Modeling The Flow Characteristics During Start-Up of Natural Circulation Systems from Rest State

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Abstract

Most of the advanced reactor designs propose to use natural circulation for core heat removal either during normal operational states and/or during off-normal operational states because of simplicity and higher reliability of natural circulation. However, these systems do have their own set of challenges with respect to assessment of their reliability. Simulation of start-up from rest and performance evaluation using conventional models are the key issues in assessing their reliability. In view of their wide importance to nuclear safety, the performance of these systems during start-up from rest is addressed in the present study.

Introduction

Passivity and simplicity are the hallmark of any good engineering design. These features are perhaps nowhere more desirable than in nuclear systems where safety of both the plant and the public at large is of prime concern. Passivity, the ability to operate without reliance on any external power source, enhances the safety. Simplicity, avoidance of complex piping and logics, makes the system operation and maintenance considerably easy. A natural circulation system has both passivity and simplicity inbuilt in it. It is because of these desirable features that these systems find wide application in many industrial systems like nuclear power plants, solar heating and cooling systems, geothermal systems, electrical machine rotor cooling, turbine blade cooling, electronic device cooling and process industry. Many of the next generation nuclear power plants propose to use natural circulation as the heat removal mode either during normal operational states and/or during off-normal operational states. Indian Advanced Heavy Water Reactor (AHWR) and Prototype Fast Breeder Reactor (PFBR), Westinghouse's AP-1000, General Electric's Economic Simplified Boiling Water Reactor (ESBWR), Russian VVER-1000 and Argentina's CAREM, are just to name a few. Despite the philosophy of defense-in-depth, electromechanical active components and/or human operations are likely to fail as exemplified by the accidents at Three Mile Island-2, Chernobyl-4 and Fukushima.

Decay heat, if not removed, can result in overheating and damage to the fuel. Hence, most of the advanced reactor designs propose to use natural circulation for decay heat removal during the off normal conditions.Like any other system, the natural circulation systems also do have their own set of challenges. Though passive systems are highly reliable, in reality there exists a non-zero probability that these system fail to perform the intended function. Therefore, a reliable prediction of their performance is of utmost importance for their successful deployment. Natural circulation systems have low driving force and need to be started from the state of rest. Start-up from rest is one of the key issues in assessing the reliability of these systems. There is always a finite time lag before these systems attain their optimum/intended performance level. Further, the models applicable under low flow conditions can be quite different from those applicable under high flow conditions. For example, under low flow conditions, the flow is essentially multi-dimensional and hence classical 1-D may not predict their behaviour with reasonable satisfaction. Also the performance of these systems is strongly dependent on the operating conditions and system geometry. Owing to the very dependence of system flow rate (and hence heat removal capability) on operating conditions, any change in operating conditions may have a bearing on their overall performance. This article addresses some of these issues.

Model for start-up from rest

During the start-up of a natural circulation system, the flow is essentially single-phase and hence, single-phase natural circulation is important to all the natural circulation systems. Unlike forced circulation systems where flow gets established as soon as the fluid moving device is put on, in a natural circulation system, the flow is generated by the temperature difference between the source and the sink. In a forced circulation system, the flow through the system is established first by putting on the fluid moving machinery. This is followed by activation of heater power, e.g. chain reaction in a nuclear power plant and electrical heating in experimental facilities. However, in a natural circulation system, the sequence is reverse. Since both source and the sink are at the same temperature to start with, initially there is no flow through the system. Hence, the heater power needs to be raised to some positive non-zero value before flow gets established through the system. The problem is further complicated by the orientation of the heat source and sink. Heat source can be vertical as in AHWR or ESBWR or horizontal as in PHWRs. Similarly sink can also be horizontal as in VVERs or vertical as in PHWRs.While in systems having vertical heater the onset of bulk fluid circulation is almost immediate for moderate heater powers, in loops having heaters in horizontal leg, the bulk circulation is always preceded by some quiescent period. In systems having horizontal heaters, the flow does not take place till the hot fluid reaches the vertical leg. This has a bearing on the peak heater element temperature and hence on the integrity of the fuel or heating element. Present 1-D models, which continue to be the workhorse for reactor thermal hydraulics and safety analysis, fail to predict the start-up of natural circulation systems having heat source in horizontal leg. Under low or no flow (circulation) conditions, the heat is transferred from the heated sections to unheated sections by natural convection. These currents were referred as secondary currents by Bau and Torrence[1]. These currents are essentially multi-dimensional in nature. They are relevant not only during start-up but also during the transients whenever low flow conditions are encountered. It is because of the inability of present 1-D models to account for these local convection currents that these models fail to predict the start-up of these systems from rest reliably.Numerical simulations carried out using system codes like ATHLET [2], RELAP5 [3] and CATHARE [4] show that start-up of these loops cannot be simulated using classical 1-D models. Fichera and Pagano [5] accounted for these currents by taking an arbitrarily high value of molecular thermal diffusivity in their numerical simulations.In the present study, a pseudo-conductivity model has been presented to account for these secondary convection currents. The proposed model is based on scale analysis. It has been validated against CFD simulations and incorporated in a 1-D model. The results of numerical simulations for start-up of single-phase natural circulation systems having different heater and cooler configurations are presented.

