ANALYSIS OF NATURAL CIRCULATION IN A THERMAL HYDRAULIC LOOP SIMILAR TO A PASSIVE NUCLEAR REACTOR COOLING SYSTEM

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ABSTRACT

The main topic in this paper is a new device being considered to improve nuclear reactor safety employing the natural circulation. Aware of the need for more precise and stable numerical models we developed new computation software employing a finite element formulation. In it a stable pressure equation is derived from the mass balance, where all pressure contributions arising from the momentum and energy balances, and the state-equation are accounted implicitly in time. The thermal equilibrium homogeneous model with the vapor velocity explicitly calculated using the Drift-Flux model is employed for a good representation of the physics of the two-phase flow. A thermal-hydraulics loop in reduced scale similar to a Pressurized Water Reactor (PWR) passive heat removal system was built at IEN to obtain typical natural circulation two-phase flow data and to validate the numerical model. Two experimental tests in single and two-phase flows were made to begin the validation of the model. Comments on the results and a discussion of the importance of software validation are presented.

Keywords: natural-circulation, two-phase-flow, passive-safety, finite-elements, and similarity.

I. INTRODUCTION

Some of the Westinghouse’s AP600 PWR [1] safety systems will rely on natural circulation to perform the same functions as the active, powered, safety systems found in current operating PWR. A reduced-scale Advanced Plant Experiment (APEX) was constructed at Oregon State University (OSU) to demonstrate that the thermal hydraulic computer codes used to evaluate the performance of the AP600 emergency core cooling system give realistic descriptions [2] [3]. An important part of the evaluation is a long natural circulation phase in the primary circuit that is expected to happen under constant pressure after the blowdown phase of a Loss of Coolant Accident (LOCA).

A nodal refinement is required by the usual computer codes [4] [5] to represent the flow oscillations that occur in a single-phase flow natural circulation that operates in an unstable condition. Compressible two-phase flow natural circulation are more subjected to oscillations even when there are favorable geometry and physic conditions for stability in single-phase flow. Very often when no experimental evidence is available, the uncertainty is high and there is no assurance if a nodal refinement is necessary and sufficient, especially when no instability is seen in a two-phase flow calculation.

Because of several modeling advantages, the use of computational fluid dynamics (CFD) is growing in several industrial and environmental fields. Thus the starting point to develop an efficient thermal-hydraulic model of a typical natural circulation installation was provided by a promising methodology developed at IEN [6].

In this paper we focus on the mathematical and numerical models of a computer program we developed for our better understanding of single-phase and two-phase flow driven by natural circulation.

II. THE CONTINUUM FLUID FLOW MODEL

Due to the fast pressure wave propagation that characterizes a nearly incompressible fluid flow, most numerical methods used to calculate compressible flow present a difficulty when applied to the low speed limit. The accuracy of the wave representation in the implicit schemes and the numerical stability of the explicit numerical methods, which are usual in high speed compressible fluid flows, require extremely small and thus impractical time-steps. Therefore, nearly incompressible flows are usually approximated as fully incompressible (the density is no function of pressure). For mass conservation, the divergence of the fluid velocity field must be zero. In a compressible fluid flow simulation, for mass conservation the divergence of the fluid velocity field must be non-zero when dependence of density with pressure is retained. The
different accurate and stable numerical methods developed for calculation of each case do not keep these desirable qualities when single flow and two-phase flow appear successively in a location. Thus, the simultaneous representation of the incompressible and the compressible flow with the same accuracy requires another unified methodology. The methodology we developed begins defining the compressibility or the pressure coefficient of fluid density, \( \alpha \), and the energy coefficient of fluid density, \( \beta \), as

\[
\alpha = \left( \frac{\partial \rho}{\partial p} \right), \quad (1)
\]

\[
\beta = -\left( \frac{\partial \rho}{\partial e} \right), \quad (2)
\]

For convenient application of the initial and the boundary conditions, a new pressure variable is defined as

\[
p = p_{o} + p_{o} g s \cos \theta - p_{o}
\]

