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Steady-State Performance Analysis of Pressurizer and Helical Steam Generator for SMART

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ABSTRACT

System-Integrated Modular Advanced Reactor (SMART), where major primary components such as modular helical steam generator and self regulating pressurizer are integrated into reactor vessel, is currently under development. The pressurizer is designed to control the primary pressure mainly with partial pressure of nitrogen gas and to maintain the fluid temperature as low as possible for the purpose of minimizing steam contribution. The steam generator (SG) is designed to produce super-heated steam inside tube at power operation. Because the in-vessel pressurizer and in-vessel SG are classified as the characteristic components of SMART, it is important to perform a steady state calculation of these components in order to evaluate the adoption of these components. A steady state analysis of the in-vessel pressurizer and in-vessel SG has been performed under normal power operation and the results show an acceptable performance of the components.

I. Introduction

SMART is an integral-type reactor producing thermal power of 330 MWt [1]. SMART adopts new design features in components such as helical once-through SGs, canned-type main circulation pumps (MCPs), and self regulating pressurizer installed inside the reactor vessel. Major design parameters are shown in Table 1. In Fig. 1, in-vessel SG, in-vessel pressurizer with cooler, and MCP are presented schematically.

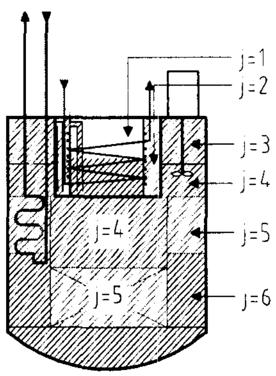

Fig. 1 Schematic geometry of primary system with nodalization for S/S analysis

Table 1 Major design parameters of SMART

nominal thermal power of the reactor, MWt	330.0
pressure in primary circuit, nominal/design, MPa	15.0/17.0
coolant temperature at nominal power operation, outlet/inlet, °C	310.0/270.0
coolant flow rate via the core, kg/s	1556.0
number of MCPs	4
steam output, kg/s	152.4
nominal steam header pressure maintained by control system, MPa	3.3
feed water temperature, °C	190.0

The in-vessel pressurizer is divided into three (3) compartments as shown in Fig. 2. Table 2 shows the comparison of pressure behavior in the pressurizer for the cases of using gas only and using steam-gas according to the change of programmed power levels. The steam in the pressurizer is the main contributor of pressure change at high temperature. So, the pressurizer should maintain fluid temperature in the pressurizer as low as possible in order to minimize pressure changes due to steam. For this purpose, wet thermal insulator and cooler are installed in the pressurizer. Heat exchanger is cooled with water supplied from equipment cooling system. Table 3 shows design data for cooler assumed in this analysis.

Table 2 Pressure behavior with temperature change

Power level (%)	0	20	40	60	80	100
Core exit temp. (°C)	290	294	298	302	306	310
Pressure. in gas PZR (MPa)	14.49	14.59	14.69	14.79	14.90	15.0
Pressure. in steam-gas PZR (MPa)	12.40	12.88	13.37	13.90	14.43	15.0

Table 3 Major design parameters of cooler for analysis

outer/inner cooler tube diameter, m	.018/.015
number of tubes	6.0
tube length of one turn /two turns/three turns per one tube, m	6.6/13.2/19.8
cooling water pressure at cooler inlet, MPa	0.5
total flow rate of cooling water, kg/s	3.4

The reactor system adopts once-through helical SG where secondary coolant flowing inside tube is heated along the tube length. Thermal sizing calculation for SG is carried out to determine whether required power and superheating of steam are produced. Heat transfer surface is arranged into tubes in the form of 12 cylindrical steam generating cassettes. These cassettes are located in the reactor vessel annulus formed by the core barrel outer shell and inside of the reactor vessel mid-section. Each steam generating cassette consists of tubes with the feed water supply and superheated steam removal units, throttling devices, and an outer casing. The throttling device is installed at inlet of each SG tube for the purpose of providing hydrodynamic stability of the SG for the whole range of operating conditions [2].

