Mathematical Analysis of the Transient Dynamic of Surge-In or/and Surge-Out of the Pressurizer of PWR

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Abstract **- We study and analysis the dynamic transient behavior of the pressurizer of pressurized nuclear water reactor PWR. In PWR the accurate prediction of system pressure is essential since system pressure is a key parameter in controlling the system transient behavior, hence a fundamental understanding and a reliable modeling of surge tank phenomena during transients are vital. The dynamic pressure of the system and the transient behavior are investigated. Overpressure necessary to limit the bulk boiling is maintained by this pressurizer. The volume of the main coolant expands and contracts with average temperature variations causing in- surges and out- surges in the surge tank volume (using a spray, relief system and electric heater) which is called "pressurizer". A pressurizer with a steam light water interface is used to maintain the sensitive pressure-temperature balance in the primary system by using heaters to make more steam and increase pressure or spraying cool water to condense steam and reduce pressure. Increase and decrease coolant mass flow rate or specific volume leads to in-surge or out-surge transient, which changes pressurizer tank's pressure so tow region thermodynamic model release to analyzing the dynamic transient behavior in-surge and out-surge tanks. The basic balances equations of mass and energy conservation with boundary condition constant heat flux, the number of flashing drop and variation temperature to casing in or/ and out surges. Theoretical thermodynamic modeling which generates a comprehensive influence of each change was analyzed. It is shown that the agreement between the results is rather good for different sets of conditions.**

Keywords - pressurizer; transint; nuclear power plant; modelling.

I. INTRODUCTION

Since the early 1950s, nuclear fission technology has been explored on a large scale for electrical power generation and has developed into the modern nuclear power plants. Nuclear Power provide over 11% of the world's electricity as continuous, reliable power to meet base-load demand, without carbon dioxide emissions.as apart of these nuclear reactor a pressurized water reactors so it is clear that the demand for safety nuclear power will be increased in the future, so to emphasize the safety operation for the PWR, it is absolutely need more investigation and researchers in the PWR and all the systems.

The importance of safety in nuclear facilities necessitates the continuous improvement of the accurate models for analyzing the dynamic behavior of all components specifically those, which are responsible for controlling the plant normal operation. In this respect, pressurizer has two major roles of maintaining a prescribed system pressure and a constant coolant mass inventory in the primary loop of a pressurized water reactor (PWR). Therefore, accurate investigation of the pressurizer behavior is crucial in safety evaluation of a PWR reactor. The main function of the pressurizer is providing a means of controlling the system pressure. Pressure is controlled by the use of electrical heaters, pressurizer spray, power operated relief valves, and safety valves. The pressurizer is particularly important in nuclear power plant in a pressurized water reactor and in all industrial application. It is used to maintain the pressure of primary coolant within allowed range because the sharp change of coolant pressure affects the security of reactor therefore the study of pressurizer's pressure control methods is very important. For this purpose, the steam-gas pressurizer model is introduced to predict the accurate system pressure. The proposed model includes bulk flashing, rainout, inter-region heat and mass transfer.

The pressurizer: In a pressurized water reactor plant, the pressurizer is a cylindrical pressure vessel with hemispherical ends, mounted with the long axis vertical and directly connected by a single run of piping to the reactor coolant system. It is located inside the reactor containment building. Although the water in the pressurizer is the same reactor coolant as in the rest of the reactor coolant system, it is basically stagnant.

Any pressurizer consists of:

1- Surge tank
2- Spray water system
3- Relief & safety steam system
4- electric heating elements

The pressurizer (Figure 1) maintains the Reactor Coolant System (RCS) pressure during steady-state operation and limits pressure changes during transients and through the water level control the pressurizer is responsible for the constant coolant mass in the primary circuit.

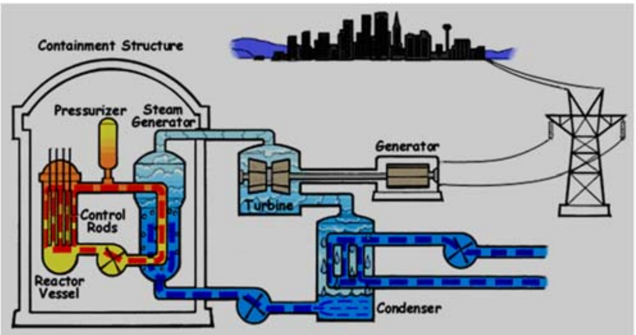

Fig.1.Schematic of a PWR showing the location of the pressurizer.

Replaceable immersion heaters and a spray nozzle are located in the pressurizer. Safety and relief valves discharge to a pressurizer relief tank. Principal design data for the pressurizer are listed in Table 1. During steady-state operating conditions, approximately 60 percent of the pressurizer volume is occupied by water and 40 percent by steam. Electric immersion heaters, located in the lower section of the vessel, keep the water at saturation temperature and maintain a constant system operating pressure.

A reduction in plant electrical load causes a temporary increase in average reactor coolant temperature with an attendant increase in coolant volume. The expansion of the reactor coolant raises the water level in the pressurizer. This increase in water level compresses the steam, raising the pressure and actuating valves in the spray lines. Reactor coolant from the cold leg of a coolant loop sprays into the steam space and condenses a portion of the steam. This quenching action reduces pressure and limits the pressure increases. An increase in plant electrical load results in a temporary decrease in average coolant temperature and a contraction of coolant volume.

