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Study on CO_2 Recovery System Design in Supercritical CO_2 Cycle for SFR Application

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1. Introduction

As a part of Sodium-cooled Fast Reactor (SFR) development in Korea, the supercritical CO_2 (S-CO_2) Brayton cycle is considered as an alternative power conversion system to eliminate sodium-water reaction (SWR) when the current conventional steam Rankine cycle is utilized with SFR.

However, leakage in a turbo-machinery cannot be avoided because S-CO_2 power cycles are highly pressurized. The parasitic loss caused by the leakage flow should be minimized since this greatly influences the cycle efficiency. Thus, a simple model for estimating the critical flow in a turbo-machinery seal was developed to predict the leakage flow rate and calculate the required total mass of working fluid in a S-CO_2 power system to minimize the parasitic loss. In this work, study on CO_2 recovery system design was conducted by finding the suitable recovery point with the developed simple CO_2 critical flow model and sensitivity analysis was performed on the power system performance with respect to multiple CO_2 recovery process options.

2. CO_2 Recovery System Design

2.1 Seals for S-CO_2 Power Cycle

A suitable shaft seal technology is required to prevent the unnecessary leak of working fluid (CO_2) in turbo-machinery since the S-CO_2 power cycle is a highly pressurized system and fluid naturally flows from high pressure to low pressure. To apply the mechanical seal to the S-CO_2 Brayton cycle, following three aspects should be considered.

1) No interaction between seal material and CO_2
2) No causing of the CO_2 pollution
3) Long life time (no contact with shaft surface)

The Barber-Nichols Inc. which is the first mover in the S-CO_2 turbo-machinery area suggested two applicable mechanical seals: 1) Labyrinth seal, 2) Dry gas seal. They are most widely used mechanical seals in high speed rotating machines. The labyrinth seal is non-contact sealing action and it is composed of many grooves, so that the fluid has to pass through a long and difficult path to escape. Leakage amount is proportional to the gap area and inversely proportional to the tooth number. The general geometry of labyrinth seal is shown in Fig. 1. The dry gas seal or dry lift off seal is non-contacting, and dry-running mechanical face seal which consists of a mating (rotating) ring and a primary (stationary) ring. General geometry of the dry gas seal is shown in Fig. 2. When operating, lifting geometry in the rotating ring generates a fluid-dynamic force causing the stationary ring to separate and create a gap between the two rings. Although sealing performance of the dry gas seal is better than the labyrinth seal, the labyrinth seal was preferentially selected because it is easier to analyze the internal flow due to the geometry simplicity and it is more economically feasible.

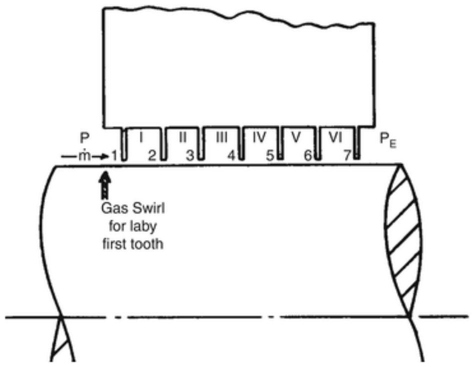

Fig. 1. General geometry of labyrinth seal [1]

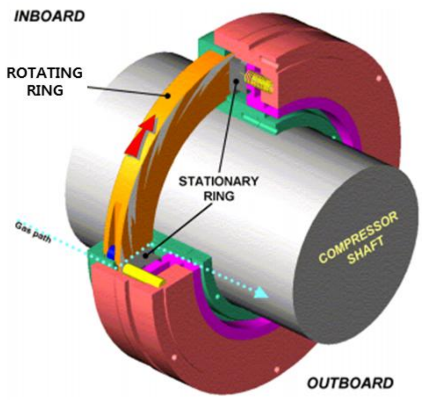

Fig. 2. General geometry of dry gas seal [2]

2.2 Seal Configuration

To calculate the exact mass flow rate of leakage in a turbo-machinery, selecting the seal configuration is important. GE Global Research in collaboration with Southwest Research Institute is working on development of a S-CO_2 turbo-expander for application to a S-CO_2 based power cycle for concentrated solar power (CSP) conversion [3].

