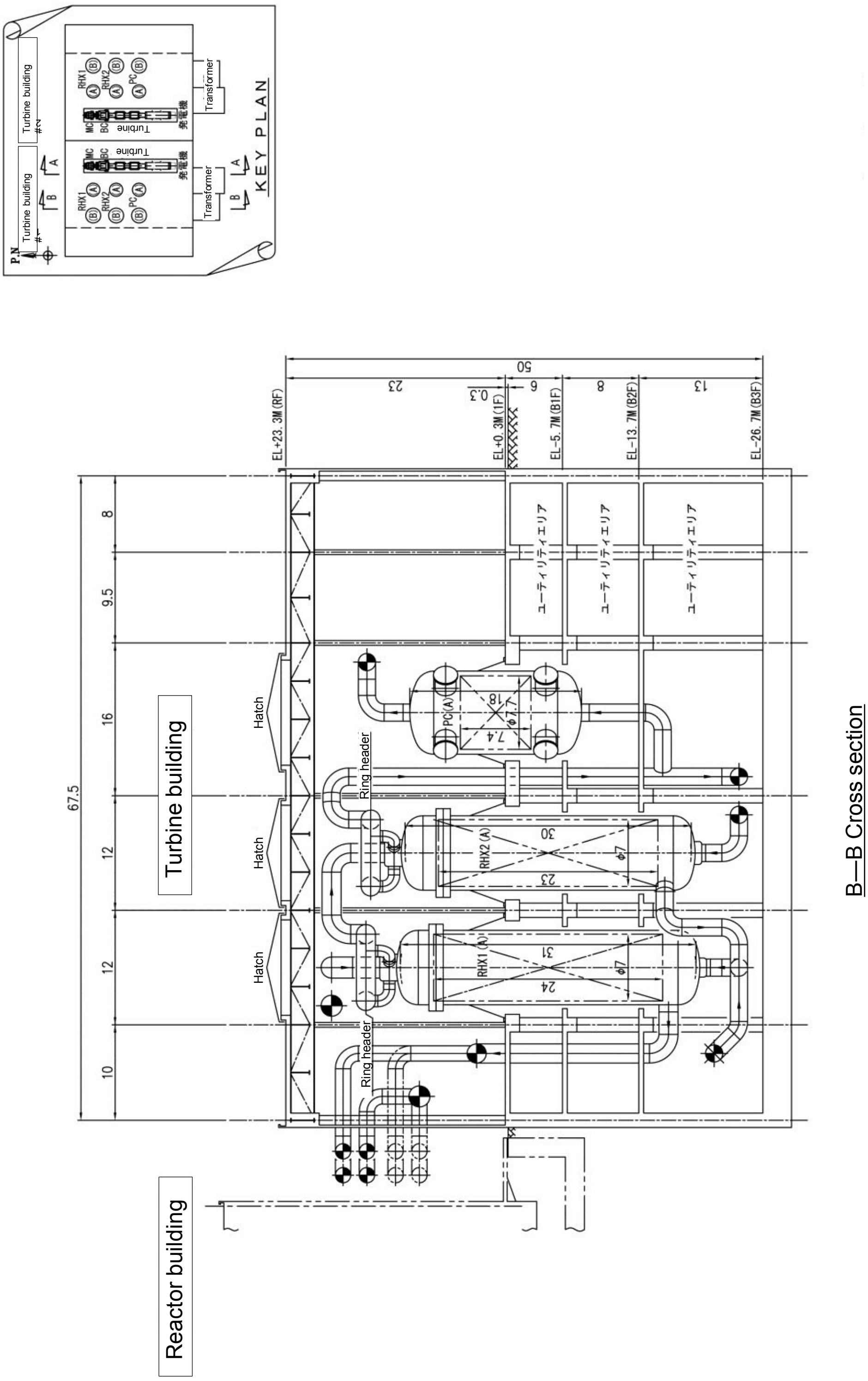
Fig. 4.1-2 S-CO_2 turbine system layout (Higher area)



A—A Cross section

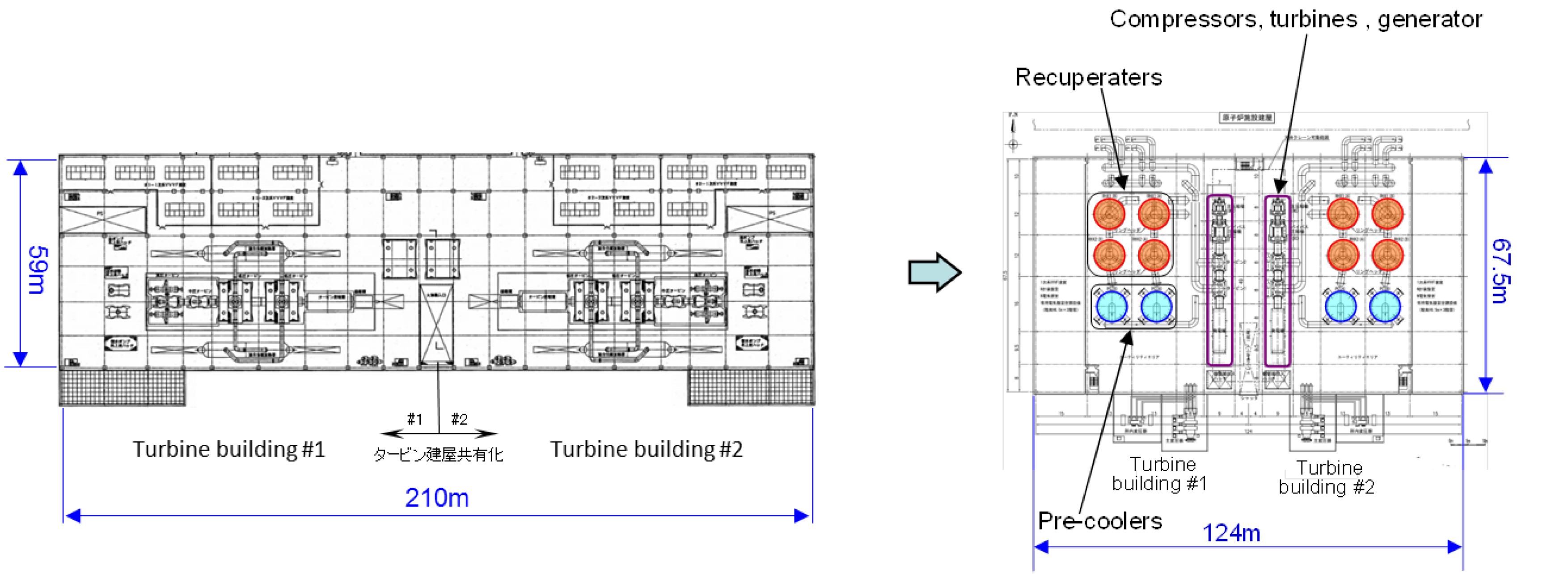
MC : Main compressor
BC : Bypass compressor

Fig. 4.1-3 S-CO_2 turbine system layout (Vertical cross section A-A)



RHX : Recuperater
PC : Pre-cooler

Fig. 4.1-4 S-CO_2 turbine system layout (Vertical cross section B-B)



Reference JSFR (Twin plant)
[Secondary circuit + Steam Rankin cycle]
303,000m^3

Secondary circuit elimination JSFR (Twin plant)
[S-CO_2 cycle]
178,000m^3

Fig. 4.1-5 Comparison of turbine building size

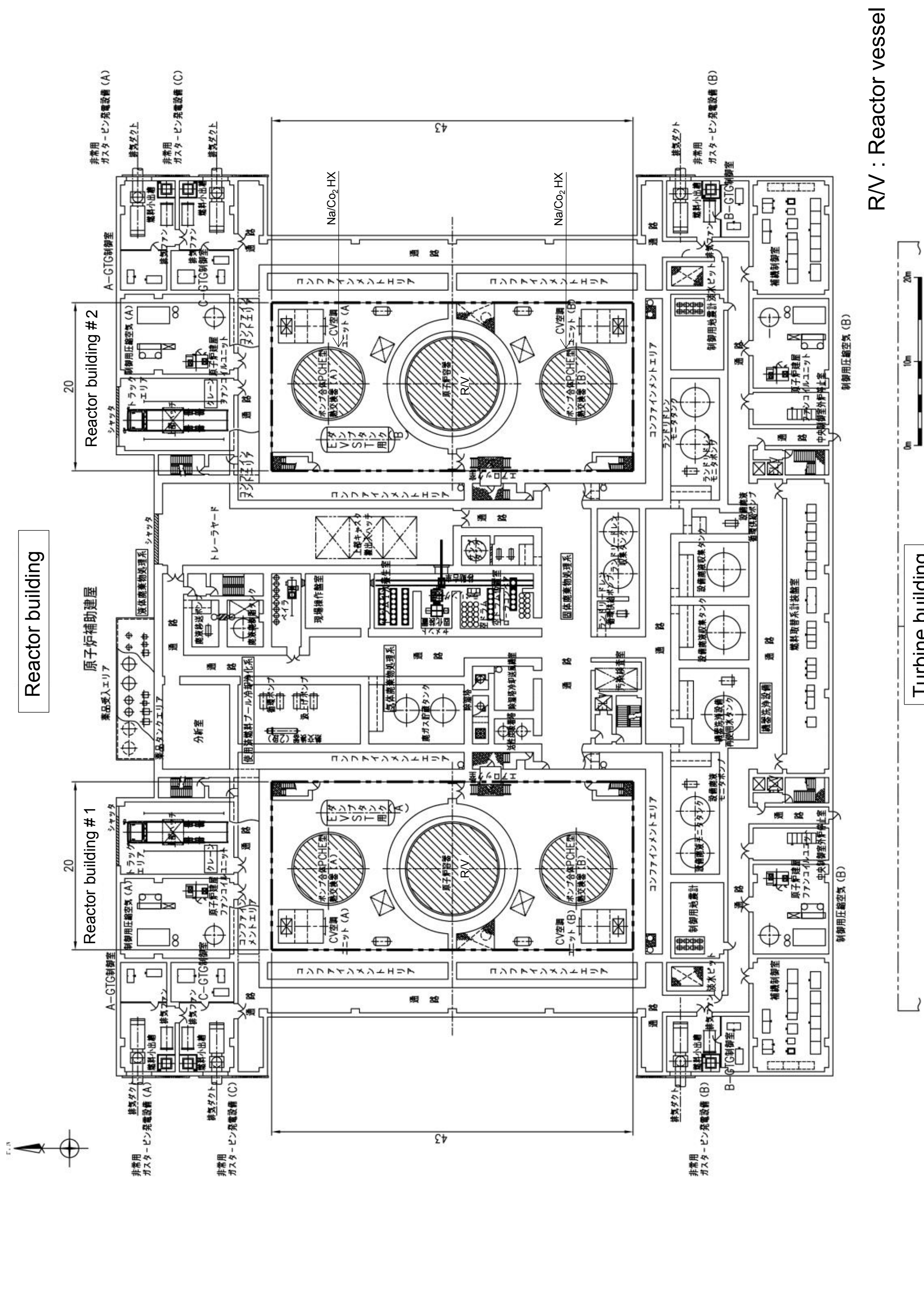


Fig. 4.2-1 Reactor building layout (1st floor)