Coolant then flows from the pressurizer into the loops, thus reducing the pressurizer level and pressure. Water in the pressurizer flashes to steam to limit the pressure reduction. This reduction in pressure also energizes the immersion heaters, heating the remaining water in the pressurizer to further limit the pressure reduction. Reductions in plant electrical load with resultant pressure increases beyond the pressure-limiting capability of the pressurizer spray system cause the air-operated relief valves to open. These valves are automatically opened at a pressure below system design pressure. They can also be opened manually from a control console in the control room.

If system pressure continues to rise, self-actuating ASME-code safety valves will open. Steam from the safety and/or relief valves is piped to the pressurizer relief tank which contains sufficient water to condense the steam. Cold water can be sprayed into the pressurizer relief tank to increase the heat sink capacity. A rupture disc vents the tank to the containment if design pressure is exceeded. [1] This arrangement also provides a means of pressure control for the reactor by increasing or decreasing the steam pressure in the pressurizer using the pressurizer heaters.

II. LITRETURE REVIEW

From the beginning of using nuclear power in demands of consumption power in the world from 1950s and the needing for more safety systems in nuclear power plants especially after the accidents in some nuclear power plants like Chernobyl and three mile island, the fact that states it is self-there is a luck in the investigations and researchers and experimentally experience in two phase flow system which is very complex to describe in manner approximately may be recommended as real systems. So the demands of more mathematical comprehensive modeling equations for all phenomena appears in the pressurizer of pressurized nuclear reactor power. In this research a thermodynamic mathematical description of a pressurizer is based on energy equations that relate internal energy to external energy inputs.

All the models and researchers dealing with determine the pressurizer thermodynamic states of the vapor phase and predict the pressure changes based on assumption some variables and fixed another. The analysis is done in two parts: in-surge and out-surge. In the previous researches from earlier 1940 s, [3, and 4] the first central station nuclear power plant in the United States will be the Pressurized Water Reactor (PWR), nearing completion at Shippingport，Pennsylvania. Produce design data for steam pressurizing systems, by electronic analog simulation of the thermodynamic transients which occur in the pressurizer vessel.

The response of the primary coolant pressurizing system to rapid changes in Steam plant load is studied by means of analog computer techniques. Results are plotted for a range of values of the pressurizer design parameters. In the old researchers view of the scanty experimental information published, which has led to numerous assumptions in pressurizer model descriptions, [5] described The various steady state and transient phenomena, occurring in a (Nuclear) Pressurized Water Reactor pressurizer, three region I =steam region II=saturated water region and III=subcooled water region. Most of the information presented was obtained by experiments performed with a $1m^3$ pressurizer, working at a steady state pressure of 125 atmosphere. The desired perturbations of the steady state condition were effected by a reactor simulating circuit. [6] Utilized the theoretical models and computational techniques, for predicting pressure transients during prescribed rates of increase or decrease of tank level.

The analytical approach is a rigorous one and is somewhat unique in that no empirical system constants are required. Included is a documented digital computer program which is based on the theoretical models and can be used to predict pressure transients for any steam surge tank containing light or heavy water. Experimental data obtained from numerous tests on an actual heavy water reactor surge tank, for both in-surge and out-surge, is included. It is found that the isentropic models for compression and expansion of the vapor are highly in appropriate, and very good agreement between experiment and predictions is obtained with the above program. While the analysis and development of computational methods is conducted primarily to solve liquid in-surge and out-surge problems associated with nuclear power plant surge tanks, it is obvious that the results apply to any vessel where compression or expansion of a vapor is brought about by changes in level of the liquid phase. After many theoretical models and analog simulation of pressurizer behavior, the first digital simulation model was developed by [7] makes no prior assumption concerning the thermodynamic processes used in pressurizers.

Then in 1973 [8] presented a digital model of two region control volume, for the application of analyzing the primary pressure transient of nuclear power plant and a comparison with experimental data from three sources indicates good overall agreement. The model can be used as a design tool for pressurizer vessels with pressure control devices. [9] Presented predicting pressure' transients of a pressurizer during prescribed rates of level changes. Included is a documented S360/CSMP program which is based upon the theoretical models described and can be used to predict pressure transients of any steam surge tank. Another investigation in [10] derived Mathematical equations from basic principles describing the dynamic behavior of pressure, water mass, etc. in a steam drum. The resultant model includes such effects as steam superheating and water sub-cooling as well as spontaneous flashing of liquid and condensation of vapour.

Experimental data from a pressurizer are adequately predicted by the model. The analysis described in this paper was carried out during the development of dynamic simulations of advanced CANDU-BLW nuclear generating stations. The pressure rise following a turbine trip can be predicted by the isentropic-compression model but not by the thermodynamic-equilibrium model. M 'Pherson was probably the first to derive very detailed equations for both phases in a steam drum. He included the possibility of - steam superheat - spontaneous flashing - spontaneous condensation - heat transfer at the liquid surface - heat transfer between each phase and the vessel walls. His general model is highly complex, requiring some inputs based on engineering judgment, hence hi suggested that the steam be considered permanently saturated, to reduce the complexity of the model. In [11] Not only derived and programmed a detailed set of equations describing the dynamics of a pressurizer, but also conducted a series of precise measurements cm a large-scale test apparatus. He was able to accurately predict multiple in-surges and out-surges by assuming - isentropic compression of the steam during an in-surge - negligible heat transfer between the steam and vessel walls - negligible heat transfer across the liquid surface - heat conduction from the liquid to the vessel. In [12] investigated a series of experiments which provides a fundamental understanding of the phenomena which are important to the analysis of a PWR pressurizer has been performed. The scope of the analysis accompanying these experiments includes in-surges to a partially-full tank, out-surges, in-surges to a tank with hot walls, empty tank in-surges, a combined in-surges and out-surges, the effect of non-condensable gases, and free surface heat transfer.