In this steady state performance analysis, modeling of in-vessel pressurizer with cooler and wet thermal insulator in conjunction with modeling of primary system including SG are developed in order to determine nitrogen mass, temperature distributions in each cavity of pressurizer and primary circuit, and water level in end cavity are performed for three cases of tube lengths of cooler. To evaluate performance of in-vessel pressurizer with cooler and wet thermal insulation, the whole primary system including pressurizer is nodalized into several control volumes depending on different temperature zones as shown in Fig. 1. The SG and core are further nodalized to obtain detailed distributions of parameters such as temperature. Piping lines penetrating reactor vessel and flow restricting devices are modeled appropriately. Major geometric data used in the analysis are summarized in Table 4.

Table 4 Major geometric data of SMART

height of reactor vessel, m	10.0
inner diameter of reactor vessel, m	3.6
inventory of primary coolant, kg	46318.5
volume of end/intermediate/upper annular cavity, m^3	12.46/1.54/7.70
height of end/intermediate/upper annular cavity, m	4.2/4.2/1.34
water mass remained in end/intermediate/upper annular cavity at cold condition, kg	873.4/78.0/308.4
outer/inner/average helicoiling SG tube diameters, mm	12.0/9.0/470.0
average length of one SG tube, m	14.4
number of SG cassettes/number of tubes per SG cassette	12.0/330.0
lateral/longitudinal pitch for SG tube bundle, m	.0165/.0135
SG tube roughness, mm	.03
SG tube material	titanium alloy PT-7M
flow resistance coefficient of SG orifice	50.0
flow resistance coefficient of SG header	5.0
thickness of wet thermal insulation, m	.02
number of layers of wet thermal insulation	20.0
bypass flow to upper annular cavity, kg/s	60.0

II. Modeling of Components Related to Analysis

II-1 Basic Equation

Mass continuity equation and momentum equation are as follows:

$$\dot{m} = \rho_j v_j A_j = \rho_{j+1} v_{j+1} A_{j+1} \quad (1)$$

$$-\frac{dP}{dl} = \frac{d}{dl}\left(\frac{G^2}{\rho}\right) + \left[K_j \frac{Pe_j}{A_{surj}} + \frac{f}{D} + \xi_i\right]\frac{G^2}{2\rho}\eta_{2\Phi} + \rho g \sin\theta \quad (2)$$

where,

$$\xi_i = \frac{0.1}{\pi D_{wind}} \quad (3)$$

$$\eta_{2\Phi} = 0.5\left(\frac{v_{2\Phi}}{v_{1\Phi}} - 1\right) + 1 \quad (4)$$

$\dot{m}$ ρ, v, A are mass flowrate, density, velocity, and cross-sectional area, respectively. K_j is an empirical irreversible form loss coefficient. G is mass flux and f is the friction factor for pipe. Pe, l, D are perimeter of tube, flow-directional length, and diameter of tube, respectively. θ is the angle of flow direction relative to the reference coordinate of +z. ξ_i is the correction coefficient for flow in helicoil tube. $\eta_{2\Phi}$ is the correction coefficient to account for the two-phase flow. Thermal sizing equation is shown in each modeling of components.

II-2 Modeling of Total Water Mass Conservation For Primary Circuit

$$M_{1,TOTAL} = M_{PZR} + \sum_{j \neq PZR} M_j = CONSTANT \quad (5)$$

$$M_{PZR} = M_{PZR1} + M_{PZR2} + M_{PZR3} \quad (6)$$

PZR1, PZR2, and PZR3 refers to the three pressurizer cavities as shown in Fig. 2 and named end cavity(EC), intermediate cavity(IC), and upper annular cavity(UAC) respectively.

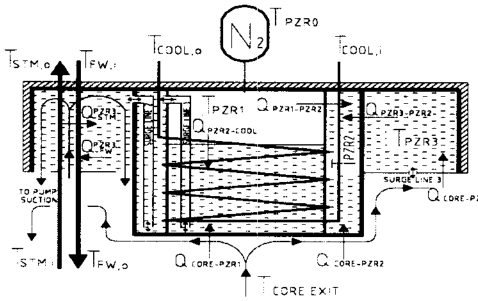

Fig. 2 Analysis model of pressurizer

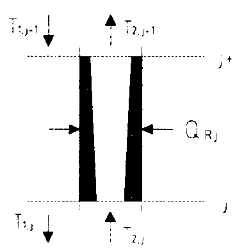

Fig. 3 Nodalization scheme used in SG modeling

II-3. Modeling of Pressurizer

Equation for Total Mass Conservation of Water and Steam

$$M_{PZRi} = M_{STEAM} + M_{WATER} \approx M_{WATER} \quad (7)$$

Equation for Total Pressure

Total fluid pressure in PZRi is expressed as a sum of partial pressures of fluid components:

