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Numerical studies of a helical coil once-through steam generator

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ABSTRACT

A Thermal Hydraulic analysis code for Helical Coil once-through Steam Generator (TH-HCSG) is developed in the present study in order to predict the heat transfer process and thermal hydraulic characteristics in vertical helical coil steam generators. The numerical model is programmed using a one-dimensional four-equation drift-flux model and evaluated according to the data on HCSG published by the International Reactor Innovative and Secure (IRIS). The results achieved under full load conditions are, then, compared with the RELAP5 codes, with the results of this comparison showing that this model is both correct and reliable and can be used to investigate the instantaneous characteristics of helical coil steam generators. This model is also used to evaluate the influence of the main representative HCSG boundary parameters on HCSG operation characteristics. It was subsequently found that, when there was a ±10% step change in the boundary parameters, the length of the single-phase water region in the secondary side of the HCSG changed considerably, so that the steam flow oscillated obviously, lagging the feedwater flow at an early stage and rapidly reaching a steady state as the water capacity of the secondary side of the HCSG was changed.

1. Introduction

Due to its compact structure, high heat transfer coefficient and good compensation for thermal expansion, helical coil tube-type once-through steam generators (HCSG) have been utilized in various reactors, especially in integral type reactors such as the CAREM-25 (Fukami and Santecchia, 2000), MRX (Kusunoki et al., 2000), IRIS (Carelli et al., 2004), MASLWR (Mascari et al., 2011), SMART (Chung et al., 2013).

In the heat transfer tubes of HCSGs, single-phase primary coolant passes outside down the tube of the helical coil through the space of helical coil tube. The cold water is heated, becoming superheated steam by using primary coolant. The thermal-hydraulic phenomena of the secondary side of the HCSG are very complex, as the flow direction is constantly changing because of their very particular spiral geometry. The centrifugal forces acting on the fluid inside the helical ducts tend to laminarize the flow, enhance the heat transfer coefficient and increase the frictional pressure drop.

In HCSGs, the geometric parameters, such as the diameters of the pipes, spiral radius and coil pitch (curvature and torsion) affect the heat transfer coefficient and pressure drop (Hardik et al., 2015). Many researchers have investigated the thermal performance of helical coil heat exchangers experimentally. Naphon (2007) reported that the temperature of the cold water outlet, the effectiveness of the heat exchanger and the average heat transfer rate increased with an increase in the mass flow rate of hot water. Ghorbani et al. (2010) found that the convection heat transfer coefficient of the shell-side increased when the coil pitch was increased, and the overall heat transfer coefficient of the heat exchanger increased with an increase in the heat transfer rate. Hardik et al. (2015) researched the influence of curvature and Reynolds number on the local heat transfer coefficient in helical coils, suggesting correlations for the overall averaged and local circumferentially averaged Nusselt number for their inner side, outer side and total surface. Saffari et al. (2014) reported a correlational equation to predict the length of the hydrodynamic entrance as a function of various helical coil parameters.

The development of simulation models suited to HCSGs is more complex than for straight tube once-through steam generators. In recent years, numerous studies have been carried out on the single-phase and two-phase pressure drop and heat transfer characteristics of HCSGs. A large number of empirical relationships have been able to provide strong support for the preparation of simulation program. In order to calculate the steady state of

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Nomenclature

A	heat transfer area (m^2)
c_p	constant pressure specific heat (J/kg/°C)
d	inner diameter of spiral pipe (m)
D_c	curvature radius (m)
f_{vis}	pressure drop gradient by wall friction
g	gravitational acceleration (m/s^2)
G_m	mass flux ($kg/m^2/s$)
h	enthalpy (J/kg)
h_{fg}	latent heat of vaporization (J/kg)
h_p	primary heat transfer coefficient ($W/m^2/K$)
h_s	secondary heat transfer coefficient ($W/m^2/K$)
k	thermal conductivity (W/m/°C)
N_u	Nusselt number
p	pressure (Pa)
P_r	Prandtl number
q	heat flux (W/m^2)
Q	heating power (W)
R_e	Reynolds number
R_f	foul resistance coefficient (m^2·K/W)
t	calculation time (s)
T	fluid temperature (K)
z	length (m)
$\triangle T_{sat}$	subcooling degree (K)