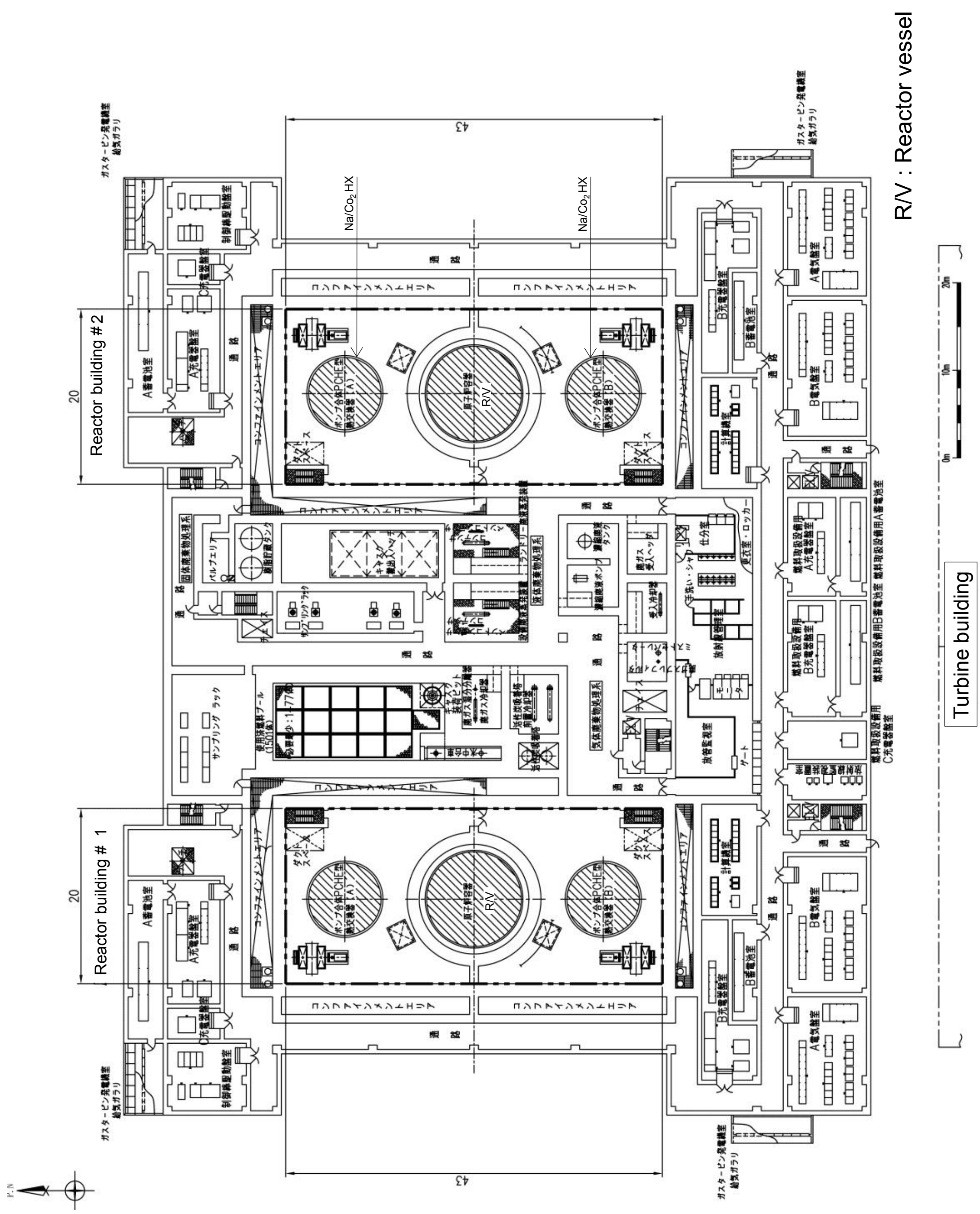


Fig. 4.2-2 Reactor building layout (2nd t floor)

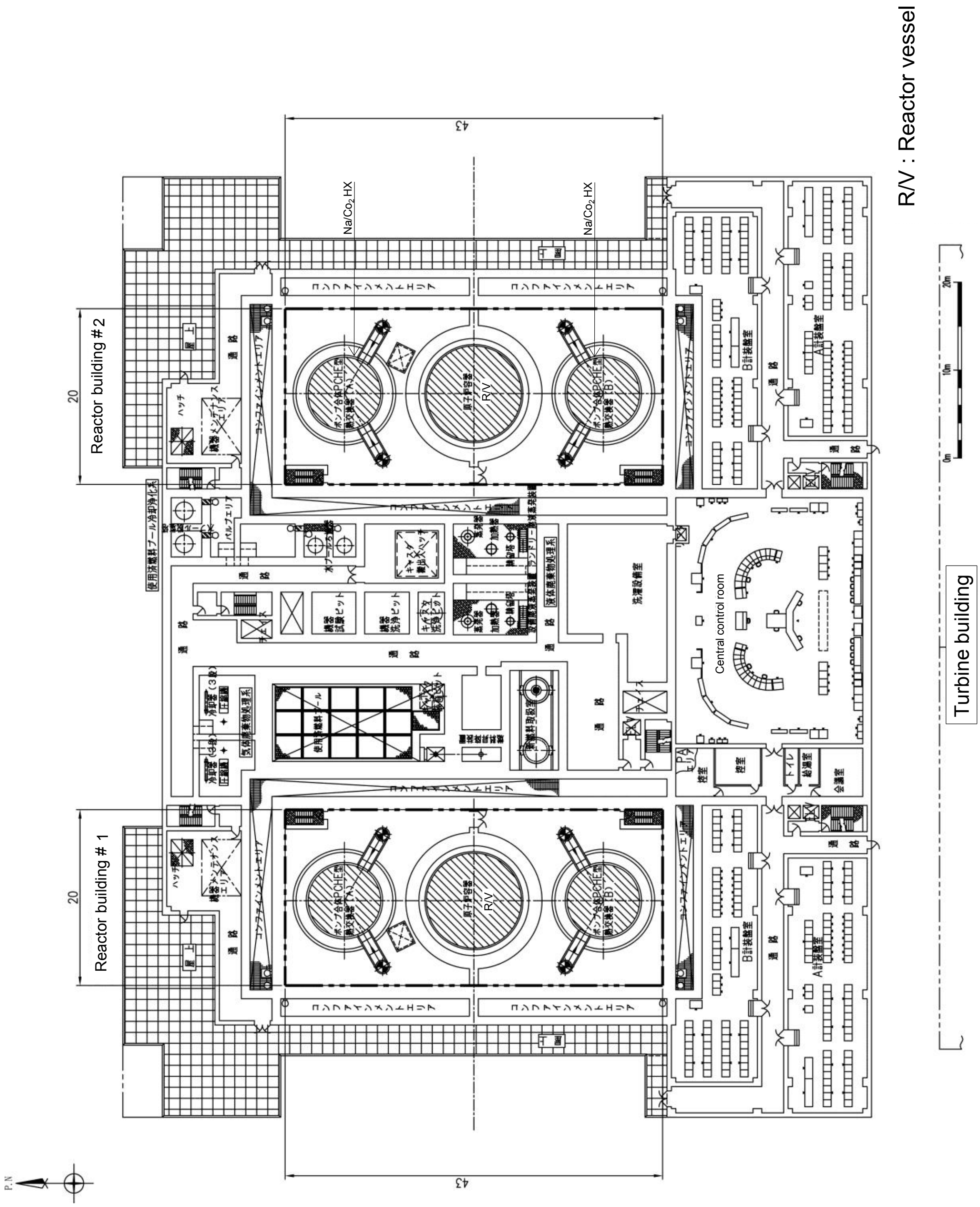


Fig. 4.2-3 Reactor building layout (3rd floor)