Governing Equations

The mathematical model is based on the following simplifying assumptions:

(a) The flow can be represented by a 1-D description that does not take into account for radial variation of the fluid properties.

(b) Heat generation in the fluid due to frictional dissipation and macroscopic terms, i.e. kinetic and potential energy are neglected in the energy equation.
(c) Each fluid element is in thermal contact with a heat structure and a convective heat transfer boundary condition can be assigned to the outside surface of the structure, specifying the outer fluid temperature and the outer heat transfer coefficient.
(d) For a nearly incompressible fluid, natural convection consists of two equal and opposite streams moving between cold and warm sections. There is zero net mass transfer between the two sections and a net energy exchange. The energy change can be accounted in the area averaged energy balance equation.

With the above assumptions the governing mass, momentum and energy balance equations for flow through a channel are written as:

$$\frac{\partial \rho}{\partial t}+\frac{1}{A}\frac{\partial W}{\partial s}=0 \tag{1}$$

$$\frac{1}{A}\frac{\partial W}{\partial t}+\frac{1}{A}\frac{\partial}{\partial s}\left(\frac{W^2}{\rho A}\right)=-\frac{f}{D}\frac{W^2}{2\rho_l A^2}-\frac{\partial p}{\partial s}-\rho g\sin\theta \tag{2}$$

$$\frac{\partial}{\partial t}(\rho h)+\frac{1}{A}\frac{\partial}{\partial s}(Wh)-\frac{\partial p}{\partial t}=\frac{\lambda_i P\left(T_w-T_f\right)}{A}+\frac{1}{A}\frac{\partial}{\partial s}\left(AK_{eff}\frac{\partial T}{\partial s}\right) \tag{3}$$

$$\rho=\rho\left(p,h\right) \tag{4}$$

For the solution of governing equations, these are discretized using staggered mesh. The computational domain is divided into non-overlapping rigid control volumes, called cells. The field variables, h, p and ρ are defined at the cell centers, while the fluid mass velocity, W is defined at cell interfaces also called junctions. The discretized equations are presented in Naveen Kumar [6].

Evaluation of K_{eff} – Pseudo-conductivity Model

K_{eff} in equation (3), known as effective conductivity, is evaluated using the pseudo-conductivity model described here. In a natural circulation loop having heater in the horizontal section, the fluid is motionless and is at same temperature to start with. When the heater is switched on and power is made greater than zero, the fluid in the heated portion starts getting heated up. This hot fluid spreads sideways into the unheated section by local natural convection. For an incompressible fluid, these currents lead to net energy transfer from the heated section to unheated section without any net mass transfer across the pipe cross section. Hence, these currents can be thought like pseudo-conduction. During the start-up of natural circulation loops having horizontal heater, the natural convection currents are confined to the horizontal pipe section only and the fluid behavior is similar to that in a slender horizontal cavity closed at both the ends. During this phase, the area averaged net flow at any cross section remains zero and there is a net heat transfer at the interface between hot and cold sections. The rate of heat transfer across any cross-section depends upon the net forward or backward (both are equal in magnitude for an incompressible fluid) mass flow rate at that cross section and the difference in temperatures of two stream. Naveen Kumar et al. [6-7] studied the natural convection in slender pipes closed at both the ends. In this study it was shown that the magnitude of cross section area averaged forward or backward velocity (backward and forward currents are equal in magnitude but opposite in direction) for horizontal and vertical pipes is given by the following expression:

$$u_{nc,j}=\begin{cases}0.99\left(\alpha/D\right)\left(Ra_D\right)^{1/2} & \textit{for horizontal cells}\\ 0.0018\left(\alpha/D\right)Ra_D & \textit{for vertical cells}\end{cases} \tag{5}$$

It is worth noting here that the velocities given by equation (5) are area averaged forward or backward (in case of a vertical cavity, it is upward or downward) velocities at any cross section under steady state conditions. However, during natural convection, the forward current is expected to be confined to half of the pipe cross section and the backward current is expected to be confined to the other half of pipe cross section. Therefore, the convection currents will move twice as fast as that given by equation (5). For a loop having horizontal heater, the heat transfer by natural convection can exist in horizontal section whenever natural circulation flow is small e.g., during start-up of such systems from no flow condition. However, for a vertical pipe section, this can happen whenever a hot fluid pocket lies below a cold fluid pocket e.g., during the transient the hot pocket may be lying below the cold pocket, and the hot and cold fluid packets may be distributed in two legs such that the loop flow is near zero. These conditions are expected during the transient unstable flow conditions and have been observed experimentally by Vijayan et al. [8]. Naveen Kumar [6] showed that the net energy exchange per unit area by these local convection currents is given by the following expression:

$$q''_{nc} = \begin{cases} 0.99\left(k_f/D\right) Ra_D^{1/2} \Delta T & for\,horizontal \quad cells \\ 1.8x10^{-3}\left(k_f/D\right) Ra_D \Delta T & for\,vertical \quad cells \end{cases} \tag{6}$$