With these definitions and in terms of the new pressure variable, the fluid and solids conservation equations are:

**Conservation of mass (continuity).**

\[
A \left( \alpha \frac{\partial p}{\partial t} - \beta \frac{\partial e}{\partial t} \right) + \frac{\partial m}{\partial s} = 0 \tag{4}
\]

**Conservation of momentum.**

\[
\frac{\partial m}{\partial t} + u \frac{\partial m}{\partial s} + \frac{\partial u}{\partial s} m
\]

\[
+ \frac{\partial}{\partial s} \left( A \frac{\alpha \rho_{0} P_{s} u_{s}}{\left( 1 - \alpha_{s} \right) \rho} \right) + A \frac{\partial p}{\partial s} + \left( p - p_{o} \right) g \cos \theta = P_{s} \Gamma_{s} = 0 \tag{5}
\]

**Conservation of internal energy.**

\[
\rho A \left( \frac{\partial e}{\partial t} + u \frac{\partial e}{\partial s} \right) - P_{0} q_{e}^{*} - u P_{s} \Gamma_{s}
\]

\[
+ \frac{\partial}{\partial s} \left( A \frac{\alpha \rho_{0} P_{s} u_{s} e_{s}}{\rho} \right) \tag{6}
\]

\[
+ \left( p + p_{o} g s \cos \theta \right) \frac{\partial}{\partial s} \left( u A \right) = 0
\]

Employing the conservation of mass to eliminate the velocity gradient term from the energy equation, a second form of the energy conservation equation is

**Conservation of internal energy (second form).**

\[
\begin{align*}
1 + \frac{\beta \rho p}{\rho^{2}} \rho A \left( \frac{\partial e}{\partial t} + u \frac{\partial e}{\partial s} \right) - P_{0} q_{e}^{*} \\
+ \frac{\partial}{\partial s} \left( A - \frac{\alpha \rho_{0} \rho_{s} \rho_{e} e_{s} u_{s}}{\rho} \right) - u P_{s} \Gamma_{s}
\end{align*}
\]

\[
= \left( \frac{\alpha \rho}{\rho} \right) A \left( \frac{\partial p}{\partial t} + u \frac{\partial p}{\partial s} - u \rho_{1} g \cos \theta \right) \tag{7}
\]

**Conduction of heat in the solid bodies and walls.**

\[
A_{s} \rho_{s} c_{s} \frac{\partial T_{w}}{\partial t} - A_{s} \frac{\partial}{\partial s} \left( \kappa_{s} \frac{\partial T_{w}}{\partial s} \right) + P_{s} q_{w}^{*} = 0 \tag{8}
\]

**Constitutive relation for friction.** The momentum flux on the solid bodies and walls, \( \Gamma_{w} \), is defined as

\[
\Gamma_{w} = \frac{f_{w}}{8 A} \left| u \right| \dot{m} \tag{9}
\]

**Constitutive relation for heat flux.** The heat flux on the solid bodies and walls, \( q_{w}^{*} \), is given by

\[
q_{w}^{*} = h_{w} \left( T_{w} - T \right) \tag{10}
\]

**Thermodynamic state equations.**

For the sub-cooled liquid:

\[
\rho = \rho \left( p, e \right) \tag{11}
\]

\[
e = e \left( p, T \right) \tag{12}
\]

For the saturated mixture of liquid and steam:

\[
T = T_{sat} \left( p \right) \tag{13}
\]

\[
e = x e_{s} \left( p \right) + \left( 1 - x \right) e_{l} \left( p \right) \tag{14}
\]

\[
\rho = \alpha_{s} \rho_{s} \left( p \right) + \left( 1 - \alpha_{s} \right) \rho_{l} \left( p \right) \tag{15}
\]

