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The Thermal-hydraulic U-tube Steam Generator Model and Code UTSG-3 (Based on the Universally Applicable Coolant Channel Module CCM)

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1. Introduction

The development of effective theoretical models and (based on it) digital codes which have the potential to describe in a detailed way the overall transient and accidental behaviour of both the reactor core but also the main components of different NPP types has already very early been an important task in the field of reactor safety research.

Within this context an own special thermal-hydraulic model for one of these components, namely the natural circulation U-tube steam generator together with its main steam and feedwater systems, has been derived. The resulting digital code UTSG (Hoeld, 1978) could be used both in a stand-alone manner but also as a part of more comprehensive transient codes. As such an overall code the thermal-hydraulic GRS system code ATHLET (Burwell et al., 1989, Hoeld 1990b, later-on also Austregesilo et al., 2003 and Lerchl et al., 2009) has been chosen. Along with the ATHLET code a special high level simulation language has been derived too (Austregesilo & Voggenberger, 1990; Austregesilo & Hoeld, 1991), resulting in the basic module GCSM (General Control Simulation Module) provided for the flexible description of balance-of-plant (BOP) actions as caused, for example, by the control, limitation and protection systems of NPP-s, thereby connecting basic functional blocks (e.g., switches, function generators, adders, differentiators, integrators) in an appropriate way.

Due to the rising demands coming from safety-related research studies and based on the efforts of many years work of application both at the GRS but also at a number of other institutes of different countries the UTSG theory and code had to be and had been continuously extended, yielding in a first step as a very satisfactory version the code UTSG-2 (Hoeld 1990a).

From the experience gained during the continuous development of this code it arose after a while the question how to establish an own basic element which is able to simulate the thermal-hydraulic situation in a cooled or heated coolant channel in an as general as possible way so that it can be applied for any modular construction of complex thermal-hydraulic assemblies of pipes and junctions. As a result finally a theoretical model has been derived based on a theoretical drift-flux related three-equation mixture fluid approach. The

resulting (coolant channel module) package CCM should then allow to calculate automatically all the needed characteristic thermal-hydraulic parameters of a coolant channel within a complicated system of channels and loops and can thus be a valuable tool for the establishment of complex codes (Hoeld 2000a and b, Hoeld, 2007b).

By means of this generally applicable code package CCM the original digital code UTSG-2 (Hoeld, 1990a) could now be extended to a new advanced version, named UTSG-3. Thereby the module CCM has been implemented for the three characteristic channel elements of an U-tube steam generator, namely the primary and secondary side of the heat exchange region and the riser, distinguished by their key numbers (KEYBC = 1, 2 or 3). Besides this essential change the new UTSG version contains compared to the previous one a number of outstanding improvements, for example by providing a more advanced simulation of the top plenum, main steam line, feedwater injection and downcomer region together with a better simulation of the natural circulation situation. This new code could then also be (and has been) used to check the performance and validity of the code package CCM (and to verify it).

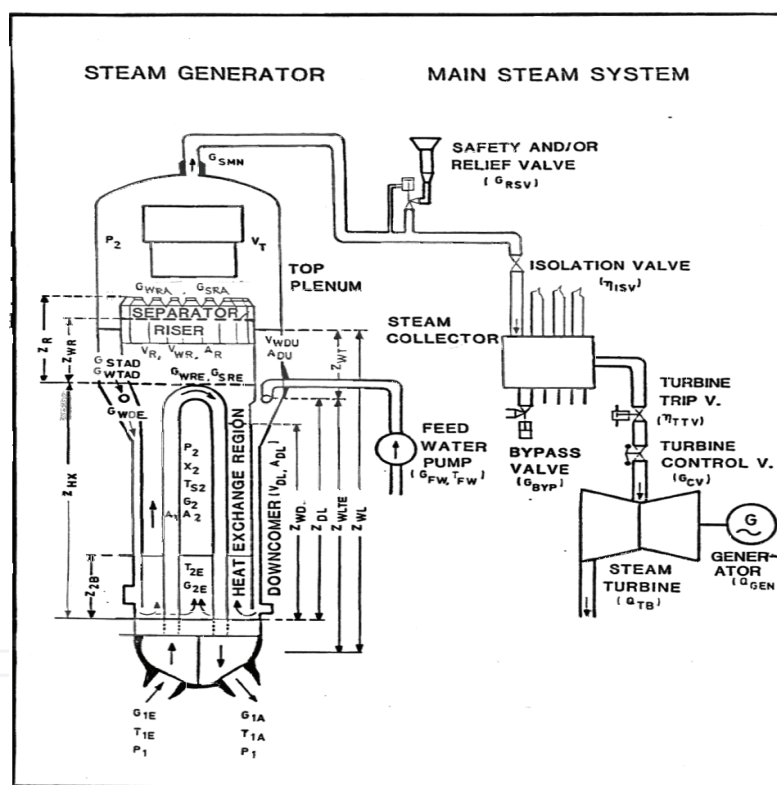


Fig. 1. Layout of a natural-circulation u-tube steam generator with its corresponding main steam system

The code UTSG-3 is, similarly as the code UTSG-2, based on the same U-tube and main steam system layout as sketched in fig.1. This means, the vertical natural-circulation U-tube steam generator is considered to consist of the following main parts:

- Primary or secondary heat exchange (HEX) region (evaporator)
- Bundle of U-tubes (index TW if restricting to a representative single U-tube)
- Riser (index R)
- Top plenum (index T) with its main steam system (index MS)

- Downcomer (index $D=DU+DL$) with feedwater system (index FW)
- Closed secondary loop (natural circulation)

The HEX region is separated by a number (N_{TUBES}) of vertical U-tubes into a primary and secondary loop, with the primary coolant flowing on the inner side (index 1) up and down, the secondary coolant on the outside (index 2) of these tubes upwards. The mixture flow from the HEX region is transported through a riser (index R) into the top plenum (TPL) where due to a separator the mixture flow is split into each phase, the water being yielded to the downcomer (DCM), the steam into the main steam system. The downcomer consist of an upper and (below the FW entrance) lower part (indices DU and DL). A feedwater system (index FW) transports sub-cooled water into the DCM where it is mixed with the downwards flowing saturated water from the TPL. To each of the U-tube steam generator systems (four in the case of a PWR) belongs a main steam line (with an isolation valve and a sequence of relief- and safety valves), all of them ending in a steam collector (SC) with steam lines to the bypass and steam turbine system (with steam bypass, turbine-trip and turbine control valves).

The flexibility of the finally established code combination UTSG-3/CCM will be demonstrated on a characteristic example by post-calculating with this combination a complicated transient situation after an initiation of the signal 'Loss of main feedwater in a PWR NPP with turbine trip and reactor scram'. A comparison with already existing (and tested out) UTSG-2 calculations for the same case has given an insight into the validity of the new code version UTSG-3 and helped thus to verify the general coolant channel module CCM too.

The here presented paper is concentrated on a very detailed description of the newest and advanced status of the theoretical U-tube steam generator model and its code version UTSG-3. Its theoretical background has already been partially published in (Hoeld, 2005 and 2007a). About the very important part of the UTSG-3 code, the coolant channel module CCM, here only a short review of its main elements can be given, an overall presentation will be placed in (Hoeld, 2011).

2. Basic equations of the thermal-hydraulic model

The thermal-hydraulic module CCM, part of UTSG-3, is based on the classical three-equation mixture fluid theory, starting from the conservation equations for mass, energy and momentum with respect to both single- and/or two-phase water/steam flow. Such conservation equations are mostly given in form of partial differential equations (PDE-s). In a closed loop further conservation equations are demanded, among them the 'volume balance equation', needed for the determination of the time-derivatives of the system pressure, and the TPL steam and water volumes. The solution of the momentum balance yields then local pressure difference terms over each channel element. It allows thus (in combination with the system pressure) to determine the behaviour of the (absolute) local pressure along a channel. A fifth physical law which is based on the fact that the sum of all pressure decrease terms along a closed loop must be zero. This gives the basis for the determination of the needed mass flow terms along a closed channel due to natural circulation by adapting these terms to the demand of this law.

The conservation equations are supported by adequate constitutive equations. These can be provided by tables for thermodynamic and transport properties of water and steam, a correlation package for single- and two-phase friction coefficients etc. In case of mixture

flow) the right choice of an adequate drift-flux package plays an important role among these constitutive equations, yielding not only the necessary equation for the fourth variable within the overall system of differential and constitutive equations but being also responsible for the possibility to simulate in a very detailed way stagnant or counter-current flow conditions or the appearance of entrainment within such a coolant channel.

2.1 Conservation equations for single- and two-phase flow

The three conservation equations (with a cross flow area A and perimeter U_{TW}) describe both the steady state ($L_{STS}=1$) and transient ($L_{STS}=0$) behaviour of three or (at two-phase flow) four characteristic fluid variables. These are the total mass flow G , the fluid temperature T (or enthalpy h) at single-phase or void fraction α at two-phase conditions and the local pressure P . At two-phase conditions for a fourth variable, i.e. the steam mass flow G_S , an own relation is asked. This can, for example, be a drift-flux correlation which yields (together with an adequate correlation for the phase distribution parameter C_0) the drift velocity v_D and thus also the steam mass flow G_S , hence closing the set of equations. They are interconnected by their definition equations.

2.1.1 Mass balance

$$\frac{\partial}{\partial t} \{A[(1-\alpha)\rho_w + \alpha\rho_s]\} + \frac{\partial}{\partial z} G = 0 \quad (1)$$

with the density terms ρ_w and ρ_s for saturated or sub-cooled water and saturated or superheated steam, the void fraction α and the cross flow area A which can eventually be changing along the coolant channel. It determines, after a nodalization, the total mass flow $G=G_w+G_s$ at node outlet in dependence of its node entrance value.

2.1.2 Energy balance

$$\frac{\partial}{\partial t} \{A[(1-\alpha)\rho_w h_w + \alpha\rho_s h_s - P]\} + \frac{\partial}{\partial z} (G_w h_w + G_s h_s) = q_L = U_{TW} q_F = A q \quad (2)$$

with the enthalpy terms h_w and h_s for saturated or sub-cooled water and saturated or superheated steam. As boundary values there have to be demanded either the 'linear power q_L ' or the 'heat flux q_F ' along the tube wall surface being directed (if positive) into the coolant, yielding thus the local 'power density term q ' (See also section 4.2.3).

As explained in very detail in connection with the establishment of CCM (Hoeld, 2011), after an appropriate finite-difference nodalization procedure it follows in the transient case then a set of nodal ordinary differential equations (ODE-s) of 1-st order for

- the mean nodal enthalpies (h_{wMn} , h_{sMn}) of either sub-cooled water (if $L_{FTYPE}=1$) or superheated steam ($L_{FTYPE}=2$) in the case of a single-phase flow situation and thus, by applying water/steam tables (for example the package MPP, see section 2.2.1), corresponding coolant temperature terms (T_{wMn} , T_{sMn}) too, or, at two-phase flow conditions ($L_{FTYPE}=0$), the mean nodal void fraction α_{Mn} over each node n and
- at the transition from single- to two-phase (and vice versa) for either the boiling boundary z_{BB} (if $\alpha=0$) or the mixture (or dry-out) level z_{ML} (if $\alpha=1$). Thereby it can be taken advantage

of the fact that these positions are marked by the fact that either the coolant enthalpy or temperature are limited by their saturation enthalpy or temperature values ($h_W = h'$ or $h_S = h''$ and $T_W = T_{SAT}$ or $T_S = T_{SAT}$), the void fractions by the limits 1 (or 0).

2.1.3 Momentum balance

$$\frac{\partial}{\partial t} (G_F) + \left(\frac{\partial P}{\partial z} \right) = \left(\frac{\partial P}{\partial z} \right)_A + \left(\frac{\partial P}{\partial z} \right)_S + \left(\frac{\partial P}{\partial z} \right)_F + \left(\frac{\partial P}{\partial z} \right)_X \quad (3)$$

describing either the pressure differences (at steady state) or (in the transient case) the change in the total mass flux ($G_F = G/A$) along a channel.

The general pressure gradient $\left(\frac{\partial P}{\partial z} \right)$ can be determined in dependence of

- the mass acceleration

$$\left(\frac{\partial P}{\partial z} \right)_A = - \frac{\partial}{\partial z} [(G_{FW} v_W + G_{FS} v_S)] \quad (4)$$

with v_S and v_W denoting steam and water velocities,

- the static head

$$\left(\frac{\partial P}{\partial z} \right)_S = - \cos(\Phi_{ZG}) g_C [\alpha \rho_S + (1-\alpha) \rho_W] \quad (5)$$

with Φ_{ZG} denoting the angle between upwards and flow direction. Then $\cos(\Phi_{ZG}) = \pm |z_{EL}| / z_L$ with the relative elevation height z_{EL} being positive at upwards flow

- the single- and/or two-phase friction term

$$\left(\frac{\partial P}{\partial z} \right)_F = - f_R \frac{G_F |G_F|}{2 d_{HW} \rho} \quad (6)$$

with the friction factor derived from corresponding constitutive equations (See section 2.2.2) and finally

- the direct perturbations $\left(\frac{\partial P}{\partial z} \right)_X$ from outside arising either by operating an external pump or the pressure adjustment due to mass exchange between parallel channels.

For more details see (Hoeld, 2011).

2.1.4 Volume balance

Volume balance considerations yield, in the case of a closed loop, to a fourth conservation equation. This is based on the (trivial) fact that the sum of water and steam volume must be equal to the total available volume. This is required for the determination of an absolute pressure parameter, e.g., the system pressure P_{SYS} in the top plenum of a steam generator (see section 4.5). From the pressure differences over different nodes of the loop following the discretization of the momentum balance equation the transient behaviour of the (absolute) local pressure values can then, in combination with P_{SYS} , be determined.