Greek letters

$\triangle$	difference operator
α	vapor volume fraction
ρ	density (kg/m^3)
v_r	relative velocity (m/s)
θ	spiral pipe rotation angle (°)
μ_f	viscosity coefficient (Pa·s)
λ	thermal conductivity (W/m/K)

Subscripts

f	liquid
g	vapor
i	inner
m	mixture
o	outer
p	primary side
s	secondary side
sat	saturation
w	wall

HCSGs, Yoon et al. (2000) wrote an ONCESG program based on the movable boundary method. Yang et al. (2008) validated the HCSG model using the system transient analysis program TASS/SMR developed by Chung et al. (2012). Lee and Park (2013a–c), Lee et al. (2012) developed the TAPINS thermal hydraulic analysis program, which was based on four drift flow models for the integration of natural circulation in PWR, validating the model developed for HCSG by comparing the simulation results with experimental data. By taking the HCSG of HTR-10 as their research objectives, Li et al. (2008) established a lumped-parameter dynamic mathematical model based on the movable boundary method. Abdalla (1994) suggested a dynamic mathematical model for advanced liquid metal reactors, their method consisted in adopting the four-region model and the movable boundary method and the drift flow model method of calculation. RELAP5 codes were utilized to build their system model (2006, 2011, 2012), and the operating characteristics of HCSGs can be analyzed in the whole system.

By examining the relevant literature, we can see that research has achieved a detailed analysis of the steady state and two-phase flow within HCSG, but that it has failed to take the overall characteristics of HCSGs and their transient operating process into account. In this paper, a mathematical model for HCSGs is established. Utilizing the established model, the operating characteristics of HCSGs under a steady state and transient conditions are investigated, and the calculated results are compared with the RELAP5 codes. Finally, the step changes of the representative boundary parameters of HCSGs are studied. It is hoped that the results of our research can be used to improve the design and operation of HCSGs.

2. Mathematical models

Based on the characteristics of HCSG, the following assumptions are made to simplify the mathematical model.

1) HCSG is configured as one flow channel.
2) Each side is treated as a single flow channel according to the heat transfer area and equivalent diameter.
3) One-dimensional approach is used and neglecting axial heat conduction.
4) Ignoring fluid expansion works and works force by wall friction and fluid viscous force.
5) Ignoring interphase kinetic energy exchange and the work done by gravity.
6) Assuming uniform pressure distribution in the same cross section.
7) Thermal equilibrium model for two phase flow.

The basic conservation equations of mass, energy, and momentum according to the abovementioned basic assumptions can be given as follows.

2.1. Basic mathematical models

Conservation equations for mass, energy and momentum are utilized for the primary and secondary sides.

Two phase mixture continuity conservative equation:

$$\frac{\partial}{\partial t}\rho_m + \frac{\partial}{\partial z}(\rho_m v_m) = 0 \tag{1}$$

Single phase gas conservative equation:

$$\frac{\partial}{\partial t}(\alpha\rho_g) + \frac{\partial}{\partial z}(\alpha\rho_g v_m) + \frac{\partial}{\partial z}\left[\frac{\alpha(1-\alpha)\rho_g\rho_f}{\rho_m}v_r\right] = \Gamma \tag{2}$$

Two phase mixture momentum conservative equation:

$$\frac{\partial}{\partial t}(\rho_m v_m) + \frac{\partial}{\partial z}(\rho_m v_m^2) + \frac{\partial}{\partial z}\left[\frac{\alpha(1-\alpha)\rho_g\rho_f}{\rho_m}v_r^2\right] = -\frac{\partial p}{\partial z} - f_{vis} - \rho_m g\sin\theta \tag{3}$$

Two phase mixture energy conservative equation:

$$\frac{\partial}{\partial t}(\rho_m h_m) + \frac{\partial}{\partial z}(\rho_m v_m h_m) + \frac{\partial}{\partial z}\left[\frac{\alpha(1-\alpha)\rho_g\rho_f(h_g-h_f)}{\rho_m}v_r\right] = \frac{Q}{A\Delta z} + \frac{\partial p}{\partial t} \tag{4}$$