Preliminary Analysis of Coupling Supercritical CO_2 Cycle and Natural Draft Dry Cooling Tower for Small Modular Reactor Application

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1. Introduction

Due to a large amount of heat rejection, the site of commercial nuclear power plants has been limited to coastal and riverside areas that serve as an ultimate heat sink. However, as the severity of global warming and its damage rises, the importance of carbon-free fuels such as nuclear energy continues to increase. As the solution, small modular reactors (SMR) with a scale of less than 50% of commercial reactors are attracting attention. Thus, state-of-the-arts PWR-type SMR projects such as NuScale (USA) have been evaluating the application of the air-cooling systems.

In addition to the amount of cooling required, the temperature gradient of the heat exchanger is also an important factor affecting the feasibility of the air-cooling application. The isothermal profile of the steam condenser is accompanied by exergy losses due to large temperature differences in heat exchange with air.

Fortunately, the exergy loss issue is expected to be greatly mitigated by direct air cooling of the supercritical CO_2 (SCO_2) Brayton cycle [1]. In their study, a comparison of air-cooling performance between the steam Rankine cycle and SCO_2 Brayton cycle under the fixed cycle maximum temperature of 550°C and thermal power of 500MW. They also reported a six times less air flow rate required for cooling the SCO_2. However, the air-cooling feasibility of the SCO_2 cycle in the PWR-type SMR with a maximum temperature of 310°C is not clearly verified as the maximum temperature of PWR-type SMR is not within the proven range from 500 °C to 800 °C where the efficiency of the SCO_2 cycle is optimized.

In this study, a preliminary assessment on the feasibility of coupling SCO_2 cycle and NDDCT for PWR-type SMR was carried out. The amount of heat transferred by the natural draft dry cooling tower was calculated by a 1-dimensional numerical computing code. SCO_2 cycle performance with the results was also evaluated.

2. Method & Results

2.1 SCO2 recompression Cycle

The SCO_2 recompression cycle has improved performance compared to the Simple recuperated SCO_2 Cycle by using an additional compression process. In a simple recuperated cycle, the SCO_2 passed through the turbine is cooled through a recuperator and precooler before being compressed again. However, a pinch-point problem occurs due to the difference in specific heat according to the pressure of SCO_2 in the recuperator. The Recompression cycle resolves this specific heat mismatch with a SCO_2 mass split. It compresses a portion of the less cooled SCO_2, thereby reducing the mass flow rate of the cold part of the relatively high specific heat value.

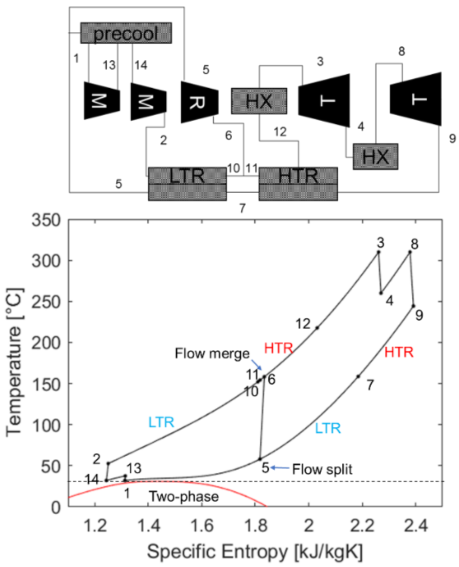

Fig. 1 SCO2 recompression cycle with intercooling and reheating [6]

Fig.1 shows the layout of recompression cycle. In the previous study of our research team [6], a SCO_2 recompression cycle analysis model and validation results were reported. Mass split ratio and pressure ratio of turbomachinery was optimized by given system maximum pressure and temperature.

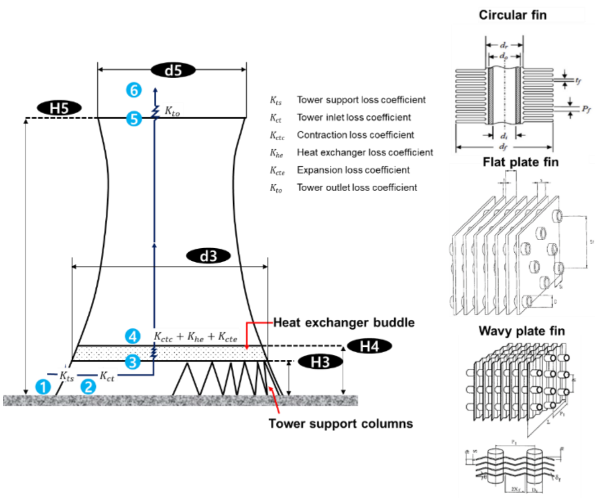

Fig. 2 Schematic diagrams of NDDCT [2], circular fins [3], flat plate fins [4], and wavy plate fins [5]