2.1.5 Balance of pressure decrease over channels within a closed loop

In a network of channels within a closed loop (for example for the simulation of natural circulation or the case of a 3D representation) a fifth conservation equation has to be taken

into account. This is founded on the physical law that the sum of all pressure decrease terms over all these channels must be zero. This is the basis for the treatment of the thermal-hydraulics of a channel according to the 'closed channel concept' (For more details see section 3.4).

2.2 Constitutive equations

For the exact description of the steady state and transient thermal-hydraulic behaviour of single- or two-phase fluids there are needed, besides the conservation equations, a number of mostly empirical (and pseudo-stationary) constitutive relations. Naturally, any effective correlation package can be used for this purpose. A number of such correlations have been developed at the GRS, adapted to the special requirements of the models by the author and thoroughly tested, showing very satisfactory results (Here only a short review of them will be given, for more details see Hoeld, 2011).

2.2.1 Thermodynamic and transport properties of water and steam

The different thermodynamic and transport properties for water and steam demanded by the conservation and constitutive equations have to be determined by applying adequate water/steam tables. This is, for light-water systems, realized in the code package MPP (Hoeld, 1996). It yields the wanted values such as the saturation temperature T_{SAT} , densities (ρ' , ρ''), enthalpies (h' , h'') for saturated water and steam with respect to their local pressure (P) and corresponding densities (ρ) and enthalpies (h) for sub-cooled water or superheated steam (index W and S) again with respect to their independent local parameters T and P (but also h and P).

For the solution of the conservation equations also time-derivatives of these thermodynamic properties which respect to their independent local parameters are demanded. They get, for example for the case of an enthalpy term h , the form

$$\begin{aligned} \frac{d}{dt} h(z,t) &= \frac{d}{dt} h[T(z,t), P(z,t)] = \left(\frac{\partial h}{\partial T} \right) \frac{d}{dt} T_{Mn}(t) + \left(\frac{\partial h}{\partial P} \right)_{Mn} \frac{d}{dt} P_{Mn}(t) \\ &= h^T \frac{d}{dt} T_{Mn}(z,t) + h^P \frac{d}{dt} P_{Mn}(z,t) \end{aligned} \quad (7)$$

Hence the thermodynamic water/steam tables should provide also the derivatives (T_{SAT}^P , ρ'^P , ρ''^P , h'^P , h''^P) for saturated water and saturated steam but also the partial derivatives (ρ^T , ρ^P , $c_P = h^T$, h^P) for subcooled water or superheated steam with respect to their independent parameters T and P (but also h and P). Additionally, corresponding thermodynamic transport properties such as 'dynamic viscosity' and 'thermal heat conductivity' (and thus the 'Prantl number') are asked from some constitutive equations too as this can be stated, for example, for the code packages MPPWS and MPPETA (Hoeld, 1996). All of them have been derived on the basis of tables given by (Schmidt and Grigull, 1982) and (Haar et al., 1988). Obviously, the CCM method is also applicable for other coolant systems (heavy water, gas) if adequate thermodynamic tables for this type of fluids are available.

2.2.2 Single- and two-phase friction factors

In the case of **single-phase flow** with regard to equation (6) the friction factor f_R will, as recommended by (Moody, 1994), be set equal to the Darcy-Weisbach single-phase friction factor f_{DW} being represented by

$$f_R = f_{DW} = \frac{1}{\xi^2} \quad (\text{at single-phase flow}) \quad (8)$$

with the parameter ξ depending on the Reynolds number $Re = Gd_H / (\Delta\eta)$ and the relative roughness (ε/d_H) of the wall surface. The factor ξ can be approximated by the relation

$$\begin{aligned} \xi &= 2 \log_{10} \left(\frac{d_H}{\varepsilon_{TW}} \right) + 1.14 & \text{if } Re > Re_{CTB} = 441.19 \left(\frac{d_H}{\varepsilon_{TW}} \right)^{1.1772} \\ \xi &= -2 \log_{10} \left(2.51 \frac{\xi}{Re} + \frac{\varepsilon_{TW}}{3.71 d_H} \right) & \text{if } Re < Re_{CTB} \end{aligned} \quad (9)$$

For **two-phase flow** conditions this factor can be extended to

$$f_R = f_{DW} \Phi_{2PF}^2 \quad (\text{at two-phase flow}) \quad (10)$$

with the single-phase part f_{DW} to be determined under the assumption that the fluid moves with the total mass flow G (= 100 % liquid flow) and the two-phase multiplier Φ_{2PF}^2 as given by (Martinelli-Nelson, 1948) as measured curves (dependent only on steam quality and pressure). For more details see (Hoeld, 1996 and 2010)

2.2.3 Drift flux correlation

In the case of two-phase flow, the three conservation equations (1), (2) and (3) have to be completed by an additional two-phase relation in order to obtain an adequate representation of the needed fourth variable G_s (Note: These correlations can be seen as a 'bridge' between G_s and α). This purpose can be achieved by any two-phase correlation, e.g. a slip correlation. However, to take care of stagnant or counter-current flow situations too an effective drift-flux correlation seemed here to be more appropriate.

For this purpose an own drift-flux correlation package has been established, named MDS (Hoeld, 2001 and 2002a). It is based on the result of a very comprehensive study (Hoeld et al., 1992) and (Hoeld, 1994) comparing different slip (6) and drift-flux (3) correlations with each other and also with a number (5) of available experimental data in order to check their validity over a wide range of application and to find which of them is most suited for incorporation into the CCM. Due to different requirements in the application of CCM it turned out that the drift-flux correlation package in the form of the 'flooding-based full-range' Sonnenburg correlation (1989) should be preferred. This correlation combines the common drift-flux procedure, being formulated by (Zuber-Findlay, 1965), and expanded by (Ishii-Mishima, 1980 and Ishii; 1990 etc.), with the modern envelope theory. The correlation in the final package MDS had to be rearranged in such a way that also the special cases of $\alpha \rightarrow 0$ or $\alpha \rightarrow 1$ (where its absolute values but also their gradients are demanded by CCM) could be treated. It should be also possible to install an inverse form (needed for example for steady state considerations) or to take care of considerations with respect to a possible entrainment.

Having now calculated for the steady state case G_s in dependence of α and in the transient situation α in dependence of G_s then all the other characteristic local two-phase parameters can be determined too (as shown, for example, in the tables of Hoeld, 2001 and 2002a). Especially the determination of the steam mass flow gradient plays an important part for the case that the entrance or outlet position of a SC is crossing a BC node boundary ($\alpha \rightarrow 0$ or \rightarrow

1). This term is thus an indispensable part in the nodalization procedure of the mixture-fluid mass and energy balance.

2.2.4 Heat transfer coefficients along a tube wall

The needed heat transfer coefficients (α_{TW1} , α_{TW2}) along different flow regimes (into and out of a tube wall) can be calculated automatically if applying an appropriate heat transfer coefficient package.

Hence, in connection with the development of the UTSG code (and thus also CCM) for this purpose a method how to get the necessary correlations for heat transfer coefficients along different flow regimes within such a channel had to be established, yielding finally an own very comprehensive heat transfer coefficient package, called HETRAC (Hoeld 1988a or already 1978). It combines, for example, especially for this purpose chosen HTC correlations for each possible flow situation within LWR-s and steam generators (i.e., into or out of heated or cooled tube walls or fuel elements) in a very effective way. Thereby adequate correlations for the cases of sub-cooled water, sub-cooled and nucleate boiling, onset of critical heat flux, transient or instable film boiling, stable film boiling, onset of superheating and superheated steam for different geometry constellations and over a wide range of input parameters (pressures, total and steam mass flows, coolant temperatures, wall temperatures or heat fluxes etc.) had to be selected. Special attention had to be given also to the case of a counter-current flow situation within a mixture flow. Some correlations (and thus also the package) take not only heat transfer from wall to the different phases but also between these phases into account. This can, for example, be clearly explained on hand of the Chen correlation (being a part of the HETRAC package) for the case of nucleate boiling. There, because of different conditions with regard to the heat input and mass flow along the channel, the entire heat transfer consists of two overlapping fundamental mechanisms, a macro- and a micro-convective part. Thus it is distinguished between a heat transfer mechanism where in the first case heat is inserted directly into a thin water film along the wall and transported through it to the other part of the mixture flow or where (at small mass velocities) already along the wall surface void is produced transporting then the heat by means of bubbles into the inner region of the fluid until they collapse there. The Chen correlation has been tested versus a number of 10 test series with an average deviation between calculated and measured heat transfer coefficients of about $\pm 12\%$.

In contrast to this classic method in a 'separate-phase' approach the heat transfer into the entire mixture fluid has to be assumed to be split completely (and not only for special correlations) into different contributions, i.e., into a part which is transferred directly from the wall to each of the two possible phases but also an inner heat transfer between these phases. This arises then to the difficult question what such special heat transfer coefficients for each phase should look like.

2.2.5 Material properties of a metallic tube

Density (ρ_{TW}) and specific heat (c_{TW}) of a metallic tube wall (needed in section 4.2.1) can be assumed to be known from input. They are (almost) independent from changes in the wall temperature. Since the heat conduction coefficients (λ_{TW}) show linear behaviour from tube wall temperatures their surface values can be described as

$$\lambda_{TWi}(z,t) \cong \alpha_{\lambda T} + \beta_{\lambda T} T_{TWi}(z,t) \quad (i=1, 2) \quad (16)$$

These parameters can then be determined on the basis of state parameters (temperatures, pressures etc.) taken from a recursion or time step before.

3. Coolant channel model and module CCM

The (double-precision) thermal-hydraulic coolant channel module CCM (and its single-precision version CCMS) belong to the essential parts of the UTSG theory and code. Within the code UTSG-3 this module is used for the description of the thermal-hydraulic situation of the single- and two-phase fluids flowing along the primary and secondary part of the HEX region and the riser.

The module has been developed with the aim to establish an effective and very universally applicable thermal-hydraulic digital code which should be able to simulate the thermal-hydraulic steady state and transient behaviour of a heated or cooled single- or a two-phase fluid flowing (in upwards, stagnant or even downwards direction) along a coolant channel and, normally, be also applicable for varying cross sections along the BC.

It can thus generally be an important basic element for the construction of complex thermal-hydraulic codes. Distinguished by corresponding key numbers (KEYBC =1, 2, etc.) it can, for example, be applied for the simulation of the steady state and transient behaviour of different types of steam generators with sometimes very complex primary and secondary loops (vertical U-tube, vertical once-through or horizontal VVER-440 ensembles) but also for the construction of 3D thermal-hydraulic codes which are needed for the simulation of non-symmetric single- or two-phase flow situations within large NPP (PWR or BWR) cores (Jewer et al., 2005). Special attention is given to the sometimes very complicated mass flow situations in these types. There the module can especially be significant for the main demand on such 3D codes, namely the automatic calculation of flow distribution into different parallel coolant channels after a non-symmetric perturbation of the entire system.

Together with the constitutive equations the discretization of the PDE-s yields a set of non-linear ordinary differential equations (ODE-s) of first order for the characteristic parameters of each of these SC and finally also BC nodes. This resulting set can then be combined with other sets of ODE-s and algebraic equations coming from additional parts of a complex theoretical model.

A very detailed description of the module CCM is given in (Hoeld, 2011), here only a short review of its main properties can be given.

3.1 Basic channel, subdivided into sub-channels. Spatial discretization by means of the PAX procedure

One of the fundamental assumptions of the code package CCM is that a coolant channel, called 'basic channel (BC)', can, according to their flow characteristics, be subdivided into a number of 'sub-channels (SC-s)' with saturated ($L_{FTYPE}=0$), sub-cooled ($L_{FTYPE}=1$) and superheated ($L_{FTYPE}=2$) flow conditions. All such SC-s can, however, belong to only two types of them, a SC with an only single-phase fluid (sub-cooled water or superheated steam) or a SC containing a saturated water/steam mixture. Theoretical considerations can thus be restricted to only these two types of subchannels.

Discretizing now (spatially) such a BC means that the main channel is further-on subdivided into a number (N_{BC}) of (not necessarily equidistant) BC nodes. This has the consequence that

each of the (N_{SCT}) SC-s within the BC is subdivided too, namely each of them into a number (N_{CT}) of SC nodes. The corresponding conservation equations for mass, energy and momentum (given in form of PDE-s of 1-st order, as shown in section 2.1) can then be discretized along these SC by means of a special spatial 'modified finite volume method' with the consequence that also the possibility of either a time-varying SC entrance or outlet positions has to be considered. Integrating the PDE-s over these SC nodes three types of discretization elements can be expected:

- Integration of functions within the PDE-s yields nodal mean values,
- integration over a gradient yields functions values at the boundaries of the node and finally
- integration of time-derivatives over eventually time-varying nodal boundaries (marking the SC entrance or outlet positions) yields time-derivatives of mean function values together with time-derivatives of these positions.

Appropriate methods had to be developed to connect the relations between the mean nodal and the node boundary function values. There exist different possibilities and concepts to solve this problem in an adequate way. Within the scope of CCM this has been done by a specially developed quadratic polygon approximation procedure (named 'PAX'), based on the assumption that the solution function of a PDE along a SC can be approximated by a quadratic polygon function over a segment which reaches not only over its node length but also (in order to avoid saw tooth like behaviour) over the adjoining one.

The resulting PAX procedure plays a very important role for the establishment of CCM, especially with respect to the fact that boiling or superheating boundaries can (in the transient case) move along the coolant channel and then also cross BC boundaries (For more details see Hoeld, 2011).

3.2 Data transfer between the calling program and CCM

The list of the needed BC input parameters to CCM, demanded as boundary conditions, and the resulting output data to be transferred from CCM to the main program is presented in very detail in (Hoeld, 2007c and Hoeld, 2011). The allocation of these data to the input data of the different SC-s and the collection of the resulting SC data stored into corresponding BC output data is then done automatically within the module CCM. The user does not need to undertake any special actions in this context.