Thus in the absence of any bulk fluid motion, the steady state heat transfer between the cold and hot sections, maintained at a temperature difference of ΔT, consists of two components: conduction and natural convection. Mathematically, it is written as

$$q'' = q''_{cond} + q''_{nc} = -k_f\left(\Delta T/\Delta s\right) + h_{nc}\Delta T \tag{7}$$

where $h_{nc} = \begin{cases} 0.99\left(k_f/D\right) Ra_D^{1/2} & Horizontal \quad pipe \\ 0.0018\left(k_f/D\right) Ra_D & Vertical \quad pipe \end{cases}$ (8)

For ease of interpretation, equation (7) is written as

$$q'' = -K_{eff}\left(\Delta T/\Delta s\right) \tag{9}$$

where $K_{eff} = k_f + h_{nc}\Delta s$ (10)

K_{eff} is the effective thermal conductivity which takes into account the heat transfer by both conduction and natural convection. The first term in equation (10) is the molecular conductivity of the fluid and accounts for axial heat transfer by conduction in the fluid. The second term accounts for heat transfer by local natural convection between two volumes (pipe sections) filled with fluids having different temperatures. However, unlike the fluid thermal conductivity, which is a property of fluid, the magnitude of pseudo-conductivity depends on the temperature difference between the two fluid volumes. Thus, the effect of natural convection can be accounted in 1-D model by using an effective thermal conductivity as given by equation (10). Advantage of this approach is that it can easily be incorporated in the conventional 1-D models without any major modifications. Also, this approach has the advantage of being computationally cost effective.

Local convection is the dominant mechanism of heat transfer till the natural circulation flow is zero. With the onset of bulk flow (circulation), the fluid starts moving either in clockwise or anti-clockwise direction throughout the loop. The fluid flow is now just like forced convection through a pipe. With the onset of bulk fluid motion, these currents start diminishing and eventually get suppressed completely when loop flow rate increases sufficiently. Hence, the heat transport mechanism in a natural circulation loop passes from the phases of local natural convection to mixed convection to natural circulation. A proper simulation of loop dynamics requires not only proper models for each regime but also the knowledge of criterion for transition from one regime to another. Naveen Kumar et al. [7] presented the following criterion for transition between natural convection and natural circulation:

$$h_{nc} = \begin{cases} \left(1-\left|\bar{u}_j/u_{nc,j}\right|\right) h_{nc} & if \quad \left|\bar{u}_j\right| \le \left|u_{nc,j}\right| \\ 0 \quad if & \left|\bar{u}_j\right| > \left|u_{nc,j}\right| \end{cases} \tag{11}$$

The above criterion is based on the relative strength of natural convection and natural circulation currents. The relative strength of natural convection to natural circulation is given by the ratio of the characteristic natural convection velocity, $u_{nc,j}$ and the natural circulation velocity, $\bar{u}_j$. It can be seen from equation (11) that pure natural convection is the heat transfer mode when natural circulation ($\bar{u}_j$) velocity is zero and natural circulation is the heat transfer mode when natural circulation velocity ($\bar{u}_j$) is greater than natural convection velocity ($u_{nc,j}$). The mixed convection regime lies between these two regimes. This regime is taken into account in the above criterion by introducing the factor $\left(1-\left|\bar{u}_j/u_{nc,j}\right|\right)$ in equation (11). The reasons for adopting such a criterion are explained below. The natural convection currents are essentially multi-dimensional in nature even in small diameter pipes and develop fully only in the absence of forced convection (global natural circulation) currents. However, forced convection currents are strongly directional in nature and involve either movement of fluid from the cold section to the hot section or from the hot section to the cold section. There is net flow across the interface. As one moves away from natural convection regime to mixed convection regime, these natural circulation (bulk loop flow) currents start interfering with natural convection currents. This impedes the development of natural convection currents thereby hindering the energy transfer by natural convection across the interface. This has been taken into account in equation (11) by introducing the factor $\left(1-\left|\bar{u}_j/u_{nc,j}\right|\right)$. The criterion given by equation (11) blends the switching from natural convection to natural circulation in a continuous fashion. The pseudo-conductivity model has been validated by comparing the model predictions with CFD simulations for circular horizontal and vertical cavities closed at both the ends [6-7].

