

Thermal Sizing of Multi-Module PCSG for a Light-Water SMR

Taeil Kim^a, Nakjun Choi^b, UngJin Na^a, Jeyeong Cheon^b, Sungkun Chung^c, Sungjin Kwon^d and HangJin Jo^{a,b*}

^aDepartment of Mechanical Engineering, POSTECH, Pohang, 790-784, Republic of Korea

^bDivision of Advanced Nuclear Engineering, POSTECH, Pohang, 790-784, Republic of Korea

^cKorea Atomic Energy Research Institute, Daejeon, 989-111, Republic of Korea

^dKorea Institute of Fusion Energy, Daejeon, 34133, Republic of Korea

*Corresponding author: jhj04@postech.ac.kr

***Keywords : PCSG, Thermal sizing, Multi-module, SMR**

1. Introduction

The Printed Circuit Steam Generator (PCSG) has been regarded as a promising candidate for substituting conventional steam generators due to its compactness and high heat transfer performance. However, its practical design is constrained by manufacturing limitations: the diffusion bonding press used to fabricate the PCSG has a certain size limit in the lateral, transverse, and axial dimensions. The size limit in the lateral and transverse directions restricts the cross-sectional area of an individual PCSG, which requires multiple modules to be arranged in a parallel configuration. Likewise, the size limit in the axial direction restricts the length of an individual PCSG, necessitating a series configuration to satisfy the target thermal duty. This study develops a thermal sizing code for PCSG modules arranged in both parallel and series configurations. As a representative application, a hypothetical light-water SMR is considered, and its operating conditions are summarized in Table I.

Table I: Target SMR operating condition

Parameter		Value
Thermal duty		200 MW _{th}
Reactor coolant system	Mass flow rate	997.5 kg/s
	Inlet temperature	325 °C
	Outlet temperature	290 °C
	System pressure	15 MPa
Steam generator	Mass flow rate	111.52 kg/s
	Inlet temperature	232.2 °C
	System pressure	6 MPa

2. Thermal sizing code

A thermal sizing code was developed to design the PCSG that meets the target thermal duty under the imposed conditions. To simplify the analysis, only a semicircular channel with a single-banking configuration was considered. Counterflow was adopted because it can maximize the temperature difference along the channel. Since the hot and cold sides are arranged periodically in both the lateral and transverse directions, the thermal sizing problem can be reduced to a one-dimensional thermal resistance model, as shown in Fig.1. The thermal resistance network consists of the

hot side channel, the solid wall, and the cold side channel. For each domain, heat transfer, pressure drop, and thermophysical properties are accounted for by discretizing the unit cell in the axial direction.

In the thermal sizing code, total mass flow rate, temperature, and pressure are utilized as inlet boundary conditions. The diameter and number of channels are fixed as input data, and the PCSG length is initialized with a certain value. Each discretized node is treated as a control volume, in which the outlet conditions of the preceding node are used as the inlet conditions of the subsequent node. The thermal resistances of the hot side, solid wall, and cold side are calculated for each node. The heat flow is obtained using the LMTD method, and the energy balance is enforced by matching the enthalpy change between the inlet and outlet of the node with the heat flow. As shown in Fig. 1, the calculation propagates from the hot side inlet (cold side outlet) to the hot side outlet (cold side inlet); therefore, the cold side outlet conditions must be initially guessed. Once the calculation reaches the hot side outlet, the cold side inlet condition is updated based on the residuals of enthalpy and pressure, and this process is iterated until the residuals fall below a certain limit. Based on the discrepancy between the thermal duty and heat flow accumulated over the entire nodes, the initially assumed PCSG length is iteratively updated until the target thermal duty is satisfied within $\pm 0.05\%$. The determined PCSG length is compared against the axial size limit to determine whether a series configuration is required, or a single module is sufficient in the axial direction.

Table II summarizes the thermal-hydraulic packages used in the thermal sizing code [1-4]. In the hot side, the flow remains in the subcooled liquid, so only whether the flow is laminar or turbulent needs to be determined. In contrast, in the cold side, the flow starts as subcooled liquid, transitions to saturated two-phase, and finally becomes superheated vapor. Furthermore, the onset of nucleate boiling (ONB) and subcooled boiling heat transfer must also be considered. In this sense, a dimensionless group (Re , Bo , Ja , Co , x_e) is calculated for each node to determine the regime and the corresponding correlation. To handle the extensive discretization and iterative calculations, the code was implemented in Python, with fluid properties evaluated using the CoolProp library.