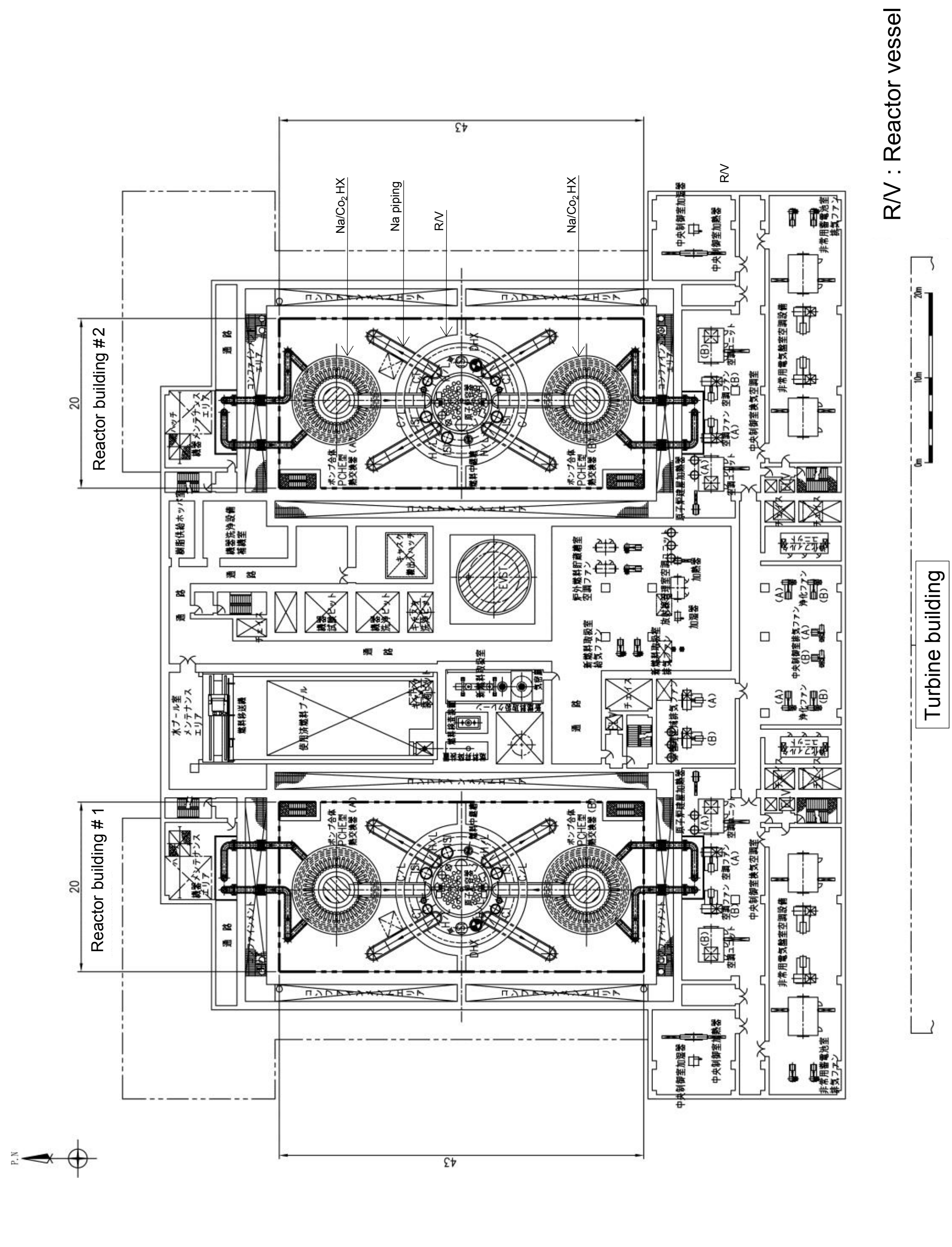


Fig. 4.2-4 Reactor building layout (4th floor)

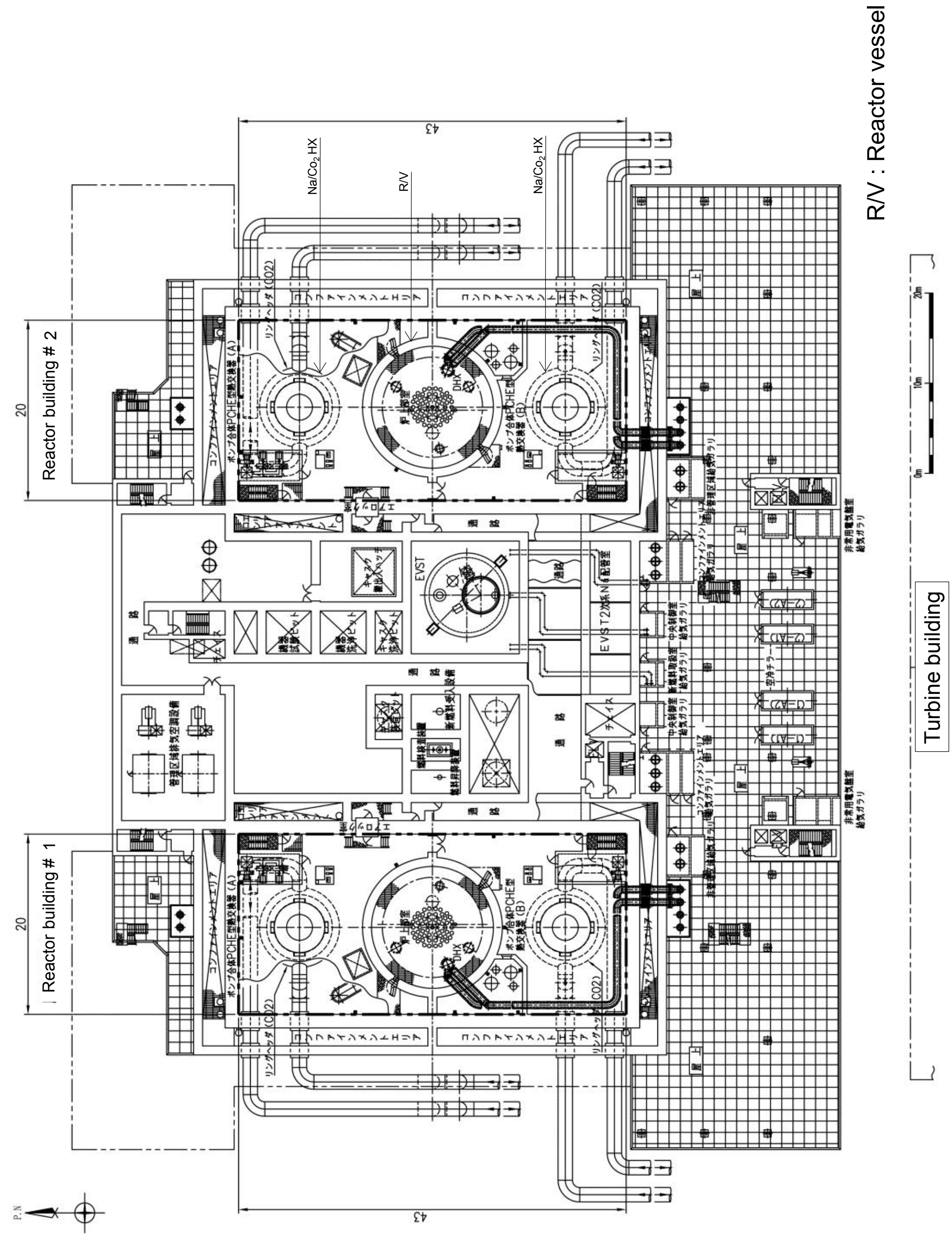


Fig. 4.2-5 Reactor building layout (5th floor)

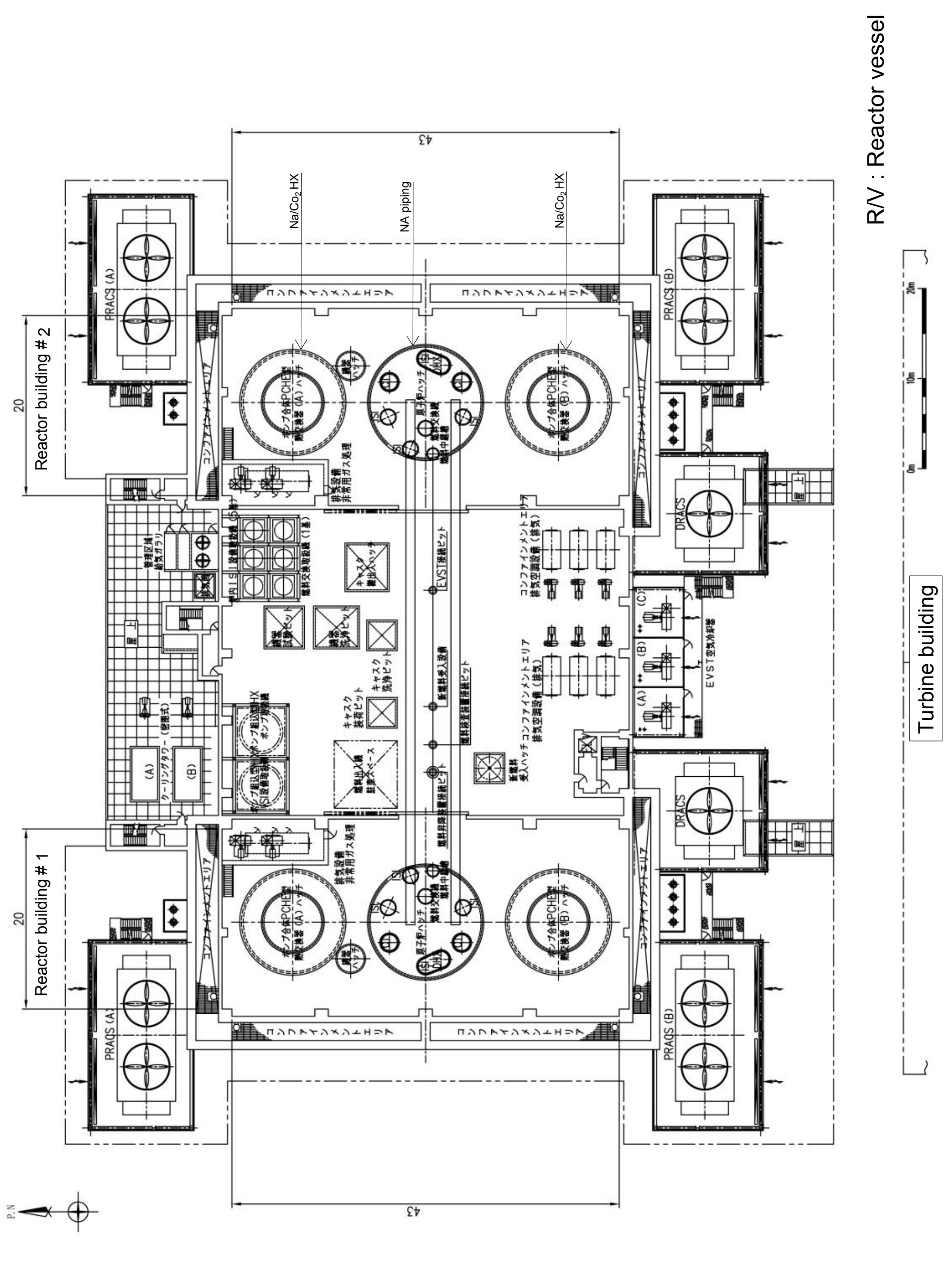


Fig. 4.2-6 Reactor building layout (6th floor)