Role of constitutive laws for wall friction

The inability of conventional friction factor correlations applicable for forced circulation under adiabatic conditions to predict pressure drop under diabatic conditions was recognized long back by Deissler[9]. Ambrosini and Ferreri[10] showed that accurate and reliable prediction of loop stability and transient behaviour is strongly dependent on the choice of friction factor. Naveen Kumar [6] proposed the following correlation for friction factor for flow through horizontal pipes:

$$f_F = f_L\left[1+\left(f_{Tt}/f_L\right)^{6.4}\right]^{1/6.4} \tag{12}$$

where

$$f_{Tt} = f_T\left[1+\left(f_t/f_T\right)^{-2}\right]^{-\frac{1}{2}} \tag{13}$$

$$f_L = \left(16/\mathrm{Re}\right)\left\{1+\left(1.56Ra_f^{\,0.15}\right)^{15}\right\}^{1/15} \tag{14}$$

$$f_T = 0.0791\,\mathrm{Re}^{-0.25}\left(\mu/\mu_w\right)^{-0.2} \tag{15}$$

$$f_t = 0.03862\left(\mathrm{Re}/2000\right)^{6} \tag{16}$$

The correlation given by equation (12) was derived from steady state experimental data by Naveen Kumar [6]. A comparison of the proposed correlation with conventional forced convection laws is shown in Fig. 1. It is seen from the Fig 1 that the conventional forced friction factor laws overpredict the loop mass flow rate (higher Ress for a given Grm/NG). A similar type of correlation showing dependence on local Rayleigh number is expected for vertical pipe sections also. However, in the absence of any such correlation, the wall friction in vertical sections has been evaluated using conventional forced friction correlation. To sum up, the model uses the following correlation for wall friction:

$$f_F = \begin{cases} Eq.(12) & Horizontal \quad pipes \\ \left[16/\mathrm{Re}, \quad 0.079/\mathrm{Re}^{0.25}\right] & Vertical \quad pipes \end{cases} \tag{17}$$

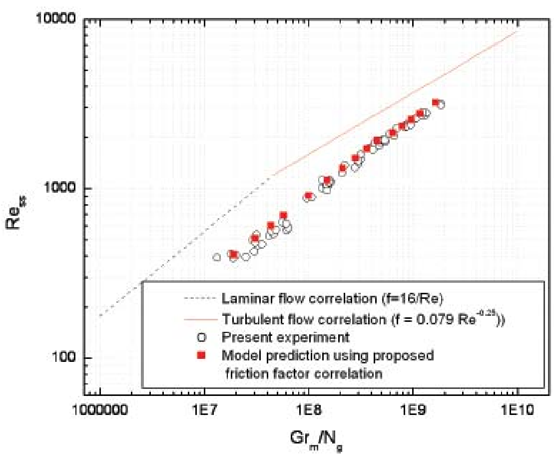


Fig. 1: Comparison of model predictions made using proposed friction factor correlation with experimental data.

Flow characteristics of single-phase natural circulation during start-up

The model developed has been applied to simulate the start-up of natural circulation loop shown in Fig. 2. The loop consists of a uniform diameter rectangular natural circulation loop. The details of the experimental setup are given in Vijayan et al. [4]. It has two heaters and two coolers. One of the heaters is at lowest elevation and the other one is in vertical section. Similarly, one of the coolers is placed in the horizontal section at the uppermost elevation and the other cooler is placed in vertical pipe section. It is possible to run the loop in any of the following four configurations:

(a) Horizontal Heater and Horizontal Cooler (HHHC)
(b) Horizontal Heater and Vertical Cooler (HHVC)
(c) Vertical Heater and Horizontal Cooler (VHHC)
(d) Vertical Heater and Vertical Cooler (VHVC)

The addressed loop with different combinations of heaters and coolers can be considered as representative of different natural circulation systems encountered in the industry.