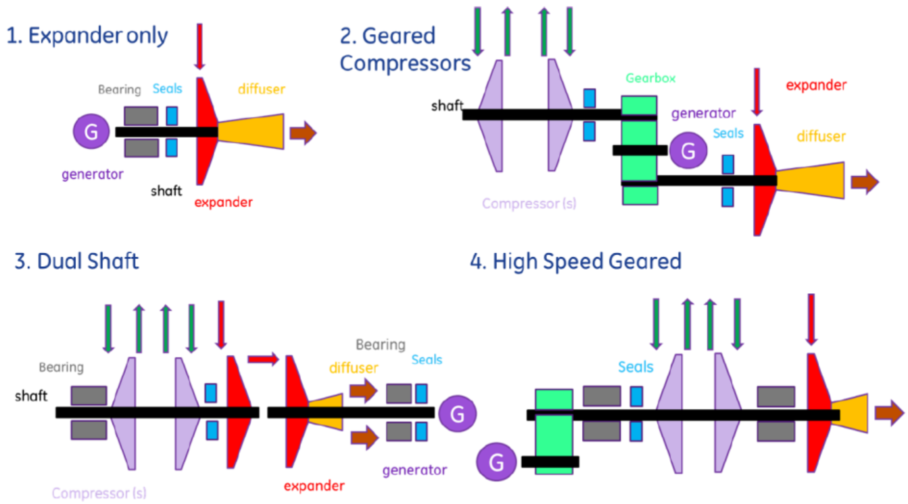

Fig. 3. Block schematic of the four feasible layouts for recompression CO_2 Brayton cycle based modular CSP power blocks [3]

The overall power block for CSP installation using recompression CO_2 cycle has the following rotating machinery: expander, main compressor, re-compressor, and generator. They organized these components in various different layouts and rotational speeds and the system configuration would provide different overall thermal conversion efficiencies for the same primary component designs. The block diagram schematic of 4 feasible designs is shown in Fig. 3. They are 1) 'expander only' - direct drive or geared turbo-generator with undefined motor driven compressor, 2) 'geared compressors' - geared compressor train with direct drive or pinion geared generator, 3) 'dual shaft' - a dual shaft concept with a single expander stage driving the compressors, while a second shaft with turbo-generator – direct drive or geared, and 4) 'high speed geared' - a single shaft concept with both expander and compressor train running at the same speed with a geared generator.

Firstly, the 'expander only' and 'high speed geared' designs are applied to the triple shaft design of the S-CO_2 recompression cycle.

2.3 Thermal Efficiency Loss with CO_2 Leak Rate and Recovery Point

For economics of the system, designing a process for CO_2 recovery to maintain the system mass at constant is important because this is directly connected to the cycle efficiency. Before calculating the CO_2 leak rate in a turbo-machinery, the analysis of thermal efficiency loss with CO_2 leak rate and recovery point was conducted. By applying the 'expander only' and 'high speed geared' designs to the triple shaft design for the S-CO_2 recompression cycle, three seal points were assumed. Initial pressure and temperature of seal leakage are the same as the inlet of turbine and the outlet of compressor.

The triple-shaft design of S-CO_2 recompression cycle for the SFR application in Korea is shown in Fig. 6. Through the cycle optimization, it was found that the cycle mass flow rate is 1190 kg/s and the optimal mass fractions to RT and to RC are 0.572 and 0.36, respectively. It is assumed that the leakage rate of each shaft is the same and the leakage rate is proportional to the mass flow rate through each turbo-machinery. The mixed enthalpy of the total leakage flow was calculated with the following equations.

$$h_{leakage} = \frac{1}{3}h_1 + \frac{1}{3}\{(\frac{0.572}{0.572+0.36})h_3 + (\frac{0.36}{0.572+0.36})h_{15}\}$$
$$+\frac{1}{3}\{(\frac{0.428}{0.428+0.64})h_5 + (\frac{0.428}{0.428+0.64})h_{12}\} \qquad (1)$$

The inventory recovery system which discharges the leakage to ambient and refills the CO_2 from a gas tank was considered and the thermal efficiency losses for obtained CO_2 leak rate and different recovery points were calculated. It is noted that the conditions of CO_2 tank were assumed to be 25 ℃, 7.0 MPa. The first candidate of the recovery point is on section 16 which is the cold side inlet of high temperature recuperator and the conditions of this section are 198.5 ℃, 19.9 MPa.

