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2014 J. Phys.: Conf. Ser. 547 012034

(http://iopscience.iop.org/1742-6596/547/1/012034)

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Abstract. A model has been developed in Matlab environment for the thermal hydraulic analysis of helical coil and shell steam generators. The model considers the internal flow inside one helix and its associated control volume of water on the external side, both characterized by their inlet thermodynamic conditions and the characteristic geometry data. The model evaluates the behaviour of the thermal-hydraulic parameters of the two fluids, such as temperature, pressure, heat transfer coefficients, flow quality, void fraction and heat flux. The evaluation of the heat transfer coefficients as well as the pressure drops has been performed by means of the most validated literature correlations. The model has been applied to one of the steam generators of the IRIS modular reactor and a comparison has been performed with the RELAP5/Mod.3.3 code applied to an inclined straight pipe that has the same length and the same elevation change between inlet and outlet of the real helix. The predictions of the developed model and RELAP5/Mod.3.3 code are in fairly good agreement before the dryout region, while the dryout front inside the helical pipes is predicted at a lower distance from inlet by the model.

Introduction

Helical coil pipes have been widely used in the past in many industrial applications such as thermal process plants or power plants for the steam generation. In nuclear industry, helical coil steam generators (SG) and intermediate heat exchangers (IHX) have been installed in the fast reactors like Monju (Japan) [1] and Super-Phénix (France). These components are still considered for the future reactors projects such as the small modular reactors (SMR), like IRIS [2], and the fourth generation reactors like the Russian project BREST.

Coiled geometries are of particular interest because they allow to obtain high values of the inner heat transfer coefficients and guarantee an efficient power removal with a very high degree of compactness. Secondary motions in the velocity field are due to the centrifugal forces induced by the pipe curvature: these forces produce a shift of the fluid towards the outer part of the pipe, reducing the boundary layer and increasing the heat transfer. Centrifugal forces induce also a displacement of the maximum values of temperature and velocity from the centre of the axis to the outer side.

Helical coil fluid dynamics has been studied by several authors as regards both the laminar to turbulent transition [3] and the heat transfer and pressure drops in single-phase flow. Many reviews related to the fluid dynamics and heat transfer of curved tubes are available in literature, also recently (Naphon and Wongwises [4] and Vashisth [5]). Most of the work already done considers single-phase heat transfer with constant wall temperature or fixed heat flux boundary conditions, concentrating the efforts on the inner side of the helix. In order to characterize efficiently the heat transfer in steam generators is also important to develop models of heat transfer between fluids, characterized by their own temperature profile, with variable heat flux and wall temperature distribution in space. Escha et al. [6] studied a helical coil steam generator for a gas cooled reactor with the system code TRACE and the results obtained with different heat transfer coefficient correlations were compared with experimental data. In the present study a model has been implemented in Matlab environment with the aim to evaluate temperature and pressure profiles, heat transfer, pressure drops and two-phase flow dynamic considering an helical coil and shell steam generator. The semiempirical correlations for the evaluation of the heat transfer coefficients and the friction factors have been taken from literature. The model has been applied to the steam generator configuration of the SMR IRIS and the profiles of temperature, pressure and heat transfer coefficients have been compared with the ones obtained with the commercial code RELAP5/Mod.3.3.

Flow dynamics

The correct characterization and modelling of compact steam generators requires the knowledge of the flow field of the fluids, both on the inner and on the outer side. The flow field inside helical coil pipes is a complex result of the balance between inertia, gravity and centrifugal forces. The centrifugal forces are strongly dependent on the geometry of the pipe: considering a constant value of the pipe diameter, its influence is inversely proportional to the radius of the helix. The presence of secondary motions within the pipe cross section affects both velocity and thermal boundary layers: this explains the observed enhancement of the heat transfer coefficient and increase of the friction pressure drops. The helical pipes in the SG are disposed in a parallel configuration at increasing radii as in figure 1, and this involves a different importance of the centrifugal force with respect to inertia and gravity moving from the inner tube bundle to the outer ones.

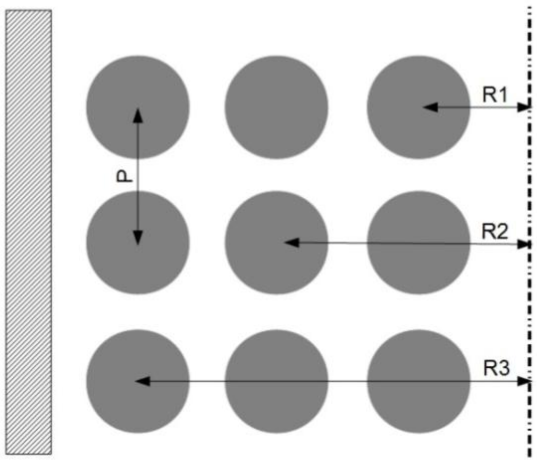

Figure 1. Helical bundle configuration.