$$P_{TOTAL} = P_{N_2} + P_{STEAM} \quad (8)$$

State Equation for N_2 Gas

$$P_{N_2} V_{N_2} = m_{PZRi} R T_{PZRi} \quad (9)$$

Total Mass Conservation of Nitrogen Gas

$$m_{N_2,TOTAL} = \sum_i m_{N_2}, PZRi = CONSTANT \quad (10)$$

Model for Predicting Temperature Variation in PZRi(i=1,2,3) for S/S Condition

The fluid temperature in each PZRi can be determined considering heat transfer through interface boundary surfaces of each PZRi and through heating or heated objects such as insulator, cooler tube, steam/feedwater piping as shown in Fig. 2. Heat addition due to bypass flow from MCP to PZR3 is also considered. Wet thermal insulation consists of water gaps between thin metal layers, which restrains the stagnant water inside the gap from developing free convection within gap and thus provides effective thermal insulation. Following effective thermal conductivity is used [3]:

$$K_{eff} = k \cdot 0.105 \cdot Ra^{0.3} \text{ for } Ra < 1.0^5$$
$$K_{eff} = k \cdot 0.4 \cdot Ra^{0.2} \text{ for } Ra > 1.0^5 \tag{11}$$

Free convection heat transfer near heating/heated surface of each PZRi and outer wall surface of cooler tube are modeled with following heat transfer coefficients:

Forced convection heat transfer on heating or heated surface such as inner and outer wall surfaces of steam line, inner and outer wall surfaces of feedwater line, inner wall surface of cooler tube, and outer surface of bottom wall of each PZRi are also modeled with following heat transfer coefficients:

$$\begin{aligned} Gr &= \frac{g \cdot (1 - v_{us}/v) \cdot z^3}{\nu^2} \\ Nu &= 0.135 \cdot Ra^{1/3} \text{ at } Ra > 2. \cdot 10^7 \\ Nu &= 0.54 \cdot Ra^{0.25} \text{ at } Ra > 5. \cdot 10^2 \\ Nu &= 1.18 * Ra^{0.125} \text{ at } Ra > 1. \cdot 10^{-3} \\ Nu &= 0.45 \text{ at } Ra \le 1. \cdot 10^{-3} \end{aligned} \tag{12}$$

where

$$\begin{aligned} h_2 &= 0.021 Re^{0.8} \Pr^{0.43} (\Pr/\Pr_{us})^{0.25} (k/d)\varepsilon \quad \text{at } Re > 10000 \\ h_2 &= K_0 \Pr^{0.43} (\Pr/\Pr_{us})^{0.25} (k/d)\varepsilon \quad \text{at } 2200 \le Re \le 10000 \\ h_2 &= 0.17 Re^{0.33} Gr^{0.1} \Pr^{0.43} (\Pr/\Pr_{us})^{0.25} (k/d)\varepsilon \quad \text{at } Re < 2000 \end{aligned} \tag{13}$$

$$\begin{aligned} K_0 &= K_{01} + (Gr^{0.1} - 1)K_{02} \\ K_{01(02)} &= A_{01(02)} Re_2^3 + B_{01(02)} Re^2 + C_{01(02)} + D_{01(02)} \end{aligned} \tag{14}$$

Values of A, B, C, D are given in Table 5.

Table 5 Values used in evaluating heat transfer coefficients

Ratio	K_{01}	K_{02}
A	$0.0648632188 \cdot 10^{-9}$	$-0.0217583558 \cdot 10^{-9}$
B	$-1.36670743 \cdot 10^{-6}$	$0.458461777 \cdot 10^{-6}$
C	$12.4373174 \cdot 10^{-3}$	$-3.10968662 \cdot 10^{-3}$
D	-19.2828924	7.00904421

