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Experimental investigation and prediction of post-dryout heat transfer for steam-water flow in helical coils

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ABSTRACT

Post-dryout heat transfer for steam-water flow in helical coils is experimentally investigated. The inner diameters of the helical coils are 12.5 mm and 14.5 mm, and the coil diameters range from 180 mm to 380 mm. The experimental system pressure covers the range 2–8 MP. The results indicate that the centrifugal force induced secondary flow has a great influence on post-dryout heat transfer in helical coils. In the low quality post-dryout region, because of the high redeposition rate induced by secondary flow, the heat transfer coefficient of helical coils is significantly higher than that of straight tubes. As the total droplet mass decreases with increasing quality, the heat transfer performance of helical coils decreases and continuously approaches that of straight tubes. In high quality post-dryout region, the bulk fluid in helical coils is closer to thermodynamic equilibrium state due to the secondary flow, and the heat transfer coefficient may be exceeded by straight tubes. For the effects of flow parameters, the post-dryout heat transfer performance decreases in heat flux and increases in mass flux. For system pressure, due to the change of the dominant heat transfer mechanism, the post-dryout heat transfer decreases in the low quality region but increases in the high quality region with increasing system pressure. Based on a detailed analysis of the effects of various dimensionless parameters, a new correlation containing a quality correction is proposed and provides good prediction accuracy.

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1. Introduction

Due to the high heat transfer performance and compact design, helical coil heat exchangers find widespread use in industries [1–6]. Particularly in nuclear industry, Helical Coil Steam Generators (HCSGs) are adopted by many Small Modular Reactors (SMRs) [7,8]. As a kind of once through steam generators, accurate heat transfer models in post-dryout heat transfer region is very important for the design and optimization of HCSGs. Generally, in post-dryout heat transfer, a significant amount of mechanical and thermal non-equilibrium is present which leads to six possible heat transfer paths among the liquid phase, the vapor phase and the heated wall [9]: 1. Convective heat transfer from the wall to the vapor, 2. Radiative heat transfer from the wall to the vapor, 3. Interfacial heat transfer from the vapor to the droplets, 4. Radiative heat transfer from the vapor to the droplets, 5. Direct contact heat transfer from the wall to the droplets, and 6. Radiative heat transfer from the wall to the droplets.

For post-dryout heat transfer phenomenon in straight tubes, many investigations have been conducted and a large number of correlations have been developed. Equilibrium type correlations are assumed equilibrium or empirically derived, including Dougall-Rohsenow [10], Groenveld [11], Slaughterbeck [12], Polomik [12] and Condie-Bengston [13]. These correlations are variants of the Dittus-Boelter type equations [14]. Non-equilibrium type correlations attempt to predict the degree of thermal non-equilibrium of fluids, which have been developed by Groeneveld and Delorme [15], Chen et al. [16], Saha [17], Tian [18], Meholic et al. [19] and Nguyen et al. [20]. Most thermodynamic non-equilibrium correlations are also based on Dittus-Boelter type correlations, and a few correlations are mechanistic [14].

However, for helical coils, a few investigations have been performed on the post-dryout heat transfer in the literature [21]. Due to the centrifugal force exerted upon the fluid, secondary flows arise and impacted the post-dryout heat transfer in helical coils. Guo et al. [22] investigated the post-dryout heat transfer for steam-water two-phase flow in a helical coil at pressure of 0.4–3.0 MPa experimentally. Experiments were conducted in four different inclination angles, which were upwardly vertical, 45 degree upwardly inclined, horizontal and 45 degree downwardly inclined. A correlation for

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Nomenclature

Cp	specific heat (J/kg•K)
D	diameter of tube (m)
Dc	coil diameter (m)
g	gravitational acceleration (m/s^2), gas
G	mass flux (kg/m^2·s)
h	heat transfer coefficient (W/m^2•K), enthalpy (kJ/kg)
k	thermal conductivity (W/m•K)
L	length (m)
P	pressure (Pa)
P_h	pitch (m)
q	heat flux (kW/m^2)
T	temperature (K)
x	quality of steam
Xtt	Lockhart Martinelli parameter

Greek symbols

δ	curvature ratio δ = D/De
ρ	density (kg/m^3)
μ	viscosity (Pa·s)
σ	surface tension (N/m)
θ	helical angle($\circ$)