As a result of the integration procedure and the application of CCM the most characteristic steady state and transient SC (and thus BC) parameters of single- and two-phase fluid can be expected. Additionally, the transient calculation with CCM yields the time-derivatives

$$\frac{dz_{\text{CA}}}{dt} \text{ (for each SC within a BC), } \frac{dT_{\text{BMk}}}{dt} \text{ and } \frac{d\alpha_{\text{BMk}}}{dt} \quad (k=1, N_{\text{BT}}) \quad (12)$$

They are, together with other characteristic channel parameters, then needed within the overall set of differential and constitutive equations in the main code.

3.3 Pressure profile along a BC

An important chapter had to be devoted to the handling of the pressure distribution along the channel. Among a special renormalization procedure has been introduced in order to compensate also pressure drop contributions from spacers, tube bends etc., terms which are analytically difficult to be represented.

After having solved within each intermediate time step the mass and energy balance equations separately (and not simultaneously) from the momentum the (now exact) nodal SC and BC pressure difference terms ($\Delta P_{Nn} = P_{Nn} - P_{Nn-1}$ and ΔP_{BNn}) can (for both single- or two-phase flow situations) be determined if discretizing the momentum balance eq.(3) by integrating over the corresponding SC nodes. The total pressure difference $\Delta P_{BT} = P_{BA} - P_{BE}$ between BC outlet and entrance follows then from the relation

$$\Delta P_{BT} = \Delta P_{PBT} - \Delta P_{GBT} \quad (\text{Note: } \Delta P_{GBT,0} = 0 \text{ at steady state conditions}) \quad (13)$$

with the parameter

$$\Delta P_{PBT} = \Delta P_{SBT} + \Delta P_{ABT} + \Delta P_{XBT} + \Delta P_{FBT} + \Delta P_{DBT} \quad (14)$$

(Note: $\Delta P_{PBT,0} = \Delta P_{BTIN,0}$ at steady state conditions)

consisting of terms from static head (ΔP_{SBT}), mass acceleration (ΔP_{ABT}), wall friction (ΔP_{FBT}) and external pressure accelerations (ΔP_{XBT} , pump or other perturbations from outside) and (in the transient case) the term ΔP_{GBT} taking care of the time-dependent changes in total mass flux along a BC.

Regarding, however, the friction correlations, there arises the problem how to consider correctly contributions from spacers, tube bends, abrupt changes in cross sections etc. as well. The entire friction pressure decrease (ΔP_{FBT}) along a BC can thus never be described in a satisfactory manner solely by analytical expressions. Hence, a special renormalization procedure had to be derived in order to compensate also pressure drop contributions from spacers, tube bends etc., terms which are analytically difficult to be represented. To minimize these uncertainties a further friction term will be included into these considerations having the form

$$\Delta P_{DBT} = (f_{FMP,0} - 1) \Delta P_{FBT} + \Delta P_{FADD} \quad (15)$$

This means that eq.(14) is either supplemented with an additive term (index FADD) or the friction parts are provided with a multiplicative factor $f_{FMP,0}$. Which of them should prevail can be governed from outside by an input coefficient $\varepsilon_{DPZ} = \varepsilon_{DPZI}$. Thereby, the additive part will be assumed to be proportional to the square of the total coolant mass flow (e.g., at BC entrance)

$$\Delta P_{FADD} = -f_{ADD,0} Z_{BT} \left(\frac{G_F |G_F|}{2\rho d_{HW}} \right)_{BE} \quad (16)$$

For steady state conditions the pressure difference term over the entire BC is given by the input ($\Delta P_{BT,0} = \Delta P_{BTIN}$). Since $\Delta P_{GBT,0} = 0$ the steady state total additional pressure term ΔP_{DBT} follows from eq.(14). If defining the additive steady state pressure difference $\Delta P_{FADD,0}$ to be the $(1 - \varepsilon_{DPZ})$ -th part of the total additional friction term

$$\Delta P_{FADD,0} = (1 - \varepsilon_{DPZ}) \Delta P_{DBT,0} \quad (17)$$

the corresponding additive friction factor $f_{ADD,0}$ follows directly from eq.(16), the multiplicative one ($f_{FMP,0}$) from the combination of the eqs.(15) and (17).

The validity of both friction factors can, for example, be expanded to transient situations too by assuming that they should remain time-independent. Then, finally, the wanted nodal pressure decrease terms can be determined for both steady state but also transient situations. The absolute nodal pressure profile P_{Bk} along a BC (needed at the begin of the

next time step for the determination of the constitutive equations) can then finally be established by adding now the resulting nodal BC pressure difference terms to the (time-varying) system pressure $P_{\text{SYS}}(t)$ (given from outside as boundary condition with respect to a certain position of the BC, e.g. at the TPL, see section 4.4)

In the transient case (as this can be seen from the momentum balance eq.(3)) a further pressure difference term (ΔP_{GBT}) has to be considered. This term has to take care of the time-dependent changes in total mass flux along a BC (caused by the direct influence of changing nodal mass fluxes). It can be described (in an approximate way) in dependence of the change of a fictive mean mass flux term (G_{FBMT}) over the entire BC. The term ΔP_{GBT} is defined as follows

$$\begin{aligned}\Delta P_{\text{GBT}} &= \int_0^{z_{\text{BT}}} \frac{d}{dt} G_{\text{FB}}(z,t) dz \cong \sum_{n=1}^{N_{\text{GCT}}} \sum_{n=1}^{N_{\text{CT}}} \Delta z_{\text{Nn}} \frac{d}{dt} G_{\text{FBMn}} \\ &= z_{\text{BT}} \frac{d}{dt} G_{\text{FBMT}} = \sum_{k=1}^{N_{\text{BT}}} \Delta z_{\text{Bk}} \frac{d}{dt} \frac{G_{\text{BMk}}}{A_{\text{BMk}}}\end{aligned}\quad (18)$$

As explained below, the resulting total pressure drop along the entire BC is then the key for the application of the module within an ensemble of channels.

3.4 BC entrance mass flow (Open and closed channel concept)

Normally, at transient conditions, the BC entrance mass flow $G_{\text{BE}} = G_{\text{BEIN}}$ and one of the BC entrance or outlet pressure terms (P_{BEIN} or P_{BAIN}) can be expected to be known from input. Then also the ΔP_{GBT} can be determined according to eq.(18). This constellation can be seen as an 'open channel' situation, the procedure based on it as an 'open channel' concept. This allows then to calculate directly the nodal SC and thus also BC pressure profiles and, finally, from eq.(13), the total pressure difference ΔP_{BT} over the entire BC, fixing then the second missing pressure term (P_{BE} or P_{BA}) too.

In contrast to this normal situation a new concept has been implemented into CCM, called 'closed channel concept'. This concept should take care of special situations where only the two BC entrance and outlet pressure terms (P_{BEIN} and P_{BAIN}) can be expected to be known from input, thus also the difference over the entire BC pressure ($\Delta P_{\text{BTIN}} = \Delta P_{\text{BT}} = P_{\text{BAIN}} - P_{\text{BEIN}}$). Then, according to eq.(14), also the term ΔP_{PBT} is given. Since now the BC entrance mass flow G_{BE} is not explicitly known this parameter has then to result from adequate considerations, by establishing a different concept (the 'closed channel concept'). This follows from eq.(18) by deriving a relation which is the centre of this concept', demanding that

$$z_{\text{BT}} \frac{d}{dt} G_{\text{FBMT}} = \Delta P_{\text{GBT}} = \Delta P_{\text{PBT}} - \Delta P_{\text{BTIN}} \quad (\text{'closed channel' conditions}) \quad (19)$$

This means that the total mass flow terms along a BC (and thus also at its entrance) must be adapted in such a way that this for the 'closed channel criterion' important condition remains valid. This means, if it is possible to find a relation between the term $\frac{d}{dt} G_{\text{FBMT}}$ and $\frac{d}{dt} G_{\text{BE}}$, then the wanted mass flow time-derivative term at BC entrance can be determined. One practical method how this problem can, at present, be solved is shown for the case of the establishment of UTSG-3 code, later-on, in section 4.7.

It has to be noted that the application of the 'closed channel' method can, however, be restricted to only one 'characteristic' channel among a sequence of channels within a closed loop. The pressure drops of the remaining (eventually very complex) parts of the loop can then be calculated by means of the normal 'open loop' concept so that the demanded pressure decrease part of such a characteristic 'closed' channel is determined from the fact that the sum of all pressure decrease terms along the closed loop must be zero. It is thus equal to the sum of the negative values of the remaining terms of these channels.

This method makes sure that measures with regard the entire closed loop do not need to be taken into account simultaneously but can be treated separately. In contrast to this procedure in the most other thermal-hydraulic approaches (see for example the 'separate-phase' models) all the pressure differences have to be handled for all the elements of the entire closed loop (together with their BOP systems) together, with the consequence of a sometimes very CPU-time consuming method.

Similar considerations can be undertaken for the case that the automatic mass flow distribution into the different entrances of a set of parallel channels is asked. Thereby the friction coefficients can be determined with respect to a representative average channel. These factors can then be assumed to be valid for all the parallel channels (which will be of the same geometry and thus friction type). In order to obey the demands of the equal total pressure difference over these channels the wanted mass flow distribution can then be calculated by applying the above described 'closed channel criterion'. (See e.g. Hoeld, 2004a and Jewer et al., 2005).

4. Theoretical U-tube steam generator model

4.1 Primary and secondary HEX coolant channels

The heat exchange (or evaporator) region (HEX) is assumed to consist of a number of equidistant vertical U-tubes (N_{TUBES}) of the (average) length of $2z_{\text{HXU}}$. The primary fluid flows on the inner side of these tubes (with the constant cross section A_1) upwards and then downwards. The secondary side (with the total length z_{HX} and constant cross section A_2) can be subdivided into N_{ZHX} equidistant nodes with the nodal (BC) length $\Delta z_{\text{B2k}} = \Delta z_{\text{HX}} = z_{\text{HX}}/N_{\text{ZHX}}$ (with $k=1, N_{\text{ZHX}}$). The primary nodes of the U-tubes have the same length except the two upper nodes (Δz_{HXU}) which take the bend of the U-tubes into account.

In the here presented advanced UTSG-3 version the wanted differential (and constitutive) equations for the primary and secondary HEX coolant channels are now automatically determined by calling the coolant channel module CCM (Hoeld, 2000, 2001 and 2010). As already explained in chapter 3 for this purpose only a number of easily available boundary conditions have to be provided to the two CCM modules (distinguished by their key numbers KEYBC=1 and 2). These are

- the primary and secondary HEX inlet temperatures $T_{1\text{E}}$ and $T_{2\text{E}}$ (or enthalpies $h_{1\text{E}}$ and $h_{2\text{E}}$), mass flows ($G_{1\text{E}}$ and $G_{2\text{E}}$) and pressures ($P_{1\text{E}}$ and $P_{2\text{E}}$),
- the heat power profile along the primary and secondary HEX side (i.e., the mean nodal power values and the power densities at both sides of the HEX entrance, as determined in combination of the heat transfer considerations (explained in the section below)

and (in the steady state case)

- the total nominal (steady state) heat power $Q_{\text{NOM},0}$ (needed for normalization purposes).

Knowing the primary and secondary nodal HEX fluid temperatures the corresponding nodal primary and secondary heat flux values (into and out of a single U-tube) can then be determined (section 4.3.3 below) and thus also the nodal heat power terms being needed as input to the coolant channel module CCM.

4.2 U-tube ensemble (Heat transfer between primary and secondary HEX side)

An effective description of the heat transfer between the primary and secondary HEX side (index 1 and 2) across a U-tube bundle has already very early been an interesting task within the theoretical treatment of steam generator models (Hoeld, 1978, 1990a, 2002b). Thereby the U-tube ensemble may consist of a number (N_{TUBES}) of cylindrical U-tubes (now fixed inner and outer cross sections A_1 and A_2). The primary fluid is assumed to flow (both up- and then downwards) on the inner, the secondary fluid (only upwards) on the outer side of the tubes.

Among the U-tube bundle a representative single U-tube (index TW) has to be ascertained, subjected to the same discretization as applied for the primary HEX channel. This means its length ($2z_{\text{HXU}}$) is subdivided axially in the same way, i.e., into $2N_{\text{ZHX}}$ nodes ($n=1, 2 \cdot N_{\text{ZHX}}$). The node positions of the channels are then given by $z_{\text{TW}in}$ (with $i = 1, 2$ and $n = 0, 1, N_{\text{ZHX}}$), having the node length $\Delta z_{\text{TW}in} = \Delta z_{\text{HX}}$ (for both inner and outer tube wall sides $i=1, 2$) but, at the positions $n = N_{\text{ZHX}}$ and $n = N_{\text{ZHX}}+1$ (where the tube bow has to be taken into account), $\Delta z_{\text{TW}1n} = \Delta z_{\text{HXU}}$. The (metallic) wall of such a single U-tube (index TW) with the (now also fixed) inner and outer radii $r_{\text{TW}1}$ and $r_{\text{TW}2}$, a wall thickness $\Delta r_{\text{TW}} = r_{\text{TW}2} - r_{\text{TW}1}$ and the perimeters $U_{\text{TW}1} = 2r_{\text{TW}1}\pi$ and $U_{\text{TW}2} = 2r_{\text{TW}2}\pi$ can be assumed to consist of N_{RT} layers. The nodal inner and outer (wetted) surfaces of the tube wall over each axial node n can then be represented by

$$A_{\text{TW}2n} = A_{\text{TW}1n} \frac{r_{\text{TW}1}}{r_{\text{TW}2}} = U_{\text{TW}2} \Delta z_{\text{TW}n} \quad (n = 1, 2N_{\text{ZHX}}) \quad (20)$$

The original number of U-tubes can then be estimated from the known inner cross flow area

$$N_{\text{TUBES}} = \frac{A_1}{\pi r_{\text{TW}1}^2} \quad (21)$$

It will be assumed that (in correspondence with the BC node boundary positions) at each nodal position $n = 0 (=BE), 1, 2 \cdot N_{\text{ZHX}}$ heat will be transported from the primary coolant to the wall surface (with its temperature $T_{\text{TW}1n}$), conducted through the wall and finally send from the secondary wall surface (with the temperature $T_{\text{TW}2i}$) to the secondary coolant side. Characteristic nodal power parameters such as the 'nodal linear power values q_{Lin} ' and thus also 'nodal heat flux values $q_{\text{Fin}} = q_{\text{Lin}}/U_{\text{TW}i}$ (with $i=1, 2$ describing the inner and outer tube wall surface) have then to be determined from corresponding heat transfer considerations, based on heat transfer coefficients and material properties of the metallic wall as described in the sections 2.2.4 and 2.2.5.

