Flow Splitting and Oscillations in Parallel Channels Simulated with System Codes

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FLOW SPLITTING AND OSCILLATIONS IN PARALLEL CHANNELS SIMULATED WITH SYSTEM CODES

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ABSTRACT
Static and dynamic instabilities, generated by feedback between mass flow rate, pressure drop and the vapor generation in boiling systems are of special interest in the nuclear industry. Complex system codes, like RELAP5 or TRACE5, are commonly used to simulate the thermal-hydraulic behavior of reactors that may have a manifold of boiling (heated) channels or cooled channels like in steam generators. In the present paper, we study flow oscillations in parallel channel configurations with system codes for diverse configurations. Different models, calculation options and, in particular, in-phase or out-of-phase oscillations were studied, both in heated and cooled parallel channels. The emphasis is on the effects of concentrated irreversible pressure losses coefficients at the inlet and at the outlet of the channels. In the case of cooled steam generator channels, the results of the Semiscale Integral Test Facility operating in natural circulation conditions are revisited. Some results coming from a more detailed nodalization set up for verification purposes lead to unexpected flow splitting asymmetries in parallel, two-pipes systems. An approximate analysis for flow splitting is presented and verified using RELAP5. These results may be considered as an addition to the usual verification data base for systems codes.

KEYWORDS
Natural Circulation, reduced inventory scenario, flow regimes, Semiscale NC2-A, RELAP5
1. INTRODUCTION

This paper is aimed at showing the path followed to elucidate the cause of some unexpected behavior in the prediction of the flow distribution in parallel boiling/condensing channels with common inlet and outlet plena. This configuration is common in the core of boiling water reactors, inverted U tube steam generators (SG) in pressurized water reactors of nuclear power plants and in other installations that typically consist of a set of parallel, vertical tubes with the physical above mentioned configuration and a common downcomer.

As it may be expected, there is plenty of consolidated literature on the subject since many decades ago but the analysis will be of restricted scope, showing the way followed to perform the verification and validation (V&V) in the particular field of natural circulation, related to a specific integral test facility (ITF) in a reduced mass inventory scenario experiment produced by a small break loss of fluid. The Semiscale ITF (S-ITF) [1] is the reference installation under analysis and the experiment is S-NC02-A. This experiment is representative of a class of small break loss of coolant accidents (SBLOCA) in a scaled model (1/1705) of a four loop pressurized water reactor.

Just a few aspects of the analysis performed to elucidate flow splitting in two parallel channels configurations will be reported in this paper using systems code such as RELAP5 and theoretical analyses. On the basis of the results conclusions are drawn, showing that the steps followed were necessary, given the cases for analysis defined in an inductive way from complex to simpler, representative scenarios.

2. ANALYSIS

2.1. Motivation: Revisiting the S-NC02-A experiment

Experiment S-NC02-A in its single loop geometry and 60 kW power has been documented three decades ago, as a part of a series of reports on natural circulation and a final one may be found in [1]. Figure 1 shows a sketch of the S-ITF as reported in said reference. This experiment consists of a controlled, stepwise reducing mass inventory natural circulation transient (referred as stepwise scenario from hereon). The proposed assessment nodalization, as reported in the manuals of the code used (RELAP5/MOD3.3gl and MOD3.3iy Patch4 [2], was considered. Figure 2 shows the natural circulation flow map (NCFM) [3] resulting from the simulations of experiment S-NC02-A using the above mentioned nodalization. This figure deserves some consideration because a special sensitivity analysis was considered. Simulations have been performed allowing random multiplying factors read once at the beginning of a calculation run. This analysis has been detailed in [4], giving consideration to the modifications to the code that allowed multiplying, by a factor chosen randomly, five heat transfer correlations and the wall friction factor. It seemed natural to the present authors to seek the effect of random variations of multipliers (ranging from 0.5 to 1.5 with a uniform distribution) affecting several closure correlations. Five heat transfer correlations were chosen based on a reference run that detailed the ones used in the thermo-siphon oscillations regime during the transient. Since friction is balanced by buoyancy in quasi steady flows, the bulk friction factor correlation was also magnified by a factor. The same was considered for the gravity acceleration that could also be modified by a factor (this may seem strange but its interest will be shown later in this paper).
Setting all multipliers equal to unity implies that results must be the same than the ones obtained from the original NRC’s executable RELAP5 code recompiled from its code source using DIGITAL© VISUAL FORTRAN 6.1 by the authors. This was appropriately verified from runs associated to the S-ITF NC02-A experiment.