Heat Balance Equation for PZRi

Following equations are used to calculate heat transfer quantity and overall heat transfer coefficient for plane wall surface of PZRi:

$$U_{PZRi-X} = \frac{1}{\frac{1}{h_{PZRi}} + \left(\frac{d_m}{k_m}\right) + \frac{1}{h_X}} \tag{15}$$

$$Q_{PZRi-X} = U_{PZRi-X} A (T_{PZRi} - T_X) \tag{16}$$

II-4. Modeling of Steam Generator, Cooler, and Piping

Equation for rate of quantity of heat removed from outside tube (primary side) along SG tube to inside tube (secondary side) can be written as follows:

$$\frac{d\dot{Q}_{SG}}{dz} = \dot{m}_1 \frac{di_1}{dz} \tag{17}$$

$$\frac{d\dot{Q}_{SG}}{dz} = \dot{m}_2 \frac{di_2}{dz} \tag{18}$$

$$\dot{Q}_{SGj} = \dot{m}_2 (i_{j+1} - i_j) = U_{ij} A_{ij} \triangle T_{LMTDj} \tag{19}$$

$$U_{ij} = \frac{1}{\frac{1}{h_{ij+1/2}} + \frac{r_i}{k_{mj+1/2}} \ln\left(\frac{r_o}{r_i}\right) + \frac{r_i}{r_o h_{oj+1/2}}} \tag{20}$$

where j+1/2 denotes average quantity over the region of interest.

$$\triangle T_{LMTDj+1/2} = \frac{(T_{1,j} - T_{2,j}) - (T_{1,j+1} - T_{2,j+1})}{\ln\left(\frac{T_{1,j} - T_{2,j}}{T_{1,j+1} - T_{2,j+1}}\right)} \tag{21}$$

Notation and symbol used above are shown in Fig. 3. To evaluate the convective heat transfer coefficients for primary side, h, following correlations are used [3]:

$$h_1 = 0.18Re^{0.64}\frac{k}{d} \quad (22)$$

Boiling heat transfer coefficients from inner wall of the tube to the secondary coolant in the evaporation section is defined according to the following equation [4].

$$h_2 = 2.9075(10.197P)^{0.2}(q/(1.163\pi d))^{0.7} \quad (23)$$

III Result and Discussion

Separate calculation for obtaining thermal-hydraulic parameters of SG along SG tube at normal power is performed and its results are shown in Fig. 4 & 5. A nature of physical phenomena of such results for in-vessel SG are well known [5].

Calculational results of performance of pressurizer for the case of three turned length of cooler tube are summarized in Table 6. Secondary pressure drops through SG throttling device and along SG tube are about 1 MPa and about 0.3 MPa, respectively, which are within the intention of a design to prevent flow instability problem. Table 6 shows that the reactor system adopting in-vessel helical SG can produce 60 °C superheat at steam piping penetrating reactor vessel header at normal power operation and that water temperature in end cavity is 95.5 °C and steam partial pressure is 0.086 MPa, which means that the behavior of pressure in pressurizer is determined mainly by nitrogen gas because steam partial pressure is negligibly small. Water temperature in upper annular cavity is high at 306.4 °C due to bypass flow from MCP and therefore temperature of steam flowing in steam piping penetrating this cavity is not affected significantly. Temperature increase of feedwater along feedwater piping penetrating this cavity is about 1.7 °C. Further study is needed to check the possibility of boiling in feedwater piping at low flow condition. Fig. 6 shows trends of water temperature in each cavity of pressurizer and cooler outlet with the change of number of turns of cooler tube. Fig. 7 shows nitrogen partial pressure and nitrogen mass required to maintain required nominal system pressure of 15.0 MPa with the change of number of turns of cooler tube. As the number of turns of cooler tube increases, cooling effect of cooler on pressurizer becomes larger and therefore water temperature is lowered. It is revealed that the change of cooler tube length does not affect the performance of SG including its connected piping line significantly. Therefore from these sensitivity evaluations of the effect of cooler tube length on steady state performance of pressurizer and SG, it is concluded that in-vessel pressurizer with wet thermal insulation and 2 or 3 turns of cooler tube and in-vessel helical SG are expected to be well performed at normal power operation. However, for justification of the conclusion and the modeling developed in this paper, large-scale experiment is needed.