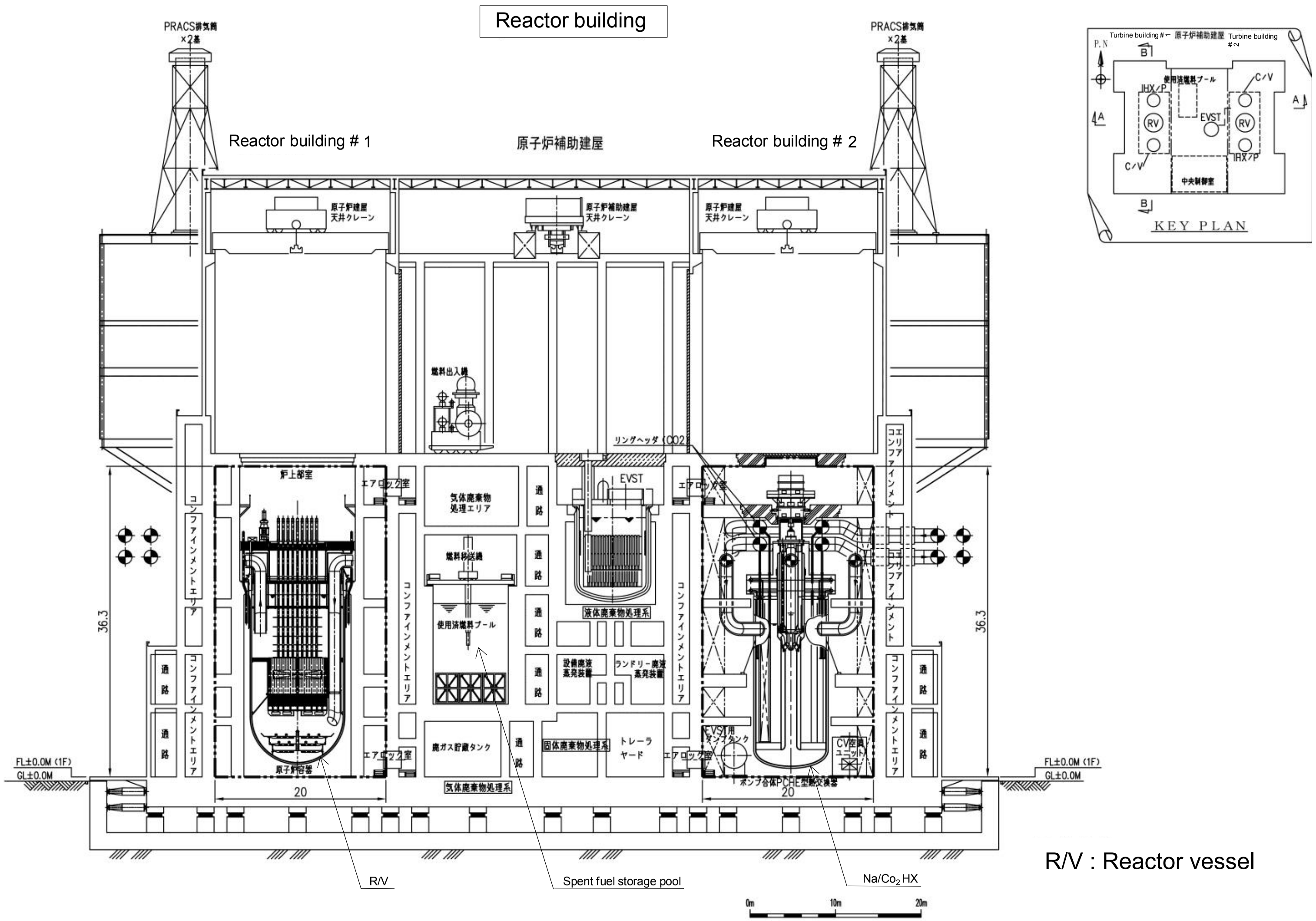


Fig. 4.2-7 Reactor building layout (Vertical cross section A-A)

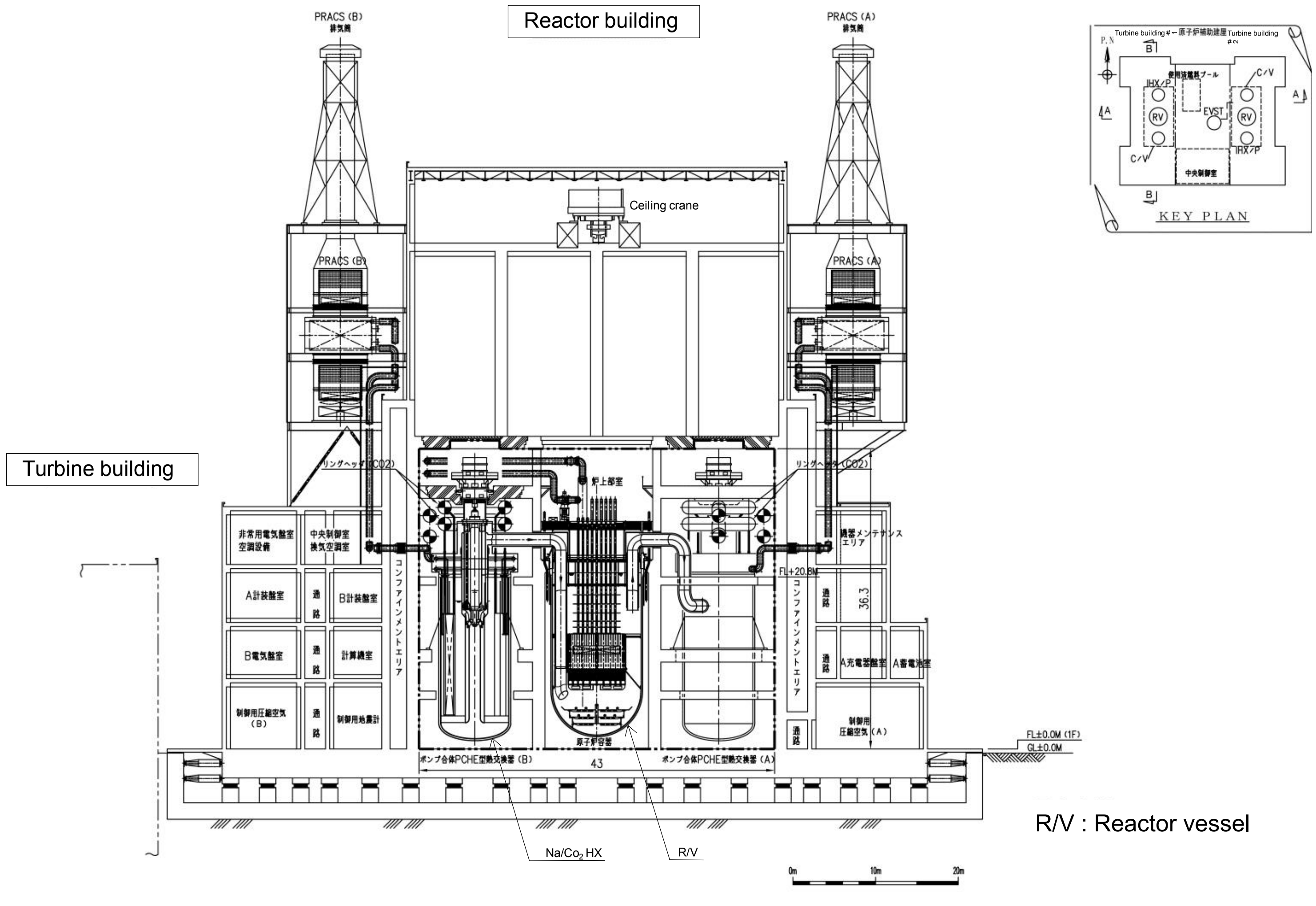


Fig. 4.2-8 Reactor building layout (Vertical cross section B-B)

