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NATURAL CIRCULATION IN PRESSURIZED WATER REACTORS

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SUMMARY

This paper treats the problem of Natural Circulation in a complex geometry similar to that of Nuclear Power Plants. A first experiment has been done at the integral test facility of COPESP for several heat flux conditions. The results obtained were compared with numerical simulations for the steady-state regime.

INTRODUCTION

The understanding of the natural circulation phenomena is one of the major steps in developing safer and simpler to operate nuclear power plants. Analytical modelling has been done by Zvirin (1981) and Botelho (1992) for steady-state and slow transients conditions. More sophisticated studies have been done by Lavine (1986) and Mertol (1982) considering two and three dimensional effects in a very simple geometry. Recently Westinghouse developed a code to analyze natural circulation under severe accident conditions for PWR (1991).

In order to get some insight in this area, an experimental and analytical program started at COPESP in 1991. This project includes fundamental studies for one and two phase flows and the application of the current techniques of simulation to engineering problems.

This activity is divided in three main phases occurring simultaneously:

1. fundamental experiments;
2. integral test facility experiments and
3. analytical and numerical studies, leading to the development of a computer code for design.

Phase one is a cooperation task between PUC/RJ and COPESP focusing the physics of the problem for different flow regimes (single and two phase flows) and the transient behavior of the system.

In phase two, experiments will be held at the integral test facility of COPESP, named CTE-150, to validate the code in complex geometries like those of nuclear power plants. This phase is especially important to establish appropriate correlations for friction factors and heat transfer coefficients.

Phase three concerns the development of a numerical tool to simulate one and two phase natural circulation problems in transient conditions.

As a first approach to the problem, an experiment was designed to verify if measurable conditions are obtained for the electric power available.

Simultaneously to the experiment, numerical simulations have been done for different values of heat flux. Although, at this stage we are just interested on temperatures and flow rates at the steady-state regime, transient behavior of the system is presented to allow the estimation of stabilization times.

GOVERNING EQUATIONS AND NUMERICS

Natural circulation occurs in loops where a heat source is connected to a heat sink in a higher level. The flow is generated by the buoyancy force due to density variations, as shown in Figure 1.

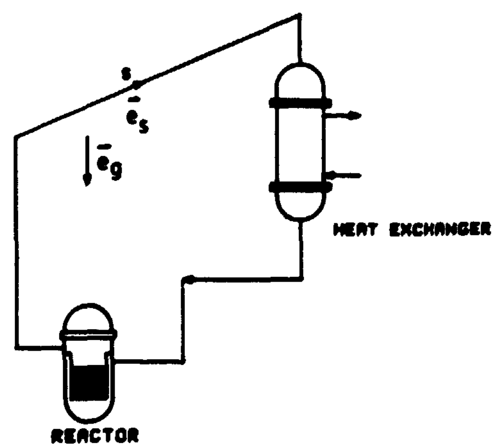

Figure 1. Schematic termosyphon.

The analysis presented here is based on a one-dimensional formulation of the conservation laws. Viscous dissipation, axial conduction and heat losses to the surroundings are neglected. The Boussinesq approximation is adopted for the driving force term. The momentum and energy balances for the fluid are presented below.

Momentum equation:

$$\rho\left(\frac{\partial v}{\partial t}+v\frac{\partial v}{\partial s}\right) = -\frac{\partial p}{\partial s}-\frac{\partial \tau}{\partial s}+\rho g e_s \cdot e_g \qquad (1)$$

where:

ρ density
v velocity at the fluid flow direction
t time
p pressure

τ shear stress

g gravitational acceleration

Energy equation:

$$\rho C_p \frac{\partial T}{\partial t} + \rho C_p v \frac{\partial T}{\partial s} = q \quad (2)$$

where:

C_p specific heat

T temperature

q''' volumetric heat flux

Considering only radial heat conduction at the tubes, it was obtained:

$$\rho C_p \frac{\partial T}{\partial t} = k \left[\frac{\partial^2 T}{\partial r^2} + \frac{1}{r}\frac{\partial T}{\partial r}\right] + q \quad (3)$$

where:

k thermal conductivity

r radial direction

Empirical Correlations. Correlations to estimate friction losses and heat transfer coefficients are necessary due to the one-dimensional approach of the problem. Unfortunately, for natural circulation, the correlations are functions of loop's geometry and this is the biggest source of code's errors. At this time, as there are no experimental results, it was decided to adopt classical correlations for the Nusselt number, Nu, as a function of Grashof, Gr, and Prandtl, Pr, numbers:

$$Gr<10^9 \qquad Nu = 0.59 \ (Gr\,Pr)^{0.59}$$

$$\text{if} \quad 10^9<Gr<10^{13} \qquad Nu = 0.021 \ (Gr\,Pr)^{0.4}$$

$$Gr>10^{13} \qquad Nu = 0.10 \ (Gr\,Pr)^{0.33}$$

The form losses are evaluated with forced circulation correlations from Idelcik (1960). The friction factor at all regimes was predicted by Churchill (1977) expression:

$$f = [(8/Re)^{12} + 1/(A+B)^{1.5}]^{1/12}$$

where:

$$A = [2.457 \ln [(7/Re)^{0.9} + 0.27\varepsilon/D]]^{16}$$

$$B = (35{,}725/Re)^{16}$$

Numerical Procedure. The set of coupled differential equations is solved by the nodal method. The number of volumes for the fluid and the tubes can be arbitrarily set and is used to solve the energy equations (for the fluid and tubes). The momentum equation is integrated around the loop (just one volume) due to the assumption of incompressible fluid. A semi-implicit algorithm is chosen for its flexibility for the time steps. The continuity of heat flux at the solid/fluid interfaces must be satisfied at each time step. A detailed description of the code can be found in Bastos at alii (1992).

EXPERIMENTAL FACILITY

The CTE-150 is the first Brazilian integral test facility of high pressure and high temperature, that simulates the actual performance of a Pressurized Water Reactor (PWR). Its main systems are the high pressure system, the secondary system, the coolant system and the coolant purification system. Silva et alii (1991) give further information about the facility.

Figure 2 presents an isometric view of part of the primary system of the CTE-150 showing the main equipments and components involved in the experiment: the horizontal shell-and-tube heat exchanger (T1), the electrical heater (A4) and the piping system with valves and tubes.

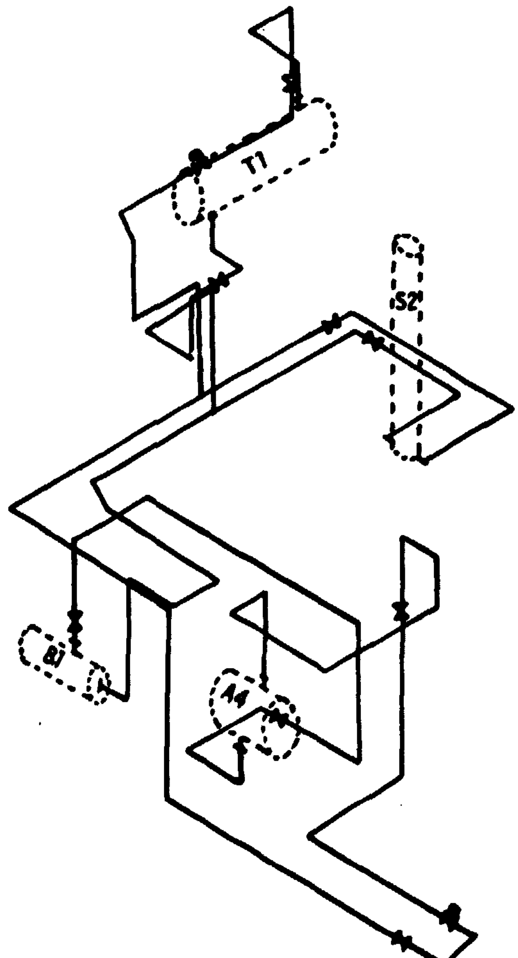

Figure 2. Isometric view.