In [13] investigated the Pressurizer transients modeling for predicting the dynamic pressure of the primary coolant system. In [14] due to lack of understanding on non-thermodynamic equilibrium and other local two-phase phenomenon in the pressurizer, worked to prepare necessary tools for a systematic study of the pressurizer and to investigate pressurizer phenomena under quasi-steady-states and to determine the effects of major pressurizer control parameters to the behavior of the pressurizer. A rate form of equation of state is analytically derived. It is used as an analytical expression of pressurizer pressure response as well as to support and to guide the rest of the work. IDRIFF, a two-phase simulation code consisting of a lumped homogeneous model and a differentially formulated drift-flux model, was developed to accommodate any physical assumptions in pressurizer modelling.

Some general-purpose system codes, such as RETRAN, RELAP4, SOPHT and FIREBIRD. On some occasions, the 'home' model of the system codes such as RELAP5 and SOPHT are directly used to simulate pressurizer transient .in [15] gives a thorough study of a case of normal transient phenomena resulting from a failure in the pressurizer controlling elements in a PWR system. Also study heavy transient phenomena resulting from pressurizer vessel rupture. The analysis of these cases have given the mode of pressure variation in the system. A new expression has been derived for the spray and condensate enthalpy in case of failure of the spray and relief valves.in 1981 [15] analyzing the transient behavior of the pressurizer of PWR.

In [16] Presented The basic mathematical model is derived from mass and energy conservation equations and includes all the important thermal-hydraulic processes which can occur inside the pressurizer. These processes are: spray condensation, bulk and surface condensation and evaporation, condensate fall and heat transfer from heaters. Furthermore, the model takes into account heat exchange processes between vapor and liquid regions and thermal dissipations between the entire pressurizer and the external ambient. To obtain the best achievable performances, the model is declined into three variants: complete lumped parameter and quasi 1D representations. Each different version has been developed as a natural evolution of the previous one, therefore, in the following pages, zero-dimensional, two-volumes and three-volume pressurizer models are presented in order of increasing complexity and accuracy.

Finally, the three models are compared with experimental data coming from Shippingport pressurizer tests and with the RELAP5® simulations. In [17, 23 and 24] Analyses the transient behavior of the surge tank in a WWER type pressurized water nuclear power plant. An analytical method is developed for predicting pressure and water level variations in the surge tank following in or out surge processes. The surge tank volume is divided into three regions according to phase condition and energy. Region 1 is the vapor and non-condensable gas (if any) region containing dropping liquid droplet equipped with spray nozzles and relief and safety valves. Region 2 is the saturated liquid region containing rising bubbles. Region 3 appears when the surge water enters the surge tank. This region is equipped with electric heating elements. Region 2 and 3 are the solution phase with boron as a solute. The analysis also takes into consideration the variation of the thermodynamic mass qualities inside the surge tank. The results are compared with RELAP5/Mod3 and RELAP5/Mod2 as valid codes. D'Auria's model (FFT method) is employed for this comparative analysis.

In [18] Introduced two independent volume steam-gas pressurizer model to predict the accurate system pressure for REX10. Liquid and gas mixture, separated with an interface. The model includes bulk flashing, rainout, inter-region heat and mass transfer and wall condensation with non-condensable gas. However, the ideal gas law is not applied because of significant interaction at high pressure between steam and non-condensable gas. The results obtained from this proposed model agree with those from pressurizer tests. In [19 ,29,21] develop a non-equilibrium models based on the two-region or three-region concept were developed t pressurizer model, and to assess its pressure transients using the TRACE code version 5.0. The benchmark of the pressurizer model was performed by comparing the simulation results with those from the tests at the Maanshan, MIT, PACTEL as well as a full-scale pressurize, The SPACE code input for MIT pressurizer experiment is developed and simulations are performed.

In [20] presents the modeling of two-region model nonlinear state-space takes the basic thermo-hydraulic processes into consideration in order to obtain a simple model structure. Real transient measurement data from the plant has been used for the identification that is based on standard prediction error minimization. And identification procedure for a pressurizer of a VVER-440/213-type pressurized water reactor. In [22] obtaining a dynamic two-phase imbalance mechanism model. Considering the influence of the spray flow, heater and safety valve, a pressurizer system was also established upon the Star-90 simulation platform by means of module packaging. The two step disturbances of heater power and spray flow rate were used to test model's dynamic characteristic while the pressurizer was at full running. The simulation results were compared with the original data obtained by a foreign PWR nuclear power plant simulator which show that the mechanism model of pressurizer has good dynamic performance and high accuracy.