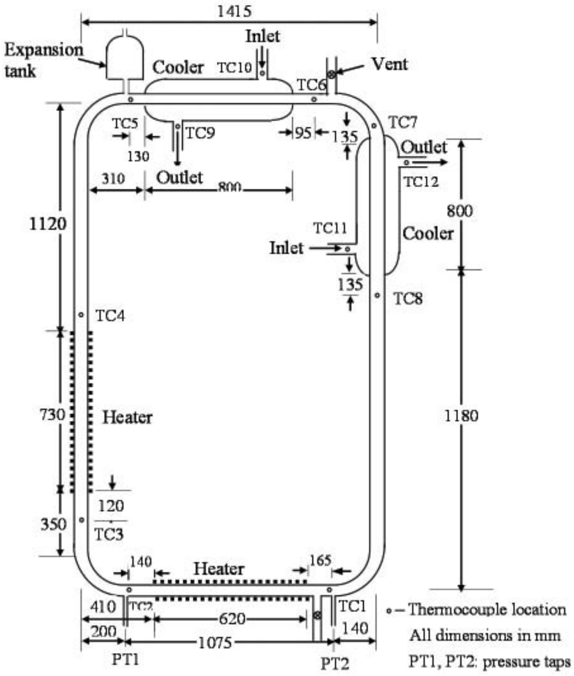
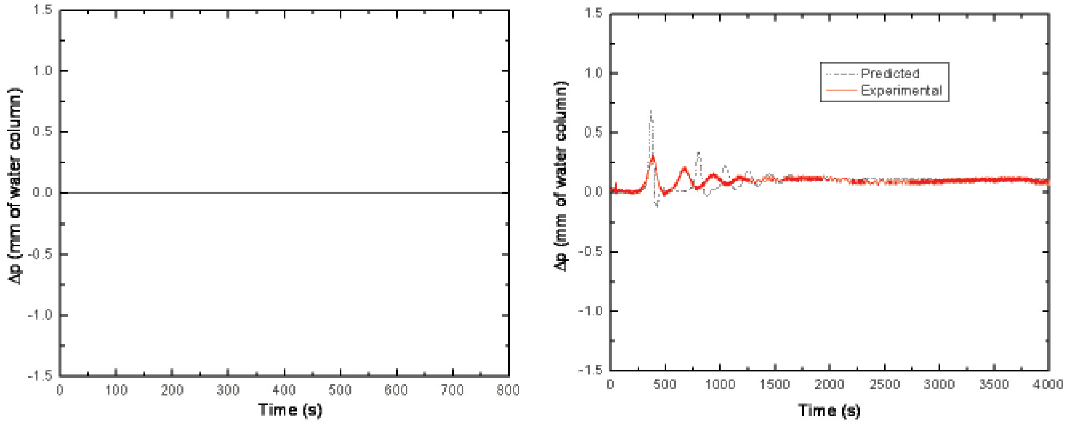


Fig. 2: Schematic of natural circulation loop [8]

Studies for horizontal heater configuration

The model developed has been applied for the simulation of loop transient behavior from state of rest conditions. The initial fluid temperature is assumed to be same throughout the loop and the expansion tank. In the test facility, the heater is located symmetrical with respect to loop centre line. In such a facility, there is equal probability of flow getting initiated in either direction. However, in numerical simulations the direction of flow initiation is dictated by the sign of initial fluid velocity which is assumed to be 1.0E-9 kg/s.

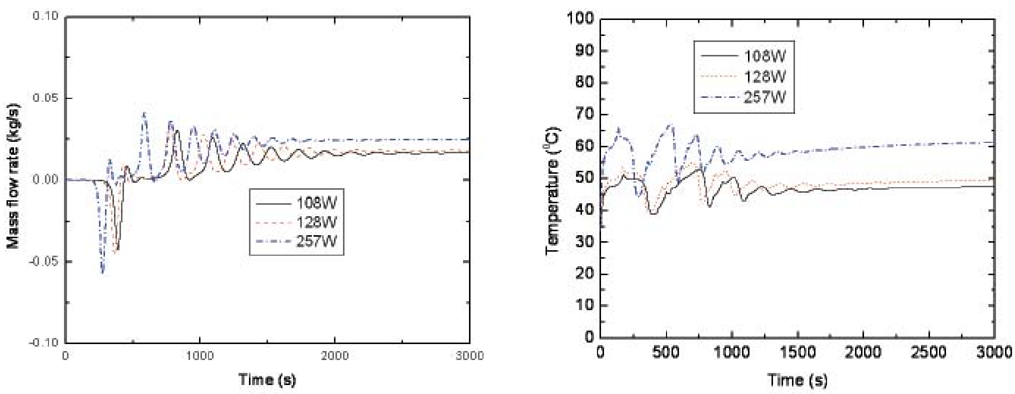
Fig. 3 shows the flow initiation transient predicted by the present model with the fluid axial heat conduction taken same as that the molecular conductivity of the fluid (h_{nc} = 0) for the HHVC



(a) hnc= 0 (b) With pseudo-conductivity model

Fig. 3: Prediction of flow initiation transient by the model developed for HHVC configuration for a heater power of 128 W.

configuration for a heater power input of 128 W. It is clear from Fig. 3(a) that the flow initiation does not take place even after a lapse of more than 800 seconds since switching on of the heater power. It was noted by Bau and Torrance [1] that mere inclusion of fluid and wall axial conduction does not help in simulating the start-up from the rest state. The same simulation is now carried out using the pseudo-conductivity model explained earlier ($K_{eff}= k_f+ h_{nc}\Delta s$). The results shown in Fig 3(b) indicates that the flow gets initiated at t = 300 seconds. This behavior has been found to be in agreement with that observed experimentally. However, the magnitude of the initial peak is greater than that observed experimentally. Also there is a mismatch in the time at which peak occurs. The mismatch between the model prediction and experimental observation can be attributed to the simplification adopted in modeling heater. The heater in the experimental test facility consists of a nichrome wire evenly wound on the glass tube. The nichrome wire makes a line contact with the glass tube and is surrounded by mineral wool from all other sides. In the heater model adopted in the numerical simulations, all the heat is assumed to be generated inside the glass tube, while in reality a portion of the heat also goes to the insulating material. This has a bearing on the flow initiation time and the magnitude of initial peak. This becomes clear from Fig 4 where the model predictions for different initial heat power inputs are compared.It is clear from Fig 4(a) that initial peak flow increases with increase in heater power input. Also the heater temperature (Fig 4(b)) increases with increase in heater power.