The characteristics and conditions of these equipments during the experiment are:

Electric Heater - A4

- capacity: 500 liters

- capacity on tube side: 140 liters

Piping System

- pipes of 80 mm of internal diameter.
- T1 is 6 meters above A4.
- pipe lengths from A4 to T1 and from T1 to A4 are, respectively 54 and 78 meters.
- capacity of pipes used in the experiment: 660 liters.

The following data were registered at a sampling rate of approximately two minutes, for 150 minutes in each run:

- three RTD temperature indications at two different places on the hot leg (lines leading from A4 to T1);
- three RTD temperature indications at two different places on the cold leg (lines leading from T1 to A4);
- two mass flow indications from two full-flow turbines on the primary system and
- inlet/outlet temperatures and mass flow of the T1 cooling water.

The heater electric power was calculated for each run by measuring the applied electric current and tension.

The mass flow rate was determined by the heat balance at the A4. Assuming adiabatic, steady state flow, follows:

$$m = q/(C_p(T_{out}-T_{in}))$$

where:

m is the mass flow rate through A4;

q is the electric power at A4;

C_p is the specific heat value at $(T_{out}+T_{in})/2$;

T_{out} is the temperature at A4 outlet and

T_{in} is the temperature at A4 inlet.

RESULTS

Experimental Results. Twenty runs, in two sequences of ten at two different days were performed. The initial electric power at A4 was 15 kW, going up to 150 kW, in steps of 15 kW.

Calculations of Departure from Nucleate Boiling (DNB), under pool boiling conditions, at the electrical heater were made. The minimum pressure in the loop to prevent reaching DNB, with maximum heat flux in the electrical heater, is approximately 9 bar. So, the experiment was made with the primary system around 15 bar absolute pressure.

The water in the loop was at 35 C at the very beginning of each sequence. All valves in the primary system are fully open to ensure the lowest hydraulic resistance. In order to ensure that all the heat introduced in the loop would be rejected to the heat sink, the cooling water flow was set constant and equal to its maximum value (26 kg/s). The inlet temperature of the cooling water was also held constant (25 C).

The temperature sensor closest to the hot source outlet indicated erratic behavior. So, the data from that instrument was disregarded. The turbine mass flowmeters signals were used only to confirm the existence of fluid motion inside the tubes. It was not possible to consider quantitatively their values since they operated out of their calibration range.

Table 1 presents, for each nominal electric power at A4, the mean values of temperature difference and mass flow rate.

Table 1. Experimental Results

Nominal Power (kW)	Measured Power (kW)	$(T_{out}-T_{in})$ (C)	Mass Flow (kg/s)
15	15.5±0.3	18.8±0.35	0.198±0.006
30	30.8±0.5	21.7±0.35	0.341±0.009
45	45.8±0.6	24.8±0.35	0.444±0.010
60	61.7±0.7	28.8±0.35	0.515±0.010
75	76.7±0.9	31.1±0.35	0.593±0.011
90	92.6±1.2	33.8±0.35	0.659±0.013
105	107.9±1.3	36.3±0.35	0.715±0.013
120	123.1±1.4	38.5±0.35	0.769±0.014
135	136.6±1.5	40.4±0.35	0.813±0.014
145	139.4±1.5	40.6±0.35	0.826±0.014

The arithmetic mean temperatures at each leg were used to make the heat balance and calculate the mass flow. It was considered uncertainties of 0.35 C for temperature and 1 % for specific heat (C_p).

Numerical Results. The calculations were performed dividing the loop in 90 control volumes, as follows:

- electric heater - 9 slices (18 nodes: 9 for the fluid and 9 for the walls) and
- heat exchanger - 12 slices (12 nodes for the fluid, 12 for the walls and 12 for the cooling water).

The A4 heating elements were not represented in this modeling. It was considered that all heat comes from the walls.

The largest uncertainties are expected to be on the evaluation of the electrical heater, pump and heat exchanger form losses.