When the fluid consists of a mixture of liquid and steam, the presence of centrifugal forces affects the flow pattern and promotes a separation of the phases due to the density difference, causing the heavier phase to move toward the external part of the pipe. From the point of view of the external flow the velocity field is disturbed by the presence of the helical coils and this promotes the formation of vortexes and radial mixing phenomena which increase as the liquid downflows. These effects are strongly dependent on the geometrical characteristics of the pipe bundle. In particular, the compactness of the whole bundle is one of the most important parameter for the velocity field in the external flow. The configuration of heat transfer from the external side lead to the decision to use the correlation of Zhukauskas for the external heat transfer coefficient [7] because it is able to predict the heat transfer in external flow on tube banks.

The thermal hydraulic model

In the reference geometry the fluid inside the helical coils has an upward flow while the fluid in the shell side flows downward in the SG. A countercurrent flow can be considered on the basis of the prevalent flow of the fluids, but in case of low helix curvature the cross-flow configuration can be assumed. The input data required by the model are the inlet conditions of the fluids in terms of pressure, flow rate and temperature and the geometrical characteristics of the helical coil.

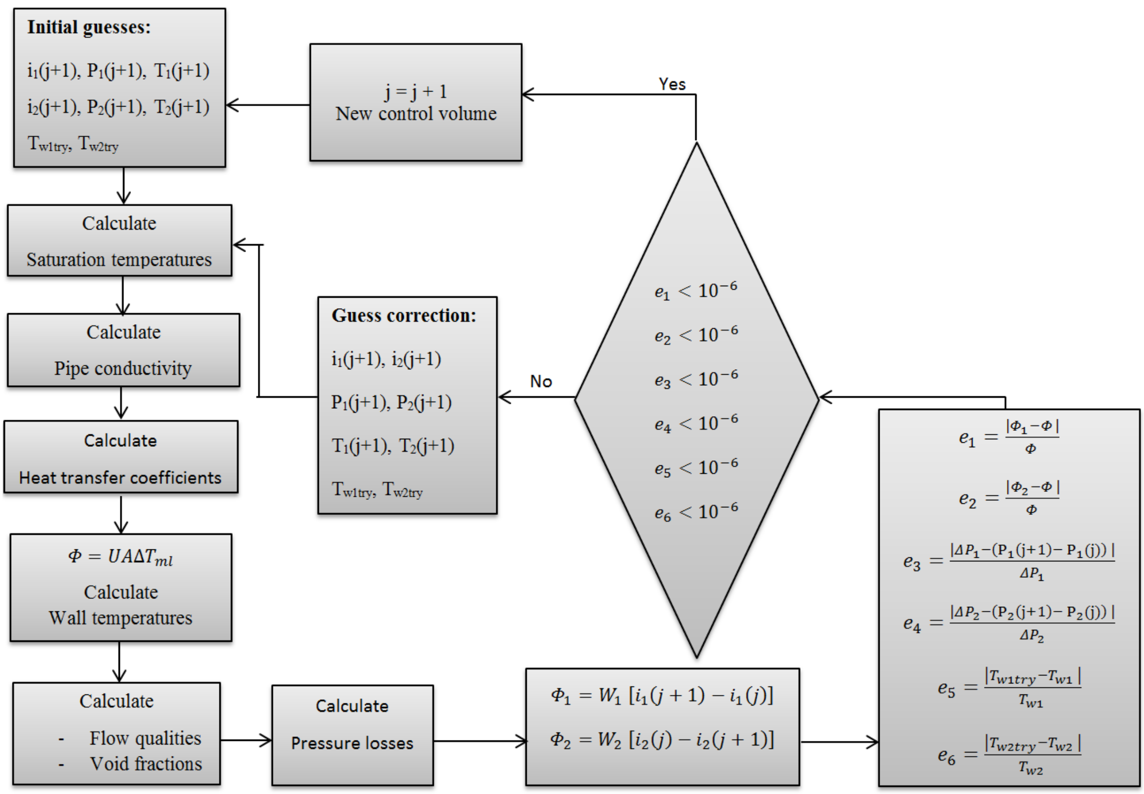

Figure 2. Control volumes flow chart.