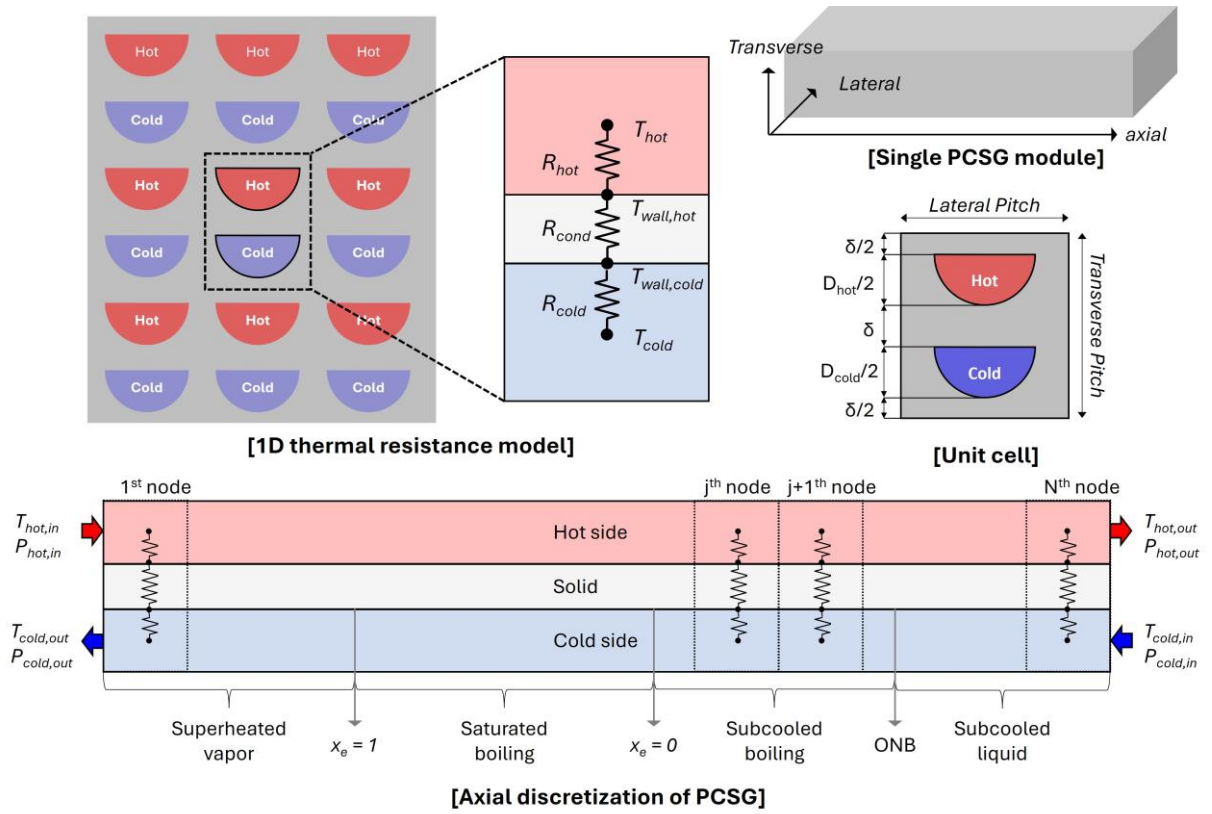


Fig. 1. Thermal resistance network and node discretization of the PCSG

Table II: Thermal-hydraulic packages

Phenomena	Correlation
steam heat transfer	McAdams (1950)
saturated boiling heat transfer	Kim and Mudawar (2013)
subcooled boiling heat transfer	Guo et al. (2023)
turbulent flow heat transfer	Gnielinski (1976)
laminar flow heat transfer	Shah and London (1978)
ONB superheat	Bergles and Rohsenow (1964)
two-phase frictional and accelerational pressure drop	Lockhart-Martinelli (1949) and Chisholm (1967)
turbulent flow pressure drop	Blasius equation
SS316L thermal conductivity	Kim (1975)

3. Results and discussions

The channel diameter and solid wall thickness of the unit cell were set to 4 mm and 1 mm, respectively. The manufacturable size limit of the PCSG was adopted from [5], in which the lateral, transverse, and axial dimensions are specified as 0.6, 0.6, and 1.5 m, respectively. Based on these geometric constraints, a single PCSG was configured with a 100×100 array of unit cells. Thermal sizing was conducted over a wide range of parallel configurations, and the results for the

configuration with six parallel branches are presented in Fig. 2 as a representative case.