Table 1 Loss coefficients of NDDCT [2-5]

Loss coefficient	Correlations
Tower support loss coefficient	$K_{ts} = \frac{C_{Dts}L_{ts}d_{ts}n_{ts}}{\pi d_3 H_3}, K_{tshe} = \frac{C_{Dts}L_{ts}d_{ts}n_{ts}A_{fr}^2}{(\pi d_3 H_3)^3}\left(\frac{\rho_{a34}}{\rho_{a1}}\right)$
Tower inlet loss coefficient	$K_{ct} = 0.072\left(\frac{d_3}{H_3}\right)^2 - 0.34\left(\frac{d_3}{H_3}\right) + 1.7, K_{cthe} = K_{ct} * \left(\frac{\rho_3}{\rho_{34}}\right)\left(\frac{A_3}{A_{fr}}\right)^2$
Contraction loss coefficient	$K_{ctc} = \left(1 - \frac{2}{\sigma_c} + \frac{1}{\sigma_c^2}\right), K_{ctche} = K_{ctc} * \left(\frac{\rho_3}{\rho_{34}}\right)\left(\frac{A_3}{A_{fr}}\right)^2$ $\sigma_c = 0.6144517 + 0.04566493\,\sigma_a - 0.336651\,\sigma_a^2 + 0.4082743\,\sigma_a^3 + 2.672041\,\sigma_a^4 - 5.963169\,\sigma_a^5 + 3.558944\,\sigma_a^6$
Heat exchanger loss coefficient	For circular fin (Robinson and Briggs [3]) $K_{heC} = 2 * 18.93 n_r * Re_{d_o}^{-0.316} * \left(\frac{P_t}{d_o}\right)^{0.927} * \left(\frac{P_t}{P_d}\right)^{0.515} + \frac{2(\rho_{a4})}{\rho_{a4} + \rho_{a3}} * \left[\left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + 2K_{ci}^{0.5}\right] * \left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + \frac{2K_d \rho_{a3}}{\rho_{a4} + \rho_{a3}}$ Validated range: 1000<Re <18000, For flat plate fin (N.H. Kim et al. [4]) $\Delta P = f\frac{A}{A_c}\frac{G_c^2}{2\rho_{a34}} = \Delta P_{fin} + \Delta P_{tube}$ $\Delta P_{fin} = f_{fin}\frac{A_f}{A_c}\frac{G^2}{2\rho_{a34}}, \Delta P_{tube} = f_{tube}\frac{A_t}{A_{c,tube}}\frac{G^2}{2\rho_{a34}}$ $K_{heFP} = f\frac{A}{A_c} * \sigma_a^2 + \frac{2}{\sigma_a^2} * \frac{\rho_{a3} - \rho_{a4}}{\rho_{a3} + \rho_{a4}} + \frac{2(\rho_{a4})}{\rho_{a4} + \rho_{a3}} * \left[\left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + 2K_{ci}^{0.5}\right] * \left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + \frac{2K_d \rho_{a3}}{\rho_{a4} + \rho_{a3}}$ $f_{fin} = 1.455\, Re_D^{-0.656}\left(\frac{S_t}{S_l}\right)^{-0.347}\left(\frac{s}{D}\right)^{-0.134}\left(\frac{S_t}{D}\right)^{1.23}$