The thermal hydraulic model is applied to four characteristic heat transfer regions from the point of view of the fluid inside the helix, namely the liquid single-phase region, the subcooled boiling region, the saturated boiling region and the steam single-phase region. Each region is divided in several control volumes containing the volume of the helical side fluid, the pipe volume itself and the equivalent annular control volume of the shell side fluid. The calculation convergence is guaranteed by a power balance and error flags of the relative error between attempted and calculated temperatures and pressures values. The flow chart describing the steps for the control volumes calculations is shown in figure 2. The parameters calculated in each control volume are heat transfer coefficients, acceleration, friction and elevation pressure drops, as well as flow quality and void fraction. At the end of the calculations the model is able to show the spatial behaviour of the characteristic parameters of the flow, such as temperature, pressure, flow quality, void fraction, heat flux and heat transfer coefficients. The semiempirical correlations and models that have been adopted to evaluate the heat transfer coefficients, the friction pressure drops and the void fraction are taken from the most validated literature correlations taking also into account the effects of centrifugal forces inside the helical coils and the mixing induced by their presence in the flow path of the shell side fluid [7-11]. The adopted correlations are reported in table 1.

Table 1. Empirical correlations adopted in the present model.

Author	Application	Correlation
Mori and Nakayama [8]	Helix side liquid single phase region	$Nu = \frac{1}{41}\left(\frac{Pr}{Pr^{0.6}-0.062}\right) Re^{\frac{5}{6}} \left(\frac{d_i}{D_{hx}}\right)^{\frac{1}{12}} \left(1+\frac{0.061}{\left\{Re\left(\frac{d_i}{D_{hx}}\right)^{2.5}\right\}^{\frac{1}{6}}}\right)$
Mori and Nakayama [8]	Helix side steam single phase region	$Nu = \frac{1}{26.2}\left(\frac{Pr}{Pr^{0.6}-0.074}\right) Re^{\frac{4}{5}} \left(\frac{d_i}{D_{hx}}\right)^{\frac{1}{10}} \left(1+\frac{0.098}{\left\{Re\left(\frac{d_i}{D_{hx}}\right)^{2}\right\}^{\frac{1}{5}}}\right)$
Owhadi [9]	Helix side subcooled and saturated boiling	$h = S\, h_b + F\, h_c$ $F = 2.35\,(\chi_{tt}^{-1} + 0.213)^{0.736}$ $F = 1\ in\ subcooled\ boiling$ $S = \begin{cases} \left(1+0.12\, Re_{tp}\right)^{-1.14} & Re_{tp} < 32.5 \\ \left(1+0.42 Re_{tp}^{0.78}\right)^{-1} & 32.5 < Re_{tp} < 70 \\ 0.0797 & Re_{tp} \geq 70 \end{cases}$ $Re_{tp} = \frac{G\,(1-x) D_h}{\mu_f} F^{1.25}\, 10^{-4}$ $h_b = 0.00122 \left(\frac{k_f^{0.79} C_{pf}^{0.45} \rho_f^{0.49}}{\sigma^{0.5}\, \mu_f^{0.29} \lambda^{0.24} \rho_g^{0.24}}\right) \Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75}$ $h_c = \frac{k_f}{41D}\left(\frac{Pr}{Pr^{0.6}-0.062}\right) Re^{\frac{5}{6}} \left(\frac{d}{D}\right)^{\frac{1}{12}} \left(1+\frac{0.061}{\left\{Re\left(\frac{d}{D}\right)^{2.5}\right\}^{\frac{1}{6}}}\right)$
CISE [10]	Helix side void fraction	$S = 1 + E_1\left(\frac{y}{1+yE_2} - yE_2\right)^{0.5}$ $y = \frac{\beta}{1-\beta}$ $\beta = \frac{\rho_l x}{\rho_l x + (1-x)\rho_g}$ $E_1 = 1.578\, Re^{-0.19} \left(\frac{\rho_l}{\rho_g}\right)^{0.22}$ $E_2 = 0.0273\, We\, Re^{-0.51} \left(\frac{\rho_l}{\rho_g}\right)^{-0.08}$
Zhukauskas [7]	Shell side heat transfer coefficient	$Nu = 0.033\, Re_{D,max}^{0.8} Pr^{0.36} \left(\frac{Pr}{Pr_s}\right)^{\frac{1}{4}}$
Ito [11]	Helix side friction factor	$Re_{crit} = 20000 * \left(\frac{d_i}{D_{hx}}\right)^{0.32}$ $f = \begin{cases} \frac{344\left(\frac{D}{d}\right)^{-0.5}}{\left\{1.56+\log_{10}\left[Re\left(\frac{D}{d}\right)^{-0.5}\right]\right\}^{5.73}} & Re < Re_{crit} \\ 0.076\, Re^{-0.25} + 0.00725 \left(\frac{D_h}{d_i}\right)^{-0.5} & Re > Re_{crit} \end{cases}$
Chen [12]	Helix side two phase multiplier	$\Phi_{l0}^2 = 2.06 \left(\frac{d_i}{D_h}\right)^{0.05} Re_{tp}^{-0.025} A$ $A = \left[1+\alpha\left(\frac{\rho_g}{\rho_f}-1\right)\right]^{0.8} \left[1+x\left(\frac{\rho_f}{\rho_g}-1\right)\right]^{1.8} \left[1+\alpha\left(\frac{\mu_g}{\mu_f}-1\right)\right]^{0.2}$
Blasius [13]	Shell side friction factor	$f = 0.079 * Re^{-\frac{1}{4}}$