Considered losses are 1) $W_{net,loss}$ – loss due to the mass flow rate change of turbines and compressors, 2) $W_{comp,loss}$ – loss due to the pumping work of additional compressor. Fig. 5 (upper) shows the thermal efficiency loss for varying leak rate percent. When the leak rate percent is 1 %, the thermal efficiency loss was evaluated to be 2.15 %. This result re-confirms that the CO_2 inventory recovery system design is important to the cycle thermal efficiency and it is essential for considering economics of the cycle. The analysis for the thermal efficiency loss of recovery point in different section was also conducted by using the same calculation logic to find the best recovery point. The second and third candidates of recovery point are on section 9 and 10, respectively.

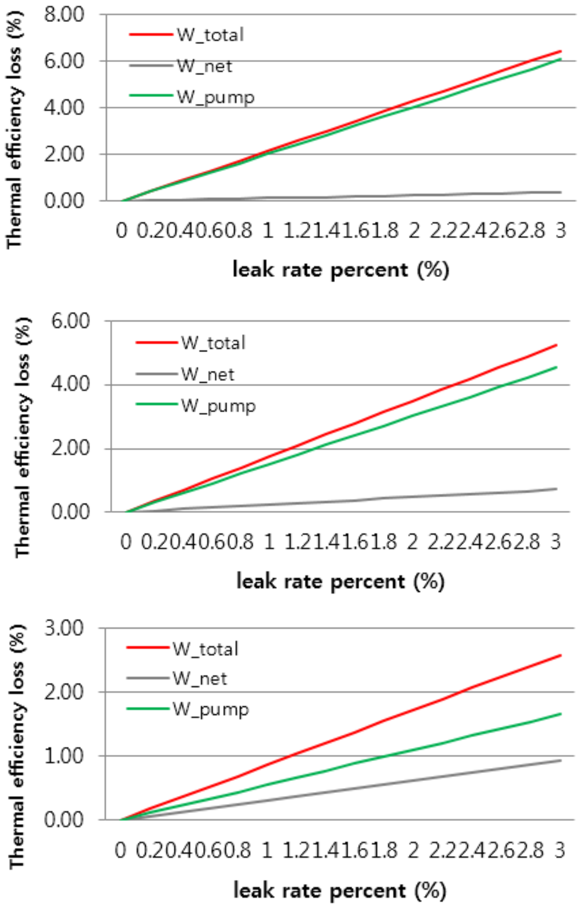

Fig. 5. Thermal efficiency loss with leak rate percent (lower) when recovery point on section 16 (upper), 9 (medium), and 10 (lower)

Section 9 is the hot side outlet of the low temperature recuperator and the conditions of this section are 92.9 ℃, 7.65 MPa. Section 10 is the inlet of the pre-cooler and the conditions of this section are 92.8 ℃, 7.62 MPa. Fig. 5 (medium) and Fig. 5 (lower) show the analysis results and they indicate that the pre-cooler inlet is the best point for the CO_2 inventory recovery point when the leakage was discharged to ambient and CO_2 from the gas tank was refilled.

2.4 Thermal Efficiency Loss with CO_2 Leak Rate and Recovery Point

For the next step, calculating the leak rate in turbo-machinery by using the CO_2 critical flow model was conducted to estimate how CO_2 inventory recovery system affects cycle thermal efficiency. To calculate the leak rate through the CO_2 critical flow model, the conditions of storage tank should be set. From the previous section, it was identified that the pre-cooler inlet is the best point for the CO_2 inventory recovery point in the S-CO_2 triple shaft recompression cycle. Therefore, the conditions of storage tank is set as 92.77 ℃ and 7.7 MPa. 92.77℃ is the same with the temperature of the pre-cooler inlet but 7.7 MPa is a bit higher than 7.62 MPa which is the same with the pressure of pre-cooler inlet to prevent the back flow. Previously, CO_2 critical flow model was designed to calculate the mass flow rate while changing the condition over time for comparison with experimental results. However, a calculation option assuming the tank size to be infinite for the high-pressure and low-pressure tanks were added to CO_2 critical flow model to reflect the real condition in a turbo-machinery. It was assumed that there is no pressure loss and heat loss in the connecting pipes from rotor cavity to the storage tank.