The main conclusion that may be drawn from Figure 2 is that the stepwise scenario is satisfactorily represented, with predicted behavior included in the experimental uncertainty band, even considering the aforementioned variation of parameters. This was somewhat surprising to the present authors and it was decided to go in a further detailed geometrical representation of the SS-ITF for this experiment.

2.2. Refining the S-NC02-A experiment: multi-tube SGs

Figure 3 (see [5]) is a drawing of the real S-ITF steam generator for both, i.e. intact and broken, loops. This figure shows that the broken SG is constructed with two tubes of different height. The nodalization as proposed in [2] represents the intact SG loop, constructed with six tubes of three different heights, whose average is the height in the nodalization. In passing, it must be noted that this lumping from the real geometry to the one tube configuration in [2], implies that the flow and heat transfer areas considered correspond to the intact loop, i.e. six SG tubes. The obvious way to proceed would be representing the tubes with their specified geometries.
However, instead of firstly following this way and for the sake of symmetry verification, it was decided to represent the SG by two equal tubes of average height, i.e. splitting the original SG (the one in the original nodalization) tube in two identical tubes, as specified in the S-NC02-A experiment with half flow area. Heat slabs have been appropriately redefined. The effects of distributed friction should not influence too much the results when compared with concentrated pressure losses.

Additionally, it is usually accepted that a very SBLOCA (with an approximate linearly decreasing mass inventory) can be representative of a stepwise varying mass inventory. This has been established three decades ago and also shown as a verification example in [4] in relation to the experiment under analysis. Then, the simulations to be considered from hereon will also show results obtained for a 1 mm diameter break size, as in the stepwise experiment considered. The geometrical details of the break have also been checked with the ones reported in [5].

Figure 3 Two-tube steam generator geometrical details (broken loop SG at left).

Figure 4 Mass flow rates in the tubes of the SG for stepwise reduction of mass inventory. Concentrated pressure losses as in [2].

Figure 4 shows the flow rate in the tubes of the SG as a function of time for a stepwise mass inventory reduction. As may be observed, coherently with the results summarized in Figure 2, the system presents high amplitude oscillations. For the particular values of the relevant concentrated pressure losses, coincident with the one SG tube nodalization, the oscillations are in phase. The nomenclature in what follows for these losses is: Ki and Ke represent the concentrated pressure loss factors at the inlet and outlet of the SG tubes while KI and KE represent the corresponding values at the SG plena. Code (RELAP5) calculated abrupt area
changes loss coefficients are denoted by “a=1”. In a consistent way with the analysis in [6], it was decided to impose concentrated pressure losses as given by the “abrupt area change” in the SG tubes. Figure 5 shows the results obtained for a SBLOCA simulation. It is evident from Figure 5 that the flow splits non-symmetrically in the tubes. Its sum is constant and equal to the loop flow rate. It must be also considered that in one SG tube simulations, these oscillations are usually considered as thermo-siphon oscillations and this interpretation is kept here too. Out-of-phase flow oscillations after splitting may be noted in Figure 6.

As an additional simulation, all concentrated loss coefficients in the SG were removed, except those calculated by the code for computing the “abrupt area change”. Figure 7 shows the mass flow rate for both channels as function of inventory in the primary system. Once again, there is a flow splitting and the out-of-phase oscillations which remain for inventories lower than 75 %. For inventory of about 79 % the flow is still oscillating but the splitting disappears.

The observed behavior could be avoided, for instance, increasing the inlet concentrated loss coefficient. This reduces the relevance of gravity term in pressure drop, as it will be shown in the following section.