The RELAP5/Mod.3.3 nodalization
RELAP5/Mod.3.3 is a commercial thermal hydraulic system code able to analyse accidents in nuclear power plants [14]. The code is based on a nonhomogeneous and nonequilibrium model for the two-phase system that is solved in a fast, partially implicit numerical scheme to permit economical calculation of system transients [14]. Thanks to its capability to model the fluid dynamics and the heat transfer in various conditions it has been chosen as a comparison of the present model. It is currently not able to predict the heat transfer inside helical pipes because of a lack of correlations for the heat transfer coefficient. As suggested by the Idaho National Laboratory (INL) [15] the helical coil geometry is simulated in the code with an inclined pipe with the same length as the real helix and an equivalent inclination created to obtain the same elevation between inlet and outlet. The equivalent channel for the external flow is simulated by an annulus. The annulus and the inclined pipe are thermally linked with a heat structure (HS) made of T91. The annulus, the inclined pipe and the heat structure have been nodalized with 99 volumes. Time dependent volumes (TMDP VOL) and time dependent junctions (TJ) are used to impose pressure and flow rates at the beginning and at the end of the heated channels. Single junctions (SJ) initialized with temperature, pressure and flow rate initial conditions are used at the outlet of the annulus and of the inclined pipe. The layout of the two hydrodynamic systems is presented in figure 3.

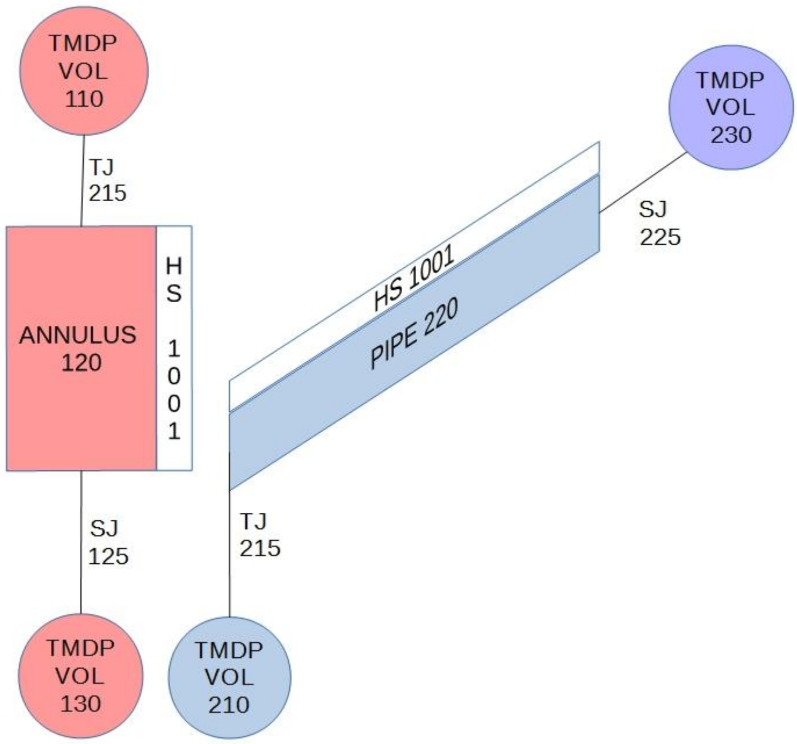

Figure 3. Layout of the hydrodynamic components.

