Effects of TTD and DCA on Cycle Efficiency and Heat-Transfer Requirements of PWR-Type SMR Power Cycle

Takhyun Chun [a], Gyudong Kim [a], Jeong Ik Lee [a*]
[a]*Department of Nuclear and Quantum Engineering, Korea Advanced Institute of Science and Technology (KAIST), 291 Daehak-ro, Yuseong-gu, Daejeon 34141, Republic of Korea*
[*]*Corresponding author: jeongiklee@kaist.ac.kr*

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1. Introduction

Small modular reactors (SMRs), with their compact and modular configurations, have been proposed as nuclear power systems with greater siting flexibility, modular construction capability, and reduced scale of major plant components [1]. To realize these advantages, compactness should be considered not only in the reactor coolant system but also in the secondary power-conversion system. In particular, the thermal design of major secondary-system heat exchangers should achieve an appropriate balance between cycle performance and heat-transfer requirements.

In a PWR-type SMR power cycle, regenerative feedwater heating plays an important role in determining the feedwater condition entering the steam generator (SG). The feedwater heaters (FWHs) recover thermal energy from turbine extraction steam and increase the SG inlet feedwater temperature. At the same time, the thermal design conditions of the FWHs determine the heat-transfer conductance required to achieve the intended feedwater heating. Therefore, the design of the regenerative feedwater-heating system should be evaluated from both cycle-performance and component-level heat-transfer perspectives.

Among the principal design parameters of FWHs, the terminal temperature difference (TTD) and drain cooler approach (DCA) characterize the thermal performance of FWHs. Reducing TTD or DCA can enhance feedwater heating and improve cycle efficiency. However, smaller design values consequently increase the required heat-transfer conductance and equipment size. Therefore, the selection of TTD and DCA involves a trade-off between thermodynamic performance and the heat-transfer requirements of the secondary system, which is particularly relevant to compact PWR-type SMR designs.

Accordingly, this study investigates the effects of TTD and DCA on the net cycle efficiency, and heat-transfer requirements of the FWHs and SG. 600 MW_{th} PWR-type SMR Rankine cycle with three closed FWHs was analyzed for 16 combinations of TTD and DCA. Under fixed SG thermal power, primary-side boundary conditions, secondary-side outlet conditions, the high-pressure turbine pressure ratio was optimized to maximize the net cycle efficiency. From this approach, the sensitivity of cycle performance and component heat-transfer requirements to TTD and DCA were assessed.

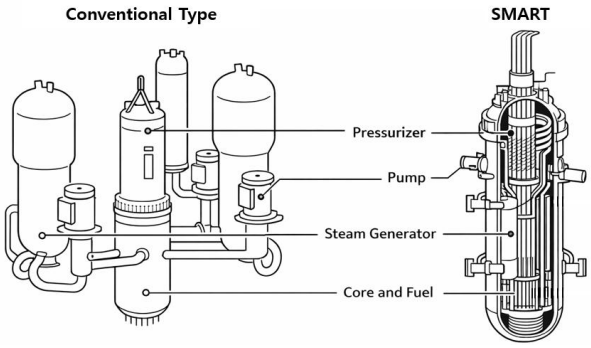

Fig. 1. Comparison of the configurations of conventional PWR (left) and SMART-PWR type SMR (right) [1]

2. SMR Power Cycle

2.1 Reference cycle

Fig. 2 presents the schematic configuration of the SMR power conversion system adopted in this study. The secondary-side cycle was developed with reference to the NuScale steam-cycle design [2], which represents a Rankine cycle configuration for a PWR-type SMR. Based on this reference, a hypothetical cycle model was established by explicitly representing the principal components, including the turbine stages and feedwater heaters, to describe the complete power conversion system.

The steady-state operating conditions, design parameters, and cycle performance are summarized in Table I. Using the established steady-state solution, the thermodynamic states at the turbine extraction points were determined, and the corresponding heat-transfer characteristics were evaluated based on the local steam properties.