By \( u, p, e, \rho, T, \) and \( T_{w} \), we denote the fluid velocity, pressure, specific internal energy, density, temperature, and the wall temperature, respectively. The drift velocity is \( u_{o} \). The reference value for the fluid density is \( \rho_{o} \) and for pressure is \( p_{o} \). The transverse flow area is represented by \( A \), \( s \) is the line coordinates, and \( t \) is the time. The gravity acceleration field is \( g \) and \( \theta \) is the angle of a component axis with the vertical. The void fraction is \( \alpha_{s} \) and \( x \) is the steam quality. The thermodynamic variables for liquid and steam phases are
\[ e_i, e_r, \rho_i, \text{ and } \rho_r. \] The vaporization internal energy is \( e_v = e_i - e_r \) and \( \Delta \rho = \rho_i - \rho_r. \) The friction factor is \( f_r \) and the heat transfer coefficient between the fluid and wall is \( h_r. \) The hydraulic perimeter is \( P_h \) and the heated perimeters are \( P_g. \) The cross sectional area of a solid is \( A_w. \) The thermal properties (density, specific heat, and conductivity) of the solids are \( \rho_w, c_w, \text{ and } K_w, \) respectively. The mixture mass flow rate is \( \dot{m} = \rho \cdot u \cdot A. \)

**Drift-Flux Model.** The mixture volumetric flux, \( j, \) is defined as

\[
j = \alpha_x u_x + (1 - \alpha_x) u_i,
\]

(16)

The drift velocity is the vapor velocity relative to the mixture volumetric flux, \( j: \)

\[
u_d = u_x - j = (1 - \alpha_x)(u_x - u_i)
\]

(17)

From the product of the Eq. (15) and the Eq. (16) result

\[
\rho_j = \rho u + \alpha_x \Delta \rho (1 - \alpha_x)(u_x - u_i)
\]

(18)

\[
\rho_j = \rho u + \alpha_x \Delta \rho u_i
\]

(19)

Employing the vapor mass balance

\[
\alpha_x \rho_i u_x = \chi \rho u
\]

(20)

and the Eq. (19) it can be shown that

\[
j = \left( \frac{1 - \chi}{\rho_j} + \frac{\chi}{\rho_r} \right) \rho u
\]

(21)

In the up-flow locations of the circuit (\( \cos \theta = +1, u > 0 \)) and (\( \cos \theta = -1, u < 0 \)) the steam velocity is calculated using the formula [7]

\[
u_s = c_o j + V_g
\]

(22)

in which the vapor terminal velocity \( V_g \) is given by [7]

\[
V_g = c_v \left( \frac{g \sigma \Delta \rho}{\rho_j^2} \right)^{1/25}
\]

(23)

The constant \( c_v = 1.13 \) is used for the distributed parameter in the drift-flux model, \( c_1 = 1.41 \) is the terminal velocity coefficient, and \( \sigma \) is the surface tension.

Computing the \( j \) from the Eq. (21) and \( u_g \) from the Eq. (22) and Eq. (23) \( \alpha_x \) is calculated using the Eq. (20).
The density $\rho^{*\alpha/2}$ is calculated employing a Taylor series expansion and the mass balance equation:

$$
\rho^{*\alpha/2} = \rho^{*} - \frac{\Delta t}{2A} \frac{\partial \bar{m}^{*}}{\partial s}
$$

**Space representation.** The mass flow rate, $\bar{m}$, the pressure, $p$, and the internal energy, $e$, are interpolated employing finite elements as $\bar{m} = \bar{m}_{j}$, $p = p_{j}$, and $e = e_{j}$, respectively. The $N_{j}$ represents the linear Lagrange shape functions, and $\bar{m}_{j}$, $p_{j}$, and $e_{j}$ are the corresponding nodal values. With the divergence term of mass flow rate $\bar{m}^{*\alpha/2}$ integrated by parts and substituted from a finite-difference time-explicit scheme for the momentum Eq. (5) a Galerkin [10] weighted residual approximation is obtained for the Eq. (29). The integral of the squared residual of the incompressible part of the energy Eq. (28) is minimized using the Petrov-Galerkin [11] weighted method. The integral of the squared residual of the time-implicit momentum Eq. (5) is minimized and the pressure term is modified using the integration by parts, thus obtaining the corresponding Petrov-Galerkin weighted residual momentum equation. The integral of the squared residual of the time-implicit energy Eq. (7) is minimized and the corresponding Petrov-Galerkin weighted residual for the second form energy equation is obtained. The connectivity of the elements is established adding the equations of adjacent elements. The solution sequence of the resulting system of equations is as follows: a) the incompressible form of the first energy equation and the heat conduction equations, b) the pressure equation derived from the mass equation, c) the momentum equation, and d) the second form of the energy equation and the heat conduction equations. In the solution of each equation of this sequence a LU factorization procedure with periodic boundary condition is employed to calculate the field variables of the complete closed circuit.