Thermal hydraulic simulation of the IRIS steam generator
The proposed model and the RELAP5/Mod.3.3 code have been used to simulate the thermal hydraulic behaviour of one of the steam generators of the IRIS SMR and the results of the simulations are compared with respect to the nominal values available in literature [2,16]. IRIS is a light water SMR, designed within an international collaboration. The reactor is equipped with 8 once-through helical coil steam generators installed in the annular region between the vessel and the riser above the reactor

core. The simulations have been performed by using an average coil in terms of length, pitch and helical radius whose data are shown in table 2.

Table 2. IRIS SG nominal data [2,16].

Parameter	Value
Power [kW]	190.5
Primary inlet temperature [°C]	328.4
Feedwater temperature [°C]	223.9
Primary pressure [bar]	155
Internal diameter [mm]	13.24
Primary flow rate [kg/s]	0.893
Helix length [m]	32.0
Primary outlet temperature [°C]	292.0
Steam temperature [°C]	317.0
Steam pressure [bar]	58
External diameter [mm]	17.46
Feedwater flow rate [kg/s]	0.0958

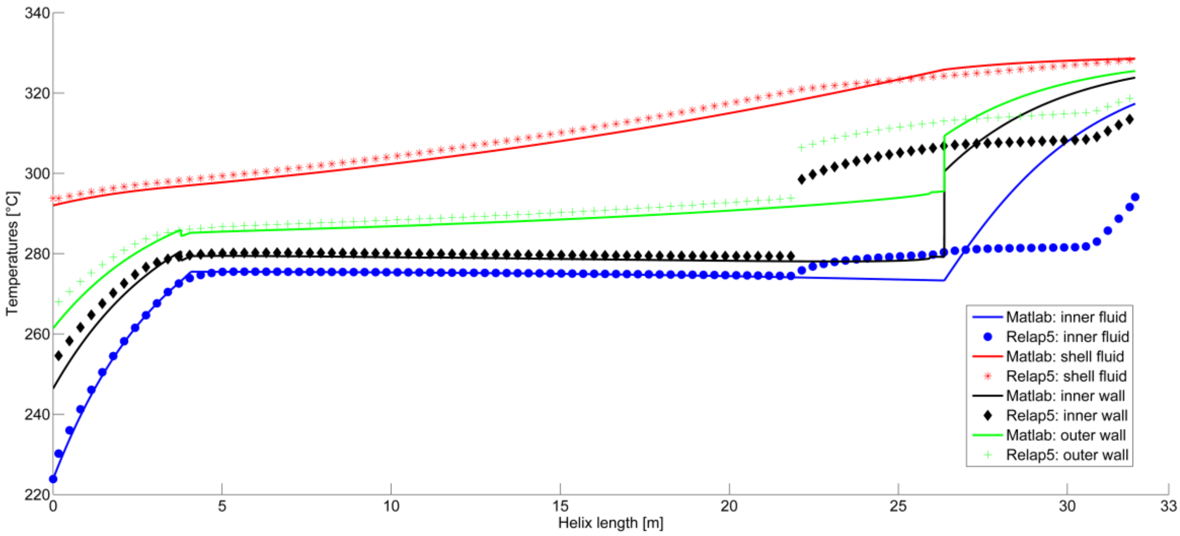

Figure 4. Temperature profiles.

Figure 4 shows the temperature profiles of the fluids as well as the temperature profile of the internal and external walls of the pipe. The main differences between the models are the two-phase nonequilibrium considered by RELAP5/Mod.3.3 and different heat transfer coefficients correlations, which are specific for helical coiled pipes in the proposed model. Nevertheless, the present model shows a good agreement with the code prediction of the temperature profile on the shell side with a maximum relative difference lower than 1%. The wall temperatures predicted by the present model are slightly lower than the code ones in the liquid single-phase and low quality regions. The dryout front is predicted at two different distances from inlet, respectively 22 meters for the code and 27 meters for the proposed model. The dryout occurrence in helical coil pipes is still an open issue and few studies