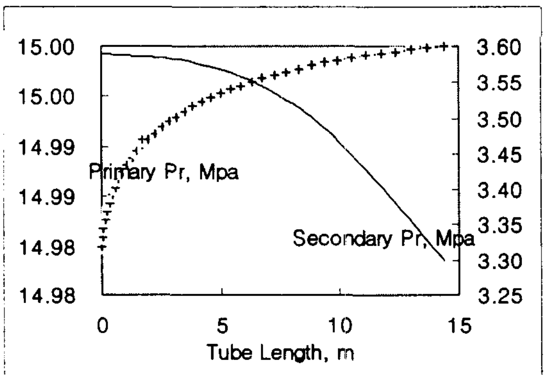

Fig. 4 Pressure distribution along SG tube length

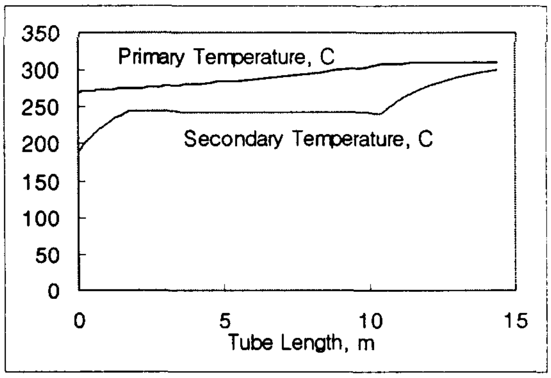

Fig. 5 Temperature distribution along SG tube length

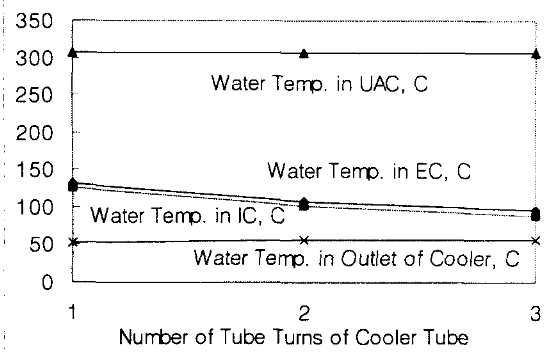

Fig. 6 Trends of water temperature in PZR with number of turns of cooler tube

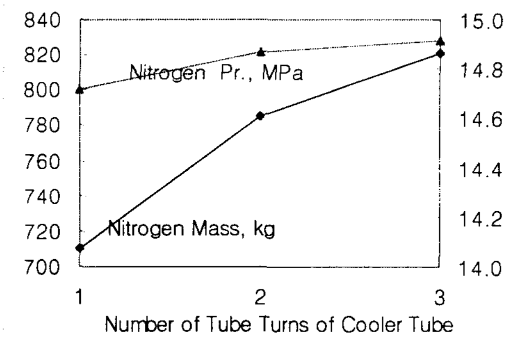

Fig. 7 Trends of N_2 mass and N_2 partial pressure in PZR with number of turns of cooler tube

Table 6 Summary of steady state calculational result for the case of 3 turns of cooler tube

water temperature at steam generator inlet/outlet from primary side, °C	**<u>310.00</u>**/279.80
mass of nitrogen gas, kg	821.27
water level in end cavity from bottom, m	2.06 (49.1 % of span)
N_2 partial pressure/steam partial pressure/total pressure in pressurizer, MPa	14.913/0.087/**<u>15.000</u>**
feedwater pressure at reactor vessel head, MPa feedwater pressure before flow restricting device, MPa feedwater pressure after flow restricting device, MPa	4.551 4.562 3.586
feedwater temperature at reactor vessel head, °C feedwater temperature before flow restricting device, °C feedwater temperature after flow restricting device, °C	**<u>190.00</u>** 191.70 191.80
controlled steam pressure at SG steam header, MPa steam pressure at bottom of upper annular cavity, MPa steam pressure at top of upper annular cavity, MPa	**<u>3.300</u>** 3.240 3.237
steam temperature at SG steam header, °C steam temperature at bottom of upper annular cavity, °C steam temperature at top of upper annular cavity, °C	300.06 299.38 299.47
water temperature in end cavity/intermediate cavity/upper annular cavity, °C	95.52/88.97/306.44
pressure at cooler inlet/outlet, MPa	**<u>0.500</u>**/0.324
temperature at cooler inlet/outlet, °C	**<u>40.00</u>**/56.61

Note: Bold and underlined values denote controlled or input parameters.

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