Abbreviation

Avg	Average
HCSG	Helical Coil Steam Generator
HTC	Heat Transfer Coefficient
RMS	Root Mean Square
SMR	Small Modular Reactor

Dimensionless number

Nu	Nusselt number Nu = h·D/k
Pr	Prandtl number $Pr = \mu c_p/k$
Re	Reynolds number Re = G·D/μl

Subscripts

cal	calculation
cr	critical
DB	Dittus-Boelter
Do	dryout
e	equilibrium state
exp	experiment
g	gas
go	gas only
l	liquid
lo	liquid only
tp	two phase

predicting the post-dryout heat transfer coefficient was proposed based on an X_{tt} correction. They also indicated that the system pressure has little effect on post-dryout heat transfer under their experimental conditions.

To understand the post-dryout heat transfer phenomenon of steam-water flow in helical coils, experimental investigations are performed at system pressure ranging from 2 to 8 MP. The coil diameters range from 180 to 380 mm and the inside diameters are 12.5 and 14.5 mm. The effects of flow parameters and geometric parameters on post-dryout heat transfer are presented and analyzed. The correlations for post-dryout heat transfer in the literature are evaluated. Based on an analysis of the effects of various dimensionless parameters, a new heat transfer correlation, containing a quality correction, for post-dryout regime of helical coils is developed.

2. Experimental apparatus and test sections

A scheme diagram of the SWAMUP-II test facility (Supercritical WAter MUltiPurpose loop II) is presented in Fig. 1. The working fluid of the facility is deionized water and the design pressure is 35 MPa. The working fluid through the loop is driven by two parallel plunger pumps. Following each plunger pump, an accumulator is equipped to absorb fluctuations in pressure and mass flow rate. The main flow goes through the re-heater and the pre-heaters and is heated up. AC powers are used in the pre-heater 1 and 2, and the maximum electrical powers are 600 kW and 200 kW, respectively.

Under all target experimental conditions in this work, the inlet of the pre-heater 2 is subcooled and the outlet is saturated due to the short lengths of the test sections. The pre-heater 2 is well insulated thermally and the heat loss is calibrated accurately for each experimental condition via a special calibration test [23]. There are three steps in a calibration. In the first step, maintaining all other test parameters of pre-heater 2 the same as the target two-phase experimental condition, the system pressure parameter is increased and a subcooled outlet of pre-heater 2 is obtained. A heat loss could be obtained based on the enthalpy rise and the heating power of the pre-heater 2. Because of the higher outer wall temperatures, the heat loss is greater than the actual heat loss under the target two-phase condition. In the second step, maintaining all other test parameters of pre-heater 2 the same as the target two-phase experimental condition, the heating power parameter of pre-heater 2 is decreased to obtain a subcooled outlet. A heat loss could be derived too, which is less than the actual heat loss due to the lower outer wall temperatures. In the final step, the heat loss of pre-heat 2, under the target two-phase experimental condition, is considered as the average of the heat losses derived in the two previous steps in the data processing. The uncertainty of the pre-heater 2 heating efficiency given by uncertainty analysis is no more than 0.2%. Based on the inlet enthalpy, heating power and heat loss, the outlet enthalpy of the pre-heater 2, which is also the inlet enthalpy of the test section, can be derived under saturated conditions [23].

Six test sections were investigated in this work as shown in Table 1 [24]. The inner diameters of the helical coils are 12.5 mm and 14.5 mm, and the coil diameters range from 180 mm to 380 mm. The helical coils are made of stainless steel and directly heated by a DC power with capacity of 900 kW. Two Venturi flow meters, with different measuring ranges, are equipped to measure the mass flow rate. The inlet pressure is measured by a capacitance-type pressure transducer. Pressure drops between the pressure taps (Fig. 2) and over the test section are measured by a set of differential pressure transducers [24]. The inlet temperature, outlet temperature and the outer wall temperature distribution are measured via N-type thermocouples. The cross sections for outer wall temperature measurement are evenly distributed and the gap is one fourth of the coil perimeter. There are four temperature attached in each cross section, as presented in Fig. 3. A National Instrument data acquisition system is equipped to record all the data mentioned above.

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