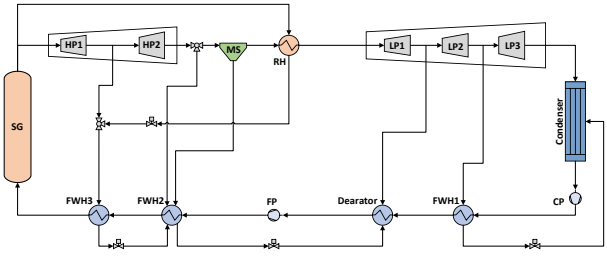

Fig. 2. Layout of SMR used in this study

Table I. Cycle parameters/conditions for on-design reference cycle

Cycle Conditions	Value
SG thermal power	600 MWth
SG outlet pressure	5.82 MPa
SG outlet temperature	304.8 ℃
SG inlet temperature	220.1 ℃
Condenser outlet temperature	40 ℃
Feedwater mass flow rate	305.4 kg/s
Cycle Parameters	Value
Turbine efficiency	90%
Pump efficiency	80%
Generator efficiency	95%
HPT work	77.8 MW
LPT work	140.6 MW
Pump work	2.49 MW
Net efficiency	34.2 %

2.2 Cycle analysis for on-design point

In this study, an in-house MATLAB code was developed to establish the Rankine cycle under specified SG boundary conditions and FWH design conditions. The SG thermal input was fixed at 600 MW_{th}, while the primary-side inlet and outlet conditions, secondary-side main-steam pressure, and design hot-end pinch temperature difference were prescribed. The main-steam outlet temperature was consequently determined from the specified primary-side inlet temperature and SG pinch condition.

For each combination of TTD and DCA, the high-pressure turbine (HPT) pressure ratio was treated as a design variable. The two HPT stages were assumed to have the same pressure ratio. For a given HPT pressure ratio, the turbine expansion states and extraction pressures were first determined. The closed FWHs were then solved by simultaneously satisfying the specified TTD and DCA conditions and the component energy balances, while the deaerator was modeled using an open-heater mixing energy balance. The required extraction flow rates were calculated accordingly.

The resulting outlet condition of the final high-pressure FWH determined the feedwater state entering the SG. The main steam/feedwater mass flow rate was then calculated from the fixed SG thermal duty.

$$\dot{m}_{SG,FW} = \frac{Q_{SG}}{h_{SG-steam,out} - h_{SG-FW,in}} \quad (1)$$

Since the extraction flow rates depend on the main steam flow rate, the turbine, FWH, and SG calculations were iterated until the mass-flow and component energy balances converged. For the LPT, equal stage pressure ratios were assumed between the reheater outlet and the condenser pressure.

After convergence at each candidate HPT pressure ratio, the turbine work, pump work, net electric output, and net cycle efficiency were evaluated. The calculation was repeated over the prescribed HPT pressure-ratio range, and the pressure ratio yielding the maximum net cycle efficiency was selected as the on-design solution for each TTD–DCA combination. The SG and FWH heat-transfer requirements were subsequently evaluated from the converged thermodynamic states as design metrics rather than as constraints in the cycle optimization.

Fig. 3 presents the on-design T–s diagram of the SMR power cycle considered in this study. The major thermodynamic processes include regenerative feedwater heating through the FWH train, steam generation and superheating in the SG, staged expansion through the HPT and LPT, and heat rejection in the condenser.

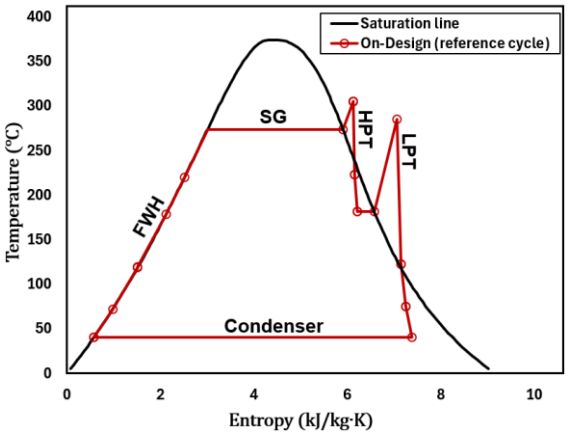

Fig. 3. Steady-state T-s diagram of the SMR reference cycle in this study

2.3 Parameters of FWHs

The terminal temperature difference (TTD) and drain cooler approach (DCA) are key design parameters and thermal performance indicators of feedwater heaters (FWHs). During operation, the water level and operating conditions of an FWH are controlled so that the TTD and DCA remain close to their design values or the target values corresponding to a given load condition [3]. Maintaining appropriate TTD and DCA values is important not only for achieving the intended thermal performance but also for maintaining a proper water level within the heater. Improper water level may result in steam ingress into the drain cooling zone, drain flashing, erosion, and vibration, which can adversely affect long-term component reliability.