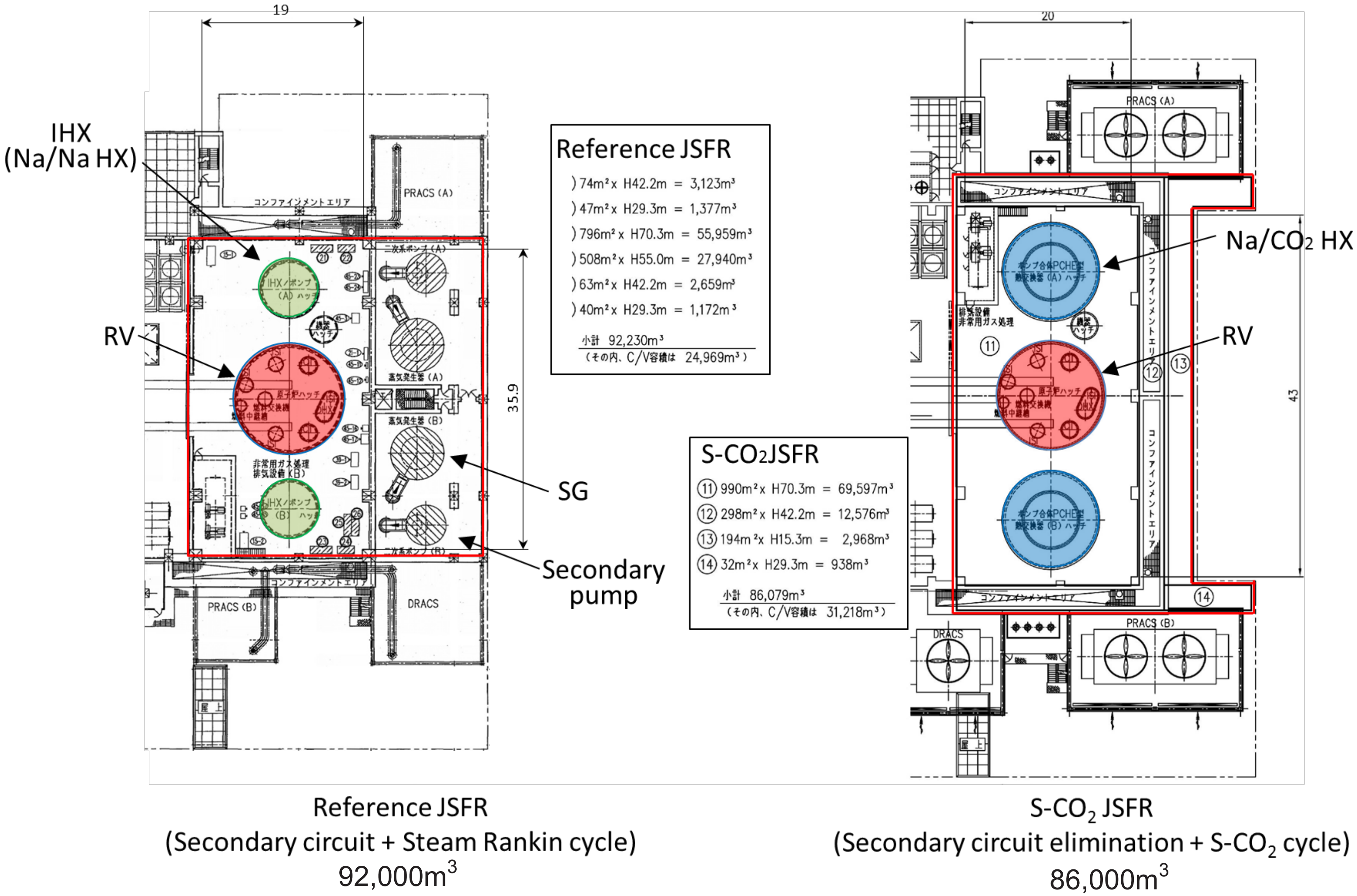


Fig. 4.2-9 Comparison of reactor building size

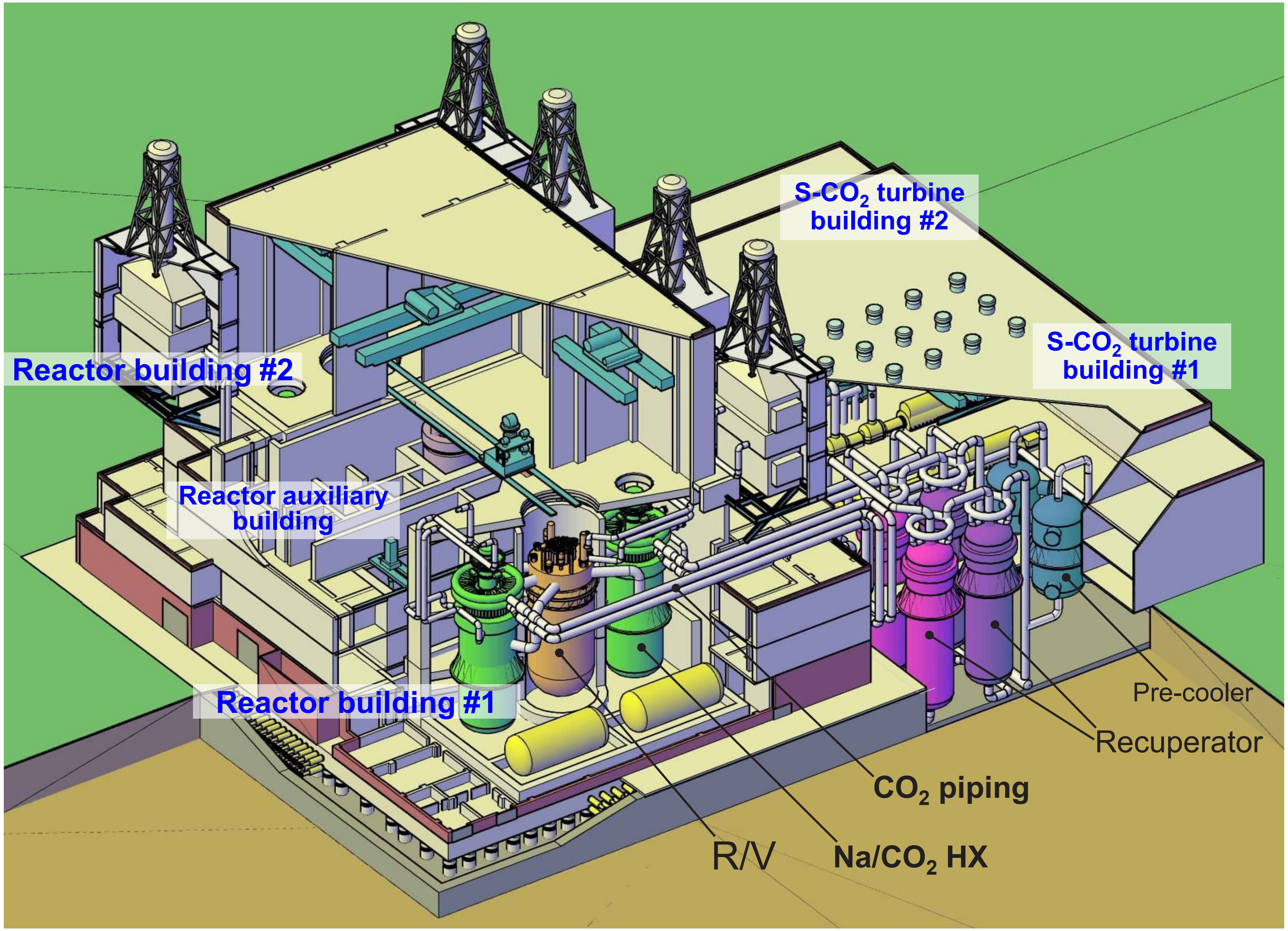


Fig. 4.3-1 Reactor building and S-CO$_2$ turbine building (3D view)

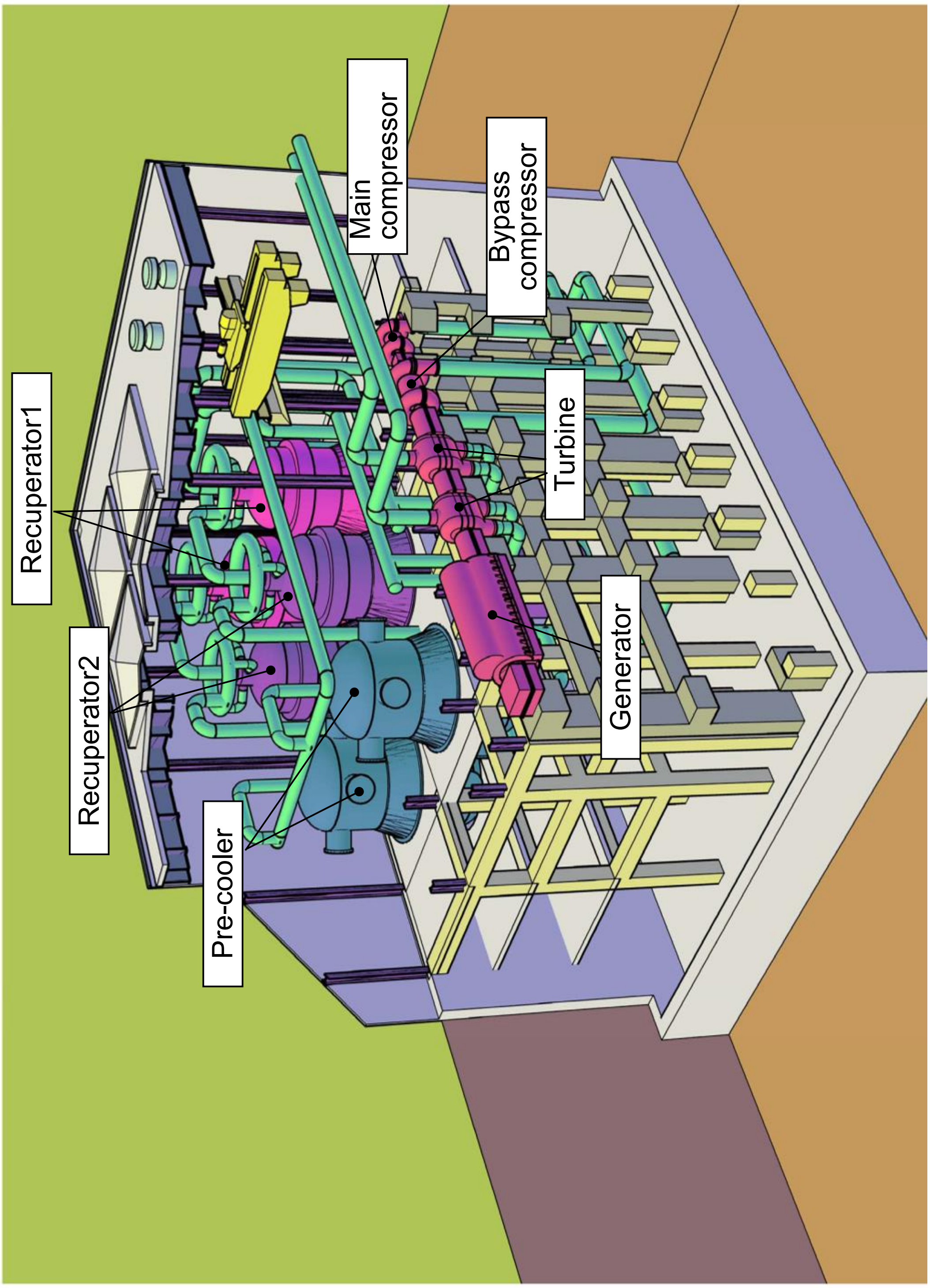


Fig. 4.3-2 S-CO_2 turbine building (3D view)