![Figure 5](image.png)

**Figure 5** Mass flow rate in the SG tubes, showing a period of “flow split” for a SBLOCA, heavy lines (cntrlvar809 and 810) correspond to time-averaged mass flow rates.

It is important to record the mass inventory at which the flow splits because this value will be considered as boundary condition for an analytical treatment of the problem. In the case shown the flow splits when the inventory is about 83%. At this condition, the SG inlet is a mixture of two phase flow with a quality close to 0.08, and the fluid comes out subcooled. The pressure oscillates between 6.2 and 7.0 MPa, approximately. The following simplified analysis will be based on these conditions.
2.3. Approximate description of splitting flow in two parallel pipe systems

Given the flow peculiarities for the considered geometry, as depicted in Figure 8, an approximate model was set up. In order to proceed, oscillations in two parallel boiling pipes were firstly considered. This subject has received plenty of attention since three decades ago, given the behavior of said systems. This interest has persisted up to recently and an interesting example may be found in [7]. Since the starting point in this work was [8], the present authors decided to extend the results of [8] to two parallel pipes. The results confirmed the ones in [7], showing in-phase and counter-phase density waves oscillations as a function of the values of the concentrated pressure losses.

These simulations, through suitable modifications of governing parameters, did not show the flow splitting behavior, so the simplified analysis to be considered and the verification using RELAP5 that follows, may give a way to the understanding of this situation.

Let us firstly consider a single boiling pipe and homogeneous, equilibrium fluid model (HEM). As it is well-known, the pressure drop along a boiling channel can be written,

\[ \Delta P = g Z_B \rho_f + g (L - Z_B) \rho_m + \frac{k_i \dot{m}^2}{2A^2 \rho_f} + \frac{f \left( \frac{Z_B}{\rho_f} - \frac{L - Z_B}{\rho_m} \right) \dot{m}^2}{2A^2} + \frac{ke \dot{m}^2 \Phi_{2p}^2}{2A^2 \rho_f} \]  

(1)

where \( Z_B \) is the boiling length, \( \dot{m} \), is the mass flow rate, \( A \) is the channel area, \( g \) the gravity acceleration, \( L \) the channel length, \( f \) the friction factor (assumed constant in this analysis), \( \rho_f \) is the liquid density at saturation and \( \rho_m \) a mean density in the two phase region calculated at half outlet quality. Parameters \( k_i \) and \( k_e \) are the concentrated friction values at the inlet and outlet of the individual channels, respectively. \( \Phi_{2p}^2 \) is the two phase friction factor defined by \( (v_f + v_g x)^{-1} \) that acts as mixture density, and it is equal to the unity when the fluid is saturated with \( x = 0 \).

Some useful (and commonly used) definitions are:
where $N_{pch}$, $N_{sub}$ and $N_{fr}$ are the phase change, subcooling and friction dimensionless numbers; $v_{fg}$ is the difference between the specific gas and liquid volumes and $h_f$ and $h_{lg}$ are the saturation and latent heat, respectively. Static instabilities exist depending on the intersection between the system curve and the driven force curve. This means that the system becomes unstable if

$$\frac{\partial \Delta P}{\partial m} < 0$$

It can be proven that depending on the system parameters (pressure, temperature, $k$ values) the acceleration and gravity terms may be neglected in the pressure drop equation. So, the maximum contributions to be considered are the distributed friction and the concentrated irreversible pressure drop terms.

Suppose now two parallel (twin channels) as shown in the Figure 8. The pressure drop through them should be equal; i.e. $\Delta P_1 = \Delta P_2$. Following the idea used in [9] let us try to find if there are solutions that allow the twin (identical) channels to have different mass flow rates and equal pressure drop (for a given inlet mass flow rate) during boiling. This is a static analysis.