The GMN Inc. which is one of the major companies for seals indicated that the clearances of the labyrinth seals of turbo-machinery in power plant are about 3mm and 5mm when bore diameter is 100 mm and 200 mm, respectively [4]. Therefore, it was also assumed that clearances of labyrinth seal are 3 mm and 5 mm when shaft diameter are 100 mm and 200 mm, respectively. In this study, not only three seal points but also five seals are considered. Second case means that each turbo-machinery has one seal. The detailed calculation results of three and five seals are described in Table I.

Consequently, the minimum and maximum total mass flow rate of the leakage flow are 35.5 kg/s and 299.0 kg/s. It is noted that this is very conservative results since a real labyrinth seal has multiple tooth to minimize the leak which will have at least an order of magnitude less leakage flow rate value.

Table I: Calculation results of the leak rate in turbo-machinery (seal point: 5, clearance: 3 mm/5 mm) [upper], (seal point: 3, clearance: 3 mm/5 mm) [medium], and loss calculation result of net work and thermal efficiency [lower]

Storage tank	Leakage position	T (℃)	P (MPa)	G (kg/m^2-s)	m_{dot} (kg/s)
92.77 ℃ 7.7MPa	1. PT inlet	505.26	19.59	33646	16.1/53.5
	2. RT inlet	444.08	11.65	20243	9.7/32.2
	3. RC outlet	189.65	19.92	48493	23.2/77.1
	4. MT inlet	444.08	11.65	20243	9.7/32.2
	5. MC outlet	84.78	20	65383	31.3/104
Total mass flow rate					89.9/299.0

Storage tank	Leakage position	T (℃)	P (MPa)	G (kg/m^2-s)	m_{dot} (kg/s)
92.77 ℃ 7.7MPa	1. PT inlet	505.26	19.59	33646	16.1/53.5
	2. RT inlet	444.08	11.65	20243	9.7/32.2
	3. MT inlet	444.08	11.65	20243	9.7/32.2
Total mass flow rate					35.5/117.9

N of Seal point	$W_{net,loss}$ (MWe)	$\eta_{net,loss}$ (%)
5	3.657/12.155	1.90/6.31
3	2.204/6.724	1.05/3.49

2.5 CO_2 Recovery System Design

Through the CO_2 critical flow model, CO_2 leakage flow rate could be estimated. Since the inventory recovery system which discharges the CO_2 leakage to ambient and refills the CO_2 from the gas tank had relatively high thermal efficiency losses, another recovery method was considered to reduce the thermal efficiency losses due to the CO_2 recovery process. Fig. 6 shows a schematic of the preliminary CO_2 inventory recovery system design of the S-CO_2 power cycle for 75 MWe power module for SFR application. Unlike the previous method, it is not only simple and intuitive but also requires relatively very low additional compressing work. Moreover, it does not need additional compressor to compress liquid CO_2 from the CO_2 tank. The $W_{net,loss}$ of new simple inventory recovery system was calculated through the following equation.

$$W_{net,loss} = W_{net,design} - W_{net,new} \tag{2}$$
$$= W_{net,design} - (W_{turb,new} - W_{comp,new})$$

By adopting the newly proposed simple method, the minimum and maximum $W_{net,loss}$ are estimated to be 2.204 MWe and 12.155 MWe, respectively. It means that the thermal efficiency losses caused by CO_2 inventory recovery system may become 1.05 % to 6.31 % for very conservative leak estimation. To compare these results with the thermal efficiency losses when conventional leak rate is assumed, which the conventional leak rate is less than 1kg/s per seal, the following study was conducted.

The seal configurations and assumptions are the same as previous results. Consequently, the minimum and maximum $W_{net,loss}$ were estimated to be 0.147 MWe and 0.207 MWe, respectively. This means that the thermal efficiency losses caused by CO_2 inventory recovery system can range from 0.08 % to 0.11 % when conventional leak rate is used. Therefore, this proves that developing a good seal technology for the S-CO_2 power system operating conditions are very important for the overall system performance.