	$f_{tube} = \frac{4}{\pi}\left(0.25 + \frac{0.118}{[(S_t/D)-1]^{1.08}} Re_D^{-0.16}\right)[(S_t/D)-1]$ Validated range: 505 < Re <124707 For wavy plate fin (C.C Wang et al. [5]) $f = \frac{16.67}{ln^{2.64}(Re_D)} * \left(\frac{A_o}{A_t}\right)^{-0.096} * n_r^{0.098}$ $K_{WP} = \left(f * \frac{A_o}{A_c} * n_r * \left(\frac{\rho_{a34}}{\rho_{a3}}\right) + (1+\sigma_{aW}^2) * \left(\frac{\rho_{a3}}{\rho_{a4}} - 1\right)\right) * \left(\frac{\rho_{a34}}{\rho_{a3}}\right)\frac{1}{\sigma_{aW}^2}$ $K_{heWP} = K_{WP} + \frac{2(\rho_{a4})}{\rho_{a4}+\rho_{a3}} * \left[\left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + 2K_{ci}^{0.5}\right] * \left(\frac{1}{sin(\theta_{AF_m})} - 1\right) + \frac{2K_d\rho_{a3}}{\rho_{a4}+\rho_{a3}}$ Validated range: 400< Re <8000
Expansion loss coefficient	$K_{cte} = \left(1 - \frac{A_{e3}}{A_3}\right)^2, K_{ctehe} = K_{cte} * \left(\frac{\rho_{a34}}{\rho_{a3}}\right)\left(\frac{A_3}{A_{fr}}\right)^2$

2.2 1-D analysis of NDDCT

The natural draft dry cooling tower (NDDCT) is a non-powered heat rejection device without water evaporation loss. Unlike traditional wet cooling towers, the heat rejection process relies solely on ambient air to cool the working fluid. The air mass flow rate through the cooling tower is determined through the balance of buoyancy and flow resistance; tower support loss (path 1-2), tower inlet loss (path 2-3), contraction loss, heat exchanger loss, expansion loss (path 3-4). The tower outlet loss (path 5-6) was neglected in this preliminary study. The correlations for each loss coefficients are listed in **Table.1**[2]. The pressure balance equation of NDDCT is as follows:

$$P_{a1}\left[\left(1 - \frac{dT}{dz}\frac{\overline{H_{34}}}{T_{a1}}\right)^{3.5}\left(1 - \frac{dT}{dz}\frac{H_5 - \overline{H_{34}}}{T_{a4}}\right)^{3.5} - \left(1 - \frac{dT}{dz}\frac{H_5}{T_{a1}}\right)^{3.5}\right] = \left(K_{ts} + K_{ct} + K_{he(i)} + K_{ctc} + K_{cte}\right)_{he}\frac{\dot{m}_a^2}{2A_{fr}^2\rho_{a34}}\left(\left(1 - \frac{dT}{dz}\left(\frac{H_5 - \overline{H_{34}}}{T_{a4}}\right)\right)^{3.5} + \left(\frac{\dot{m}_a^2}{2A_{fr}^2\rho_{a5}}\right)\right) \quad (1)$$

The subscript $he(i)$ indicated the fin-types of heat exchangers as three types of fin types were selected as candidates: circular fins, flat plate fins, wavy fins.

2.3 Heat transfer modeling

The heat exchanger outlet temperature of air (point 4) is needed to solve the equation (1). The temperature could be calculated from heat transfer relations by the LMTD method between in-tube working fluid and ambient air.

$$Q_a = \dot{m}_a\Delta h_{air} = \dot{m}_{w.f}\Delta h_{w.f} \quad (2)$$
$$Q_{LMTD} = UAF_T\Delta T_{LM} \quad (3)$$

For air side heat transfer, empirical correlations of the finned tube studies are also available [3-5]. Empirical correlation of S.H Yoon et al. [7] is used for in-tube SCO_2 heat transfer.

$$Nu = aRe^bPr^c\left(\frac{\rho_{pc}}{\rho_b}\right)^n \quad (4)$$

$a = 0.14, b = 0.69, c = 0.66, n = 0$ when $T_b > T_{pc}$
$a = 0.013, b = 1, c = -0.05, n = 1.6$ when $T_b \leq T_{pc}$

Table 2 Validation data of NDDCT analysis code.

Parameters	Reference data [8]	Present study	err
Tower height	50.622m	50.622m	n/a
SCO_2 inlet temp.	71°C	71°C	n/a
Ambient air temp.	20°C	20°C	n/a
SCO_2 flow rate	406.6kg/s	406.6 kg/s	n/a
SCO_2 outlet temp.	40.3°C	39.75°C	0.55
Heat rejection rate	26.03MW	26.25MW	0.8%

Eshan et al. [8] designed direct SCO_2 dry air-cooling system and evaluated the cooling performance for the 25MWe scale concentrated solar power plant. Based on system parameters such as tower height and tower inlet conditions of their study, the present code is validated. Under the given input parameters, the amount of heat transferred and the SCO_2 outlet temperature were well matched.