(a) Loop mass flow rate (b) Heater surface temperature

Fig. 4: Effect of heater power on flow initiation transient for HHVC configuration.

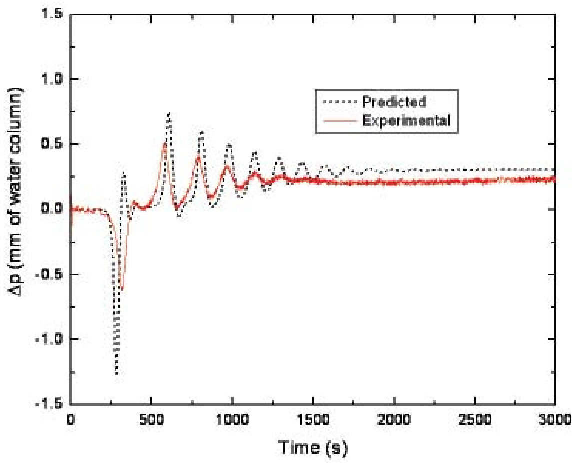


Fig. 5: Comparison of flow initiation transient predicted by the model developed with experimental behavior for HHVC configuration for a heater power of 257 W.

Vijayan et al. (2001) reported that for HHVC configuration, while there is equal probability of flow getting initiated in either clockwise or anticlockwise direction, however, only clockwise flow was found to be stable. This is shown in Fig. 5, where for a heater power input of 257 W, the flow gets initiated in anti-clockwise direction and it turns clockwise after one oscillation. A similar behavior is predicted by the model as well. Fig. 5 shows the comparison of the model predictions (made using pseudo-conductivity model) with the observed experimental behavior. The model predictions are in fair agreement with the observed experimental behavior. Fig 6 shows a comparison of model predictions with the experimental data for HHHC configuration for a heater power input of 120 W. For this configuration, both loop mass flow rate (indicated by pressure drop across heater) and fluid temperature were found to be oscillating. In fact, for this configuration, stable behaviour was observed only at powers below 55 W.

Role of expansion tank

All the closed single-phase natural circulation loops, the authors have come across are provided with an expansion tank. The tank serves the twin purposes of venting the air out during the loop filling and accommodation of the swells and shrinkages of the loop fluid during the transient. Creveling et al. [11] and Damerell and Schoenhals [12] mentioned that the interior of the loop was connected to an open reservoir; however, no details of the reservoir were reported. The loops studied by Misale et al. [4] and Fichera and Pagano [5] and Vijayan et al. [8] have expansion tank in their loops. However, only few details are available about the dimensional details of these tanks in published literature. In most works, this component is not considered important in characterizing natural circulation loop performance. In nuclear reactors also, the components like pressuriser, emergency core cooling headers etc. are also similar to expansion tank. In the present study, the expansion tank was

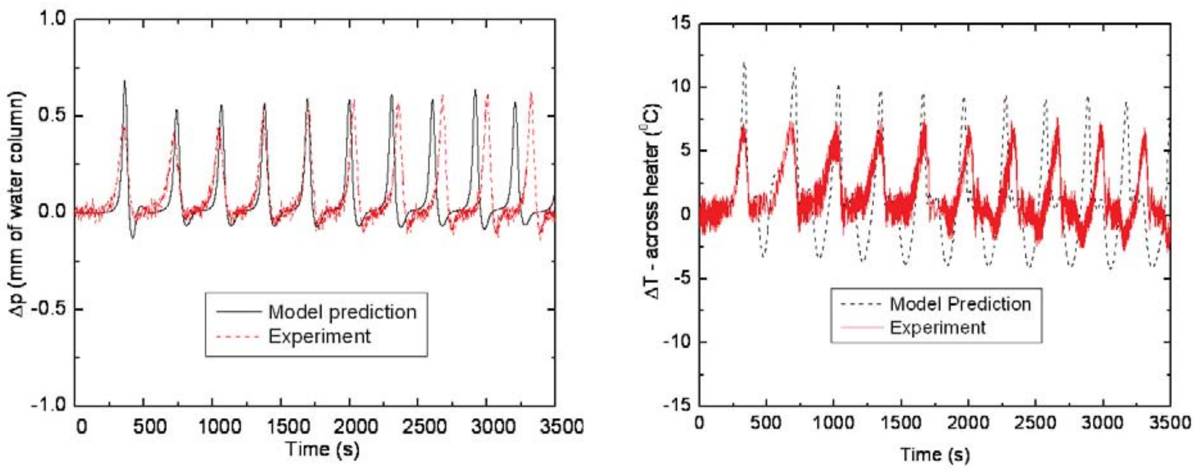


Fig. 6: Comparison of the predicted behaviour with experimental data for 120W heater power for HHHC configuration.

found to be affecting the performance of these systems.