In [25] a dynamic two-phase imbalance mechanism model of pressurizer in pressurized water reactor (PWR) nuclear power plant was built based on some reasonable simplifications and basic assumptions. The equations of energy and mass conservation are used in obtaining a mathematical model for pressurizer operation. The pressurizer is divided into two regions, steam region and liquid region but not necessary in equilibrium with each other. Considering the influence of the spray flow, surge flow, safety valve and heater, the model of pressurizer pressure control system is established by MATLAB/Simulink. In [26] a double-district equilibrium model of a nuclear power plant pressurizer dynamic characteristic was established through the approach of theoretical modelling simplifying. Moreover, the proposed model was examined in spray water disturbances and heating disturbances experiments by on-site simulator operating data. Compared the results of experiments at the proposed model with that of the simulator, two results agreed well with each other.in the recent the using of fuzzy system ,state space , artificial neural networks (ANNs); and genetic algorithms play an important role to model the pressurizer such as in [27,28].

III. MATHEMATICAL MODEL OF THE PRESSURIZER

Many researches and thesis discussing the phenomena existed in the pressurizer from early of 1960 till now with different models, thermodynamically, mechanically, analytically and experimentally from the old hand calculation through the development computer calculations and by artificial intelligence methods and finally by using the package cods like TRACE CODE etc. Also the researchers divided the pressurizer system as closed system by:

1- two region system
2- three region system
3- four region system

The transient process are

a. In-surge
b. Out-surge
c. Combination of in-surge &out-surge.

The task of the pressurizer is to keep the pressure within a predefined range. The pressurizer is a vertical tank and inside this tank there is hot water at a temperature of about 325◦C and steam above. If the primary circuit pressure decreases, electric heaters switch on automatically in the pressurizer. Due to the heating more steam will evaporate and this leads to a pressure increase. If the increasing pressure in the pressurizer reaches a certain limit, firstly the heaters are turned off and then cold water is injected into the tank (if needed) to reduce the pressure down to the predefined range .[2].In fact among analytical approaches to the dynamic analysis of a pressurizer, the two following methods are worthy of particular consideration:

- Equilibrium thermodynamic model, the two phase are saturated
- Non-equilibrium thermodynamic model. Liquid and steam in the pressurizer separately, considering a distinct temperature for each phase.[16].

In summary, the most important studies presented in the literature are based on these assumptions:

- The space inside the pressurizer is divided into two or three independent control volumes, steam and water, separated with a liquid interface. During steady state the steam and water phases are in thermal equilibrium, Saturated

- Conservation equations of energy and mass are applied to each phase
- The processes of mass transfer taking place between steam and liquid phases inside a pressurizer are due to the rate of steam condensation and the rate of bubbles rise. Variation in the average primary coolant temperature in a pressurized water reactor systems leads to a direct variation in the water volume and hence the pressure inside the pressurizer.
- Spray plus condensate mixture enters the water phase as saturated liquid
- The enthalpy of the sprayed water inside the pressurizer is the same as that of the reactor coolant cold leg
- Pressurizer is adiabatic
- The processes of steam condensation on vessel wall and water surface are neglected or not compared to other mass transfer terms
- Delay times of bubbles rise and condensate fall are neglected
- In-surge water mixes completely with water already present in the pressurizer.
- Steam discharged through the relief valve is taken zero.[16]
- The heaters must always be entirely liquid-covered, which for a fixed pressurizer geometry prescribes the minimum required liquid volume, approximately the water must be 60% of the pressurizer. [30].

All thermal hydraulic mechanisms that may control the phenomena in the pressurizer are:

1. Energy Transfer:

- Heat transfer across the steam-liquid interface;
- Interphasial heat transfer between bubbles and liquid;
- Interphsial heat transfer between droplets and steam;
- Heat transfer between the steam and the wall;
- Heat transfer between the liquid and the wall;
- Axial Heat Transfer through the Wall.
- Energy carried out of the pressurizer by the steam-bleed flow;
- Energy carried into and out of the pressurizer by the surge flow;
- Conduction heat transfer within fluids due to temperature profiles;
- Heat generation from the heaters.

2. Mass Transfer:

• Condensation a t liquid-steam interface;
• Evaporation at liquid-steam interface;
• Indirect contact condensation of steam (due to thermodynamic state change or due to spray);
• Direct contact condensation of steam on pressurizer wall;
• Nucleation process in the liquid;
• Boiling of liquid as it is heated by the heaters;
• Local boiling of liquid on pressurizer wall;
• Steam-bleed flow;
• Inflow and outflow of coolant due to in-surge and out-surge.

3. Momentum Transfer:

- motion of liquid due to in-surge and out-surge;
- Bubble rise;
- However when the flashing scares, the heat loses to the cold wall suppresses the flashing. This will be called " Suppression of Flashing"
- Condensate droplet drop;
- Local motion of the liquid due to the motion of bubbles;
- **L**ocal motion of the steam due to the motion of droplets. [14].

IV. Pressurizer Two-Phase Thermodynamic Non-equilibrium Model

The pressurizer function during power transients can be understood as follows: In the case of a decrease in turbine load the average temperature of the primary circuit coolant will increase, resulting in liquid expansion. Liquid from the primary circuit hot leg (figure 1.) is permitted to surge into the bottom of the pressurizer (the in-surge), raising the circuit pressure while compressing the vapor phase in the upper pressurizer dome .To prevent an excessive pressure rise, subcooled water from the loop cold leg is sprayed into the pressurizer steam phase, condensing part of the vapor and thus providing space for the incoming surge at a controllable pressure.