TTD is defined as the difference between the saturation temperature of the extraction steam and the feedwater outlet temperature. A higher TTD may

therefore indicate degradation in heat-transfer performance under comparable operating conditions.

DCA is defined as the difference between the drain outlet temperature and the feedwater inlet temperature. An increase in DCA relative to its design value may indicate degraded drain cooling performance caused by factors such as improper water level, insufficient flow, or internal fouling. A lower DCA generally improves regenerative feedwater heating and can increase cycle efficiency. However, achieving a lower DCA requires a larger heat-transfer area and may therefore increase the size and cost of the FWH.

$$TTD = T_{sat-steam,in} - T_{FW,out} \quad (2)$$

$$DCA = T_{drain,out} - T_{FW,in} \quad (3)$$

Typical FWH designs employ TTD values of approximately 2.8–4.4 °C and a DCA of approximately 8.3 °C [4]. For FWHs used in nuclear power plants without a de-superheating zone, minimum values of approximately 1.11 °C for TTD and 5.56 °C for DCA have been suggested [3]. In addition, the NuScale PWR design, which was used as a reference for establishing the cycle configuration in this study, employs TTD and DCA values of 2.9 °C and 5.7 °C, respectively.

Based on these values, the NuScale design conditions were adopted as the reference case, and the ranges of TTD and DCA were selected to investigate their effects on the SMR power cycle. As summarized in Table II, four TTD values and four DCA values were considered, resulting in a total of 16 combinations for the on-design analysis.

Table II. Investigated TTD and DCA values

Parameters	Value			
TTD	1.5	*2.9	3.5	4.5
DCA	5.6	*5.7	6.5	7.5

* Reference values of TTD and DCA for FWH [2]

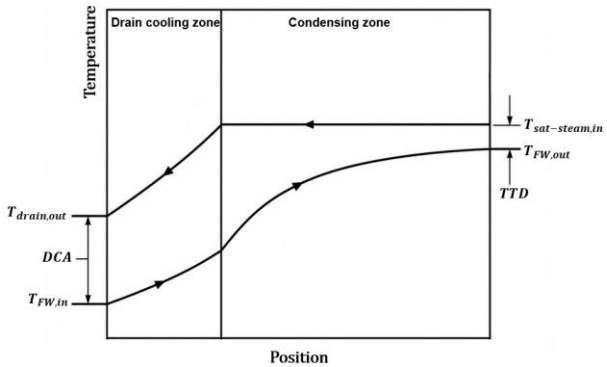

Fig. 4. Temperature distribution of closed FWH with drain cooling zone and condensing zone

3. UA Evaluation

Following the on-design cycle analysis, the required heat-transfer conductance, (UA), was evaluated for the SG and FWHs. The (UA) value represents a key thermal sizing parameter of a heat exchanger. This value was used to quantify the change in heat-transfer requirements associated with different TTD and DCA conditions. Therefore, the effects of TTD and DCA were evaluated in terms of not only cycle performance but also the required (UA) of the SG and FWHs.

Since both the SG and FWHs involve phase-change processes, their temperature profiles include regions in which the fluid temperature remains constant at the saturation temperature. Accordingly, each heat exchanger was divided into individual heat-transfer regions, and a lumped (UA) approach was applied to each region based on its inlet and outlet thermodynamic conditions.

$$(UA)_{zone} = \left(\frac{Q}{\Delta T_{LMTD}}\right)_{zone} \quad (4)$$

$$(UA)_{total} = \sum (UA)_{zone} \quad (5)$$

For SG, the heat-transfer process was divided according to the feedwater heating, evaporation, and superheating regions, whereas FWH was divided into the drain cooling and condensing regions. This approach allows the contribution of each heat-transfer region to the total (UA) to be identified.