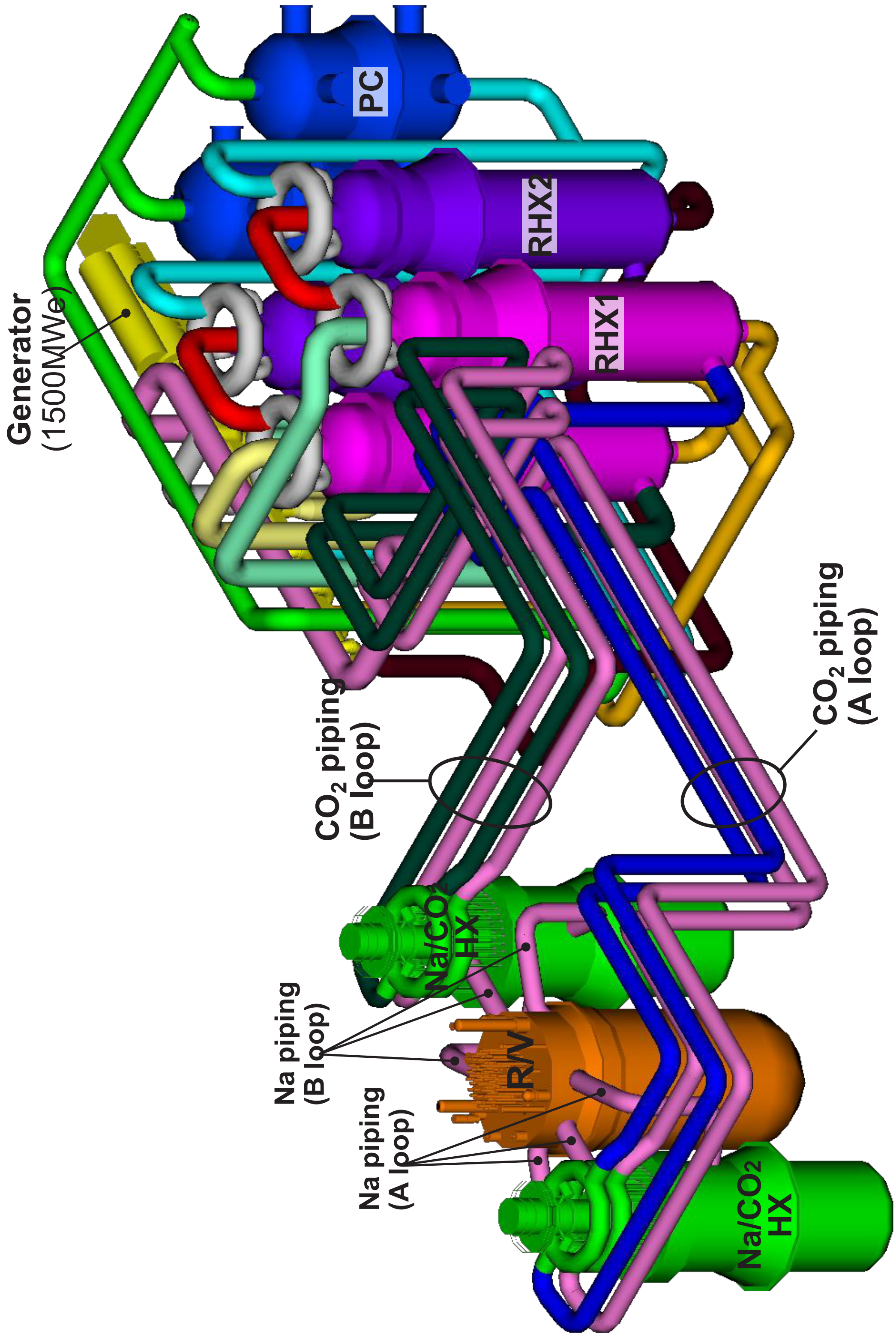


Fig. 4.3-3 Component and piping layout (3D view)

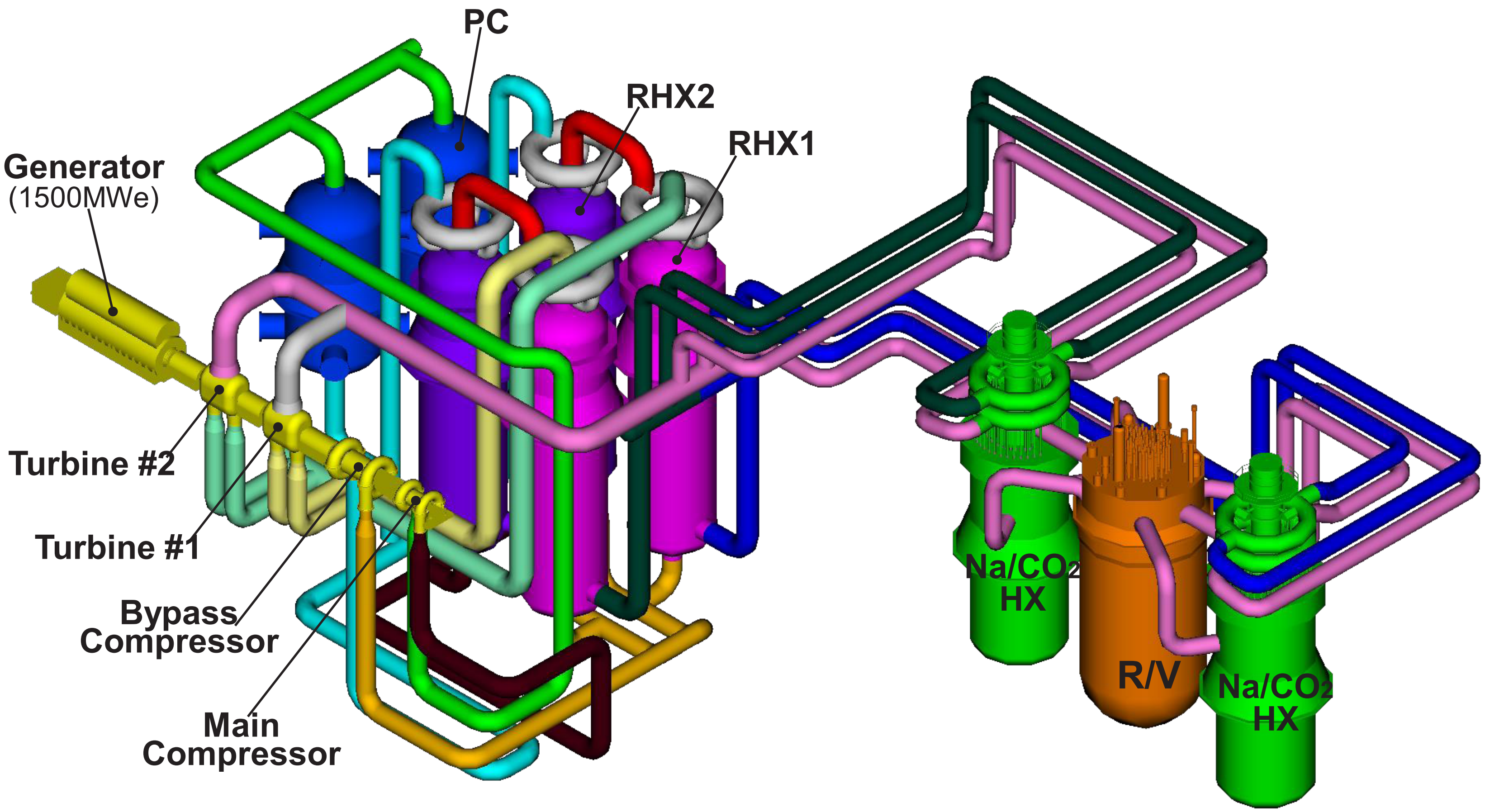


Fig. 4.3-4 Component and piping layout (3D view)