Journal of Physics: Conference Series **547** (2014) 012034 doi:10.1088/1742-6596/547/1/012034

have been performed so far. The heat transfer degradation is found to be asymmetric and strongly dependent on the weight of centrifugal forces [17], and the critical heat fluxes in the helical coil are much higher than in a vertical straight tube [18]. The temperatures of the internal fluid have a different behaviour in the high quality region and this is due to two differences between the models, which are the absence of transition boiling in the proposed model and the treatment of nonequilibrium conditions in the RELAP5/Mod.3.3 code. The outlet temperatures have a discrepancy of 20°C which is consistent with the different power exchanged in the two cases. Figure 5 shows the behaviour of the thermal resistances of the inner and outer heat transfer and the thermal resistance induced by the conduction term in the overall heat transfer. The highest thermal resistance is predicted on the outer side of the pipe in the liquid single-phase and in the low quality regions. After the dryout region the highest thermal resistance is presented by the heat transfer inside the helical pipe. There is a fairly good agreement between the two predictions except in the high quality region inside the helical pipe because of a different prediction of the dryout front.

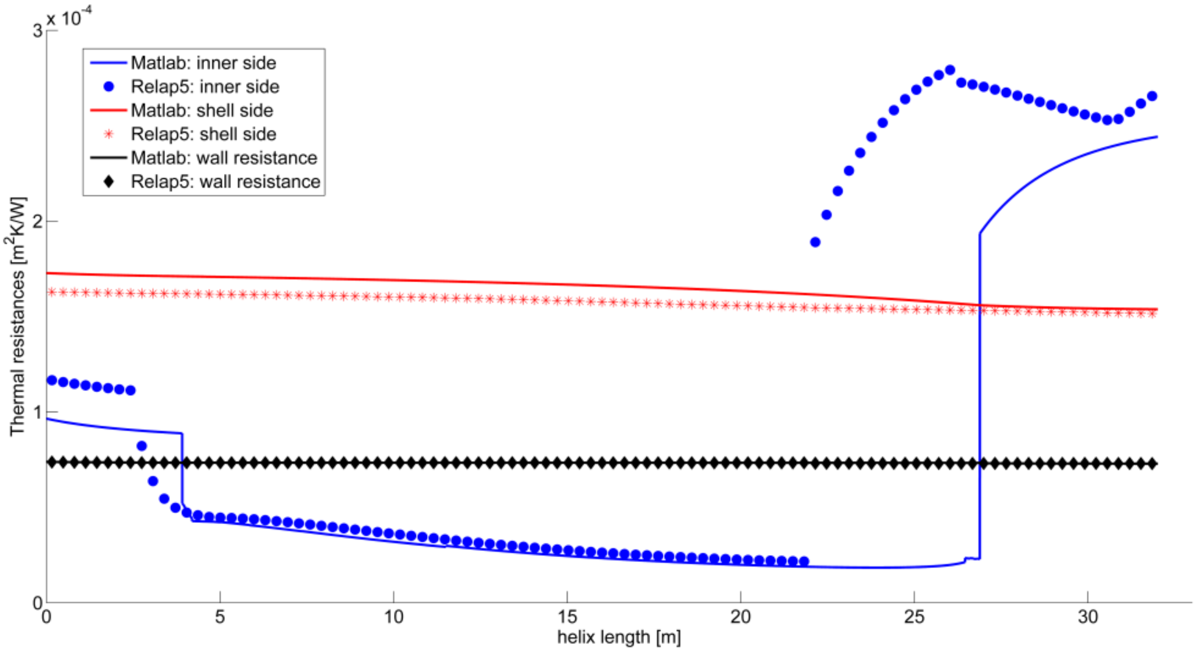

Figure 5. Thermal resistances.

Figure 6 shows the pressure profile as a function of the helix length. The present model, which is implemented with semiempirical correlations specific for the helical geometry, predicts an higher pressure loss compared with the RELAP5/Mod.3.3 code evaluation. The obtained result is consistent with the results available in literature where the frictional pressure drop for coiled geometries is higher than the one of a straight pipe both in single-phase and two-phase flow [11]. The relative difference between the predicted pressure drops is of the order of 25%.