For the reference cycle, the UA distributions of the SG and FWHs are shown in Fig. 5. The phase-change region accounted for the dominant portion of the total UA in both the SG and FWHs, and similar distributions were observed for the other TTD and DCA cases considered in this study. However, FWH2 exhibited a noticeably lower fraction of condensing-zone UA than FWH1 and FWH3. This difference was primarily attributed to the relatively large heat-transfer requirement of the drain-cooling zone. In FWH2, the drain flow from the higher-pressure FWH3 is combined with the moisture from moisture separator (MS), resulting in a relatively large liquid flow through the drain-cooling region. In addition, the extraction steam supplied to FWH2 has a lower quality than that supplied to FWH3 because of the preceding turbine expansion. These combined effects increase the sensible-heat recovery in the drain-cooling zone and consequently reduce the relative contribution of the condensing zone to the total UA.

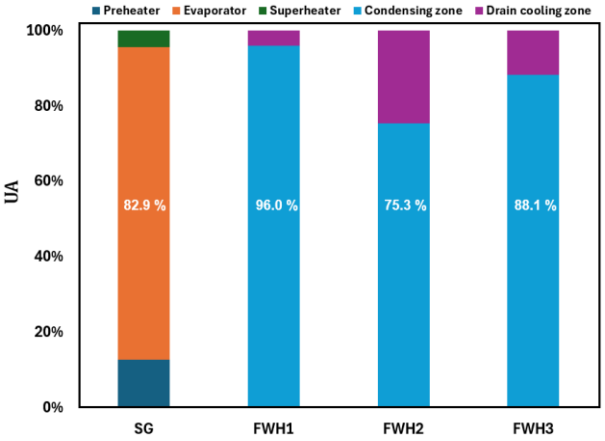

Fig. 5. Distribution of UA of each heat-transfer regions in SG and FWHs for reference conditions

4. Results and Discussions

Under the conditions investigated in this study, TTD had a more significant influence than DCA on both cycle efficiency and heat-transfer requirement. At a fixed DCA of 5.7 K, decreasing TTD from 4.5 to 1.5 K increased the SG inlet feedwater temperature from 218.65 to 221.33 °C and the net cycle efficiency from 34.133% to 34.221%, as shown in Fig. 6.

The corresponding total UA of FWHs increased from 9.89 to 15.20 MW/K, representing an increase of approximately 54%, as shown in Fig. 7. In contrast, the UA of SG increased only from 26.50 to 26.97 MW/K, or approximately 1.8%, as shown in Fig. 8. These results indicate that the heat-transfer requirement is considerably more sensitive to TTD in the FWHs than in the SG.

DCA had a comparatively smaller effect. At a TTD of 2.9 K, increasing DCA from 5.6 to 7.5 K reduced the net cycle efficiency from 34.181% to 34.162% and the total FWH UA from 11.77 to 11.54 MW/K. The SG inlet temperature and SG UA remained nearly unchanged over the investigated DCA range, indicating that the influence of DCA was mainly confined to the heat-transfer requirement of the FWHs.

Overall, decreasing TTD and DCA improved cycle efficiency but increased the heat-transfer requirement, with this trade-off being much more pronounced for TTD. The highest net cycle efficiency of 34.222% was obtained at TTD = 1.5 K and DCA = 5.6 K. However, the corresponding total FWH UA was 15.22 MW/K, approximately 29% higher than the reference value of 11.76 MW/K. Compared with the reference condition, reducing TTD to 1.5 K improved the net cycle efficiency by only 0.042 percentage points while increasing the required FWH UA by approximately 29%.