The upper pressure limit is set by structural considerations. Conversely a gain of turbine load reduces the average coolant temperature, causing a surge of liquid from the pressurizer (the out-surge) into the primary circuit hot leg. The resulting pressure drop is counteracted by the production of vapour as a result of flashing of saturated pressurizer liquid and of the switching on of additional heater elements. The lower pressure limit is set by the requirement to restrict boiling in the reactor core. If, however, the out-surge volume rate is high or if a rapid sequence of several in/out-surge cycles takes place, the effect of the heaters will be small compared with the latent heat released from the saturated liquid phase by flashing.

The adequate functioning of a pressurizer under the severest load transients is thus seen to depend heavily on the amount of flashable liquid available during an out-surge, i.e. during an increase of power demand from the reactor core. The required steam and saturated liquid volumes cause the pressurizer to be a relatively large component, typically in the order of one third of the reactor vessel by volume, containing a significant amount of stored energy. Coupled to the lack of available information pertinent to P.W.R. pressurizer operating conditions (typically: 125-150 at., 325-340 °C). [5].Generally the two phases may be saturated, the steam saturated and the liquid is subcooled, the steam superheated and the liquid saturated or subcooled.

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A. Model Description

The pressurizer system operation can be represented in flowchart form is shown in Figure 4.This figure shows the flowchart of operation of pressurizer. Initially, the pressurizer is in normal operation. When the pressure inside the pressurizer decreases, the hot water from the hot leg enters the bottom of pressurizer via surge line valve. At the same time, the heater at the bottom of pressurizer is activated to heat the water. When the pressure inside the pressurizer starts to increase, the surge line valve is closed and heater is deactivated. Then, the coolant water from the cold leg enters the pressurizer via the spray valve that located at the top of pressurizer. At the same time, the relief valve is opened to release the steam to the atmosphere. When the pressurizer is stabilized where the pressure is maintained at the set point level, the pressurizer operates at normal condition. If not, it will loop back as shown in Figure 2.

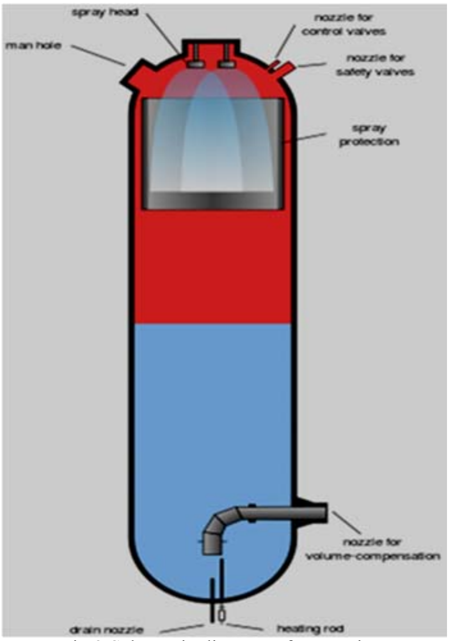

Fig.2.Schematic diagram of pressurizer

[25] In the mathematical model of the pressurizer, two phases are taken into consideration----the steam phase (including the droplets), the water phase. All three phases are individually calculated by mass and energy conservation equations, and thermal non-equilibrium is possible between any two phases. Interface flow mainly includes the evaporation flow of water, the steam condensate flow, the spray condensing flow and the wall surface condensation flow. According to the pool boiling theory, the bubbles will gradually narrow down to disappear in sub cooled boiling, so the bubble production flow should not be counted in. While water enthalpy is more than saturated one corresponding to the saturated pressure, the water will produce bubbles and the amount of bubbles will rapidly increase correspondingly. The phase diagram of pressure system in pressurizer mechanism model is shown in Fig. 3.

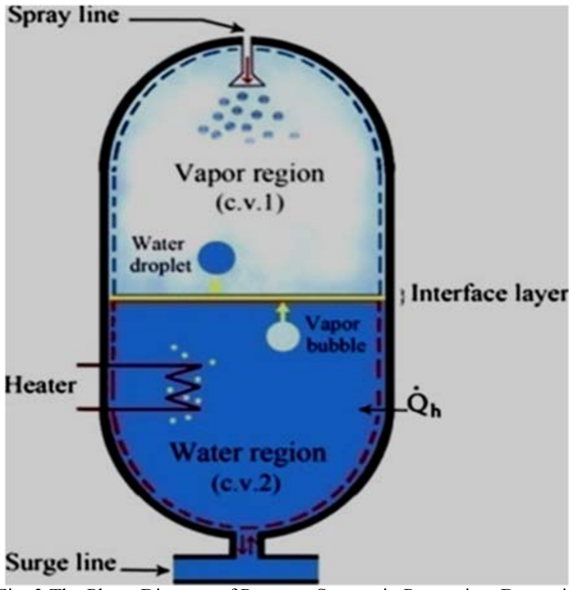

Fig. 3 The Phase Diagram of Pressure System in Pressurizer Dynamic Model

B. Analytical Analysis and Assumption

The pressurizer volume can be divided into two areas: the Steam zone and the liquid area. Under the following assumptions, a two-phase dynamic non-equilibrium stabilizer model is set up.