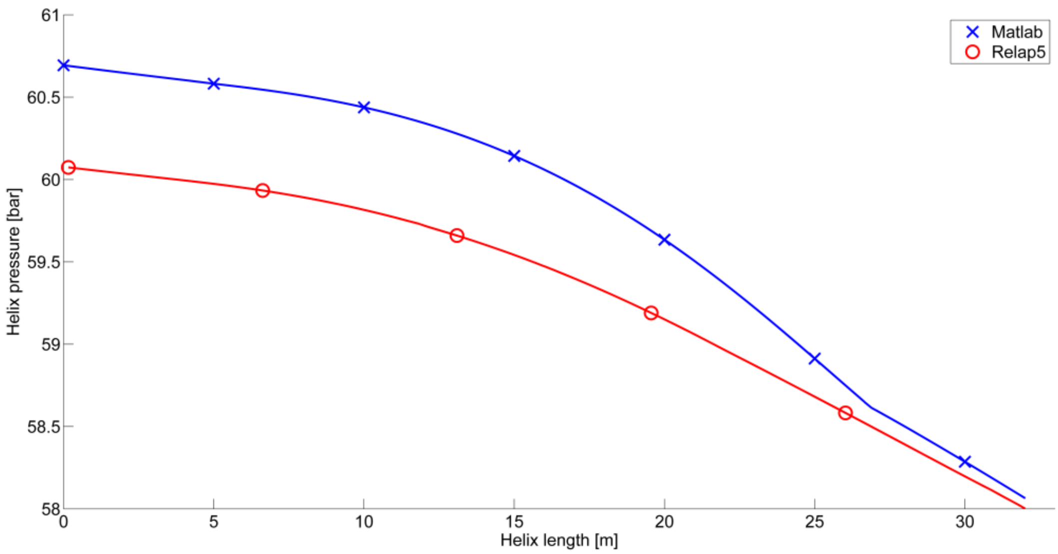

Figure 6. Internal pressure.

Figures 7 and 8 show the behaviour of the heat transfer coefficients for the fluids. The heat transfer coefficient predicted by the present model inside the helical coil is generally higher than the one calculated by RELAP5/Mod.3.3 because the correlations used in the model take into account the heat transfer enhancement due to centrifugal forces. The subcooled region is predicted at a lower distance by the code, and this explains why the heat transfer coefficients result higher than the present model prediction before the fifth meter. The high quality region difference of the temperature profiles shown in figure 5 is also present in the heat transfer coefficient prediction: the region of heat transfer degradation is predicted at an higher distance from the inlet by the present model. As far as the heat transfer coefficient on the shell side is concerned, the proposed model predicts lower values even if the discrepancy is low, with a relative difference lower than 6%.

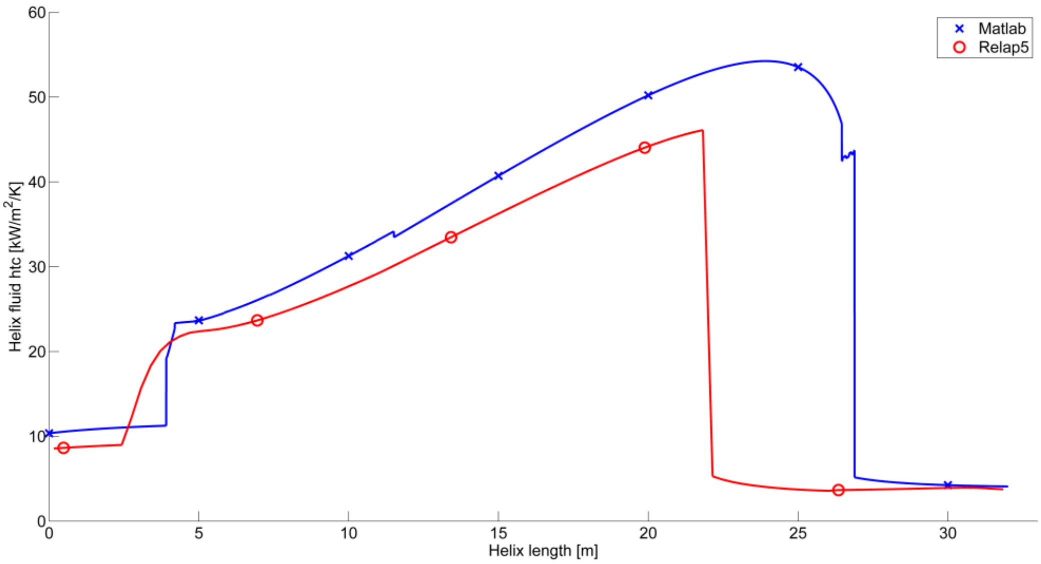

Figure 7. Helix side heat transfer coefficient.