The temperature variation on the hot side is moderate compared to that on the cold side because the flow rate is much larger on the hot side (Fig. 2a). The cold side shows a sharp variation in the subcooled liquid and superheated vapor regions, and a negligible variation in the saturated two-phase region. Fig. 2b shows the axial distribution of the fluid pressure. As expected, the pressure drop on the hot side is much larger than that on the cold side. The pressure on the cold side undergoes a sharp decrease in the saturated two-phase region because of acceleration and frictional pressure drops. The axial heat flux distribution on the hot side is presented in Fig. 2c. The heat flux is calculated to be the largest at the location where the saturated two-phase flow transitions to superheated vapor on the cold side. The thermal resistances of the hot side, solid wall, and cold side are decomposed in Fig. 2d. Throughout the axial direction, the thermal resistance of the solid wall is dominant, except in the intervals where the flow on the cold side is in the superheated vapor state. The location at which subcooled boiling occurs on the cold side shows the lowest thermal resistance. Fig. 2d shows a relatively flat thermal resistance distribution in the subcooled liquid regions on the cold side, whereas a few cases (10–12 parallel branches) show a laminar-turbulent transition in the liquid region, resulting in a large variation in thermal resistance.

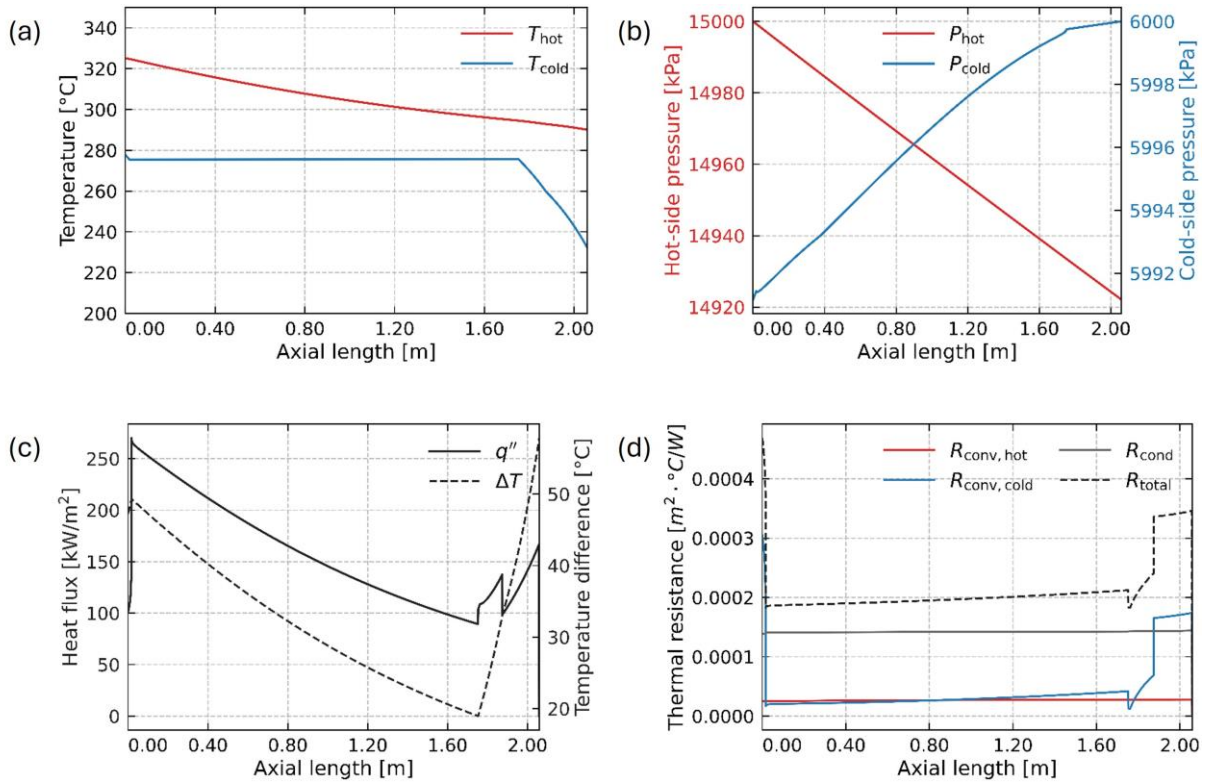


Fig. 2. Thermal sizing results for the configuration with six parallel branches: axial distributions of (a) fluid temperature, (b) fluid pressure, (c) heat flux and temperature difference, and (d) thermal resistance. The required PCSG length is 2.059 m.

As presented in Fig. 2d, the thermal resistances of the hot side and solid wall are nearly constant along the axial direction, indicating that the total thermal resistance is primarily governed by the cold side. Therefore, the determined PCSG length is decomposed into the subcooled liquid, subcooled boiling, saturated boiling, and superheated vapor regions based on the heat transfer regime on the cold side (Fig. 3a). The saturated boiling region occupies 83.8–85.3% of the PCSG length, followed by the subcooled liquid region (6.5–10.0%), and the subcooled boiling region (4.0–8.7%). The superheated vapor region accounts for merely 1% of the PCSG length due to the small thermal capacity of the vapor state. As the number of parallel branches increases, the portions of subcooled liquid and saturated boiling regions decrease, whereas those of the subcooled boiling and superheated vapor regions increase.