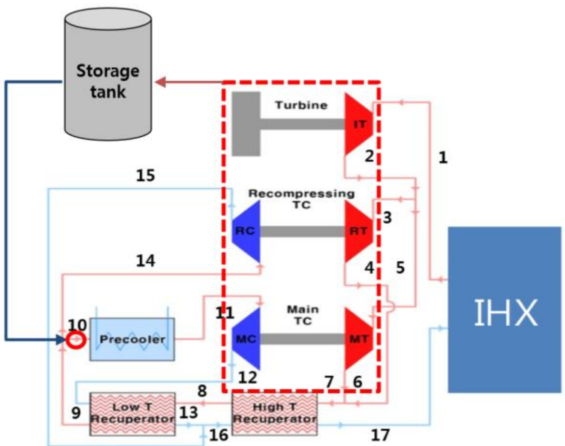

Fig. 6. Preliminary design on CO_2 inventory recovery system of S-CO_2 power cycle for 75 MWe power module for SFR application

3. Conclusions

The study of a CO_2 recovery system design was conducted to minimize the thermal efficiency losses caused by CO_2 inventory recovery system. For the first step, the configuration of a seal was selected. A labyrinth seal has suitable features for the S-CO_2 power cycle application. Then, thermal efficiency losses with different CO_2 leak rate and recovery point were evaluated. This study indicates that leakage management is very important to the cycle efficiency and pre-cooler inlet is the best location for the recovery point. To calculate the leak rate in turbo-machinery by using the developed CO_2 critical flow model, the conditions of storage tank is set to be closer to the recovery point. After modifying the critical flow model appropriately, total mass flow rate of leakage flow was calculated. Finally, the CO_2 recovery system design work was performed to minimize the loss of thermal efficiency. The suggested system is not only simple and intuitive but also has relatively very low additional work loss from the compressor than other considered systems. When each leak rate is set to the conventional leakage rate of 1 kg/s per seal, the minimum and maximum losses of thermal efficiency become 0.08 % to 0.11 %, which the values are very small. This again proves that the seal performance can be very important for maintaining high overall system performance.

Actually, the developed CO_2 critical flow model does not correctly reflect a labyrinth seal geometry effect. The real labyrinth seal has multiple tooth to further minimize the leak. Therefore, to upgrade the numerical model by applying the labyrinth seal geometry effect and conducting an experiment of a real labyrinth seal geometry nozzle will be performed.

NOMENCLATURE

S-CO_2: Supercritical carbon dioxide
SFR: Sodium-cooled fast reactor
SWR: Sodium-water reaction
CSP: Concentrated solar power
PT: Power generation turbine
RT: Recompressing turbine
MT: Main turbine
RC: Recompressing compressor
MC: Main compressor
IHX: Intermediate heat exchanger

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[2] John S. Stahley, Dry Gas Seal System Design Standards for Centrifugal Compressor Applications, The 31st Turbomachinery Symposium, 2002
[3] Chiranjeev Kalra et al., Development of High Efficiency Hot Gas Turbo-expander for Optimized CSP Supercritical CO_2 Power Block Operation, The 4th International

Symposium - Supercritical CO_2 Power Cycles, Pittsburgh, Pennsylvania, September 9-10, 2014
[4] GMN Inc. (German Manufacturer of High Precision Ball Bearings, Clutches, and Seals.): http://www.gmnbt.com/labyrinth-seals

1) How do you calculate the W_net,loss and W_comp,loss and is the h_leakage (Eq.1) necessary for calcaulating Ws?

Ans.: $W_{net,loss}$ is the work loss due to the mass flow rate change of turbines and compressors. Specifically, mass flow rate of turbo-machinery's inlet and outlet is changed since some amount of leakage is inevitable. The turbine work and compressor work are decreased due to the leakage but the proportion of reduced compressor work is smaller than that of turbine work. Therefore, changed net work is smaller than that of the on-design point. It is assumed that the leakage rate of each shaft is the same and the leakage rate is proportional to the mass flow rate through each turbo-machinery. Actually (Eq.1) shows the logic of calculation for the $W_{net,loss}$.

$W_{comp,loss}$ is the work loss due to the pumping work of additional compressor. The inventory recovery system which discharges the leakage to ambient and refills the CO_2 from a gas tank was considered. In this case, the conditions of CO_2 tank were assumed to be 25 ℃, 7.0 MPa. Therefore, $W_{comp,loss}$ is the additional pumping work to compress the working fluid from CO_2 tank condition (25 ℃, 7.0 MPa) to leakage point condition.