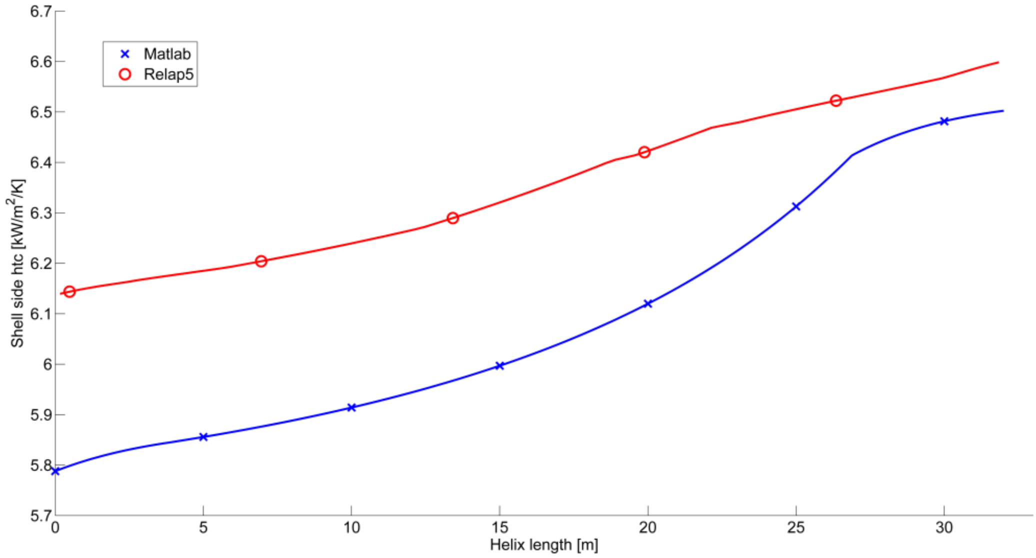

Figure 8. Shell side heat transfer coefficient.

Table 3 presents the comparison between the IRIS steam generator nominal data and the results of the two predictions. The proposed model demonstrates a good capability to reproduce the nominal data.

Table 3. Comparison between nominal and calculated values.

	Nominal data [2,16]	Present model	RELAP5/Mod.3.3
Power [kW]	190.5	189.6	182.03
Liquid single-phase linear power [kW/m]	\	5.97	5.91
Two-phase linear power [kW/m]	\	6.59	5.74
Steam temperature [°C]	317.0	315.5	297.1
Coolant outlet temperature [°C]	292.0	292.1	293.8
Steam pressure [bar]	58	58.06	58

Conclusions and future work

A model for the thermal hydraulic characterization of an helical coil steam generator has been proposed and a comparison with the commercial code RELAP5/Mod.3.3 has been done with respect to the nominal values of the IRIS SMR steam generator. The analysis shows a good agreement between the results except in the high quality region, where the dryout is predicted at two different distances from the inlet: in particular, the proposed model predicts the heat transfer degradation at an higher distance. The present model predicts higher values of in-coil heat transfer coefficient as well as higher pressure drops because of specific correlations for coiled geometries which take into account the effect of centrifugal forces. From the point of view of the shell side heat transfer coefficient the prediction of the two models is comparable. The comparison with the nominal data of the IRIS SMR steam generator show that the proposed model is able to predict the steam generator behaviour with a relative error for the transferred power lower than 1%. Future works to improve the model will be done to analyse the sensitivity of the steam generator performances with respect to the geometry and the fluid dynamics. Also, on the basis of the literature on helical coils, new heat transfer regimes like transition boiling and film boiling will be implemented, as well as by performing predictions for non-equilibrium conditions.

Acknowledgements
The present research has been supported by ENEA and by the ministry of economic development.

Nomenclature

D [m]	Diameter
d [m]	Diameter
P [m]	Pitch
R [m]	Radius
N	Number of
i [J/kg]	Enthalpy
P [bar]	Pressure
T [°C]	Temperature
c_p *[J/kg/K]*	Heat capacity at constant pressure
k [W/m/K]	Thermal conductivity
μ *[Pa s]*	Dynamic viscosity
ρ *[kg/m³]*	Density
σ *[N/m]*	Surface Tension
λ *[J/kg]*	Enthalpy of vaporization
χ_{tt}	Martinelli parameter
S	Suppression factor
F	Reynolds factor
Φ *[W]*	Heat flux
U [W/m²K]	Overall heat transfer coefficient
W [kg/s]	Flow rate

Subscripts

try	Attempt value
SG	Steam generator
IHX	Intermediate heat exchanger
c	Coil
e	External
II	Shell side
I	Inner side
p	Pipes
f	Fluid
sat	Saturation
sub	Subcooled
tp	Two phase
w	Wall

Dimensionless numbers

Re	Reynolds number
Nu	Nusselt number
Pr	Prandtl number

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