According to Fig. 3b, the required PCSG length decreases significantly with the increasing number of parallel branches. For the configuration with three parallel branches, a PCSG length of 3.689 m is necessary to satisfy the target thermal duty. Since the manufacturable axial length is limited to 1.5 m, each parallel branch must consist of three modules connected in series. For configurations with 4–8 parallel branches, two modules connected in series are required for each branch. Beyond nine parallel branches, a series configuration is no longer necessary, and a single module is sufficient for each branch. The total number

of PCSG modules is minimized at eight for the configuration with four parallel branches, each consisting of two modules connected in series.

The total PCSG volume can be obtained by multiplying the PCSG length by the number of parallel branches (Fig. 3c). The total volume exhibits a monotonic increase with an increasing number of parallel branches. The effectiveness of increasing the number of parallel branches diminishes to some extent, because the PCSG length does not decrease proportionally.

The pressure drops on the hot side and cold side are summarized in Fig. 3d. It is important to note that these results do not include the minor losses caused by the header and flow connecting sections, and the actual pressure drop would be much larger. Therefore, from a conservative point of view, the allowable pressure drop was set to 100 kPa and 80 kPa for the hot and cold sides, respectively. This indicates that below six parallel branches, the pressure drop on the hot side exceeds the allowable value, and the corresponding configuration is no longer available. In contrast, the pressure drop on the cold side remains much lower, with a maximum value of 47.9 kPa. However, the current analysis assumes a single geometric set (i.e., a fixed channel diameter and single-banking configuration), and other geometric sets should also be examined to identify the optimal design of the multi-module PCSG.