2) How do you calculated m_dot (Table I)

Ans.: Previously, CO_2 critical flow model was designed to calculate the mass flow rate while changing the condition over time for comparison with experimental results.

To simplify the expected CO_2 leak flow in a turbo-machinery, a simplified model for CO_2 leak flow simulation was constructed and shown in Figure 1.

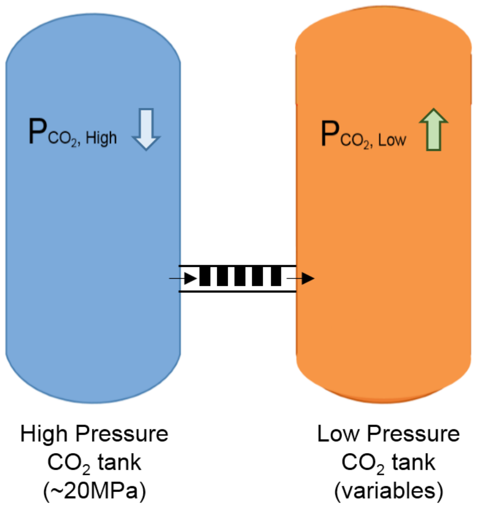

Figure 1 Conceptual diagram of the simplified model for a numerical analysis

The base calculation mechanism of CO_2 critical flow model for calculating the leak rate of CO_2 in a turbo-machinery is referred from the CO_2 leak model of Na-CO_2 heat exchanger in the S-CO_2 power cycle [1]. It was assumed that CO_2 flows through a nozzle from a high pressure CO_2 tank to a low pressure CO_2 tank, and the nozzle diameter plays the same role as the seal size.

For an isentropic flow, the frictional pressure loss and heat transfer are neglected thus the flow state can be easily calculated with the following governing equations (i.e. continuity equation, critical-pressure ratio equation, Mach number equation with pressure ratio, and mass flux equation from continuity equation):

$$G = \rho V_{elovity} = constant \tag{1}$$

$$\frac{P_0}{P_{critical}} = \left(1 + \frac{\gamma - 1}{2}\right)^{\gamma/(\gamma-1)} \tag{2}$$

$$M = \sqrt{\frac{2}{\gamma - 1}\left[\left(\frac{P_0}{P}\right)^{(\gamma-1)/\gamma} - 1\right]} \tag{3}$$

$$G = \frac{P_0}{\sqrt{RT_0}}\sqrt{\gamma}M\left(1 + \frac{\gamma - 1}{2}M^2\right)^{-\frac{\gamma+1}{2(\gamma-1)}} \quad (P_{critical} < P_{low-P\ tank}, \text{Unchoked flow case}) \tag{4.1}$$

$$G_{max} = \frac{P_0}{\sqrt{RT_0}}\sqrt{\gamma}\left(\frac{\gamma + 1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}} \quad (M_{exit} = 1.0) \ \ (P_{critical} \geq P_{low-P\ tank}, \text{Choked flow case}) \tag{4.2}$$

Based on the above governing equations, the critical pressure obtained from Eq. (2) is compared to the low pressure side at every time step. The choked condition is then checked. If the flow is not choked, Mach number is calculated from Eq. (3) and it is applied to Eq. (4.1). On the other hand, Eq. (4.2) with Mach number of unity is used to calculate the choked mass flux.

$$h_{high-P\ tank,t+1} = (m_{hP\ tank,t} * h_{hP\ tank,t} - m_{leak} * h_{hP\ tank,t})/m_{hP\ tank,t+1} \tag{5}$$

$$\rho_{high-P\ tank,t+1} = m_{hP\ tank,t+1}/V_{hP\ tank} \tag{6}$$

$$h_{low-P\ tank,t+1} = (m_{lP\ tank,t} * h_{lP\ tank,t} + m_{leak} * h_{lP\ tank,t})/m_{lP\ tank,t+1} \tag{7}$$