1) It is light water pressurizer.
2) The pressurizer is good isolated system. Adiabatic system.
3) The two areas have the same pressure at the same time; pressure is uniform throughout the pressurizer all the time;
4) Neither phase can be metastable .This means that the vapor can he either saturated or superheated but not supersaturated) while the liquid can be either saturated or subcooled (but not superheated). This assumption implies that flashing and condensation occurs spontaneously within the bulk of the liquid and vapor phases, respectively.
5) Uniform properties for each phase.
6) Rainout to maintain saturated vapor to prevent sub-cooling.
7) Flashing to maintain saturated liquid to prevent superheating.
8) The mass exchanges in the interface between the two areas are finished in an instant;

9) The steam state is either saturation or overheating, and the liquid phase could be either saturation or overcooled;
10) When the two-phase separates completely, the thermodynamic parameters of each phase maintain the consistent in the meantime;
11) The spray water is saturated before leaving the steam zone, that means, the spay efficiency is 100%;
12) The surge enthalpy is constant and the surge flow is given.
13) The non-condensable gas should be negligible;
14) The spray water reaches saturation temperature before leaving the steam phase and going into the liquid phase;
15) Spray plus condensing mixture enters the water phase as saturated liquid. The spray condensation processes at once;
16) The spray rate α pressure difference.
17) The internal transfer process from the top region to the lower region are then condensate fall, condensate on spray droplets, condensate on water surface and condensate on wall. The first process only occurs when the upper region is two phase, the transfer from the lower region to the upper region is bubble rise occurs when the lower region is two phase.
18) The mass and energy transfer from wall condensate and surface evaporation is assumed to contribute very little and may be neglected.
19) Condensate droplet velocity is taken as nominal.
20) The heater will lag by a fixed time constant when given the signal to turn on or off.
21) The thermal inertia of heater is negligible.
22) The in-surge flow of coolant water mixes perfectly with the hot water in the bottom of pressurizer.
23) The liquid phase is either saturation or subcooled, and the steam phase could be either saturation or superheated.
24) The heat transfer between pressurizer contents and pressurizer wall is proximately very small in comparison with other terms in the energy equation.
25) Mass interface at the liquid –vapor interface is a result of bubble rise or condensation droplets.
26) The spray enthalpy is constant and the spray mass at time (t+dt) will be part of the fluid phase.
27) The enthalpy of the sprayed water inside the pressurizer is the same as that of the reactor coolant cold leg.
28) Delay time of bubble rise and condensate fall are neglected.
29) Steam discharged through the relief valve
30) Wall condensation and boiling.
31) Interface heat and mass transfer.
32) Heat transfer rate between the liquid and the wall.
33) Assuming that the flashing only occur at saturation conditions and that the latent heat of evaporation is supplied by the liquid region.
34) Similarly ,under the assumption that rainout occurs only at saturation conditions and the latent heat is released only to the steam region
35) The first assumption is that heat transfer between ti\e fluids and the vessel materials and between the two phases (by conduction) is negligible, i.e. $Q_w = Q_s = 0$.

C. Thermodynamic Model

The first law of thermodynamics is a version of the law of conservation of energy, adapted for:

$$\Delta U = Q - W \;\&\; H = U + pV \;\&\; dH = dU + pdV + Vdp \quad -1$$

Where ΔU = internal energy & Q = amount of heat supplied to the system& W= work done by the system on its surroundings. H =enthalpy, p=pressure & V=volume. The total pressurizer volume remains constant. Since the volume of the pressurizer is fixed , $\mathbf{pdV} = 0 \quad -2$
The mass of each phase continually changes with time .during the time interval, dt, some or all of the following changes may occur:-
M_{SU}dt=mass of water surge line from hot leg.
M_{SUI}dt=mass of water entering surge line and mixing with the liquid phase from hot leg.
M_{SUO}dt=mass of water leaving liquid phase surge line to reactor coolant hot leg.
M_{SP} dt=mass of spray injected into the pressurizer from cold leg.
M_{CS} dt=mass of steam condensing on spray droplets.
M_{RO} dt=mass of condensate falling to liquid from steam, or rainout
M_{CW} dt=mass of condensate on the pressurizer wall.
M_{re} dt=mass of steam leaving through the relief valve.
M_{sv} dt=mass of steam leaving through the safety valve.
M_{FL} dt=mass of water leaving from liquid entering steam from bubble rise.

Where m=dm/dt, the rate change of mass.

In this paper, in the model, mentioned to pure phase by capital letter and to saturated phase by small litter. Figure 4 shows the schematic of a two-fluid pressurizer model. The external boundary conditions are spray mass flow rate and enthalpy, relief and safety valve discharge mass flow rates, surge mass flow rate and enthalpy, and heater power.

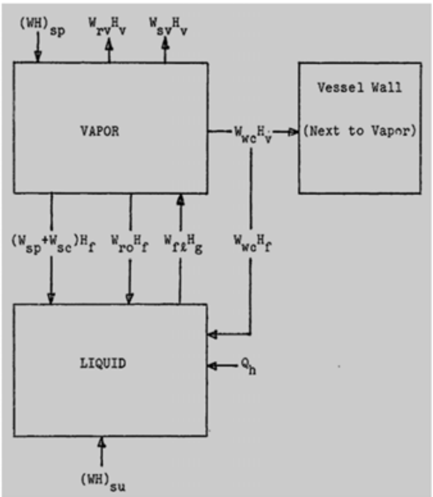

Fig. 4 Schematic of a two-fluid pressurizer model.

The interfacial boundary conditions are mass flow rates for condensation on spray and wall condensation, rainout, and flashing ,some of these boundary conditions are specified, such as spray and heater power. The rest of them are unknowns which require additional constitutive relations in order to solve the conservation equations in a closed form. To model these constitutive relations adequately is one of the major challenges in pressurizer modeling. Figure 5 gives an illustration of spontaneous flashing and condensation.