4.2.1 Fourier heat conduction equation

As input to the energy balance eq.(2) the 'linear power value q_L ' (or the corresponding 'heat flux value $q_F = q_L/U_{\text{TW}}$ along each nodal perimeter U_{TW}) are demanded. They describe the heat transferred at a certain axial position z (for example at their node boundaries) into or

out of the coolant channel, i.e. from a heated or cooled surface (for example from or into an U-tube wall or out of the casing of a fuel rod). These terms, but also the radial tube wall temperatures $T_{TW}(z,r,t)$, can be determined by solving an adequate 'Fourier heat conduction equation'. This has (by neglecting the heat transfer in axial direction) the form

$$\rho_{TW} c_{TW} \frac{\partial}{\partial t} T_{TW}(z,r,t) = \frac{1}{r} \frac{\partial}{\partial r} \left[r \lambda_{TW} \frac{\partial}{\partial r} T_{TW}(z,r,t) \right] \quad (22)$$

with its initial condition

$$T_{TW}(z,r,t=0) = T_{TW,0}(z,r) \quad (\text{Index 0: Steady state}) \quad (23)$$

and the (for a tube wall characteristic) boundary conditions for a (single) U-tube. They can be represented together with the heat transfer relations at the inner and outer surfaces

$$-\lambda_{TW1} \frac{\partial}{\partial r} T_{TW1} = -\lambda_{TW1} T_{TW1}^{(r)} = q_{FTW1} = \alpha_{TW1} (T_1 - T_{TW1}) \quad (24)$$

$$-\lambda_{TW2} \frac{\partial}{\partial r} T_{TW2} = -\lambda_{TW1} T_{TW2}^{(r)} = q_{FTW2} = \alpha_{TW2} (T_{TW2} - T_2) \quad (25)$$

The mean tube wall temperature T_{TWM} (over the entire wall thickness $\Delta r_{TW} = r_{TW2} - r_{TW1}$) can be defined by

$$\frac{1}{2} (r_{TW2}^2 - r_{TW1}^2) T_{TWM}(z,t) = \int_{r_{TW1}}^{r_{TW2}} r T_{TW}(z,r,t) dr \quad (26)$$

The (positive) local heat fluxes q_{FTW1} into and q_{FTW2} out of such a (single) TW, the local primary and secondary surface temperatures T_{TW1} and T_{TW2} and, in the transient case, the time derivatives of the mean tube layer temperatures are expected to result from the heat transfer procedure in dependence of the local primary and secondary fluid temperatures T_1 and T_2 to be known from outside.

4.2.2 Heat transfer across a (representative) single cylindrical U-tube

To solve the partial differential eq. (PDE) of 2-nd order of the Fourier heat conduction eq.(22), the experience showed that for the special case of an U-tube steam generator it is sufficient to represent the tube by a single layer ($N_{RT}=1$). It can even be assumed that in a transient situation, because of the relative thin tube wall and thus very small overall heat capacity of the material, the layer thickness Δr_{TW} can be neglected ($N_{RT} = 0$). This means that (similar to the steady state solution, $L_{STS}=1$) in this special case an overall heat transfer coefficient α_{OV1} (or α_{OV2}) can be derived by setting the time-derivative of the mean tube wall temperature within the Fourier heat conduction eq.(27) equal to 0 without losing on exactness, i.e., it can be treated in a pseudo-stationary way. Examples for the case that a tube wall is subdivided into more than one layer, e.g. $N_{RT}=2$ or 3, are presented by (Hoeld, 1978), heat transfer out of a fuel rod is described in (Hoeld, 2004).

Integrating now in the case $N_{RT}=1$ the Fourier heat conduction eq.(22) over the layer thickness and inserting from the boundary conditions (24) and (25) yields finally the (ordinary) differential equation (ODE)

$$\frac{d}{dt} T_{TWM} = \frac{q_{FTW1} - \frac{r_{TW2}}{r_{TW1}} q_{FTW2}}{(1 + \frac{1}{2} \frac{\Delta r_{TW}}{r_{TW1}}) \Delta r_{TW} \rho_{TW} C_{TW}} \quad (L_{STS} = 0 \text{ and } N_{RT} = 1) \quad (27)$$

A special polygon approximation procedure had to be established in order to get a connection between the resulting mean and boundary layer values and thus to determine all the characteristic parameters of the tube wall (q_{FTW1} , q_{FTW2} , T_{TW1} and T_{TW2}) and (in the transient case) also the time-derivative of the mean TW temperature, needed for the next integration step. Thereby it had been assumed that the radial temperature profile $T_{TW}(r)$ along the TW can (for the cases $N_{RT} \geq 1$) be approximated by a quadratic polygon. If introducing the coefficients

$$C_{H1} = \frac{1}{2} \Delta r_{TW} \frac{\alpha_{TW1}}{\lambda_{TW1}} \quad (28)$$

$$C_{H2} = \frac{1}{2} \Delta r_{TW} \frac{\alpha_{TW2}}{\lambda_{TW2}} = C_{H1} \frac{r_{TW2}}{r_{TW1}} \frac{\alpha_{TW2}}{\alpha_{TW1}} \quad (29)$$

this procedure yields with regard to the boundary conditions (24) and (25) a relation connecting the heat fluxes q_{FTW1} and q_{FTW2} with the corresponding coolant temperatures T_1 and T_2

$$(1 + C_{H1}) \frac{q_{FTW1}}{\alpha_{TW1}} + (1 + C_{H2}) \frac{q_{FTW2}}{\alpha_{TW2}} = T_1 - T_2 \quad (L_{STS} = 0 \text{ and } N_{RT} = 1) \quad (30)$$

and if applying it to the definition eq.(26) for the mean layer temperature T_{TWM}

$$(1 + \frac{1}{6} \frac{r_{TW2}}{r_{TW1} + r_{TW2}} C_{H1}) \frac{q_{FTW1}}{\alpha_{TW1}} - (1 + \frac{1}{6} \frac{r_{TW1}}{r_{TW1} + r_{TW2}} C_{H2}) \frac{q_{FTW2}}{\alpha_{TW2}} = T_1 + T_2 - 2 T_{TWM} \quad (L_{STS} = 0 \text{ and } N_{RT} = 1) \quad (31)$$

Hence, if replacing in eq.(31) the term q_{FTW2} by inserting from eq.(30) yields finally

$$\frac{q_{FTW1}}{\alpha_{TW1}} = \frac{(1 + \frac{1}{6} \frac{r_{TW1}}{r_{TW1} + r_{TW2}} C_{H2})(T_1 - T_2) + (1 + C_{H2})(T_1 + T_2 - 2 T_{TWM})}{(1 + C_{H1})(1 + \frac{1}{6} \frac{r_{TW2}}{r_{TW1} + r_{TW2}} C_{H2}) + (1 + C_{H2})(1 + \frac{1}{6} \frac{r_{TW1}}{r_{TW1} + r_{TW2}} C_{H1})} \quad (L_{STS} = 0 \text{ and } N_{RT} = 1) \quad (32)$$

and thus, by means of eq.(25), also a relation for the heat flux q_{FTW2} (Note: In the transient case and $N_{RT}=1$ the mean tube wall temperature T_{TWM} is known from the integration procedure). Finally, from the eqs.(24) and (25) the surface temperatures T_{TW1} and T_{TW2} and also their temperature gradients and from eq.(27) the corresponding mean tube wall temperature time-derivative, needed for the next integration step, can be determined.

For the special case of a **steady state situation** but also for case $N_{RT} = 0$ a pseudo-stationary treatment of the Fourier heat conduction equation (22) and thus of eq.(28) can be applied, setting there the time derivative equal to 0,. This yields then

$$q_{FTW2} = \frac{r_{TW1}}{r_{TW2}} q_{FTW1} \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \quad (33)$$

Hence, if starting from the boundary conditions (24) and (25) it is obvious that the (steady state) gradients of the TW entrance and outlet temperatures are then equal to

$$T_{TW1}^{(r)} = \frac{T_{TW2} - T_{TW1}}{\Delta r} = T_{TW2}^{(r)} = - \frac{q_{FTW1}}{\lambda_{TW1}} = - \frac{q_{FTW2}}{\lambda_{TW2}} \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \quad (34)$$

This means that the shape of the radial temperature distribution can be represented for this case by a linear function. From the relations (33) and (34) it follows

$$\lambda_{TW2} = \lambda_{TW1} \frac{r_{TW1}}{r_{TW2}} \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \quad (35)$$

Inserting from eq.(33) into eq.(35) allows deriving relations for q_{FTW1} and, if looking again at eq.(33), also for q_{FTW2}

$$q_{FTW1} = \frac{r_{TW2}}{r_{TW1}} q_{FTW2} = \frac{\lambda_{TW1}}{\lambda_{TW2}} q_{TW2} = \alpha_{OV1} (T_1 - T_2) \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \quad (36)$$

Thereby an 'overall heat transfer coefficient' α_{OV1} has been introduced having the form

$$\frac{1}{\alpha_{OV1}} = \frac{1}{\alpha_{TW1}} + \frac{1}{2} \frac{\Delta r_{TW}}{\alpha_{TW1}} \left(1 + \frac{r_1}{r_2} \frac{\lambda_{TW1}}{\lambda_{TW2}}\right) + \frac{r_{TW1}}{r_{TW2}} \frac{1}{\alpha_{TW2}} \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \quad (37)$$

Now, from the boundary conditions (24) and (25) the corresponding (steady state) surface temperatures T_{TW1} and T_{TW2} can be determined too.

Rearranging eq.(31) by inserting for the power flux terms q_{FTW1} , q_{FTW2} and C_{H1} from the eqs.(33), (37) and (29) yields finally the relation for the (steady state) mean layer temperature

$$\begin{aligned} T_{TWM} &= \frac{1}{2} (T_{TW1} + T_{TW2}) - \frac{1}{24} \frac{\Delta r_{TW}}{r_{TW1} + r_{TW2}} (T_{TW1} - T_{TW2}) \\ &= \frac{1}{2} (T_{TW1} + T_{TW2}) - \frac{1}{24} \frac{\Delta r_{TW}}{r_{TW1} + r_{TW2}} \frac{\Delta r_{TW} q_{FTW1}}{\lambda_{TW1}} \quad (L_{STS} = 1 \text{ or } N_{RT}=0) \end{aligned} \quad (38)$$

Knowing now the nodal 'linear power' terms q_{LTWin} and if assuming a linear behaviour of this linear power within a BC node the nodal power terms ΔQ_{TW1n} and ΔQ_{TW2n} (per node $n=1, 2N_{ZHX}$) into the inner side of a (single) tube wall surface A_{TW1n} and out of its outer surface A_{TW2n} are then given by

$$\Delta Q_{TWin} = \frac{1}{2} \Delta z_{TWin} (q_{LTWin} + q_{FLWin-1}) = \frac{1}{2} (U_{TWin} q_{FTWin} + U_{TWin-1} q_{FTWin-1}) \quad (39)$$

(i=1,2 ; n =1, 2*N_{ZHX})

yielding finally the corresponding total power terms into and out of a single tube

$$Q_{TWi} = \sum_{n=1}^{2*N_{ZHX}} \Delta Q_{TWin} \quad (i = 1, 2) \quad (40)$$

4.2.3 Nodal and total power out and into the adjoining coolant channels

There are always uncertainties in describing the heat transfer coefficients analytically in an exact way. This is, for example, very often caused by the very complicated geometrical situation within a complex system so that it is not possible to take care in a sufficient way of the influence of all the spacers, tube bends. It can thus be expected that the resulting primary and secondary total heat power terms (based on these HTC calculations) will not be able to simulate the real situation. Hence, to circumvent these difficulties, the number of U-tubes (N_{TUBES}) will be provided with a normalization factor $\varepsilon_{Q_{TW}}$ which should help to adapt the power terms in a correct way. Thereby it can be taken advantage from the fact that at steady state (index 0) these power terms have to correspond to a given nominal power term $Q_{NOM,0}$, usually equal to the from input known steady state power terms $Q_{1,0} = Q_{2,0}$. Hence, this factor can be determined from this steady state considerations as

$$\varepsilon_{Q_{TW}} = \frac{Q_{NOM,0}}{N_{TUBES} Q_{TW1,0}} \quad (41)$$

Since the nodal power terms ΔQ_{TWin} depend on the heat transfer and heat conduction coefficients which in turn are functions of the tube wall (and coolant channel) temperatures the parameters ΔQ_{TW1n} , Q_{TW2n} and thus also $\varepsilon_{Q_{TW}}$ have (in the steady state start calculation) to be determined in a recursive way. After the convergence the renormalization factor $\varepsilon_{Q_{TW}}$ will then be assumed to remain valid also for the transient case.

Applied these results to the U-tube bundle of the here presented steam generator yields the nodal heat flux values q_{Fin} (with $i=1$ if KEYBC=1 and $i=2$ if KEYBC=2) taken from the primary HEX channel at each BC node n , transported across the N_{TUBES} U-tubes and inserted into the second HEX coolant channel

$$q_{F1n} = \varepsilon_{Q_{TW}} N_{TUBES} q_{FTW1n} \quad (n = 0, 1, 2 \cdot N_{ZHX}) \quad (42)$$

and (because of the co- and counter-current contributions of the U-tube to the secondary side)

$$q_{F2k} = \varepsilon_{Q_{TW}} N_{TUBES} (q_{FTW2k} + q_{FTW2j}) \quad (k = 0, 1, N_{ZHX} \text{ and } j = 2 \cdot N_{ZHX} - k) \quad (43)$$

the nodal power terms

$$\Delta Q_{1n} = \frac{1}{2} A_{TW1n} (q_{F1n} + q_{F1n-1}) \quad (n = 0, 1, 2 \cdot N_{ZHX}) \quad (44)$$

$$\Delta Q_{2k} = \frac{1}{2} A_{TW2k} (q_{F2k} + q_{F2k-1}) \quad (k = 0, 1, N_{ZHX}) \quad (45)$$

and finally the total primary power terms Q_1 and Q_2

$$Q_1 = \sum_{n=1}^{2 \cdot N_{ZHX}} \Delta Q_{1n} \quad (46)$$

$$Q_2 = \sum_{k=1}^{N_{ZHX}} \Delta Q_{2k} \quad (47)$$

leaving the primary and being inserted into the secondary HEX coolant channel.