Fig.3 shows the sample cooling tower analysis results according to tower height. The inlet conditions of SCO_2 were given from SCO_2 cycle analysis results of turbine inlet temperature 310°C. The wavy plate finned tube type heat exchanger with an A-frame arrangement of incline 60° shows superior heat removal performance even though the flow rate is relatively low.

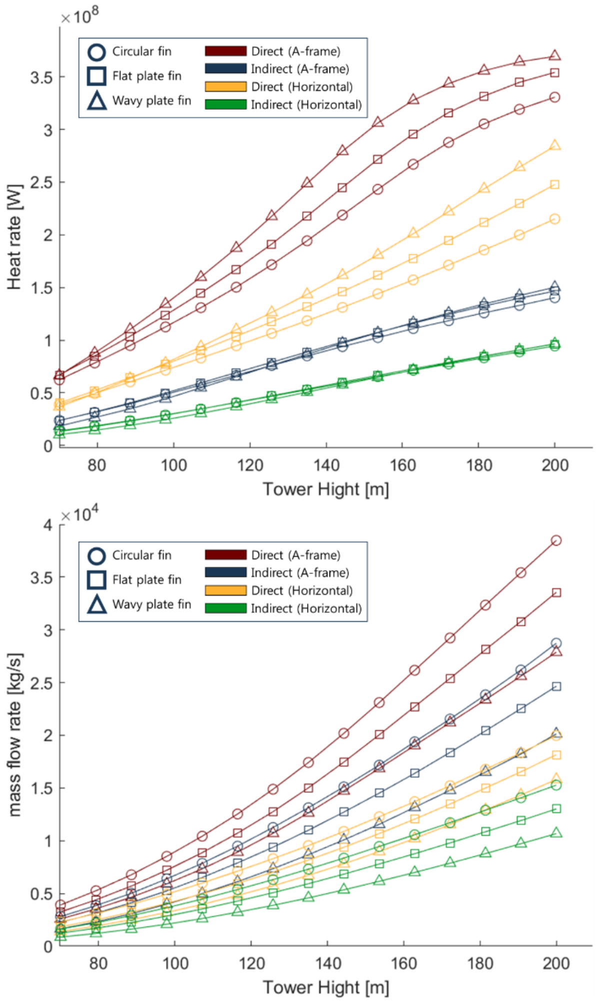

Fig. 3 Heat rejection rate and air mass flow rate of NDDCT at given inlet conditions.

Table 3 Input parameters of case calculation (**fig.3**).

Parameter	Value
SCO_2 inlet temp.	50.41°C
SCO_2 inlet pressure	8 MPa
Ambient air temp.	25°C
SCO_2 flow rate	2296 kg/s

2.4 Coupling SCO_2 recompression cycle with NDDCT

To couple with SCO_2 cycle analysis code with 1-D NDDCT analysis code, following algorithm was suggested. The SCO_2 cycle configuration was determined, and the overall cycle performance was evaluated according to the configuration. According to the SCO_2 inlet temperature and the geometry of the cooling tower, the cooling performance consistent with the results of SCO_2 cycle analysis was evaluated through iterative calculations.

As direct type wavy plate fine with A-frame arrangement shows the best cooling performance (**Fig.3**), the coupling analysis was conducted with the same heat exchanger configurations and the height of NDDCT was fixed by 200 m which is practical maximum height of the cooling towers. Considering the reactor coolant temperature of the PWR-type SMR, the maximum temperature of the SCO_2 cycle was assumed to be 310°C. And the compressor outlet pressure was considered in the range of 15 MPa to 25 MPa. In this preliminary study, single stage reheating strategy, which is available in the indirect heating cycle, was used for modifying the cycle performance.

Fig.4 shows the single stage reheating recompression cycle analysis results. According to the target cooling temperature, the maximum thermal and electrical power of the reactor were calculated based on the amount of heat rejection that can be accommodated in the 200 m NDDCT. The results of the calculation are summarized in **Fig.5**.

