Design Characteristics of N_2 Brayton Cycle Power Conversion System coupled with an SFR in terms of Thermal Balance and Cycle Efficiency

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1. Introduction

As part of efforts to improve SFR safety, research on replacing conventional Rankine cycle power conversion system (PCS) with N_2 Brayton cycle PCS has been underway at Korea Atomic Energy Research Institute (KAERI). Since most design techniques of turbo-machineries for air have been already proven well, the use of Nitrogen gas (N_2) as the working fluid would be very promising for near-term deployment of an advanced power conversion options for a Sodium-cooled Fast Reactor (SFR) [1]. Moreover, a chemically inert feature of Nitrogen is one of the most essential benefits that may be potentially free from a conventional pressure boundary rupture and consequential sodium-water reaction (SWR) event [1].

The computational code of REGACY (REgenerative GAs brayton CYcle analyzer) to set the heat balance and to obtain cycle efficiency of N_2 Brayton cycle PCS were recently developed [2] and its verification process has been performed by using the reference design parameters of N_2 PCS for ASTRID [3][4].

The purpose of this study is to investigate cycle efficiencies in N_2 Brayton cycle PCS for ASTRID [3][4] complying with various design parameters such as the working fluids and cycle configurations

2. Methods and Results

2.1 Methodology

In order to obtain key design factors of N_2 Brayton cycle PCS, one-dimensional computational code, that is called REGACY, was developed [2]. A typical Brayton cycle power conversion system with intercooling and regeneration process using Nitrogen gas (N_2) as the working fluid has been preliminarily configured as shown in Fig.1 [3]. Most of the equations of states have been reasonably derived to obtain heat balances and thermo-dynamic cycle efficiencies of the N_2 PCS that consists of two compressors, single turbine, and recuperator. The state quantities at each point of the cycle were set up and detailed parameters were practically determined as well. In this process, it was assumed that i) the efficiency of the compressor, turbine, and recuperator are kept at constant in terms of N_2 flow rate changes and that ii) the pressure drops in all pipings and each cycle component are ideally kept at constant regardless of the system flow rate variations. Moreover, exit conditions at the compressor and turbine were practically set up based on the definition of isentropic efficiency.

The heat balance equations for the Sodium-to-gas (N_2) heat exchanger and the recuperator are as follows.

$$Q_h = \dot{m}_t(h_6 - h_5)$$

$$h_5 - h_4 = h_7 - h_8$$

Where Q_h is the heat source input from the IHTS loop sodium to the N_2 gas in PCS, and the terms of h and $\dot{m}_t$ are the enthalpy and the mass flow rate, respectively. The formulas of the isentropic efficiency of the two compressors and single turbine, and those of the effectiveness for the recuperators are also obtained by using the following equations;