Considering the nodes at an initial temperature equal to 20 C and the fluid at rest, for a nominal power of 145 kw at A4, the evolution of the mass flow rate is shown in Figure 4.

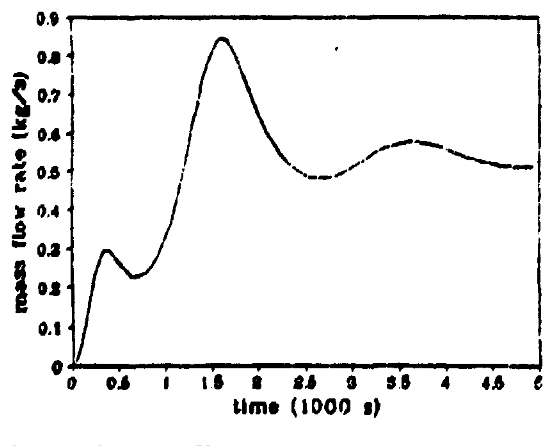

Figure 3. Mass flow rate evolution.

This figure shows the oscillatory behavior of the system, characteristic of natural circulation transient regimes. The stabilization times are approximately 5 hours. This is due to initial conditions (fluid at rest and uniform temperature) and the thermal capacities involved.

A comparison between the experimental and numerical results for steady-state conditions is shown in Figures 4 and 5.

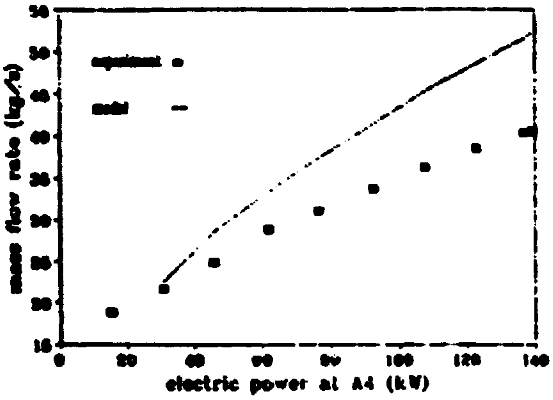

Figure 4. Mass flow rate comparison.

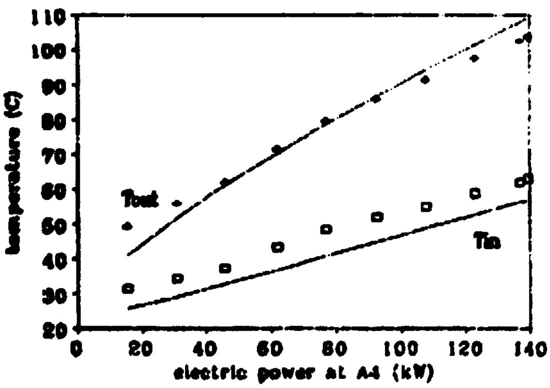

Figure 5. T_{out} and T_{in} comparison.

It is observed a good agreement between the results.

The mass flow rate is quite well predicted. The differences varies from 3 % (15 kW) to 23 % (145 kW). This increasing difference can be explained analyzing the frictional and form losses of the total flow resistance parameter. As the fluid velocity increases the frictional losses decrease and the form losses remain constant. So, the influence of the latter, which does not have a good prediction, is predominant at higher velocities.

CONCLUSIONS

Considering the experiment performed, it was concluded that measurable conditions, despite difficulties for direct flow measurement, are obtained for the operational parameters imposed. The comparison of the experimental data with the numerical has shown good agreement for the steady-state regime. The differences for higher levels of electric power may be explained by the lack of information about form losses.

In order to validate the code in transient conditions, some improvements at CTE-150 and experimental procedure are under way:

1. calibration of the turbine flowmeters in an appropriate range;
2. installation of differential pressure transducers at A4 and T1;
3. reduction of cooling water flow rate at T1 to increase its temperature variation and measure the heat losses;
4. increasing of sampling time for longer periods of runs.

Regarding the computer code evolution, detailed models for A4 and T1 are being developed in order to have a better representation of the heat transfer processes and to refine the description of the geometry.

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