In most cases, the pressure drop along of the channel is assigned and kept constant as a boundary condition. However, in this case, we will fix the inlet mass flow rate and outlet pressure. From mass conservation: $m_1 + m_2 = m_T = \text{constant}$. 

\begin{equation}
 N_{pch} = \frac{Q v_{fg}}{h_{lg} v_f m}, \quad N_{Sub} = \frac{\Delta h_{in} v_{fg}}{h_{lg} v_f}, \quad N_{fr} = \frac{fL}{2D}
\end{equation}
Defining \( N_{pm} = \frac{Q_{vg}}{h_{lg} \gamma_{m}} \) and \( N_{pi} = \frac{Q_{vg}}{h_{lg} \gamma_{m}} \). We can set that

\[
\phi = \frac{m_1}{m_T} \quad 1 - \phi = \frac{m_2}{m_T}
\]

\[
N_{pch1} = \frac{N_{PM}}{\phi}, \quad N_{pch2} = \frac{N_{PM}}{1 - \phi}
\]

The boiling channel length \( Z_{B1} \) and \( Z_{B2} \) are defined as the position at which the fluid become saturated, that is

\[
Z_{B1} = \frac{N_{sub} L}{N_{pch1}} = \frac{L N_{sub} \phi}{N_{PM}} \quad Z_{B2} = \frac{N_{sub} L}{N_{pch2}} = \frac{L N_{sub} (1 - \phi)}{N_{PM}}
\]

The different \( \Delta P \) terms for channel 1 will be evaluated in what follows. Recall that the \( \Delta P \) for channel 2 is will be same as for channel 1 but exchanging \( \phi \) by \( 1 - \phi \). The friction term in (1), using the definition of the dimensionless numbers, reads

\[
\Delta P_1 = \frac{m_T^2 \phi^2}{2A^2 D \rho_f} L (N_{sub} \phi/N_{PM} - (1 - N_{sub} \phi/N_{PM})/(1 - N_{sub} + N_{PM}/\phi)
\]

Following a similar procedure, the pressure drop for channel 1 due to form losses is

\[
\Delta P_1 = k_f \frac{m_T^2 \phi^2}{2A^2 \rho_f} + \frac{k_e m_T^2 \phi^2}{2A^2 \rho_f} (1 - N_{sub} + \frac{N_{PM}}{\phi})
\]

where \( \phi^2 \) was replaced by \( 1 + N_s + N_{PM} \phi \).

The gravity contribution, although it is being disregard in this work, results,

\[
\Delta P_1 = g Z_{B1} \rho_f + g (L - Z_{B1}) \rho_m = g N_{sub} \rho_f \phi/N_{PM} \frac{g L (1 - N_{sub} \phi/N_{PM}) \rho_f}{(1 - N_{sub} + N_{PM}/\phi)}
\]

Once again, the \( \rho_m \) may be calculated at the half exit quality. Adding (5) to (7), the total pressure drop along channel 1 become

\[
\Delta P_1 = \frac{k_f m_T^2 \phi^2}{2A^2 \rho_f} + \frac{k_e m_T^2 \phi^2}{2A^2 \rho_f} (1 - N_{sub} + \frac{N_{PM}}{\phi}) + \frac{m_T^2 \phi^2}{2A^2 D \rho_f} L (N_{sub} \phi/N_{PM} - \frac{1 - N_{sub} \phi/N_{PM}}{1 - N_{sub} + \frac{N_{PM}}{\phi}})
\]

Pressure loss \( \Delta P_2 \) has the same expression as (6), (7) and (8) changing \( \phi \) by \( 1 - \phi \).
Since $\Delta P_1 = \Delta P_2$ then

$$G(1 - \phi) = \Delta P_1 \frac{A^2 \rho_l}{N_{fr} \cdot m_T^2} = \Delta P_2 \frac{A^2 \rho_l}{N_{fr} \cdot m_T^2} = G(\phi) \quad (9)$$

$$G(\phi) = k_{im} \phi^2 + k_{en} \phi^2 \left(1 - N_S + \frac{N_{PM}}{\phi}\right) + \phi^2 \left(\frac{N_S \phi}{N_{PM}} - \frac{(1 - N_S \phi/N_{PM})}{1 - N_S + N_{PM}/\phi}\right) \quad (10)$$