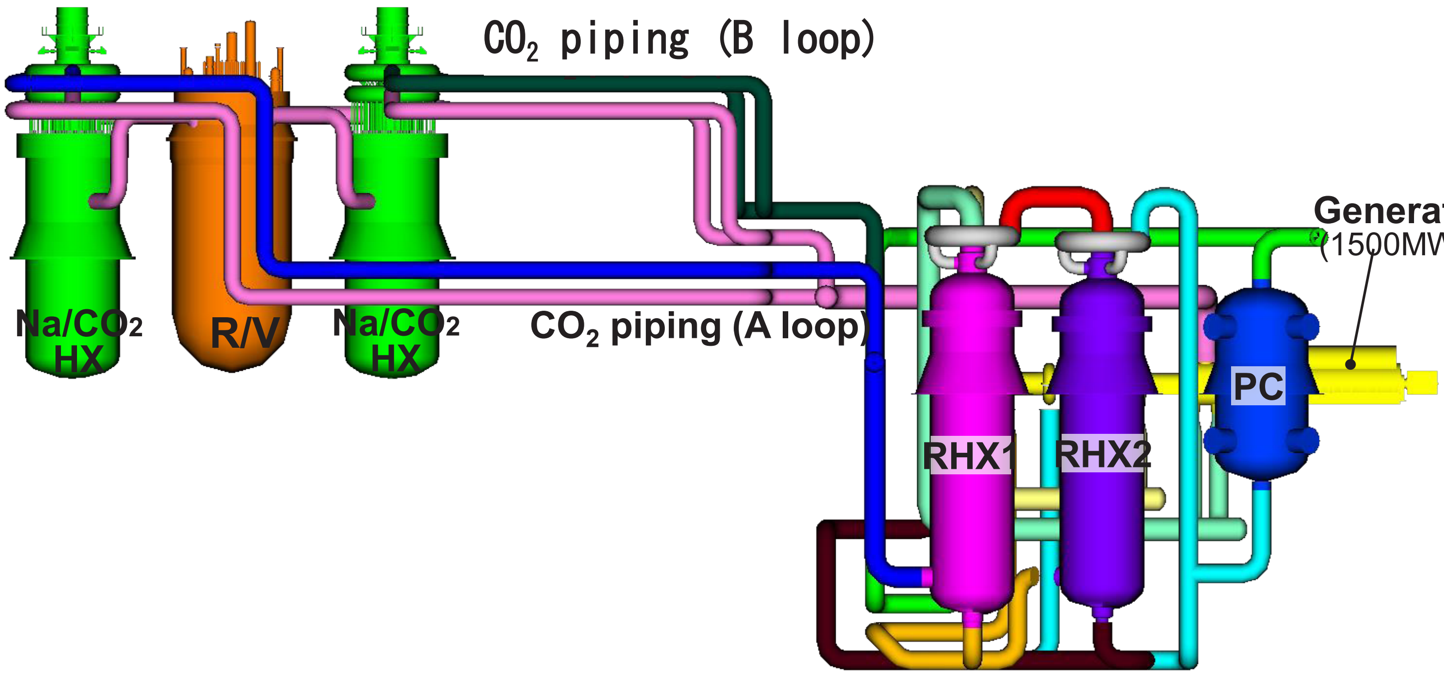


Fig. 4.3-5 Component and piping layout (Lateral view)

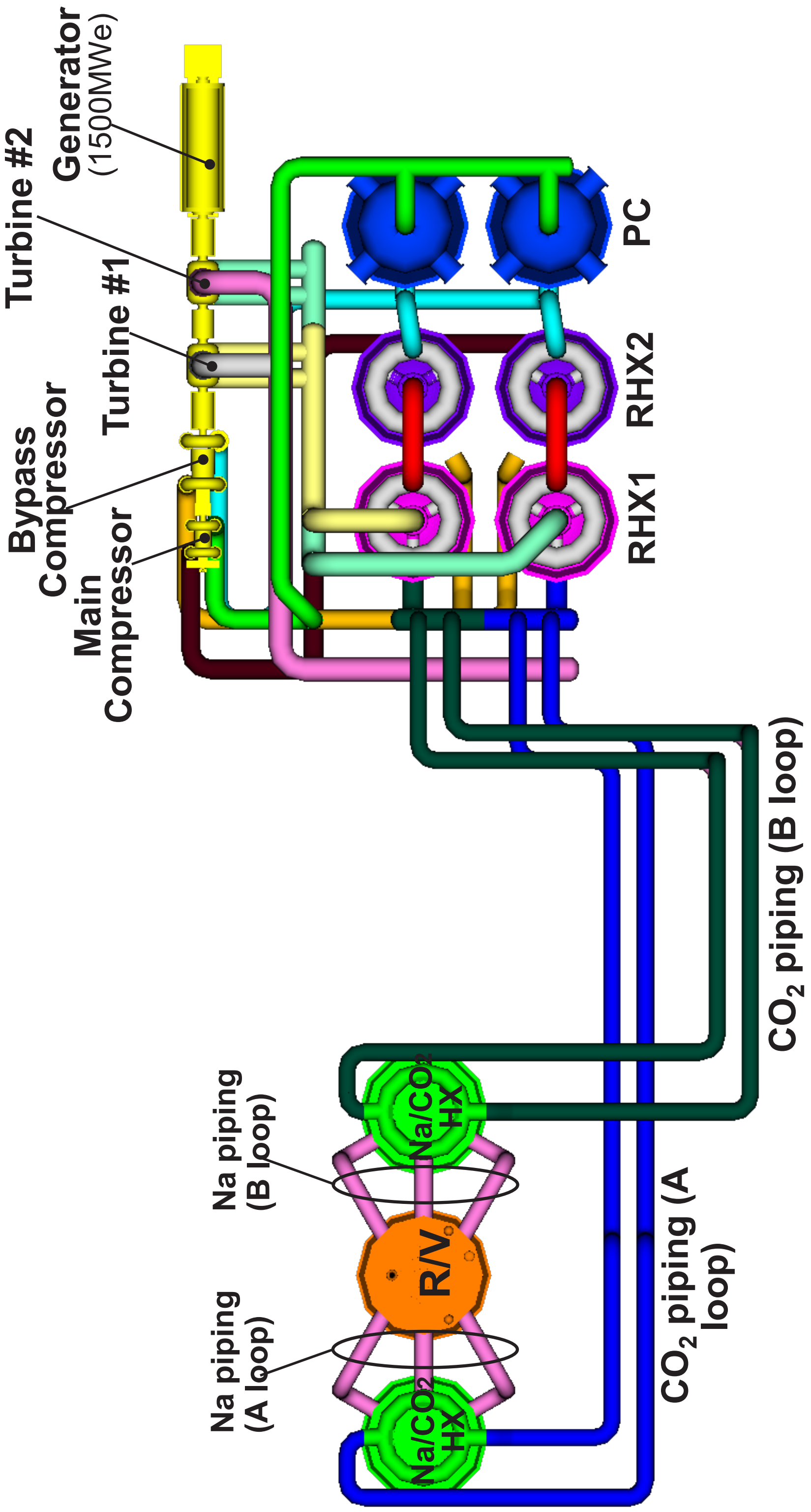


Fig. 4.3-6 Component and piping layout (Top view)

5. Conclusions

The CO_2/Na reaction is not a furious event, and failure area in the PCHE during CO_2 leak is limited by its partitioned flow paths. Taking into account these advantages, a preliminary conceptual design of the secondary sodium circuit-eliminated plant system was performed as a possible application of a supercritical CO_2 turbine system to the JFSR.

This report discusses a supercritical CO_2 turbine system for the secondary sodium circuit-eliminated 1500 MWe JSFR. The compressor CO_2 inlet temperature and compression ratio were set by the design sea water temperature and the protection of compressor from a gas-liquid mixing region. Based on these results, the heat-mass balance at a turbine inlet pressure of 20 MPa was calculated to determine the design conditions of the turbine system components. The sizes of the main and bypass compressors, recuperator and pre-cooler were estimated.

Furthermore, the layout of the components/piping of the secondary sodium circuit-eliminated JSFR was studied to clarify the plant size in combination with the Na/CO_2 heat exchanger deign in another work. The study revealed that the reactor building volume and the turbine building volume was reduced by approx. 7% and 40%, respectively, compared with those for the JSFR with the secondary sodium circuit and water/steam systems.

The turbine building volume was dramatically reduced because the supercritical CO_2 turbine is much smaller and its system configuration is simpler compared with the steam turbine. On the other hand, the reduction in reactor building volume remains only 7% because the Na/CO_2 heat exchanger is larger than the IHX and SG.

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This design study was performed by the advice of Dr. Yasushi MUTO and Dr. Takao ISHIZUKA of Tokyo Institute of Technology, and Dr. Tatsuya HINOKI of Kyoto Univ. Their advice was indispensable to accomplishing the study and is greatly appreciated.

This paper includes the outcome of collaborative study between JAEA and JAPC (this is the representative of nine electric utilities, Electric Power Development Co., Ltd. and JAPC) in the accordance with "the agreement about the development of a commercialized fast breeder reactor cycle system".

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国際単位系（SI）

表１．SI 基本単位

基本量	SI 基本単位	
	名称	記号
長さ	メートル	m
質量	キログラム	kg
時間	秒	s
電流	アンペア	A
熱力学温度	ケルビン	K
物質量	モル	mol
光度	カンデラ	cd

表２．基本単位を用いて表されるSI組立単位の例

組立量	SI 基本単位	
	名称	記号
面積	平方メートル	m^2
体積	立法メートル	m^3
速さ，速度	メートル毎秒	m/s
加速度	メートル毎秒毎秒	m/s^2
波数	毎メートル	m^{-1}
密度，質量密度	キログラム毎立方メートル	kg/m^3
面積密度	キログラム毎平方メートル	kg/m^2
比体積	立方メートル毎キログラム	m^3/kg
電流密度	アンペア毎平方メートル	A/m^2
磁界の強さ	アンペア毎メートル	A/m
量濃度 (a)，濃度	モル毎立方メートル	mol/m^3
質量濃度	キログラム毎立法メートル	kg/m^3
輝度	カンデラ毎平方メートル	cd/m^2
屈折率 (b)	（数字の） 1	1
比透磁率 (b)	（数字の） 1	1

(a) 量濃度（amount concentration）は臨床化学の分野では物質濃度（substance concentration）ともよばれる。
(b) これらは無次元量あるいは次元１をもつ量であるが、そのことを表す単位記号である数字の１は通常は表記しない。