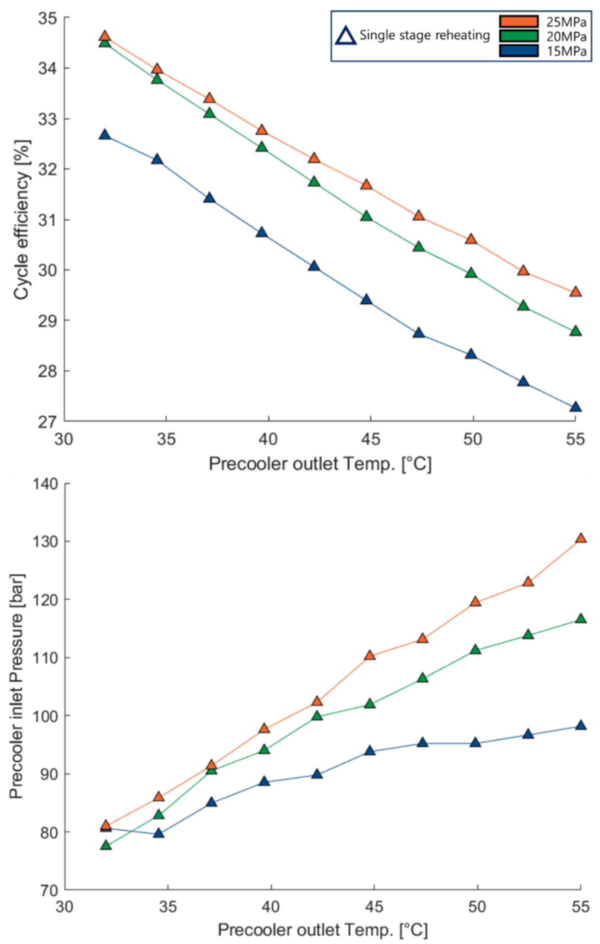

Fig. 4 SCO_2 cycle analysis results (cycle efficiency and tower inlet SCO_2 pressure)

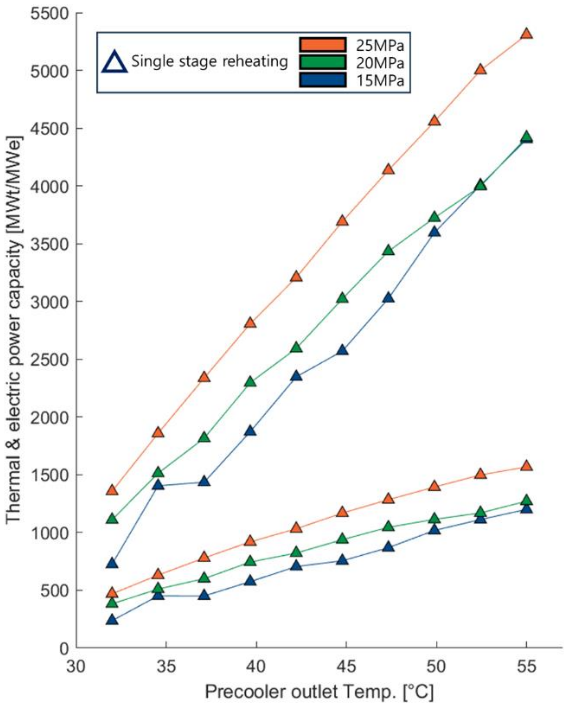

Fig. 5 Cycle analysis result according to precooler outlet temperature at 200m NDDCT height

Table 4 Reynolds number range of the coupling cycle analysis result (**fig.5**).

Reynolds number	Range
Present analysis	5135.4 – 8850.5
Wavy-plate fin validated range	400 – 8000

3. Conclusion

The feasibility of coupling NDDCT and the SCO_2 Brayton cycle with PWR-type SMR was confirmed using 1-D numerical code. Three types of finned tube heat exchangers (circular fins, flat plate fins, wavy fins) were compared and the coupled system performance was evaluated. The acceptable rejected heat and expected electrical output was evaluated by cooling performance of 200 m high NDDCT. The main finding of this research can be summarized as follows:

- ✓ Wavy plate fin type heat exchanger shows superior performance than circular or flat plate fin type under the same pitch of tubes.
- ✓ The results of the case study on the 200 m high NDDCT shows that the NDDCT can accommodate the thermal power of the reactor from about 1000MW to about 5000MW with no usage of evaporation water.
- ✓ These results implies that a sufficient amount of heat can be rejected by a practically sized NDDCT in PWR-type SMR if the SCO_2 cycle is applied.

Acknowledgments

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