In most of the studies, a lumped parameter model is used for modelling the expansion tank. The pressure in the tank is assumed to be constant. With these assumptions, mass, momentum and energy balance in the expansion tank take the following from:

$$\frac{dm_{TK}}{dt} = -\frac{dm_{loop}}{dt} \tag{18}$$

Neglecting frictional and acceleration losses, the momentum balance is written as

$$\left(p_{atm} - p_0^{n+1}\right) = -\rho_{TK}^{n}\left(L_{TK}^{n+1} + \Delta Z_L\right)g \tag{19}$$

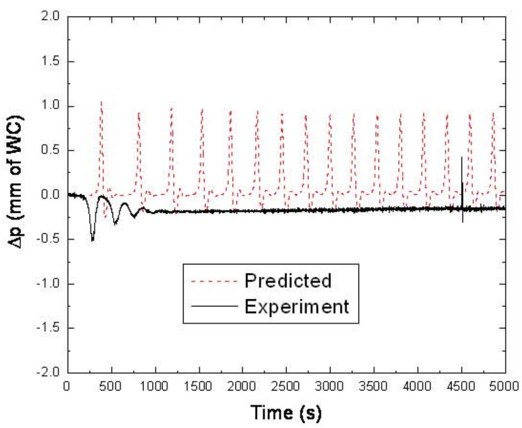
Energy balance equation is written as

$$A_{TK}L_{TK}\frac{d}{dt}(\rho_{TK}h_{TK}) = \begin{cases} h_{m+1}\dfrac{dm_{loop}}{dt} - A_C h_{nc}\Delta T & if\ \dfrac{dm_{loop}}{dt} > 0 \\ -h_{TK}\dfrac{dm_{loop}}{dt} - A_C h_{nc}\Delta T & if\ \dfrac{dm_{loop}}{dt} < 0 \end{cases} \tag{20}$$

The first term on the right hand side of Eq. (20) represent the energy exchange with the loop because of swell/shrinkage in volume of liquid contained in the loop and the last term accounts for energy exchange between the expansion tank and the main loop by natural convection. Thus, h_{nc} in equation (20) is evaluated using the following expression:

$$h_{nc} = \begin{cases} 0.0018\dfrac{k}{D}\left(\dfrac{g\beta D^3}{v\alpha}\right)\left|T_0^n - T_{TK}^n\right| & for\quad T_0^n > T_{TK}^n \\ 0 & for\quad T_0^n \le T_{TK}^n \end{cases} \tag{21}$$

D in equation (21) represents the diameter of the pipe connecting the main loop and the expansion tank and A_C is area of cross-section of the connecting pipe. In the absence of any flow through the pipe, there will be circulation of fluid between the main loop and the expansion tank if fluid temperature in the main loop is higher than that in the tank. This fluid circulation is caused by the natural convection currents and is known as natural convection heat transfer. Under pure natural convection conditions, these currents are responsible for net heat exchange between the pipe and the expansion tank. This heat exchange was found to have a bearing on loop stability behaviour by Naveen Kumar [6]. The experimental investigations in loop having geometry similar to that addressed in present study for numerical simulations (Fig. 2) showed that expansion tank temperature increases during the transient. It is worth noting that most of the numerical simulations reported in literature ignore this energy exchange between the expansion tank and the main loop by natural convection. Fig. 7 and Fig 8 show the comparison of experimentally observed behaviour with that predicted using 'Simplified Expansion Tank Model' (h_{nc} = 0) and 'Modified Expansion Tank Model' (h_{nc} given by equation (21)). It is clear from Fig 7 that simplified expansion tank model predicts unstable behaviour while modified expansion tank model predicts stable behaviour. Fig 8 shows comparison for expansion tank temperature. It is clear from Fig. 8(b) that modified expansion tank model captures the loop