Although TTD produced a clear change in FWH thermal requirements, the overall cycle efficiency remained relatively insensitive to both TTD and DCA. In the present analysis, the SG thermal input and secondary-side outlet conditions were fixed, and a constant on-design turbine efficiency of 90% was assumed. For each TTD–DCA combination, the main-steam flow rate and HPT pressure ratio were adjusted to identify the maximum-efficiency operating point. Since variations in turbine efficiency with mass flow rate and pressure ratio were not considered, changes in the optimized cycle efficiency were governed primarily by variations in the main-steam flow rate. The optimized main-steam flow rate varied by less than 1% over the investigated cases. Consequently, the resulting changes in turbine work and net cycle efficiency also remained relatively small. The limited variation in the main-steam flow rate also resulted in only a small change in the SG feedwater flow rate. In addition, the SG inlet feedwater temperature do not have significant variations. These led to a relatively small variation in the required UA of SG.

The fixed HPT inlet pressure and the limited variation in the optimized HPT pressure ratio also resulted in relatively small changes in the turbine extraction pressures. Therefore, variations in TTD were reflected directly in the feedwater outlet temperatures of the FWHs. To satisfy the FWH energy balances under these conditions, the extraction-steam flow rates were changed. The extraction flows to FWH1, FWH2, and the deaerator (DA) exhibited variations of up to approximately 10%, whereas the extraction flow to FWH3 varied by less than approximately 2%.

The smaller sensitivity of the FWH3 extraction flow can be explained by the sequential arrangement of the regenerative heating system. For FWH1 and FWH2, the feedwater inlet temperatures are primarily determined by the condenser-side condition and the DA outlet condition, respectively. A reduction in TTD consequently increases the corresponding FWH outlet temperature and requires a noticeable adjustment in extraction-steam flow. In contrast, the inlet temperature of FWH3 is the outlet temperature of FWH2. When TTD is reduced, both the inlet and outlet feedwater temperatures of FWH3 increase simultaneously. As a result, the change in the required feedwater heating duty of FWH3 is relatively small, leading to a smaller variation in its extraction-steam flow.

The limited influence of DCA on overall cycle performance can also be explained by its role in the FWH energy balance. While the FWH feedwater outlet temperature is primarily determined by the extraction-steam saturation temperature and TTD, DCA directly determines the drain outlet temperature. Therefore, variations in DCA mainly alter the sensible heat recovered from the condensate in the drain-cooling zone. Since the drain flow rate is relatively small and the sensible heat of the drain is considerably smaller than the latent heat released by the condensing extraction steam, changes in DCA have only a limited effect on the overall regenerative heating duty and cycle performance.

In contrast, UA of FWH showed a pronounced sensitivity, particularly to TTD. While the heat-transfer duty, Q, changed slightly, the corresponding ΔT_{LMTD} varied substantially with TTD. Since the

extraction steam in the condensing zone remains close to its saturation temperature, reducing TTD directly decreases the temperature difference between the condensing steam and the feedwater, thereby reducing ΔT_{LMTD}. Consequently, the required FWH UA increased markedly as TTD decreased, even though the resulting improvement in cycle efficiency remained relatively small.

It should also be noted that the present analysis assumes constant turbine efficiency regardless of changes in mass flow rate and pressure ratio. In an actual turbine design, variations in these operating parameters can alter stage efficiency. Incorporating such design-dependent turbine efficiency characteristics may therefore modify the sensitivity of the overall cycle to TTD and DCA. Future work will incorporate these turbine characteristics and extend the analysis to off-design conditions to provide a more realistic assessment of the coupled FWH and turbine design.