If a unit mass of liquid, at pressure P and saturation conditions 2 is depressurized by an amount ΔP, some of the liquid flashes into saturated steam.

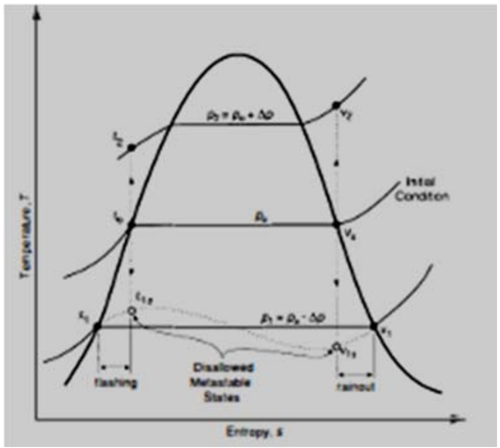

Fig. 5. Pressurizer Pressure Transients from Saturated Initial Conditions. [10]

The horizontal intercept on the temperature-entropy diagram is a measure of the an amount of flashing. Similarly, *a* unit mass of vapor, at pressure ΔP and saturation conditions 3, will partially condense when depressurized by an amount ΔP. Again, the amount of condensation is given by the horizontal intercept. Figure 5 also shows that, initially subcooled liquid at 1 or superheated vapor at 4, when depressurized, must first reach the saturation line before flashing or condensing into the other phase. Finally, a pressure increase from P to P+AP suppresses flashing and condensation, regardless of the initial state of the liquid (1 or 2) or vapor (3 or 4).

D. Abbreviations and Acronyms: See table in Appendices.

E. Flashing and Condensation

TABLE I. FLASHING AND CONDENSATION

The system has the same five prescribed input parameters: mspray , msurge, h spray, hsurge , and Qh							
Cases	Pressure	Condition of		Condensation	Flashing	Constraints	Unknown
		Vapor	Liquid				
1	Rising	Saturated	Saturated	No	No	$h_l,=h_f(p)$, $u_l=u_f(p)$ $h_V=h_g(p)$, $u_v=u_g(p)$ $W_{RO}=0$,$W_{FL}=0$	P, α_P,
2	Rising	Saturated	Subcooled	No	No	$h_V=h_g(p)$, $u_v=u_g(p)$ $W_{RO}=0$,$W_{FL}=0$	P, h_l, α_P
3	Rising	Superheated	Saturated	No	No	$h_l,=h_f(p)$, $u_l=u_f(p)$ $W_{RO}=0$,$W_{FL}=0$	P , h_V ,α_P
4	Rising	Superheated	Subcooled	No	No	$W_{RO}=0$,$W_{FL}=0$	P, h_l, h_V, ,α_P,
5	Falling	Saturated	Saturated	Yes	Yes	$h_l,=h_f(p)$, $u_l=u_f(p)$ $h_V=h_g(p)$, $u_v=u_g(p)$ $W_{RO}\neq0$,$W_{FL}\neq0$	P, α_P, ,W_{FL}, ,$W_{RO,}$
6	Falling	Saturated	Subcooled	Yes	No	$h_V=h_g(p)$, $u_v=u_g(p)$ $W_{RO}\neq0$,$W_{FL}=0$,	P, , h_l,α_P, W_{RO}
7	Falling	Superheated	Saturated	No	Yes	$h_l,=h_f(p)$, $u_l=u_f(p)$ $W_{RO}=0$, $W_{FL}\neq 0$	P , h_V, α_P, W_{FL}
8	Falling	Superheated	Subcooled	No	No	$W_{RO}=0$,$W_{FL}=0$	P, h_l, h_V, ,α_P,

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- Within the liquid region , mass transfer and energy transfer occur Owing to bulk flashing across the interface
- In general, the vapor bubbles are created by flashing of lower region liquid at enthalpy the to produce a vapor mass per unit time at enthalpy h_G•
- Neglecting work terms.
- Within the vapor region, mass and energy transfer occur at three condensation locations, bulk condensation within the vapor region called rainout (W_{RO})' on the vessel walls, (W_{CW}), and on the spray droplets (W_{CS}).
- ***First, consider the rainout drops***, In general, these drops are created by condensation of upper region vapor at enthalpy h_G, which produces a liquid mass per unit time at enthalpy h_F. The latent heat of condensation is released at the interface due to vapor and liquid heat transfer, Assuming that rainout occurs only at saturation conditions and the latent heat of condensation is released only to the vapor region:
- ***Next, consider condensation on the spray drops***, which creates the interface with the vapor region, assuming that condensation occurs only at saturation conditions and that the latent heat of condensation is released totally to the spray by conduction through the condensate shell. Finally , if it is assumed that the rate of condensation is just sufficient to raise the spray enthalpy to saturation:
- Finally, consider condensation on the pressurizer wall which creates the interface with the vapor region. Assuming that condensation occurs only at saturation conditions and that the latent heat of condensation is released to the condensate. Furthermore, it is assumed that the latent heat of condensation that is released to the condensate is transferred by conduction completely to the pressurizer wall. It is s important to note that the heat transfer to the wall is treated as heat transfer from liquid (condensate) to the wall even though physically this condensation process is occurring on the pressurizer wall in the upper vapor region, the heat transfer to the wall from the bulk vapor or liquid in the pressurizer is assumed negligible. [30]
- A rising pressure results in increased sub-cooling of the liquid and superheating of the vapor, so that a Saturated-saturated condition can occur only as a starting point of a transient. [10] The conservation of energy is given by the following expression, neglecting changes in potential and kinetic energy figure 6. And Table 2 verify the state of the pressurizer which can occur.