In order to establish the connection to the heating (or cooling) power terms of the primary and secondary BC channels in CCM (see Hoeld, 2011, section 3.4) one has to be aware that in CCM a power term leaving a coolant channel has to be provided with a negative sign.

Because of a sometimes very steep increase of the heat transfer coefficients at the transition from single to two-phase flow and steep decrease at changing to dry-out conditions the assumption of linearity of the linear power terms within a BC node is in these seldom cases somehow problematic. Then adequate precautions are to be advised (e.g., choosing a better nodalization for these nodes).

4.3 Riser

The riser (index R) is the third element within the UTSG-3 code which will be described by the coolant channel module CCM (thus setting KEYBC=3). It will be assumed to be represented by a (non-heated) coolant channel (with a constant cross section A_R and a length z_{RT}) which can be subdivided into N_{RT} nodes. Applying CCM the wanted time derivatives and characteristic steady state and transient parameters can be determined in a similar way as for the secondary HEX region, having a zero power profile.

Thereby the 4 characteristic steady state and transient parameters at secondary HEX outlet (coolant temperature, pressure, total and steam mass flow) are yielded directly to the entrance of the riser. Hence it can be stated: $T_{RE} = T_{HA}$, $P_{RE} = P_{HA}$, $G_{RE} = G_{HA}$ and $G_{SRE} = G_{SHA}$. The entrance void fraction α_{RE} can, however, be different from the HEX outlet value α_{2H} if $A_{RE} \neq A_{2A}$. This term (α_{RE}) will then be estimated within CCM from the known value G_{SRE} ($=G_{2SH}$) by applying the inverse drift-flux correlation of the package MDS (see section 2.2.3). At special situations the riser can start to dry-out with the superheating boundary z_{RSPH} (= mixture level) to be provided by CCM. This boundary can even move into the secondary HEX region (z_{2SPH}). Usually the overall parameter z_{SPH} will be equal to $z_{HX} + z_{RSPH}$ but it can be also $z_{SPH} = z_{2SPH}$ if $z_{2SPH} < z_{HX}$. Similar considerations can be performed for the boiling boundary z_{BB} ($=z_{2B}$ but $z_{BB} = z_{RB}$ if $z_{2B} \geq z_{HX}$) if moving into the riser.

4.4 Top plenum (with steam separator)

The top plenum (with its total, steam and water volumes $V_T = V_{ST} + V_{WT}$ and the pressure in the TPL being taken as system pressure, i.e. $P_{2SYS} = P_2 = P_T$) will be assumed to consist of a steam crest together with the entire main steam system and (as to be described in section 4.6) the upper part of the DCM ($V_{DU} = V_{WDU} + V_{SDU}$), i.e. the DCM volume above feedwater entrance. The TPL water volume is thus equal to the volume of this upper section ($V_{WT} = V_{WDU}$). The main steam volume V_{SMN} is counted as the volume of the steam pipe (i.e. the volume from the isolation valve on to the entrance to the steam collector, SC). Hence, in case of a trip of the isolation valve, the corresponding steam and thus total top plenum volume have of course to be diminished by V_{SMN} .

The two-phase flow mixture leaving the riser enters the top plenum (TPL) with the mass flows $G_{WTE} = G_{WRA}$ and $G_{STE} = G_{SRA}$ and their enthalpies $h_{WTE} = h_{WRA}$ and $h_{STE} = h_{SRA}$ (Note: Allowing $h_{WTE} < h'$ or $h_{STE} > h''$ means that also single-phase flow with sub-cooled water or superheated steam can be included into the considerations). By means of a separator (being situated within the top plenum entrance) which is assumed to show an ideal separation effect the entire water is yielded (as sketched in fig.1) to the upper part of the DCM, the entire steam to the main steam system. From the steam collector on the steam is then either directed to the turbine or to the steam turbine bypass system (see section 4.6).

The separator adds to the overall pressure decrease terms an additional (friction) contribution (ΔP_{FRSP}) which can be treated in the same way as done later-on with the additional terms of chapter 3.3, assuming its steady state value $\Delta P_{FRSP,0}$ being given either as input or to be already included in the overall additional part.

Caused by the natural circulation flow and the ceasing injection of feedwater it can happen that the TPL (and thus DU) water volume V_{WDL} diminishes, i.e. the DCM water level z_{WD} falls below the position (z_{DL}) of the feedwater injection. Then dry-out of the DCM can be stated ($L_{DCDRY}=1$). Hence, the originally constant TPL volume V_T has to be extended to the now time dependent parameter V_{TEX} by taking into account that

$$\begin{aligned} V_{TEX} &= V_T + V_{SDL} = V_{ST} + V_{WT} && (\text{if } L_{DCDRY}=0) \\ \text{with } \frac{d}{dt} V_{TEX} &= \frac{d}{dt} V_T = 0, \quad \frac{d}{dt} V_{ST} = -\frac{d}{dt} V_{WT}, \quad \frac{d}{dt} V_{SDL} = \frac{d}{dt} V_{WDL} = 0 \quad \text{and } V_{SDL} = 0 \\ &= V_{ST} = V_T + V_{SDL} && (\text{if } L_{DCDRY}=1) \quad (48) \\ \text{with } \frac{d}{dt} V_T &= \frac{d}{dt} V_{TEX} = \frac{d}{dt} V_{ST} = \frac{d}{dt} V_{DL} = -\frac{d}{dt} V_{WDL} \quad \text{and } V_{WT} = 0 \end{aligned}$$

The term $V_{SDL} = V_{DL} - V_{WDL}$ depends on the amount of water and steam (G_{WTDE} , G_{STDE}) leaving the TPL in direction to the lower DCM entrance (or, if having a negative sign, entering it) and the amount G_{2E} leaving the DCM at its outlet in connection with the natural-circulation flow.

Only at non-dry-out situations water (with G_{WTAD} and the enthalpy $h_{WTAD} = h'$) can leave the top plenum (in direction to DL), dependent on the natural circulation and the addition of feed water mass flow.

The dynamic situation of the characteristic parameters within the TPL and main steam system is governed by the, for both the non- dry-out and dry-out case, valid balance of in- and outgoing water and steam mass flows. Hence, the ODE-s for the change in TPL (and thus applied as system) pressure ($P_2 = P_{2SYS}$), steam and water volumes (V_{ST} and V_{WT}) within the top plenum can then be derived from corresponding conservation equations for volume, as already presented in eq.(49), and mass and energy

$$\frac{d}{dt} (V_{ST} \rho'' + V_{WT} \rho') = \Delta G_{ST} + \Delta G_{WT} = \Delta G_T \quad (49)$$

$$\frac{d}{dt} (V_{ST} \rho'' h'' + V_{WT} \rho' h') - V_{TEX} \frac{d}{dt} P_2 = h'' \Delta G_{ST} + h' \Delta G_{WT} - (h' - h_{FW}) G_{FWSAT} + Q_{TPL} \quad (50)$$

introducing an auxiliary power term Q_{TPL}

$$Q_{TPL} = (h_{STE} - h') G_{STE} - (h' - h_{WTE}) G_{WTE} \quad (51)$$

This term can take care also of possible special cases where instead of a mixture flow pure superheated steam (with $h_{STE} > h'$) or sub-cooled water (with $h_{WTE} < h'$) enter the TPL.

From the combination of the eqs. (64) and (65) it follows then the general valid differential equation

$$B_{VP} \frac{d}{dt} P_2 - (\rho' - \rho'') \frac{d}{dt} V_{ST} = \Delta G_T \quad (L_{DCDRY} = 0 \text{ or } 1) \quad (52)$$

with the coefficient

$$B_{VP} = V_{WT} \rho^P + V_{ST} \rho^{/P} \quad (53)$$

Eliminating the $\frac{d}{dt} V_{ST}$ in the eqs.(49) and (52) by inserting from eq.(48) yields finally the wanted relation for the time-derivative of the system pressure P_2

$$B_{GP} \frac{d}{dt} P_2 = [\Delta G_{ST} + \frac{\rho''}{\rho'} \Delta G_{WT} + (1 - \frac{\rho''}{\rho'}) \Delta G_{VP}] / [1 + (C_{VP} - 1) \frac{\rho''}{\rho'}] \quad (L_{DCDRY} = 0 \text{ or } 1) \quad (54)$$

with the coefficients

$$B_{GP} = \frac{1}{h_{SW}} [V_{ST} \rho^{/P} h_{SW} + \rho'' h^{/P}] + V_{WT} \rho' h^P - V_{TEX} \quad (55)$$

$$C_{VP} = \frac{B_{VP}}{B_{GP}} \quad (56)$$

The corresponding time-derivatives of the TPL steam volume V_{ST} resp., if the DCM is drying out, of the extended steam volume $V_{TEX} = V_{ST}$ follows then from eq.(49), for the water volume from eq.(48).

4.5 Main steam system (with steam collector, bypass resp. steam turbine system)

As sketched in fig.1 the main steam system will be assumed to be composed (for each secondary loop, i.e. each steam generator) of a steam pipe (beginning at the position after the isolation valve) with an isolation, a sequence of relief and safety valves. The steam of all these loops is then collected into a (single) steam collector which is split afterwards into a bypass and steam turbine line with corresponding bypass resp. turbine trip and turbine control valves governing thus the steam mass flow into the steam turbine with consequences for the steam generator (See also Hoeld, 1990a).

The steam mass flow entering the main steam system (with the steam mass flow G_{SMN} and enthalpy h_{SMN}) is governed by changes in the steam mass flow due to the closing of the isolation or turbine trip valves (characterized by the multiplication factors η_{ISV} and η_{TTV}) or restricted through a number of N_{RSV} relief or safety valves, the bypass valve G_{RSV} , the opening and closing of the turbine control valve G_{TCV} (regulating the steam mass flow G_{STB} into the steam turbine) but also by feedbacks coming from the steam turbine resp. steam generator. Additionally, in the case of a multi-loop representation (see, for example Bencik et al., 1993) the steam collector can, in turn, be also influenced by eventually non-symmetric perturbations due to irregular steam mass flow extractions coming from or going into the other loops. The sum of all these contributing mass flow terms (for example by simulating corresponding balance-of-plant actions on these valves) is then determining

$$G_{SMn} = \sum_{n=1}^{N_{RSV}} G_{RS}(n) + \eta_{ISV} G_{BYP} + G_{STB} \quad (57)$$

If the system pressure exceeds a given threshold pressure setting, $P_{OP}(n)$, the relief and safety valves n begin with a certain time delay $t_{DOP}(n)$ and an opening time $t_{OP}(n)$ to release steam. The corresponding mass flow at a fully open valve $G_{RS}(n)$ can be approximated by the function

$$G_{RS}(n) = S_{RSV}(n) P \quad (n = 1, N_{RSV}) \quad (58)$$

with the valve discharge rate $S_{RSV}(n)$ of each valve (with respect to the system pressure P) to be known. A decreasing system pressure initiates then the closing of the valves n when reaching the pressure value $P_{CL}(n)$ with the delay and closing times $t_{CDL}(n)$ and $t_{CL}(n)$.

The isolation and turbine trip valves can be closed at a certain starting time $t_V = t_{ISV}$ or $= t_{TTV}$ with a closing time $t_C = t_{ISC}$ or $= t_{TTC}$ represented by the multiplication factor $\eta = \eta_{ISV}$ or $= \eta_{TTV}$, with a value $\eta = 1$ or $= 0$ if $t \leq t_V$ or $t \geq t_V + t_C$ and a value in between if within the closing status.

Steam mass flows G_{STB} and G_{TCV} through the turbine bypass and turbine control valves are treated as outside perturbation values, e.g., given in form of corresponding polygons or as a result of corresponding BOP actions.

The steam mass flow G_{STB} through the steam turbine can be (roughly) estimated from the 'cone law' for condensation turbines

$$\frac{G_{STB}}{\sqrt{\rho_2 P_2}} = \text{const.} = \frac{G_{STB,0}}{\sqrt{\rho_{20} P_{2,0}}} \quad (59)$$

Thus, defining G_{TCV} as steam mass flow through the turbine control valve at constant (i.e., steady-state) system pressure conditions, it follows the general relation

$$G_{STB} = \eta_{ISV} \eta_{TTV} G_{TCV} \frac{\sqrt{\rho_2 P_2}}{\sqrt{\rho_{20} P_{2,0}}} \quad (60)$$

Then the steam-induced power can (very roughly) be estimated as

$$Q_{STB} = (h_{TSPH} - h_{STB,0}) G_{STB} \quad (61)$$

If neglecting thereby the change in the enthalpy of the expanded steam within the turbine system it can be set $h_{TSPH} \approx h_{TSPH,0}$ and $h_{TSPH,0}$ be determined from the fact that at steady state conditions it can be set $Q_{STB,0} = Q_0$.

From the fact that the turbine power Q_{STB} is responsible for the acceleration of the rotating masses of the steam generator and thus for the electrical power Q_{GEN} this term can finally be estimated by a first-order differential equation

$$\frac{d}{dt} Q_{GEN} = \frac{1}{\Theta_{GEN,0}} (\eta_{QSTB0} Q_{STB} - Q_{GEN}) \quad (62)$$

The characteristic time constant $\Theta_{GEN,0}$ can be determined either from theoretical considerations (using a more comprehensive theoretical model) or from adequate measurements. The efficiency η_{QSTB0} of the turbine system (which is assumed to remain unchanged also during the transient) is then defined as the ratio of $Q_{GEN,0}/Q_{GEN}$.