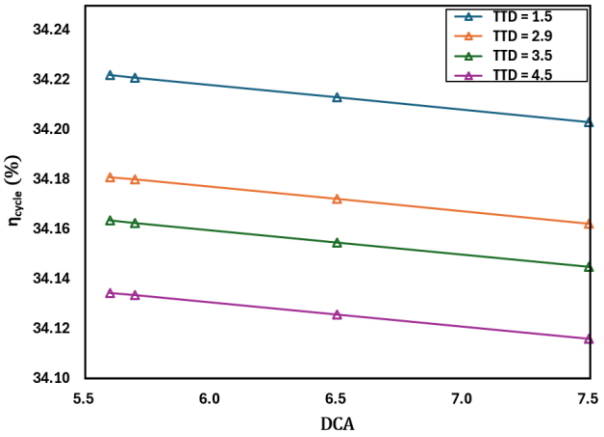

Fig. 6. Effect of TTD and DCA on net cycle efficiency

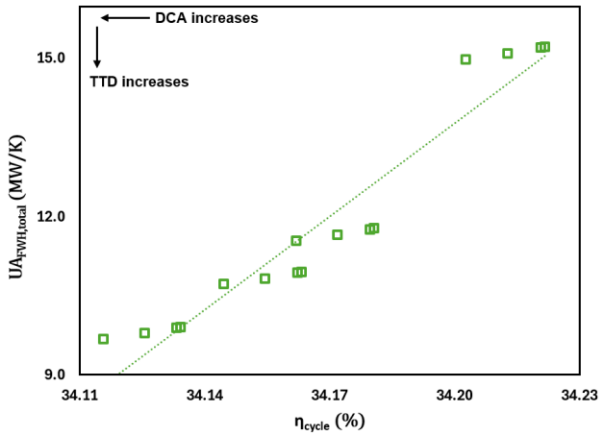

Fig. 7. Relationship between net cycle efficiency and total UA of FWH

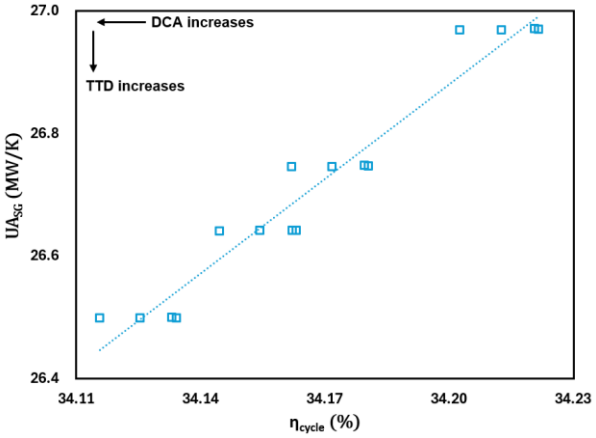

Fig. 8. Relationship between net cycle efficiency and UA of SG

5. Summary and Conclusions

This study investigated the effects of TTD and DCA on the on-design performance and heat-transfer requirements of a hypothetical 600 MW_{th} PWR-type SMR power cycle with three closed FWHs. 16 TTD–DCA combinations were analyzed under fixed SG boundary conditions. For each case, the HPT pressure ratio was optimized to maximize the net cycle efficiency. And the required UA values of the SG and FWHs also were evaluated using a heat-transfer analysis.

The overall cycle performance was relatively insensitive to both TTD and DCA within the investigated range. This limited sensitivity resulted from the small variations in the optimized main-steam flow rate and turbine expansion conditions under the fixed SG boundary conditions and constant turbine-efficiency assumption. In contrast, the FWH heat-transfer requirement was strongly affected by TTD. Reducing TTD decreased the temperature difference in the condensing region and therefore substantially increased the required FWH UA. DCA had a comparatively smaller influence because its effect was mainly confined to sensible heat recovery in the drain-cooling region. The SG heat-transfer requirement also remained relatively insensitive to both parameters.

In this study, TTD and DCA have only secondary effects on overall cycle efficiency under the present assumptions, whereas TTD is a major parameter governing FWH thermal sizing. Therefore, reducing TTD solely to obtain slight efficiency improvement may impose relatively large increase in FWH heat-transfer requirements. Future work will incorporate design-dependent turbine efficiency and extend the analysis to off-design conditions to assess whether these sensitivities are maintained during load-following operation.

ACKNOWLEDGEMENTS

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REFERENCES

[1] Giorgio Locatelli, “Small modular reactors: A comprehensive overview of their economics and strategic aspects”, Progress in Nuclear Energy, Vol 73. 2014. 75-85p
[2] U.S. NRC, Preliminary Safety Analysis Report (PSAR), Chapter 10 – Steam and Power Conversion System, U.S. Nuclear Regulatory Commission, Washington, D.C.
[3] EPRI, Level Control Guide for Feedwater Heaters, Moisture Separator/Reheaters, and Other Equipment, Electric Power Research Institute, Palo Alto, CA, 2002.
[4] EPRI, Life Cycle Management Sourcebooks—Volume 10: Feedwater Heaters, Electric Power Research Institute, Palo Alto, CA, 2003.