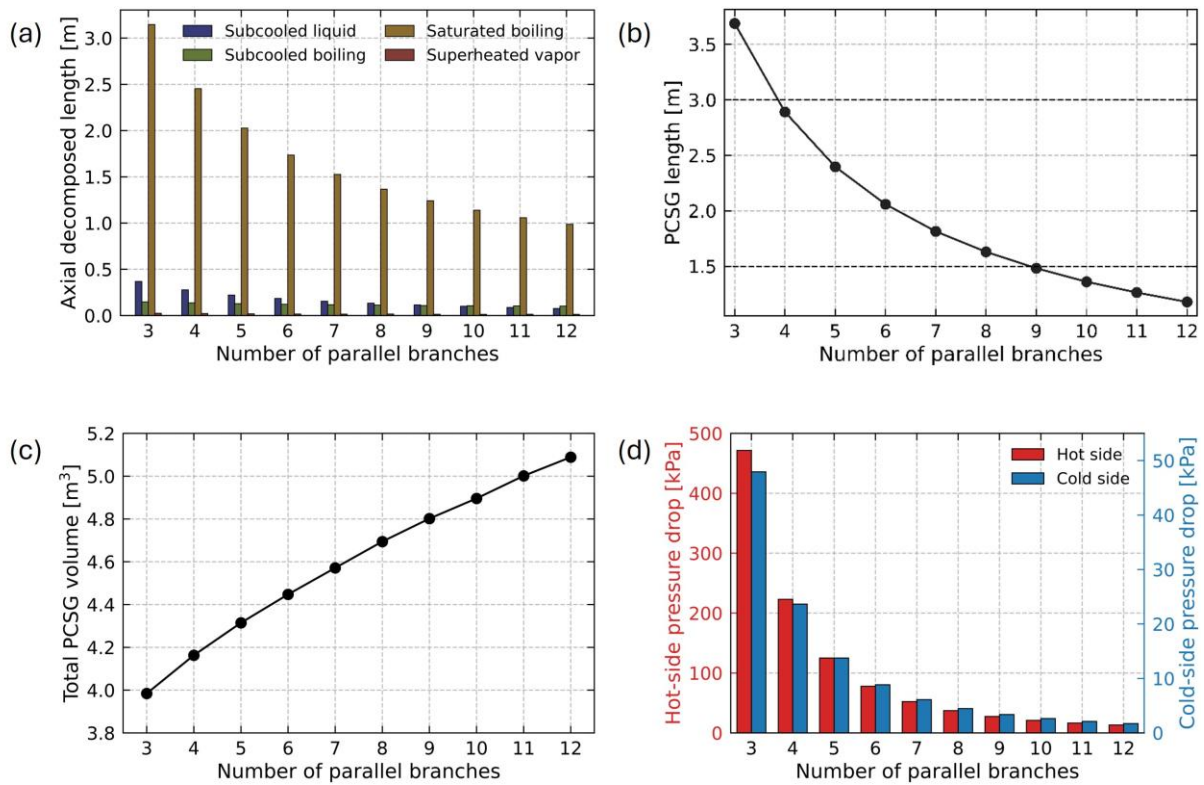


Fig. 3. The effect of the number of parallel branches on (a) the axial decomposed length, (b) PCSG length, (c) total PCSG volume, and (d) pressure drops on the hot and cold sides.

4. Conclusions

A one-dimensional thermal sizing code was developed by modeling the relevant thermal-hydraulic phenomena in the PCSG. Thermal sizing of a multi-module PCSG was conducted for a hypothetical SMR with a thermal duty of 200 MW_{th}. Both parallel and series configurations were considered over a wide range, and the number of modules, the total volume of the PCSG, and the pressure drop were determined as key design parameters. Since the code takes the target thermal duty and operating conditions as inputs, it can be applied to the thermal sizing of a multi-module PCSG for any light-water SMR.

5. Acknowledgement

This research was supported by the National Research Council of Science & Technology (NST) grant by the Korea government (MSIT) (No. GTL25031-000). Additional support was provided by the Institute for Korea Spent Nuclear Fuel (iKSNF) and Korea Foundation of Nuclear Safety (KOFONS) grant funded by the Korea Government (Nuclear Safety and Security Commission, NSSC) (No. RS-2021-KN066120).

REFERENCES

[1] McAdams, W., W. Kennel, and J. Addoms, Heat transfer to superheated steam at high pressures, Transactions of the

American Society of Mechanical Engineers.Vol. 72(4), pp. 421–428, 1950

[2] Kim, S.-M. and I. Mudawar, Universal approach to predicting saturated flow boiling heat transfer in mini/micro-channels–Part II. Two-phase heat transfer coefficient, International Journal of Heat and Mass Transfer.Vol. 64, pp. 1239–1256, 2013

[3] Guo, P., et al., Experimental investigation on heat transfer of subcooled flow boiling of water in mini channels under high heat fluxes, Experimental Thermal and Fluid Science.Vol. 142, pp. 110831, 2023

[4] Kim, C.S., Thermophysical properties of stainless steels, Argonne National Laboratory (ANL), 1975

[5] Nestell, J. and S. Sham, ASME code considerations for the compact heat exchanger, Oak Ridge National Laboratory (ORNL), 2015