For simplicity $k_{en}$ and $k_{im}$ denote $k_e$ and $k_i$ divided by $N_{fr}$. The main objective is to find the fixed points (roots) of the equation $G(\phi) - G(1 - \phi) = 0$. The equation may have at most four possible solutions. Two obvious solutions are: $\phi = 0$ and $\phi = \frac{1}{2}$ where the latter means equal flow rate in both tubes. The other two solutions may be obtained solving the quadratic polynomial, which are:

$$\phi = \frac{1}{2} \mp \frac{\sqrt{N_{sub}^4 - 4N_{sub}^2 (N_{PM} + k_{em}N_{PM} + k_{im}N_{PM} + N_{PM}^2 + k_{em}N_{PM}^2 - 2N_{PM}N_{sub} - k_{em}N_{PM}N_{sub} + N_{sub}^2)}}{2N_{sub}^2} \quad (11)$$

Other real solutions exist if the discriminant is greater than zero, i.e.:

$$N_{sub}^4 - 4N_{sub}^2 (N_{PM} + k_{em}N_{PM} + k_{im}N_{PM} + N_{PM}^2 + k_{em}N_{PM}^2 - 2N_{PM}N_{sub} - k_{em}N_{PM}N_{sub} + N_{sub}^2) > 0 \quad (12)$$

In order to check the results, the following values for the different parameters are defined [8], we set: $k_i = 23$; $k_e = 5$; $h_{fg} = 1.5039 \times 10^6$ J/kg; $h_f = 1.267 \times 10^6$ J/kg; $f = 0.03$; $L = 3.66$ m; $D_0 = 1.24$ cm. Hence $N_{fr} = 4.43$; $k_{en} = 1.13$ and $k_{im} = 5.2$. For these values, the solutions of the above inequality are shown in Figure 9. Inside the colored zone are the possible real solutions for $\phi$ and the boundary corresponds to a zero discriminant.

Figure 9  Contour plot of inequality (12).
For instance, let us select \(N_{\text{sub}}\) close to 15 (fixing liquid temperature), two values \(N_{\text{pM}}\) are obtained, namely: 6.6 and 12. These values correspond to the intersections of the red line (horizontal line) and the boundary. So, for a \(N_{\text{pM}}\) ranged between 6.6 and 12, the system will have more than one solution for each channel. Besides, fixing the inlet mass flow rate to 0.2 kg/s and the total power to 150 kW (delivered to the fluid in 2500 sec), the \(N_{\text{pM}}\) is close to 10. Figure 10 shows the results using RELAP5, with the HEM option.

As it can be seen in said figures there is no flow split (other than symmetrical). No agreement with the model exists because RELAP5 considers gravity. We recalculated using a modified RELAP5 version with variable gravity acceleration to keep simulations independent from changes in flow regimes that may arise using horizontal pipes. For this case we chose \(g=10^{-9}\), \textit{i.e.} disregarding the effects of the gravity term. Now, the simulations using RELAP5 may be observed in Figure 11.

The flow splits in a symmetric way. Please note that the bifurcation appears close to 2000 sec. The power when splitting is about 120 kW, giving a \(N_{\text{pM}} = 8\) (close to one of the boundary). Recall that the \(N_f\) number was selected arbitrarily and should be adjusted to a mean value.

Previously, it was considered two fluids using HEM model. When simulating applying the two-fluid model in RELAP5, the results are shown in Figure 12.

With the two-fluid model (now considering gravity) flow split was obtained at about 1500 sec (here the total power was delivered to fluid in 1500 sec instead of 2500 sec as in the HEM model). The most important observation is that the mass flow in both channels start oscillating about 1400 sec (\(N_{\text{pM}} = 8\)) and splits close to \(N_{\text{pM}} = 10\). These values are similar to the theoretical ones.
The analysis done might be applied to parallel channel with a common inlet and outlet identical heated and forced and constant circulating fluid. However, it cannot explain why a similar behavior is found in a condensing system such as parallel tubes in a SG. A similar analysis for two U-tubes will be considered in what follows.

Figure 1 Mass flow rate through channels (named 100 and 200) using two-fluid model in RELAP5. There is a stable flow splitting close to 1500 sec.