$$\varepsilon_r = \frac{C_{p78}(T_7 - T_8)}{C_{pm\,in}(T_7 - T_4)}$$

$$\eta_t = \frac{h_6 - h_7}{h_6 - h_{7'}}$$

$$\eta_{c1} = \frac{h_{2'} - h_1}{h_2 - h_1}$$

$$\eta_{c2} = \frac{h_{4'} - h_3}{h_4 - h_3}$$

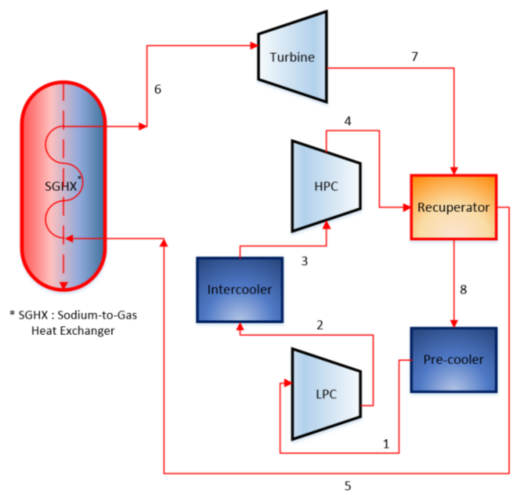


Fig. 1 Schematic of N_2 Intercooling and Regeneration Brayton Cycle

where η_t is the isentropic efficiency of the turbine, and η_{c1} and η_{c2} are those of the two compressors. The terms of ε_r and C_p represent the effectiveness of the regenerative heat exchanger and the specific heat at constant pressure, respectively. Based on the notations in Figure 1, C_{p78} and C_{p54} represent the average specific heat at constant pressure in each process. We picked the term of C_{pmin} as the smaller value between those two terms of C_{p78} and C_{p54} .

Although the cycle processes of '4-5,' '5-6,' '7-8,' and '8-1' are theoretically dealt with isobaric conditions, there are practical pressure drops in those processes. To this end, the cycle analysis of the present study has been made by taking into account practical pressure losses.

In the above equations, there are six unknowns such as h_2, h_3, h_4, h_7, T_8, and $\dot{m}_t$. All physical properties and the state quantities at each point were obtained by using the subroutines to provide appropriate properties of N_2 at the conditions of 'pressure and temperature' or 'pressure and enthalpy.' Finally, each unknown parameter was obtained with the solutions of six equations using the Gauss-Seidel method. The cycle efficiency, η_{th} was also obtained by using the following equation;

$$\eta_{th} = \frac{Work_{in} - Work_{out}}{Heat_{in}} = \frac{W_t - W_{c1} - W_{c2}}{Q_h}$$

where, W_t, W_{c1}, and W_{c2} represent the turbine work and works for two independent compressors configuring the cycle, respectively.

2.2 Results

The cycle configurations and corresponding thermal efficiency variations of N_2 Brayton cycle PCS for ASTRID were investigated with respect to the types of working fluids. All analyses were performed under the basic assumptions and boundary conditions below.

- Isentropic Efficiency (or effectiveness)
  - LPC : 89%
  - HPC : 88%
  - Turbine : 93%
  - Recuperator : 95%
- Pressure drop at each component
  - Pre-cooler : 40 kPa
  - Inter-cooler : 40 kPa
  - Recuperator (cold/hot) : 30 kPa / 60 kPa
  - Reheater : 30 kPa
  - SGHX : 120 kPa
- Mechanical efficiency of turbine : 98%
- Power generator efficiency : 98%
- Heat generated by core : 1500 MWt
- Primary pump work : 4.4 MW
- Secondary pump work : 1.8 MW
- Heat removed by RHRS : 4.6 MWt

Air, Helium, and Argon were considered as additional working fluids, and the main characteristics of each cycle using four different types of working fluids were investigated. The comparison results for the key parameters of each cycle are listed in Table 1. It was figured out that there are no significant differences in cycle efficiency for use of air and N_2 since those have similar nature of physical properties, and Helium and Argon show relatively low cycle efficiency.

Fig. 2 shows the cycle efficiency comparison with respect to the turbine inlet temperature of gas Brayton cycle for ASTRID and steam cycle according to working fluid. As mentioned above, Helium shows relatively lower cycle efficiency than other working fluids in the medium temperature range, but the highest cycle efficiency in the high temperature range.

Table 1. Comparison of key parameters of each cycle using different types of working fluids