**IV. TRANSIENT RESULTS**

**The IEN Natural Convection Circuit (CCN).** The CCN is shown in the Fig. 1. The similarity design, the description, the measurement calibration, and the operation of the IEN circuit that represents in reduced similar scale, a typical passive cooling system of innovative PWR are reported in separate papers in this Conference. Here we present the comparison of the results of the numerical simulations to the experimental measurements of two heating transients in this circuit.

**Single-phase test.** The initial condition for the IEN circuit water, the heater resistors, and the solid walls were 24.1°C. This was also the initial and boundary condition for the ambient air. The power was fixed at 1290 Watts. No feedwater was injected into the secondary side of the heat exchanger in this test. In these conditions the loop fluid remained liquid for the 4 hours duration of the test.
of the primary loop. Only for the lower plenum water of the heating vessel we had a substantial disagreement, believed to be due to a close contact between the thermocouple and the vessel wall. On the heat exchanger tube walls and for the secondary water we also noted large disagreements between the calculated and the experimental temperatures. These are probably due to the poor re-circulation model used to calculate the heat transfer in the secondary volume of the heat exchanger.

Two-phase test. The IEN loop was thermally insulated from the ambient air. The circuit initial temperature was 25°C. The heating power was 1425 W. On the secondary side of the heat exchanger the entrance water temperature was 25°C and the forced volume flow rate was fixed at 6 liter per hour. Part of the expansion tank volume was filled by air and its safety valve remained closed. Due to the pressurization of the circuit, only after 7 operating hours did the circuit attained a two-phase condition. We used an external cross-flow tube-bundle correlation [12] for heat transfer coefficient of the secondary side of the heat exchanger. No modifications in proper form-loss coefficients for area change and bends were made in this test, but their usual two-phase flow multipliers were employed.

The measured and the calculated temperatures in the upper plenum fluid are shown in the Fig. 3 as a function of the time. During the single flow period we observed a good agreement between the experimental and calculated temperature in most parts of the circuit. A separate pressurization model (employing one element only to represent the air/water volume of the circuit’s expansion tank) was very crude and did not represented correctly the pressurization of the circuit. For this reason the boiling started latter in the calculation.

Even though the amplitudes differ, the calculated lower frequency estimated from the Fig. 4 is well comparable to the experimental value obtained from the Fig. 5. This model can also be used to check the two-phase similarity criteria employed for the design of the IEN facility. As illustration, the calculated steady maximum value in the whole circuit for the steam quality was 0.003 in the end of the two-phase test. As a coarse comparison, the steady-state average design value in the hot leg of the IEN circuit was 0.00092.
**V. CONCLUSION**

From the comparison of the calculated and experimental results we conclude that our thermal hydraulic computer program gave a reasonable good description for the evaluation of natural circulation in a reduced scale circuit similar to a passive emergency core cooling system of evolutionary PWR. The good two-phase flow dissipation capacity of the IEN circuit is demonstrated by the steady operation that it reached in this test. As the IEN circuit has good similar conditions to the prototype we expect a steady two-phase flow natural circulation for the passive full-scale heat removal system.

In the recent Brazilian general electric blackout, the Angra nuclear power plants depended on the emergency diesel generators to avoid a serious accident of complete station blackout. For more safety, more redundancy or passive safety systems may be necessary in these and in other PWR. Since the performance of full-scale passive safety systems are evaluated by safety licensing computer programs which are required to be verified in experimental facilities [2] we call for more Brazilian investment in this field.

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**REFERENCES**