$$\rho_{low-P\ tank,t+1} = m_{lP\ tank,t+1}/V_{lP\ tank} \tag{8}$$

The mass flux is calculated in CO_2 critical flow model, then the amount of leaked CO_2 is obtained for each time step. Mass of CO_2 in the high pressure tank at next time step is calculated by subtracting the obtained amount of leaked CO_2 from mass of CO_2 in the high pressure tank at previous time step. This is applied to Eqs. (5) and (6). Therefore, pressure and temperature in high pressure tank at next time step can be obtained from the NIST database. Eqs. (7) and (8) show the enthalpy and density in low pressure tank at next time step, and pressure and temperature in the low pressure tank at next time step also can be obtained by using Eqs. (7) and (8). By updating the changed pressure and temperature of CO_2 in each time step, mass flux is calculated until both tanks reach equilibrium. A configuration of the nozzle is shown in Figure 2.

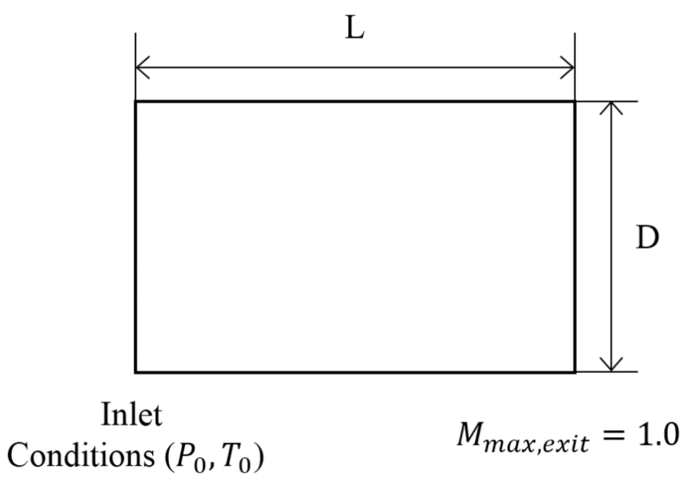

Figure 2 Configuration of nozzle for CO_2 critical flow model

To simplify the model, it was assumed that temperature and pressure of CO_2 at seal exit are at equilibrium with CO_2 in the low pressure tank. This assumption actually neglects expansion process of CO_2 at the nozzle exit although the CO_2 pressure at the exit is higher than that of CO_2 in the low pressure tank when the flow is choked.

A calculation option assuming the tank size to be infinite for the high-pressure and low-pressure tanks were added to CO_2 critical flow model to reflect the real condition in a turbo-machinery. It was assumed that there is no pressure loss and heat loss in the connecting pipes from rotor cavity to the storage tank.

3) The Eq.2 is now W_net, than what is old one.

Ans.: The logic of calculation is same as (Eq.1). The Eq.2 also indicates that changed W_{net} due to the leakage is smaller than that of the on-design point ($W_{net,design}$).

4) Can you show the reference of the conventional leakrate? Conventional leakrate is very very lower than the value that you calculated.

Ans.: Please refer to following reference.
http://www.gmnbt.com/m-series-labyrinth-seal.htm

5) Is Fig. 6 the best design, even if the conventional leakrate was used?

Ans.: It was confirmed that the newly proposed simple method (Fig.6) has relatively smaller $W_{net,loss}$ than the inventory recovery system which discharges the leakage to ambient and refills the CO_2 from a gas tank. However it does not mean that Fig.6 is the best inventory recovery system design. For future works, the study to identify the condition at which the storage tank is optimized for the CO2 recovery system will be conducted. Also, the study to identify whether the constant condition of storage tank is better or not will be performed.

6) What the RT, RC, PT, MT, MC, ... stand for ?

Ans.: Thank you for the comment. Authors added the following nomenclature.

S-CO_2: Supercritical carbon dioxide

SFR: Sodium-cooled fast reactor

SWR: Sodium-water reaction

CSP: Concentrated solar power

PT: Power generation turbine

RT: Recompressing turbine

MT: Main turbine

RC: Recompressing compressor

MC: Main compressor

IHX: Intermediate heat exchanger

7) References 1 and 2 is not referred in text.

Ans.: Thank you for the comment. References 1 and 2 are removed and mismatched number are revised.

*Reference

[1] Hwa-Young Jung. Preliminary Safety Studies of Sodium-CO_2 Heat Exchanger in SFR coupled with S-CO_2 Brayton Cycle. Thesis, KAIST (2015).