4.6 Downcomer and feedwater system

As already pointed-out the downcomer (DCM) is assumed to be subdivided into an upper and lower section (indices DL and DU), separated either by the entrance of the feedwater line (at the position z_{DL}) or, in the case of a drying-out DCM, the upper head of the water

column (z_{WDE}). The upper DCM section (with the volume V_{DU} , the length z_{DU} and water level $z_{WDU}=z_{WD}-z_{DL}$) will be treated, within this theoretical approach, as a part of the top plenum (TPL). In the case of drying out of the lower DCM section, the original TPL volume (V_T) will now be extended by the (time-dependent) steam volume V_{SDL} of the DL, hence $V_{TEX} = V_T + V_{SDL}$ (see also eq.(48)). The lower section (DL) itself consists of an annulus with an outer and inner radius r_{DA} and r_{DI} , a flow area A_{DL} , the volume V_{DL} or, at dry-out conditions, V_{WDL} and the length $z_{WDE} (= z_{DL} \text{ or } = V_{WDL}/A_{DL})$.

Knowing (after the integration procedure) the transient behaviour of the TPL water volume $V_{WT} (= V_{WDU})$ resp., in the case of DCM dry-out, extended TPL steam volume $V_{SDL} = V_{TEX} = V_{ST}$ (as discussed in section 4.4) the movement of the water level ($z_{WD} = z_{WL}$) along the upper and lower DCM section can be determined to be

$$\begin{aligned} z_{WD} = z_{WL} = z_{WDU} + z_{DL} \quad \text{with} \quad z_{WDU} = f(V_{WDU}, \dots), \text{ e.g., } z_{WDU} &= \frac{V_{WDU}}{A_{DU}} \quad (\text{if } L_{DCDRY} = 0) \\ &= z_{WL} = z_{DL} - \frac{V_{SDL}}{A_{DL}} \quad (\text{if } L_{DCDRY} = 1) \end{aligned} \quad (63)$$

Saturated water which leaves the TPL (with the mass flow G_{WTAD} and enthalpy $h_{WTAD}=h_{TA}/$) is mixed at FW entrance with feedwater (with the mass flow G_{FW} and enthalpy h_{FW}) resulting in a mixture enthalpy h_{WDE} . Since the term G_{WDE} is determined, as explained below, from considerations with respect to DCM outlet mass flow, the density changes due to the movement of the enthalpy front and eventual flashing the term is given from the relation $G_{WDE} = G_{WTAD} + G_{FW}$ with $G_{WTAD} = 0$ at DCM dry-out.

From the fact that the amount of sub-cooling power at DCM entrance (due to the injection of sub-cooled feedwater into the DL) must be equal to the corresponding term at DCM outlet (=HEX entrance)

$$Q_{SCFW,0} = G_{FW,0} (h'_{T,0} - h_{FW,0}) = Q_{SC2E,0} = G_{2E,0} (h'_{2E,0} - h_{2E,0}) \quad (64)$$

the HEX entrance enthalpy $h_{2E,0}$ and thus, if applying the package MPP (section 2.2.1) for thermodynamic properties of water/steam, also the steady state HEX entrance temperature $T_{2E,0}$ can be determined.

The change in TPL (=DU) water volume $V_{WDU} (=V_{WT})$ and thus movement of the water level ($z_{WD}=z_{WDU}+z_{DL}$) within the upper DCM part (above feedwater entrance) is governed by the difference (and eventually deficit) between the entering (saturated) water coming from TPL (G_{WTAD} and h_{STAD}), sub-cooled water coming from the feedwater system and water leaving the downcomer (due to natural circulation). If the water volume of the upper part reaches zero ($V_{WDU} = V_{WT} = 0$), partial dry-out of the lower DCM section can be stated ($L_{DCDRY}=1$). The water level ($z_{WD}=z_{WDL}$) moves then within the DL part (marked by the upper end of the water column, index DE).

The difficult task of the moving enthalpy (and thus temperature) front along the downcomer will now, different to other methods, be simulated by an analytical approach (and not be described, as usual, by a set of ODE-s, i.e. the downcomer will not be treated as a coolant channel, simulated by CCM).

Beginning with the mixture enthalpy h_{WDE} at DL-E (=FEW) the enthalpy front of each water element will move, driven by the natural circulation mass flow, downwards along the DL

section and reach, after a certain time delay, the DL (=DCM) outlet (G_{WDA} , h_{WDA} , T_{WDA}). This coolant is then assumed to be yielded immediately to the entrance of the HEX region ($G_{2E} = G_{WDA}$, $h_{2E} = h_{WDA}$, $T_{2E} = T_{WDA}$). Different to other approaches this situation will be simulated by an analytical approach (and not be described, as usual, by a set of ODE-s). Thereby the lower DCM section is subdivided into a number of (maximal 50) nodes. The enthalpies at each (DL) node can then be determined by estimating for each of these nodes the length of the way the enthalpy front has attained during each time step Δt . Taking these new positions as basic points of a polygon the corresponding enthalpy changes at the original node boundaries (and thus also at DCM outlet) can now be estimated by interpolation. No smearing effects due to the movement of the enthalpy front along the DCM have to be expected (See, for example, fig.2C). The corresponding (water) mass flow parameters along the DL and thus also at its entrance (G_{WDE}) or, in the case of DCM dry-out, at the upper end of the water column can then be determined in an analytical way too, but now by starting from the (given) DCM outlet mass flow ($G_{WDA} = G_{2E}$).

If due to a decreasing secondary system pressure (P_{2SYS}) the nodal saturation water enthalpy falls below some of the nodal DL water enthalpy values, flashing is initiated, i.e. producing steam in these nodes. This steam mass will be assumed to be transported directly to the next higher node, heating-up there eventually the still sub-cooled nodal water masses being transported to the next higher node, etc. By this very special procedure finally the amount of water flow G_{SFLS} can be estimated which disappears due to flashing from the water column (and enters directly the TPL). This term has also to be taken into account for the calculation of the mass flow (G_{WDE}) at the top of the water column

On the other side, if in the case of a partial dry-out of the DCM (with $z_{WDL} < z_{DL}$ and $V_{SDL} = A_{DL} z_{WDL} > 0$) cold feedwater (with the mass flow G_{FW} and enthalpy h_{FW}) is injected into the DL steam volume (V_{SDL}) the possibility that a part of the feedwater (G_{FWSAT}) is reacting directly with the steam in this compartment has to be taken into account. This part is then heated-up to saturation conditions, thereby condensing a corresponding part of the steam to saturated water (with the resulting condensed water mass flow G_{WDCND} being equal to the steam removed from the extended TPL). The condensed part is assumed to be mixed with the remaining feedwater part $G_{FWSUB} (= G_{FW} - G_{FWSAT})$, falling then directly and unperturbed to the top of the water column. This procedure is governed by the input parameter z_{DLMIX} which states at which falling length z_{DLMIX} the entire feedwater has condensed the steam to saturated water. This can be expressed by a mixing factor $\varepsilon_{MIX} = z_{DLMIX} / z_{DL}$ with $G_{FWSAT} = (1 - z_{WDL} / z_{DL}) G_{FW} = \varepsilon_{MIX} G_{FW}$.

The total change in water (and thus also steam) volume ($\frac{d}{dt} V_{WDL} = -\frac{d}{dt} V_{SDL}$) of a drying-out DL region ($L_{DCDRY}=1$) has to be estimated from the balance of in- and outgoing water mass flows at the top of the water column. They are, as shown in eq.(48), equal to the corresponding time-derivative of the extended TPL volume.

The corresponding steam and water masses within DL (needed for the calculation of the natural circulation situation) are then: $M_{WDL} = \rho_{WDL} V_{WDL}$ and $M_{SDL} = \rho'' V_{SDL}$.

4.7 Pressure decrease and natural circulation along secondary SG loop

The closed circuit of the secondary SG loop consists of the elements 'HEX secondary side', the riser, the top plenum and finally the downcomer. The sum of all pressure decreases should be equal to zero. Since the thermal-hydraulic behavior of the first two elements are simulated by corresponding CCM (distinguished by the logical KEYBC = 2 and 3). The

pressure decrease terms along this closed circuit are directly provided by the corresponding CCM-s. The corresponding terms for the top plenum and DCM section (mainly contributing by their static head) can be calculated in a similar way as done in CCM.

To determine in the transient case the natural circulation behaviour (for example by determining the time-derivative of the HEX entrance mass flow $\frac{d}{dt} G_{2E}$) the HEX channel has to be applied as a 'closed loop' (as described in section 3.4 resp. Hoeld, 2011). This means the mass flow terms have to be adapted in such a way that the 'closed channel criterion' claimed by eq.(19) for the secondary HEX region is always fulfilled. For this purpose the term $\frac{d}{dt} G_{BMT}$ can be taken directly as an independent variable within the entire set of ODE-s (estimating the entrance mass flow G_{2E} then after the integration from the resulting G_{FBMT}) or it must a relation be found which connects the term $\frac{d}{dt} G_{FBMT}$, as defined in eq.(18), with the time-derivative of HEX entrance mass flow $\frac{d}{dt} G_{2E}$. At present it will be assumed that the change in the overall mass flow is almost equal $\frac{d}{dt} G_{2E}$ (i.e., assuming $\frac{d}{dt} G_{BMT} \approx \frac{d}{dt} G_{2E}$). Together with the fact that the secondary cross section is constant ($A_{Bk} = A_2$) it follows from eq.(19)

$$\Delta P_{G2T} = \frac{Z_{HX}}{A_2} \frac{d}{dt} G_{BE} \quad (65)$$

This term can be eventually provided with a form factor which could help to fulfil (in a recursion procedure) the demand of eq.(24). Which of these procedures should be preferred will experience show.

As explained in chapter 3.2, the time-derivative for the overall natural circulation mass flow, for example at the entrance to the HEX region, follows from the facts that the sum of all pressure increase terms along the entire loop must be zero ($\Delta P_{AE} = 0$), the differences in static head acting as driving force. If considering that a change in total mass flow is very fast propagating along the entire loop it can be estimated that

$$\frac{d}{dt} G_{2E} \cong d_{G2E} \Delta P_{GAE} \quad (66)$$

with $\frac{1}{d_{G2E}} = \frac{Z_{HX}}{A_2} + \frac{Z_R}{A_R} + \frac{Z_{DU}}{A_{DU}} + \frac{Z_{WDL}}{A_{DL}}$

$$\text{and } \Delta P_{GAE} = \Delta P_{PAE} - \Delta P_{AE}, \quad \Delta P_{PAE} = \Delta P_{STH} + \Delta P_{ACC} + \Delta P_{XIN} + \Delta P_{FR} + \Delta P_Z, \quad \Delta P_{AE} = 0$$

The total pressure differences along the secondary HEX region and the riser are provided by the coolant channel code package CCM. The terms for the TPL and the DCM region have to be derived in a similar way. Knowing in the transient case the mass flow time-derivatives (which are assumed to be equal at each position of the loop) the pressure differences ΔP_{GHAE} , ΔP_{GRAE} and ΔP_{GDAE} and thus also the total pressure increase terms ΔP_{HAE} , ΔP_{RAE} and ΔP_{DAE} along the HEX, riser and DCM regions can be calculated too and, finally, in relation with the system pressure P_{2SYS} , also their absolute pressure parameters.

5. Digital code UTSG-3

Due to the rising demand on the thermal-hydraulic codes needed for comprehensive research studies in nuclear reactor safety the digital UTSG codes have been continuously expanded to the now very mature code version UTSG-3. Thereby it could be taken

advantage of the experiences gained during a variety of test-calculations (one example being presented in chapter 6) during the different stages of development through all the years. The resulting advanced code version UTSG-3 is based on the theoretical background as presented in the chapters above. It is, at present, applied in a stand-alone manner but is, however, constructed in such a way that it can be used also as a part of more complex transient codes.

As already discussed before, the code follows a layout as shown (and described) by fig.1. Thereby it has been assumed that the HEX region is axially subdivided into N_{ZHX} nodes (restricted by $N_{ZHXMx}=7$) and the riser into N_{RIS} nodes (restricted by $N_{RTMX}=5$). The heat transfer through a tube wall is simulated either by a single layer ($N_{RT}=1$) or described by overall heat transfer coefficients ($N_{RT}=0$). Obviously, a moving boiling (and eventually also superheating) boundary (z_{2B} and z_{2SPH}) is taken into account too.

5.1 Input to UTSG-3

As already pointed-out, the primary and secondary side of the HEX region and the riser are simulated by the digital modules CCM, distinguished by the key numbers $K_{EYBC}=1, 2$ and 3 . Thereby it has to be noted that they demand as inputs only BC parameters. These will then (within CCM) be automatically translated into the needed corresponding SC parameters. Hence, the set of input data demanded by the code consists of the following parameters

- Steady state operational conditions (such as power, temperatures, pressures, mass flows etc.),
- geometry data (now with constant cross sections A_1 and A_2) and logicals consisting, for example, of the number of axial HEX and riser nodes (N_{ZHX} and N_{RIS} , restricted by $N_{ZHXMx}=7$ and $N_{RISMx}=5$) and the number of possible tube wall layers $N_{RT} (\leq N_{RTMX}=1)$,
- characteristic data of the top plenum, downcomer and feedwater system,
- characteristic data of the valves along the main steam system,
- outside-perturbation signals

and

- option values governing the wanted output in form of tables and plots.

Looking at the (steady state) energy balance equation

$$Q_{1,0} = Q_{2,0} = G_{FW,0} (h_{T,0}'' - h_{FW,0}) \quad (67)$$

it is clear that from the 4 possible operational steady state input values $Q_{SINP,0}$, $G_{FWS,0}$, $h_{FW,0}$ and $P_{T,0}$ only 3 of them are required, otherwise the problem would be overestimated. Hence, a remaining fourth variable has to be replaced by a parameter which follows directly from the energy balance relation given above. This (and other normalization actions) will be done automatically by the code.