2.4. Pressure drop instabilities in parallel U tubes

Following the above analysis, the pressure drop along a U-tube using HEM can be written as:

\[ \Delta P = gL(\rho_h - \rho_c) + \frac{k_i m^2}{2A^2 \rho_i} + \frac{fLm^2}{2A^2 \rho_s} + \frac{k_e m^2}{2A^2 \rho_e} + \frac{k_m m^2}{2A^2 \rho_s} \]

(13)

Nomenclature is as follows: \( \rho_i \) at inlet, \( \rho_e \) at outlet and \( \rho_h, \rho_c \) are the mean densities along the upward and downward legs respectively and \( \rho_s \) is the density at the top of the U-tube. In Table 1 is a description of other variables used. The value \( k_m \) was set to zero, however it may be relevant depending on the \( k_i \) and \( k_e \) values. The enthalpy along a \( l \) coordinate of the U-tube is:

\[ h(l) = h_i - \frac{\pi D U_2 \Delta T_{ss}}{m} l = h_i - \frac{\xi}{m} l \]

(14)

\( \Delta T_{ss} \) is the difference temperature between its primary and secondary sides (kept constant) and \( U_2 \) is a global heat transfer coefficient. The mean densities along a leg may be determined as

\[ \tilde{\rho}_i = \frac{1}{L} \int_{0}^{L} \frac{dl}{\nu_l + x(l) \nu_{fg}} \]

(15)

Where \( x(l) \) is the thermodynamic quality at position \( l \). The mean density for the upward leg (assuming that both inlet and outlet are in two phase) results:
If the downward leg is full of two phase fluid, the above expression can be extended between $L$ and $2L$. However, due to the cooling conditions the downward leg will have single and two phase fluid, so the density should be a weighted mean between both regimes.

$$\bar{\rho}_h = \frac{h_g m}{L \xi} \log \left[ 1 + \frac{x_{i=0}vfg}{\nu_f} \right] \left[ 1 + \frac{x_{i=L}vfg}{\nu_f} \right]$$  \hspace{1cm} (16)

Using the value in Table 1, it could be checked that when the mass flow rate is decreased friction losses are reduced approaching to zero, whereas the gravity contribution becomes more negative, reaching its minimum as it is shown in Figure 13.

![Figure 13](image)

Figure 13 Total pressure drop, and gravity and friction contributions as a function of mass flow rate in one U-tube using the values in Table 1.

The local minimum takes place when the two phase – single phase interphase is between the middle plane and the top of the U tube of the downward leg. Finally, since both channels have the same pressure drop it results
Expression (18) can be simplified under the following approximations: $\rho_e = \rho_f, \rho_s = \frac{\rho_f + \rho_i}{2}, f = 0.015$. Given that the pressure drop depends on $m^2$ and $m$, resulting in only one possible solution having both tubes equal flow rates. Thus, the flow splitting could not be seen under these simplifications.

Despite of the above, we may calculate the flow rate at which the minimum value is reached. The calculation was performed assuming the dependence of $\rho_s$ with the mass flow rate for deriving and replacing by (15), accordingly. The mass flow rate for the minimum (where pressure drop derivative changes sign) and where it is expected a static instability (or Ledinegg type) is:

$$m_{\text{min}} = -\frac{g_h\left\{\ln\left(\frac{\rho_s}{\rho_f}\right) - \rho_c \ln\left\{\frac{\rho_s}{\rho_i}\right\} - \frac{\rho_c}{\rho_i}\right\} - \frac{\rho_c}{\rho_i}}{2 \left(\frac{k_i}{k_p^2} + 2 \frac{\xi}{\rho_i \sqrt{\rho_i}} + \frac{k_i}{\rho_i \sqrt{\rho_i}}\right)^2} + \sqrt{\frac{g_h \ln\left(\frac{\rho_s}{\rho_f}\right) + \rho_c \ln\left(\frac{\rho_s}{\rho_i}\right) + \frac{\rho_c}{\rho_i}}{\rho_i}} + \frac{2 \rho_c}{\rho_i} + \frac{k_i}{\rho_i \sqrt{\rho_i}}) (19)$$

The value obtained using (19) is close to the minimum shown in Figure 13, about 0.3 kg/s.