Parameters	N_2	Air	He	Ar
Turbine inlet temperature. ºC	515.0			
Turbine inlet pressure, MPa	18.0			
Pressure ratio (LPC), -	1.38			
Pressure ratio (HPC), -	1.56			
Pressure ratio (Turbine), -	2.10			
LPC eff., %	89.0			
HPC eff., %	88.0			
Turbine eff., %	93.0			
Recuperator eff., %	95.0			
Flow rate, kg/s	4155.1	4265.2	728.1	6364.5
Turbine work, MWt	641.1	634.7	728.1	645.7
LPC work, MWt	142.8	140.3	183.3	149.5
HPC work, MWt	206.2	201.6	264.8	213.1
SGHX Q, MWt	751.0			
Precooler Q, MWt	294.6	293.89	295.2	284.2
Intercooler Q, MWt	164.4	164.4	175.8	183.7
Recuperator Q, MWt	1339.6	1347.4	839.7	794.3
Cycle eff., %	38.88	38.98	37.28	37.69
Plant eff.(Grpss), %	37.39	37.49	35.85	36.25
Plant eff.(Net), %	36.67	36.77	35.13	35.53

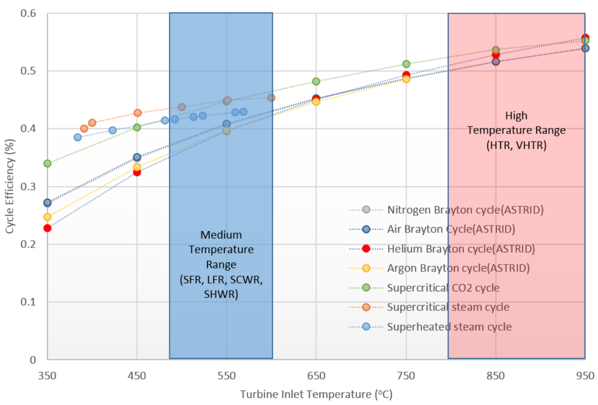
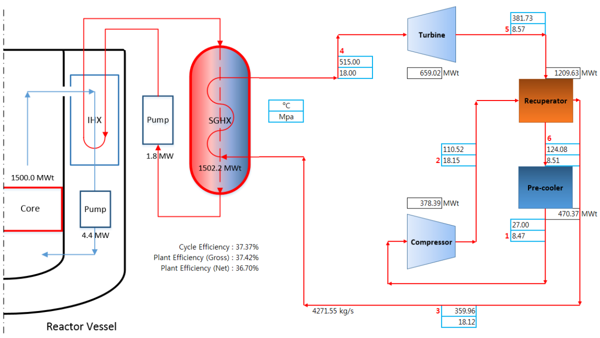


Fig. 2 Cycle Efficiency Comparison of Gas Brayton Cycle for ASTRID and Steam Cycle according to Working Fluid [5]

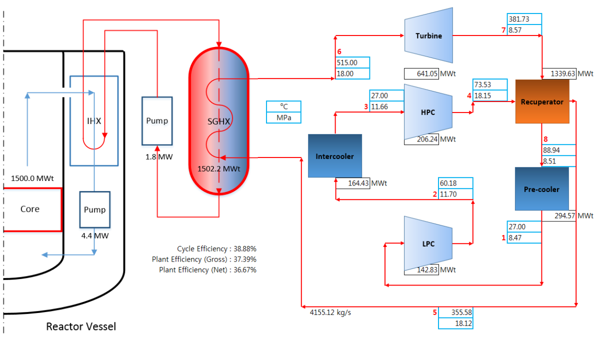
The cycle efficiency variations with respect to the cycle configuration were also investigated in ASTRID. All analyses were performed under the same assumptions and boundary conditions as before. 1C1T, 2C1T, 2C2T without reheater, and 2C2T with reheater were considered as cycle configurations. The comparison results for the key parameters of each cycle configuration are listed in Table 2 and corresponding thermodynamic state points at each cycle configuration are shown in Fig. 3. The cycle configuration with the highest cycle efficiency was found to be 2C2T with reheater. However, this is a result of a very ideal assumption that the second turbine inlet temperature is reheated to the first turbine inlet temperature by the reheater. Also, it was found that the cycle efficiency of 2C2T with reheater under the above ideal conditions is not significantly different from that of 2C1T. Thus, it can be seen that the most suitable cycle configuration in ASTRID is 2C1T.