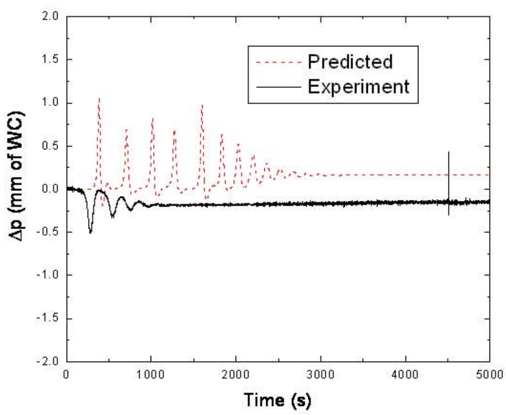
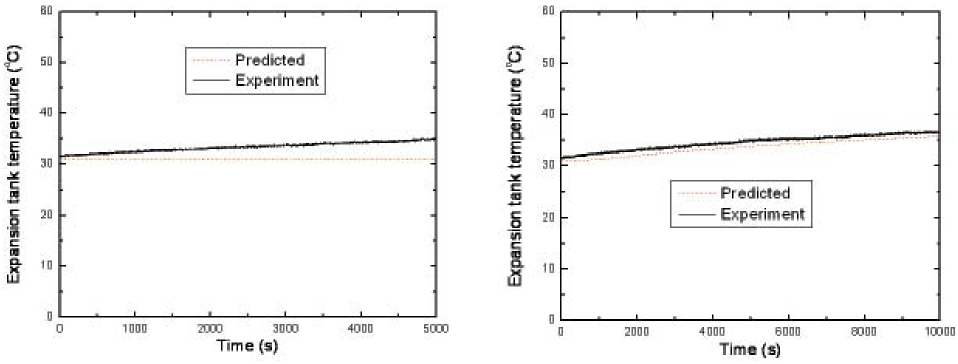


Fig. 7: Comparison of the predicted pressure drop across the heater with the experimentally observed behavior for heater power of 200W



(a) Simplified expansion tank model (b) Modified expansion tank model

Fig. 8: Comparison of the predicted fluid temperature in the expansion tank with the experimentally observed behavior for heater power input of 200W

behaviour more realistically. The results presented in Fig 7 and 8 clearly show that the tanks connected to these systems may modify the operating conditions and hence the loop dynamic thermal hydraulic behaviour. In nuclear reactors, main heat transport systems have several components like pressuriser, Emergency Cooling Headers etc. connected with it. While simulating the behaviour of these systems, due consideration should be given to interaction of these systems with main loop.

Studies for vertical heater configuration

In a natural circulation loop having vertical heater, the flow initiation occurs almost immediately after the heater power is made greater than zero. This is because the buoyancy starts developing as soon as the fluid in the vertical heated section gets heated. Typical flow initiation transient predicted with model developed for VHHC configuration for a heater power input of 530W is shown in Fig. 9. The model predictions are in reasonably good agreement with the experimental behavior. However, as discussed earlier, the secondary effects (simplified heater wall dynamic model and use of wall constitutive laws derived from steady state forced convection experiments) lead to over estimation of the magnitude of the initial peak. As the hot fluid rises through the vertical pipe, it comes in contact with cold wall. When the hot fluid comes in contact with cold wall, it gets cooled, the buoyancy acts opposite to loop flow and fluid motion near the wall has a tendency to get retarded. This alters the velocity profile near the wall and may change the wall friction factor. Saylor and Joye [13] studied pressure drop in mixed convection heat transfer in vertical tubes and observed that at low Reynolds numbers and high Grashof numbers, the pressure drop for aiding flow (heating in up flow and cooling in down flow) through a vertical tube under constant temperature conditions can be orders of magnitude higher than that expected on the basis of forced flow considerations. They also observed negative pressure drop for downflow heating in mixed convection zone. Also subtle changes in model predictions made using simplified and modified expansion tank model are quite clear.

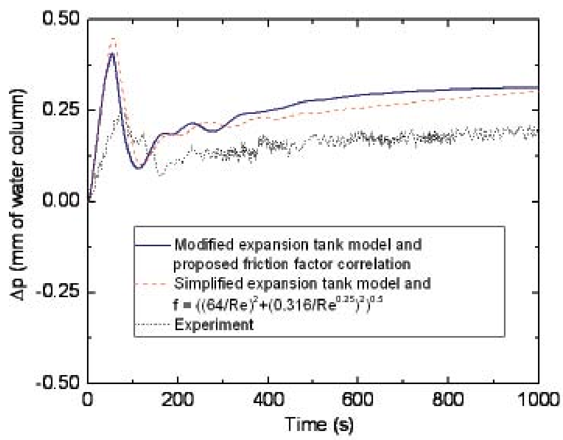


Fig. 9: Comparison of the predicted behaviour with experimental data for 530W heater power for VHHC configuration.

Conclusions

In this research, a model has been developed to predict the performance of natural circulation systems during start-up from rest. A pseudo-conductivity model has been proposed to account for multi-dimensional heat diffusion under low flow conditions. The model has been validated against the CFD simulations and successfully integrated with 1-D models which are the industry workhorse for predicting the behaviour of these systems. The model has then been used to predict the performance of a natural circulation loop having different heater and cooler orientations. The numerical predictions show that onset of flow in a NCL having horizontal heater is always preceded by some quiescent period. During this period, the heater surface temperature can rise substantially. Numerical and experimental studies show that onset of flow in a loop having vertical heater is almost immediate for moderate heater powers. The study brings out that the conventional friction factors derived from steady state forced convection experimental data are not applicable for natural circulation loops under low flow conditions. A new correlation has been proposed for wall friction for flow through horizontal tubes. In this research, the behaviour of natural circulation systems has been found to be affected by the expansion tank.

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