One of most important terms is the gravity term. Then, the difference between mean densities is:

$$\rho_n - \rho_c = h_{fg} m \frac{\ln\left(\frac{\rho_s^2(m)}{\rho_f \rho_e}\right)}{L \xi_{fg}} - \frac{\rho_f}{L} \left(2L - x_i \frac{m h_{fg}}{\xi}\right) (20)$$

Here, it can be noted the nonlinear dependence of the gravity term with the mass flow rate. Since density $\rho_s$ varies with the first power of mass flow rate, the distributed friction depends on the third power of flow. Due to the combination of these nonlinearities, other possible states may exist. Unfortunately an analytical expression could not be obtained and if some approximations are applied, other solutions may not appear.

As an example, following the values used in Table 1, Figure 14 shows the pressure drop difference between channels as function of mass flow rate. There exists more than one equally pressure drops and different mass flow rates.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Value</th>
<th>Variable / units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$A$</td>
<td>$9.0 \times 10^{-4}$</td>
<td>Channel flow area (m²)</td>
</tr>
<tr>
<td>$k_i$</td>
<td>$10$</td>
<td>Inlet concentrated pressure losses (-)</td>
</tr>
</tbody>
</table>
In previous works [10], other authors studied the pressure drop in U-tubes, focusing the analysis in conditions for flow reversal, a well-known phenomenon in vertical U-tubes of steam generators.

In this work, we point out on other possible solutions, having equal pressure drops and different mass flow rates. The contribution of the gravity term was crucial since the nonlinear dependence of mass flow rate was necessary for getting flow splitting. As it was shown, the split may be produced as a consequence of a static instability similarly as it happens with the flow reversal. In this case, when splitting occurs instead of flow direction reversal, the system may have found another stable operating state and the mass flow rates are in the same direction in both channels.
The results of the preceding analysis have been verified again using RELAP5 in the restricted geometry shown in Figure 15 and considering the above listed parameters and the additional ones depicted in said figure. The problem consists of two twin U-tubes channels with fixed mass flow rate and constant pressure at the inlet and the outlet, respectively. A heat structure was coupled to each U-tube for transferring heat from primary at constant heat transfer coefficient (“htc”) to secondary side, represented by a constant temperature boundary condition. It should be noted that the inlet quality and heat transfer coefficient when using two-phase model have been accommodated for searching the desired effect and may be different to those values used in HEM model.

The results obtained are shown in Figures 16 and 17. As may be observed, flow splitting has been correctly predicted. It should be noted that, on contrary to the HEM model, the two phase model exhibits and out-of-phase oscillations, which may be generated by different fluid velocities.

Figure 15 - Sketch of the RELAP5 nodalization for simulating two U-tubes with fixed mass flow rate.

Figure 16  Mass flow rate distribution in U-tubes using HEM model with RELAP5.

Figure 17  Mass flow rate distribution in U-tubes using two-phase flow model with RELAP5.
3. CONCLUSIONS

In this work, we have revisited the S-NC02-A experiment. Doing that, and since the number of steam generators U-tubes is small compared to a commercial SG, it was decided to split into two twin U-tubes. Uneven flow splitting, without flow reversal, at mass inventory of 83 % was found. The observed behavior was not expected, so two analytical models have been developed: one for two twin boiling pipes and another for two twin U-tubes. The analytical models showed that in both configurations it may be possible to get stable states with different mass flow rates without flow reversals and equal pressure drop. Particularly, for U-tubes it was proven that non-symmetrical splitting may be caused by appearance of a static instability which is responsible for flow reversal if no other states can exist. This analysis was also verified using RELAP5/MOD3.3. Coming back to S-ITF, the flow splitting may be a consequence of a static instability that produced the fluid splitting to other stable values. The out-of-phase oscillation has not been seen when the HEM model was used and conversely observed for two fluid velocities. The analysis herein shown and the correspondence of results obtained using RELAP5/MOD3.3 may constitute a new verification test for systems codes.

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