Table 2. Comparison of key parameters of each cycle configuration

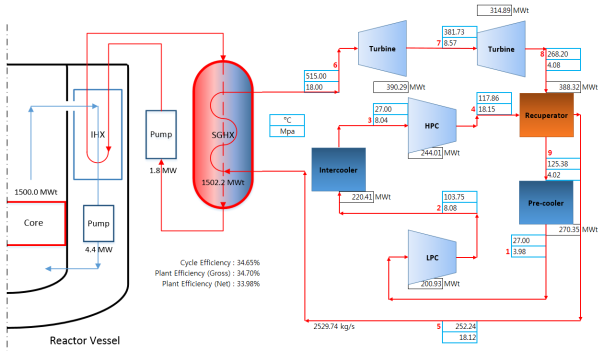
Parameters	2C2T	2C2T (w/o reheater)	2C1T	1C1T
FLOW RATE, kg/s	4187.9	2529.74	4155.22	4271.63
TURBINE INLET TEMP., °C	515	515	515	515
PRC1, -	2.04	2.03	1.38	2.14
PRC2, -	2.26	2.26	1.56	-
PRT1, -	2.1	2.1	2.1	2.1
PRT2, -	2.1	2.1	-	-
COMP1 EF., -	0.89	0.89	0.89	0.89
COMP2 EF., -	0.88	0.88	0.88	-
TURB1 EF., -	0.93	0.93	0.93	0.93
TURB2 EF., -	0.93	0.93	-	-
RECUP EF., -	0.95	0.95	0.95	0.95
TURB1 WORK, MWt	646.11	390.29	641.05	659.02
TURB2 WORK, MWt	628.35	314.89	142.83	-
COMP1 WORK, MWt	334.51	200.93	206.24	378.39
COMP2 WORK, MWt	403.95	244.01	-	-
Q_SGHX, MWt	751	751	751	751
Q_PC, MWt	473.11	270.35	294.57	470.37
Q_IC, MWt	366.87	220.41	164.43	-
Q_RH, MWt	624.99	-	-	-
Q_RC, MWt	1135.11	388.32	1339.63	1209.63
CYCLE EFFICIENCY, %	38.95%	34.65%	38.88%	37.37%
PLANT EFFICIENCY (GROSS), %	38.98%	34.70%	37.39%	37.42%
PLANT EFFICIENCY (NET), %	38.59%	33.98%	36.67%	36.70%
HEAT BALANCE, MWt	-0.00566	-0.00115	0.00547	-0.00548



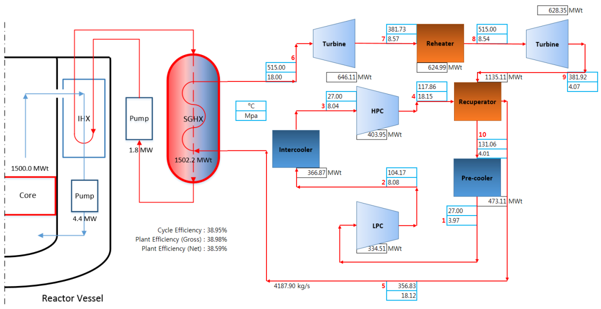
(a) 1C1T



(b) 2C1T



(c) 2C2T (w/o reheater)



(d) 2C2T

Fig. 3 Thermodynamic state points in each cycle configuration

3. Conclusions

Design characteristics of N_2 Brayton cycle power conversion system coupled with ASTRID were investigated by using the REGACY code. System performance of the specified N_2 Brayton cycle PCS according to variations of working fluids and cycle configurations were taken into account in the present study. The evaluation results from the present study are expected to be utilized to obtain the optimal system configuration with improved cycle efficiency of N_2 Brayton cycle PCS. The analysis results in terms of system thermal balance and cycle efficiency will be also used to obtain detailed design parameters of key components configuring N_2 Brayton cycle PCS such as heat exchangers as well as turbo-machineries.

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