The entire natural-circulation U-tube steam generator system can, if operated in a stand-alone manner, be perturbed from outside by the following (transient) parameters:

- Primary HEX coolant inlet temperature T_{1E} (or enthalpy h_{1E}), mass flow G_{1E} and pressure P_{1E} ,
- perturbations coming from the main steam system (see section 4.5.2)

and

- perturbations coming from the feedwater system (see section 4.6)

These outside perturbation signals can be described by means of polygons characterized by their basic points given as inputs.

If the code is a part of a more comprehensive code (for example the previous code version UTSG-2 used in combination with the thermal-hydraulic GRS system code ATHLET), entrance parameters of the primary side (temperature, total mass flow and pressure) are then taken from the main code, the resulting outlet parameters being transferred again back to it. If the overall code is operated in combination with balance-of-plant (BOP) actions the corresponding perturbation parameters of UTSG-3 are then directly provided with the corresponding BOP signals.

A more detailed input description of the code combination UTSG-3/CCM can be found in (Hoeld, 2007c).

5.2 Solution procedure

The corresponding steady state parameters will be determined by solving the resulting set of non-linear state equations by means of a linear algebraic solution procedure in a recursive way. These parameters are then needed as starting parameters for the transient calculation.

In the transient case a final system of state and differential equations has to be solved consisting of a set of maximal 53 (or 38, if choosing $N_{RT}=0$) non-linear ODE-s of 1-st order for the variables

$$\begin{aligned} &T_{1M}(i_M, k), T_{2M}(i_M) \quad \text{with } i=1, N_{ZHX} \leq 7, i_M = i - \frac{1}{2}, k=1,2: \text{ up- and downwards flow} \\ &T_{TWE} \text{ and } T_{TWM}(i, k) \quad (\text{only if } N_{RT} = 1) \quad \text{with } i_{RM}=1, N_{RT} \leq 5, i_{RM} = i_M - \frac{1}{2} \\ &\alpha_{2M}(i_M), \alpha_{RM}(i_{RM}) \\ &Z_{2B}, Z_{2SPH}, P_2, V_{WT}, G_{2E} \end{aligned} \quad (68)$$

Additionally, a number of state equations is required, describing total and nodal power terms, turbine power, nodal heat power fluxes, total and nodal steam mass flow, steam mass flow into main steam system, i.e. into steam relief and safety valves and into steam turbine, total and nodal pressure drops, steam volumes along HEX, riser, top plenum and DCM, dry-out boundary and movement of the enthalpy front along the DCM, etc.

The resulting set of equations (together with the corresponding contributions from the constitutive and momentum balance equations) can be combined with other sets of ODE-s and algebraic equations coming from additional parts of a complex model, e.g., from other basic channels which represent different thermal-hydraulic objects within an entire closed loop or a system of parallel channels, from heat transfer or nuclear kinetics considerations etc. Solving now directly the resulting set of ODE-s of such a complex physical system would have the effect that, if using an explicit integration procedure, the computation has due to the very fast pressure wave propagation to be performed with very small time steps ('stiff equation system'), with the consequence of high CPU values to be expected. This situation can partially be improved by choosing an implicit-explicit integration procedure, such as the 'forward-euler, backward-euler' routine FEBE developed by (Hofer, 1981), with a computing time being, however, still disagreeable. As already pointed out, in CCM and thus also UTSG-3 this time-consuming procedure could be circumvented by an approach which takes advantage of the fact that under the most circumstances the mass and energy balance equations can be treated separately from momentum balance without losing essentially on accuracy. Hence, if 'stiff' equations can be avoided in the final overall set of ODE-s then, for this purpose, the routine

DIFSYS can be recommended, based on a procedure established by (Bulirsch-Stoer, 1961) and (Stoer, 1974). Additionally, special precautions have been foreseen for the case that a variable is restricted (within an intermediate time-step) by a certain limit (For example a boiling boundary which should not fall below zero etc.). For more details see (Hoeld, 2000).

5.3 Output

Besides the print-out of the input data block and of the most important steady state parameters, as output governed by adequate input options interesting characteristic transient parameters (as absolute, absolute differences or relative differences) can be achieved in table form. It is, additionally, possible to store data needed for restart purposes and (if setting the input logical LSTORE > 0) to store at each time point TSTORE a selected number (NPARAM) of transient parameters in form of a 'Poly Plot Format' on the file LSTORE

WRITE(LSTORE,*) NPARAM, TSTORE, (PSTORE(I),I=1,PARAM) (69)

Then their transient behaviour can be represented as plots (as this is demonstrated, for example, in fig.2).

6. Verification and validation procedure

The continuous adaption of the computer code version UTSG-2 (see for example Hoeld, 2005) and, later-on, of the code combination UTSG-3/CCM on the rising demands coming from the reactor safety research studies and the demand for the necessary quality of the codes has been accompanied by appropriate verification and validation (V&V) procedures (Bencik et al., 1991) with feedbacks to the formulation of the theoretical model. Thereby the judgment of the feasibility and exactness of the chosen theoretical model (compared with eventual other possible approaches) counted already to the first steps of the verification. A separate validation procedure of the applied packages for drift flux, single- and two-phase friction coefficients, heat transfer coefficients, thermodynamic properties of water and steam etc. represented the next one. By means of out-of-pile calculations these packages had to undergo a very thorough study (see Hoeld et al. 1992, Hoeld 1988a, 1994, 1996, 2001, 2002a) before being applied in CCM. It had to be made sure (again by out-of-pile calculations) that the applied mathematical methods are working satisfactory (integration routine, the PAX procedure together with the calculation procedure of moving boundaries). The development of the code with the intention to demand only easily applicable and available input data of the code and to provide the user with inner quality control parameters can help to avoid errors at the application.

Since CCM is constructed with the objective to be used (similarly as done in separate-phase models) only as an element within an overall code, V&V actions could be performed only in an indirect way, i.e. in combination with such an overall code. One of these verification steps was to take some of the most characteristic UTSG-2 calculations being previously done both in a stand-alone manner but also as a part of the overall modular GRS system code ATHLET as benchmark cases. Post-calculating them with the now advanced version (UTSG-3) means the verification of not only this advanced version but also of its most sensitive part, the CCM.

Within the scope of these tests, a number of PWR transients with different initiating events followed by process sequences according to the resulting 'balance-of-plants' (BOP) actions have been post-calculated. Some of the most characteristic cases were:

- 'Loss of feedwater with turbine trip and scram' (Hoeld, 1988b, 1990b, 1999 and 2002b). (See also test calculation below),
- 'loss of feedwater with turbine trip and ATWS' (anticipated transient without scram) (Hoeld, 1988b, 2000, 2007b) with or without boron injection (Frisch et al., 1989),
- 'station blackout with turbine trip and scram' (Hoeld, 1985),
- 'loss of preferred power with turbine trip and scram' (Austregesilo et al., 1991),
- 'loss of preferred power with turbine trip and ATWS' (Hoeld, 1990b).
- 'post-calculation of 3 out of 4 start-up tests' at a German NPP station such as the cases 'operational transient by changing the nominal power from 100 to 58%.', 'reactor scram (at 60 % nom. power) together with turbine trip' (Bencik et al., 1991) and (the non-symmetric case of) 'a loss of 1 out of 4 main circulation pumps in a multi-loop simulation' (Bencik et al., 1993),
- 'coupled thermal-hydraulic and neutron kinetics behavior of a 3D core' (Jewer et al., 2005),

The good agreement of the above presented test calculations with respect to similar calculations with earlier versions applied to the same transient cases demonstrates that despite of the continuous improvements of the code UTSG and the incorporation of CCM into UTSG-3 the newest and advanced version has still preserved its validity.

Besides the V&V actions for this code combination it has to be noted that an important verification step was based on the fact that UTSG calculations have always been accompanied with strong quality control measures. One of them is the comparison of the actual masses being present at any time in the HEX, RIS, TPL and DCM region of the secondary loop of an UTSG with masses to be expected due to the balance of in- and outgoing mass flows. Differences can give valuable hints to the quality of the study and thus the validity of the code. They can point to parts of the calculation where improvements in the theoretical model or in the realisation of the resulting set of equations of the code can be recommended (see fig.2H).

Additionally, accompanying calculations of pseudo-steady state parameters can be compared with the actual transient parameters and thus contribute to the quality control too (see, for example, the boiling boundary in fig.2G).

It is obvious that own V&V actions have to be foreseen for each special application case which can thus also contribute to the further maturation the module CCM too.

7. UTSG-3 /CCM test calculation

To demonstrate the properties and validity of the advanced code version UTSG-3 a number of earlier ATHLET/UTSG-2 calculations (see for example Hoeld 1988 or 1990a) have been taken as benchmarks and been successfully post-calculated by the UTSG-3 code. They can thus contribute to the verification process of this newest code version and also of the underlying coolant channel module CCM. The corresponding UTSG-3 tests had to be based on the same input data set, despite of the fact that the philosophy about the balance-of-plant (BOP) actions in a NPP may have partially changed during the years.

As an example the process sequence of an UTSG-3 stand-alone calculation of the case 'loss of main feedwater at a PWR NPP with turbine trip and scram' (at nominal conditions) will be presented. As a result of these calculations in the figs.2A-2H some selected and most characteristic parameters are plotted. The calculation is based on an ATHLET/UTSG-2 calculation (Hoeld, 1990a) which has been performed in connection with the establishment of an general standard input data set for the ATHLET/UTSG-2 code, using the general

control simulation language GCSM of ATHLET (see for example Austregesilo et al., 1991) for the description of BOP actions.

UTSG-3 (CCM) STAND-ALONE CALCULATION
Loss of main feedw(but EMFW),TUTRI/SCRAM,max.pr.c,prt.coold.

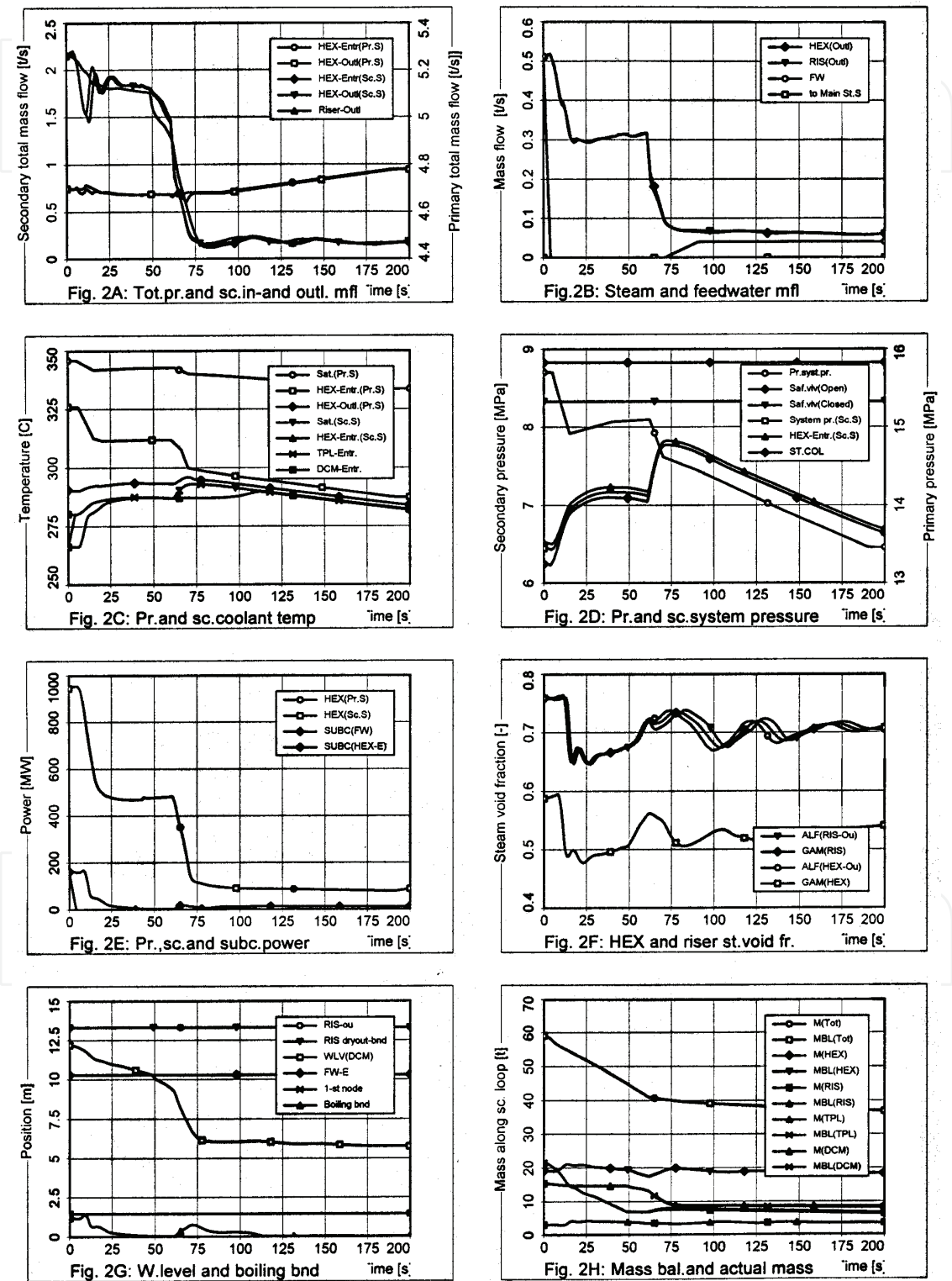


Fig. 2.