表３．固有の名称と記号で表されるSI組立単位

組立量	SI 組立単位			
	名称	記号	他のSI単位による表し方	SI基本単位による表し方
平面角	ラジアン(b)	rad	1 (b)	m/m
立体角	ステラジアン(b)	sr(c)	1 (b)	m^{2}/m^{2}
周波数	ヘルツ (d)	Hz		s^{-1}
力	ニュートン	N		$m\ kg\ s^{-2}$
圧力，応力	パスカル	Pa	N/m^2	$m^{-1}\ kg\ s^{-2}$
エネルギー，仕事，熱量	ジュール	J	N m	$m^{2}\ kg\ s^{-2}$
仕事率，工率，放射束	ワット	W	J/s	$m^{2}\ kg\ s^{-3}$
電荷，電気量	クーロン	C		s A
電位差（電圧），起電力	ボルト	V	W/A	$m^{2}\ kg\ s^{-3}\ A^{-1}$
静電容量	ファラド	F	C/V	$m^{-2}\ kg^{-1}\ s^{4}\ A^{2}$
電気抵抗	オーム	Ω	V/A	$m^{2}\ kg\ s^{-3}\ A^{-2}$
コンダクタンス	ジーメンス	S	A/V	$m^{-2}\ kg^{-1}\ s^{3}\ A^{2}$
磁束	ウエーバ	Wb	Vs	$m^{2}\ kg\ s^{-2}\ A^{-1}$
磁束密度	テスラ	T	Wb/m^2	$kg\ s^{-2}\ A^{-1}$
インダクタンス	ヘンリー	H	Wb/A	$m^{2}\ kg\ s^{-2}\ A^{-2}$
セルシウス温度	セルシウス度(e)	℃		K
光束	ルーメン	lm	cd sr(c)	cd
照度	ルクス	lx	lm/m^2	$m^{-2}\ cd$
放射性核種の放射能 (f)	ベクレル (d)	Bq		s^{-1}
吸収線量, 比エネルギー分与, カーマ	グレイ	Gy	J/kg	$m^{2}\ s^{-2}$
線量当量, 周辺線量当量, 方向性線量当量, 個人線量当量	シーベルト (g)	Sv	J/kg	$m^{2}\ s^{-2}$
酸素活性	カタール	kat		$s^{-1}\ mol$

(a)SI接頭語は固有の名称と記号を持つ組立単位と組み合わせても使用できる。しかし接頭語を付した単位はもはやコヒーレントではない。
(b)ラジアンとステラジアンは数字の１に対する単位の特別な名称で、量についての情報をつたえるために使われる。実際には、使用する時には記号rad及びsrが用いられるが、習慣として組立単位としての記号である数字の１は明示されない。
(c)測光学ではステラジアンという名称と記号srを単位の表し方の中に、そのまま維持している。
(d)ヘルツは周期現象についてのみ、ベクレルは放射性核種の統計的過程についてのみ使用される。
(e)セルシウス度はケルビンの特別な名称で、セルシウス温度を表すために使用される。セルシウス度とケルビンの単位の大きさは同一である。したがって、温度差や温度間隔を表す数値はどちらの単位で表しても同じである。
(f)放射性核種の放射能（activity referred to a radionuclide）は、しばしば誤った用語で"radioactivity"と記される。
(g)単位シーベルト（PV,2002,70,205）についてはCIPM勧告2（CI-2002）を参照。

表４．単位の中に固有の名称と記号を含むSI組立単位の例

組立量	SI 組立単位		
	名称	記号	SI 基本単位による表し方
粘度	パスカル秒	Pa s	$m^{-1}\ kg\ s^{-1}$
力のモーメント	ニュートンメートル	N m	$m^{2}\ kg\ s^{-2}$
表面張力	ニュートン毎メートル	N/m	$kg\ s^{-2}$
角速度	ラジアン毎秒	rad/s	$m\ m^{-1}\ s^{-1}=s^{-1}$
角加速度	ラジアン毎秒毎秒	rad/s^2	$m\ m^{-1}\ s^{-2}=s^{-2}$
熱流密度，放射照度	ワット毎平方メートル	W/m^2	$kg\ s^{-3}$
熱容量，エントロピー	ジュール毎ケルビン	J/K	$m^{2}\ kg\ s^{-2}\ K^{-1}$
比熱容量，比エントロピー	ジュール毎キログラム毎ケルビン	J/(kg K)	$m^{2}\ s^{-2}\ K^{-1}$
比エネルギー	ジュール毎キログラム	J/kg	$m^{2}\ s^{-2}$
熱伝導率	ワット毎メートル毎ケルビン	W/(m K)	$m\ kg\ s^{-3}\ K^{-1}$
体積エネルギー	ジュール毎立方メートル	J/m^3	$m^{-1}\ kg\ s^{-2}$
電界の強さ	ボルト毎メートル	V/m	$m\ kg\ s^{-3}\ A^{-1}$
電荷密度	クーロン毎立方メートル	C/m^3	$m^{-3}\ sA$
表面電荷	クーロン毎平方メートル	C/m^2	$m^{-2}\ sA$
電束密度，電気変位	クーロン毎平方メートル	C/m^2	$m^{-2}\ sA$
誘電率	ファラド毎メートル	F/m	$m^{-3}\ kg^{-1}\ s^{4}\ A^{2}$
透磁率	ヘンリー毎メートル	H/m	$m\ kg\ s^{-2}\ A^{-2}$
モルエネルギー	ジュール毎モル	J/mol	$m^{2}\ kg\ s^{-2}\ mol^{-1}$
モルエントロピー, モル熱容量	ジュール毎モル毎ケルビン	J/(mol K)	$m^{2}\ kg\ s^{-2}\ K^{-1}\ mol^{-1}$
照射線量（Ｘ線及びγ線）	クーロン毎キログラム	C/kg	$kg^{-1}\ sA$
吸収線量率	グレイ毎秒	Gy/s	$m^{2}\ s^{-3}$
放射強度	ワット毎ステラジアン	W/sr	$m^{4}\ m^{-2}\ kg\ s^{-3}=m^{2}\ kg\ s^{-3}$
放射輝度	ワット毎平方メートル毎ステラジアン	$W/(m^{2}\ sr)$	$m^{2}\ m^{-2}\ kg\ s^{-3}=kg\ s^{-3}$
酵素活性濃度	カタール毎立方メートル	kat/m^3	$m^{-3}\ s^{-1}\ mol$

表５．SI 接頭語

乗数	接頭語	記号	乗数	接頭語	記号
10^{24}	ヨタ	Y	10^{-1}	デシ	d
10^{21}	ゼタ	Z	10^{-2}	センチ	c
10^{18}	エクサ	E	10^{-3}	ミリ	m
10^{15}	ペタ	P	10^{-6}	マイクロ	µ
10^{12}	テラ	T	10^{-9}	ナノ	n
10^{9}	ギガ	G	10^{-12}	ピコ	p
10^{6}	メガ	M	10^{-15}	フェムト	f
10^{3}	キロ	k	10^{-18}	アト	a
10^{2}	ヘクト	h	10^{-21}	ゼプト	z
10^{1}	デカ	da	10^{-24}	ヨクト	y

表６．SIに属さないが、SIと併用される単位

名称	記号	SI 単位による値
分	min	1 min=60s
時	h	1h =60 min=3600 s
日	d	1 d=24 h=86 400 s
度	°	1°=(π/180) rad
分	’	1’=(1/60)°=(π/10800) rad
秒	”	1”=(1/60)’=(π/648000) rad
ヘクタール	ha	$1ha=1hm^2=10^4m^2$
リットル	L，l	$1L=11=1dm^3=10^3cm^3=10^{-3}m^3$
トン	t	$1t=10^3$ kg

表７．SIに属さないが、SIと併用される単位で、SI単位で表される数値が実験的に得られるもの

名称	記号	SI 単位で表される数値
電子ボルト	eV	$1eV=1.602\ 176\ 53(14)\times10^{-19}J$
ダルトン	Da	$1Da=1.660\ 538\ 86(28)\times10^{-27}kg$
統一原子質量単位	u	1u=1 Da
天文単位	ua	$1ua=1.495\ 978\ 706\ 91(6)\times10^{11}m$

表８．SIに属さないが、SIと併用されるその他の単位

名称	記号	SI 単位で表される数値
バール	bar	$1\ bar=0.1MPa=100kPa=10^5Pa$
水銀柱ミリメートル	mmHg	1mmHg=133.322Pa
オングストローム	Å	$1\ Å=0.1nm=100pm=10^{-10}m$
海里	M	1 M=1852m
バーン	b	$1\ b=100fm^2=(10^{-12}cm)2=10^{-28}m^2$
ノット	kn	1 kn=(1852/3600)m/s
ネーパ	Np	SI単位との数値的な関係は、対数量の定義に依存。
ベル	B	
デジベル	dB	

表９．固有の名称をもつCGS組立単位

名称	記号	SI 単位で表される数値
エルグ	erg	$1\ erg=10^{-7}\ J$
ダイン	dyn	$1\ dyn=10^{-5}N$
ポアズ	P	$1\ P=1\ dyn\ s\ cm^{-2}=0.1Pa\ s$
ストークス	St	$1\ St=1cm^2\ s^{-1}=10^{-4}m^2\ s^{-1}$
スチルブ	sb	$1\ sb=1cd\ cm^{-2}=10^4cd\ m^{-2}$
フォト	ph	$1\ ph=1cd\ sr\ cm^{-2}\ 10^4lx$
ガル	Gal	$1\ Gal=1cm\ s^{-2}=10^{-2}ms^{-2}$
マクスウェル	Mx	$1\ Mx=1G\ cm^2=10^{-8}Wb$
ガウス	G	$1\ G=1Mx\ cm^{-2}=10^{-4}T$
エルステッド (c)	Oe	$1\ Oe \triangleq (10^3/4\pi)A\ m^{-1}$

(c) ３元系のCGS単位系とSIでは直接比較できないため、等号「 ≙ 」は対応関係を示すものである。

表10．SIに属さないその他の単位の例

名称	記号	SI 単位で表される数値
キュリー	Ci	$1\ Ci=3.7\times10^{10}Bq$
レントゲン	R	$1\ R=2.58\times10^{-4}C/kg$
ラド	rad	$1\ rad=1cGy=10^{-2}Gy$
レム	rem	$1\ rem=1\ cSv=10^{-2}Sv$
ガンマ	γ	1 γ =1 nT=10-9T
フェルミ		1フェルミ=1 fm=10-15m
メートル系カラット		1メートル系カラット = 200 mg = 2×10-4kg
トル	Torr	1 Torr = (101 325/760) Pa
標準大気圧	atm	1 atm = 101 325 Pa
カロリー	cal	1cal=4.1858J（「15℃」カロリー），4.1868J（「IT」カロリー）4.184J（「熱化学」カロリー）
ミクロン	µ	$1\ \mu=1\mu m=10^{-6}m$

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