The following transient behaviour of these parameters can be observed:

- As initiating event the switch-off of all 2 (plus 1 reserve) main feedwater pumps had been assumed.
- The abrupt coast-down in main feedwater (falling within 4 s to zero, fig.2B) created the signal 'low-feedwater-flow' which in turn caused a number of BOP actions.
- In the first phase of the transient a signal for reactor power limitation (RELEB) is initiated, causing due to an adequate drop of control rods a reduction of the nuclear kinetic and thermal reactor power (and thus HEX power, fig.2E) to about 50%, resulting in a corresponding reduction of the primary HEX entrance temperature (fig.2C) and primary system pressure (fig.2C). The primary coolant mass flow remains almost unchanged (fig.2A).
- Simultaneously, a decrease in steam turbine power to about 55 % (fig.2E) is initiated by reducing, due to the 'maximum pressure control' procedure, in a controlled way the steam mass flow through the turbine-control valve (fig. 2B) yielding consequently to an (at the begin very steep) increase in secondary system pressure (fig.2D) and thus saturation temperature (fig.2C). The corresponding pressure set point curve is a function of the part-load diagram, is limited by a maximal increase rate of 2.0 MPa/min after having reached 7.7 MPa and is kept below 8.0 MPa.
- The temperature at DCM entrance is a mixture of saturated water coming from the riser/separator and the injected feedwater (with a temperature of 218 °C). Hence, the switch-off of the main feedwater pumps and increase in saturation temperature yielded to an abrupt increase in DCM entrance temperature, reaching saturation conditions. Its temperature (or enthalpy) front (fig.2C) is moving (due to natural circulation) along the DCM until it reaches (almost at the begin of the transient, i.e. after about 8 s) the HEX entrance.
- The decrease in feedwater flow causes also a decrease in sub-cooling power at feedwater and HEX entrance (fig.2E) and thus also a decrease in boiling boundary (fig.2G).
- From fig.2F the transient behaviour of the corresponding local and total HEX and riser steam void fractions can be seen.
- Due to the deficit in incoming (feedwater) and outgoing (steam) masses the downcomer starts to dry-out (see water level in fig.2G), crossing at about 45 s the position of feedwater nozzle. This means that in the model it had to be taken into account that feedwater, being injected after this time point, will act partially with the increasing steam content of the lower DCM section (i.e., will condense a part of it). The falling DCM water level causes at about 10.2 m (with a delay of about 10 s) the activation of the auxiliary FW pumps (fig.2B) and at 9.0 m a turbine trip (TUTRI) and a reactor scram, switching down the turbine and reactor power (fig.2E) (leaving only the power decay heating term), withdrawing, however, power from the primary loop due to steam removal through the bypass valve.
- Despite of turbine trip steam can still be removed through bypass valves (the valves being part of the main steam system, fig.1) . Steam mass flow through these valves is governed by the 'partial cool-down procedure'. It acts at first in combination with the 'maximum pressure control'. If the pressure exceeds 8.6 MPa the corresponding 'cool-down set point curve' is lowered from 8.6 to 8.3 MPa and decreases then in correspondence to its saturation temperature value with 100 K/h until it reaches the

mark 7.5 MPa, resulting in a pressure transient as can be seen from fig.2D. It should be noted that the difference between the secondary system pressure and the pressure at channel entrance (i.e. HEX entrance or DCM outlet) is an important basis for parallel channel assemblies. This difference stays in the first phase of the transient almost unchanged as long as no essential changes in the DCM (temperature, dry-out) appear. Hence the pressure difference between HEX outlet and steam collector entrance is still present but diminishing. This parameter plays an important role for the determination of different in- and outflows into the steam collector in case of a multi-loop application of the ATHLET/UTSG code (Bencik et al., 1991)

- The natural-circulation flow (e.g., at secondary HEX, fig.2A) is continuously decreasing, stagnant flow will, however, not be reached because of steam removal by the bypass valves.
- UTSG calculations have always been accompanied with strong quality control measures. The control of the actual masses being present at any time in the HEX, riser/separator, top plenum and downcomer regions with masses which should be expected due to the balance of in- and outgoing masses can give valuable hints to the quality of the calculations and thus the validity of the code. In fig.2H the total mass content along the secondary loop is split into its contributions from different regions and shows excellent agreement. This procedure has been a most valuable tool during the construction of the theoretical model.

The calculations showed, as expected, no noticeable differences in comparison to calculations with the code ATHLET/UTSG-2. That means, that the broad experience with the almost 20 years of UTSG application and the many verification runs with the ATHLET/UTSG-2 code combination could be transferred directly to this code and the module CCM. (See, for example, the case of a loss of one out of four main coolant pumps (see Bencik et al. [1]) within the series of post-calculations of start-up tests of a German PWR NPP).

8. Conclusions

The presented model fulfils the objective of constructing a reliable module which shows in many cases large flexibility with respect to other existing codes, can easier be handled (see the possibility of an automatic subdivision of a BC into SC-s) and has a much higher potential for further applications (e.g., if using it for parallel channel assemblies by taking advantage of the 'open and closed channel concept').

The procedure PAX and the drift flux correlation package are a central part of the theoretical model and module CCM and thus also of the advanced code version UTSG-3. The approximation procedure has to provide the model, apart from the above feature, also with gradients of the resulting approximation function needed for the determination of the time-derivatives of coolant temperature and void fraction and governs the movement of SC (= boiling or mixture) boundaries across BC node boundaries whereas an adequate drift-flux correlation package states in which way co- and counter-current flow in vertical, inclined or even horizontal coolant channels can be treated. In both cases it has, due to the availability of different input parameters, be distinguished between a steady state or a transient case and both methods had to be submitted to a thorough test phase outside of the code before being implemented into the code.

On hand of its application within the UTSG-3 concept it could be demonstrated that the presented theoretical drift-flux based thermal-hydraulic coolant channel model and the resulting module CCM can be a valuable element for the construction of complex assemblies of pipes and junctions. Simultaneously, it could be build a bridge to the verification status of the widely used UTSG-2 code. Experiences with other application cases will help to mature the present CCM module. As it turned-out the method to discretize PDE-s and connect the resulting mean and boundary nodal functions by means of the PAX procedure can be of general interest for similar projects too.

The knowledge of characteristic parameters of a U-tube steam generator allows also establishing some normalization procedures in order adjust the code to the real situation. Taking the steady state heat power as a nominal power helps to compensate the uncertainties in the determination of the heat transfer coefficients and the exact number of the U-tubes, the steady state natural circulation mass flow allows to adjust the pressure decrease over the entire secondary loop (see chapters 3.2 and 4.6) and, finally, from the given steady state sub-cooled power (being dependent on total power and pressure) an overestimation in feedwater entrance enthalpy or mass flow parameters can be avoided (chapter 4.4).

The resulting equations for different channels appearing in a complex physical system can then be combined with other sets of algebraic equations and ODE-s coming from additional parts of such a complex model (heat transfer or nuclear kinetics considerations, downcomer etc.). The final overall set of ODE-s can then be solved by applying an appropriate time-integration routine. Since ‘stiff’ equations could be avoided by treating the momentum balance separately from the energy and mass balance the integration routine DIFSYS can be recommended based on a procedure established by (Bulirsch-Stoer, 1961 and (Stoer, 1974). Otherwise an implicit-explicit integration procedure such as the ‘forward-euler, backward-euler’ routine FEBE (Hofer, 1981) can be chosen.

Several measures have been installed to control continuously the quality of the calculated results during a computational run. Parallel to the normal calculation a number of characteristic parameters can be called. These should then allow judging the quality of the run. Besides a high number of test-prints the presentation of characteristic pseudo-stationary parameters (such as boiling boundary, heat content values along the primary and secondary loop) turned out to be a very important tool for the comparison of the actual steam and water mass contents within the HEX, the riser and the DCM region with the mass contents as to be expected from the balance between the in- and outgoing mass flows. Already small deviations can grow during a transient and give a hint to the existence of some uncertainties in the model or in the realization in the code (see fig. 2H).

9. Nomenclature

A	m ²	Cross sectional area
C , C ₀	-	Dimensionless constant, Phase distribution parameter
d _{HW}	m	Hydraulic diameter
f _{ADD0} , f _{FMP0}	-	Additive and multiplicative friction coefficients

G, G_S, G_W	$\frac{\text{kg}}{\text{s}}$	Total, steam and water mass flow
G_F	$\frac{\text{kg}}{\text{sm}^2}$	Mass flux
$h, h^P, c_P = h^T$	$\frac{\text{J}}{\text{kg}}, \frac{\text{m}^3}{\text{kg}}, \frac{\text{J}}{\text{kg}^\circ\text{C}}$	Specific enthalpy and their partial derivatives with respect to pressure and temperature (= specific heat)
K_{EYBC}	-	Characteristic key number of each BC
$L_{\text{FTYPE}} = 0, 1 \text{ or } 2$	-	SC with saturated water/steam mixture, sub-cooled water or superheated steam
$N_{\text{BT}}, N_{\text{CT}}, N_{\text{RT}}, N_{\text{TUBES}}, N_{\text{ZHX}}$	-	Total number of BC or SC nodes, of radial U-tube layers, of U-tubes and of HEX nodes
$P, \Delta P = P_A - P_E$	$\text{Pa} = \frac{\text{Ws}}{\text{m}^3} = \frac{\text{kg}}{\text{m s}^2}$	Pressure and pressure difference (in flow direction)
$Q_{\text{SCE}} = Q_{\text{SCFWE}}$	$W = \frac{\text{m}^3}{\text{kg}}$	Subcooling power at HEX and FW entrance
$Q_i, Q_{\text{NOM}}, \Delta Q_{\text{kn}}$	$W = \frac{\text{m}^3}{\text{kg}}$	Total, nominal and nodal power into (!!) channel i
$q_{\text{in}} = q_{\text{FTWin}} A_{\text{TWin}} / U_{\text{TWin}} \frac{W}{\text{m}^3}$ $= q_{\text{FTWin}} V_{\text{Nn}} / A_{\text{Nn}} \frac{W}{\text{m}^2}$ q_{FTWin}	$\frac{W}{\text{m}^3}$ $\frac{W}{\text{m}^2}$	Local nodal power density into and out of a single and thus also all U-tubes (i=1,2) Local nodal heat flux in- and out of a single U-tube
$r, \Delta r = r_2 - r_1$	M	Radial U-tube variable and thickness
T, t	C, s	Temperature, time
U_{TW}	M	(Heated) perimeter of a single U-tube
$V_{\text{Bn}} = 0.5(A_{\text{Bn}} + A_{\text{Bn-1}}) \Delta z$	m^3	Nodal BC volume
v	$\frac{\text{m}}{\text{s}}$	Velocity
$X = G_S / G$	-	Steam quality
$z, \Delta z_{\text{Nn}} = z_{\text{Nn}} - z_{\text{Nn-1}}$	M	Local variable, SC node length ($z_{\text{Nn-1}} = z_{\text{CE}}$ at $n=0$)
$z_{\text{BT}}, z_{\text{CA}}, z_{\text{CE}},$	M	BC length, SC outlet and entrance positions
α		Void fraction
α_{TWn}	$\frac{W}{\text{m}^2^\circ\text{C}}$	Heat transfer coefficient at inner and outer TW surface
Δ	-	Nodal difference
ε_{DPZ}	-	Coefficient for choice of additional friction
$E_{\text{MIX}} = z_{\text{DLMIX}} / z_{\text{DL}}$	-	Mixing coefficient governing the condensation of saturation steam along DL due to the injection of subcooled feedwater
ε_{QTW}	-	Correction factor with respect to $Q_{\text{NOM},0}$
ε_{TW}	M	Absolute roughness of tube wall ($\varepsilon_{\text{TW}} / d_{\text{HW}} = \text{rel.value}$)

Φ_{DW}	-	Darcy-Weisbach two-phase multiplier
η_{ISV}, η_{ITV}	-	Multiplication factor (opening or closing of valves)
λ_{TW}	$\frac{W}{mC}$	Heat conductivity along tube wall
ρ, ρ^P, ρ^T	$\frac{kg}{m^3}, \frac{kg}{mJ}, \frac{kg}{m^3C}$	Density and their partial derivatives with respect to (system) pressure and temperature
∂		Partial derivative

Subscripts

0, 0 (=E)	Steady state or entrance to the HEX region (U-tubes)
A, E	Outlet, entrance
B, S	Basic or subchannel
A,F,Z,S,X	Acceleration, direct and additional friction, static head or external pressure difference (if in connection with ΔP)
D	Drift
HEX, R,T,MN, DU,DL,TAD	Heat exchanger (evaporator), riser/separator, top plenum, main steam system, upper and lower DCM part, out of TPL to DCM
i=1,2	Primary and secondary HEX side (containing all tubes)
Mn, BMk	Mean values over SC or BC nodes
Nn, Bk	SC or BC node boundaries
D	Drift
RS, ISV, BYP, STB	Relief/safety, isolation, bypass and turbine control valves
S, W	Steam, water
P, T	Derivative with respect to constant pressure or constant temperature
TW, TWin	Tube wall and primary or secondary nodal tube wall surface of a single U-tube

Superscripts

$' , ''$	Saturated water or steam
P, T	Partial derivatives with respect to P or T

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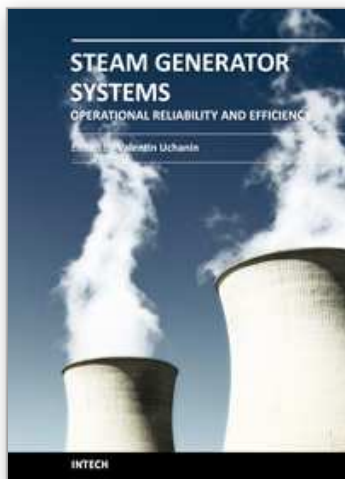
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The book is intended for practical engineers, researchers, students and other people dealing with the reviewed problems. We hope that the presented book will be beneficial to all readers and initiate further inquiry and development with aspiration for better future. The authors from different countries all over the world (Germany, France, Italy, Japan, Slovenia, Indonesia, Belgium, Romania, Lithuania, Russia, Spain, Sweden, Korea and Ukraine) prepared chapters for this book. Such a broad geography indicates a high significance of considered subjects.

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