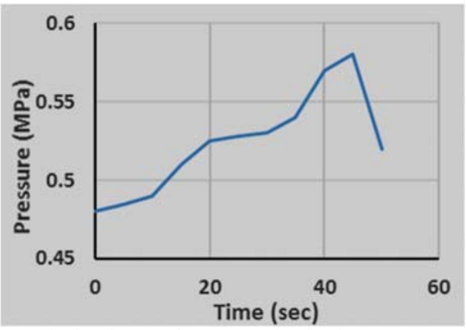

Fig. 6. Calculated Rate of Pressure Rise for the MIT Pressurizer

Refer to appendix A-2: Now rewrite the six equations of the pressurizer transients to in-surge actions, the general form of response to in-surge action is: - these equations are 29, 37,38,39,41 and 42 which are listed below.

Transport process in the pressurizer

-1. Surge flow: The essential motivation process in the pressurizer transients is the surge flow of water into or out the pressurizer which forces the pressurizer to be under disturbance, as said that the in-surge water has a constant enthalpy and at once flow into the bottom part of the pressurizer will be mixed completely with the water present there. The enthalpy of surge water will be the same as the hot leg reactor enthalpy at in-surge action and the same as the pressurizer water enthalpy at out-surge from pressurizer to the reactor vessel. $M_{SU}h_{SU}$=constant at specific time.

-2. Spray condensation: The spray condensation action is a complex operation in a real pressurizer. The process of spray condensation in a pressurizer is a complicated one. In particular both the surface area and the surface temperature of a falling spray droplets change radically as the droplet falls through the pressurizer vapor volume. On this basis, the condensation rate of steam is calculated by applying the energy equation to the open thermodynamic system consisting of the interacting steam and the sprayed subcooled water. From the first law of thermodynamics for an open system is applied to the upper region to get:

$$m_{cs} = m_{sp}\left(\frac{h_f - h_{sp}}{h_G - h_f}\right) = m_{sp}e\,, e = \left(\frac{h_f - h_{sp}}{h_G - h_f}\right) \tag{43}$$

-1. Relief valve & safety valve flow rate:

$$(m_{rv} + m_{sv}) \tag{44}$$

in this paper neglected. If the pressure increase and outstrips the safety valve set point, the relief valve flow rate is considered negative and here will be neglected.

-2. Heater thermal power: The heart works automatically when the system pressure decrease .the value of Qh initially is zero.

-3. Bubble rise (flashing) and rainout (condensation fall rate). it was assumed that the bulk heat transfer comes from bubble rise of boiling (flashing) and condensate fall (rainout) the mass leaving the lower region from boiling M_{FL} can be determined from:

$$\dot{M}_{FL} = v_b A\, \alpha\, \rho g \tag{45}$$

Then the amount of condensate that falls M_{RO} from upper region to the lower region can be calculated from the following relation:

$$\dot{M}_{RO} = v_f A\,(1 - \alpha\,)\rho \tag{46}$$

-4. Wall condensation, WC (steam condensation on the wall surface): If the wall temperature is less than steam saturation temperature, an additional condensation flow related to the wall heat transfer is computed. The flow rate $\frac{dM_{cw}}{dt}$ results from the heat balance equation: [22]. The mass flow rate due to vapor film condensation on the vessel wall where the wall temperature is lower than the steam temperature is calculated from the Nusselt model [34].

$$W_{WC} = \frac{\dot{Q}_{WC}}{h_{fg}} = \frac{\beta_{stam} A(T_v - T_{in,w})}{h_{fg}} = 0.943\left(\frac{g\rho_l(\rho_l-\rho_v)k_l^3}{h_{fg}^3 L\mu_l}A^4(T_v - T_{in,w})^3\right)^{0.25} \quad (47)$$

-5. The pressurizer wall is assumed to be perfectly insulated with negligible heat capacity. [30]

Therefore rewriting the equations 29, 37,38,39,41 and 42 by changing:

$$\frac{dp}{dt} = -[a1m_F + a2m_G + a3M_{SU} + a4M_{SP} + a5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + a6Q_h - a7m_{rvsv}] \quad (48)$$

$$\frac{dz}{dt} = [b1m_F + b2(0) + b3M_{SU} + b4M_{SP} + b5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + Q_h b6 + (0) + b8\frac{dP}{dt}] \quad (49)$$

$$\frac{dM_G}{dt} = -m_{RO} - m_{CW} - m_{CS} + m_{FL} - m_{rvsv} \quad (50)$$

$$\frac{dM_F}{dt} = M_{SU} + M_{SP} + m_{RO} + m_{CW} + m_{CS} - m_{FL} \quad (51)$$

$$\frac{dh_G}{dt} = c1(-m_{RO} - m_{CW} - m_{CS} - m_{FL} - m_G) + c2\frac{dp}{dt} \quad (52)$$

$$\frac{dh_F}{dt} = d1M_{SU} + d2\, M_{SP} + d3((m_{RO} + m_{CW} + m_{CS} - m_{FL}) - m_F) + d4\frac{dP}{dt} + d5Q_h \quad (53)$$

Refer to equations 48, 49,50,51,52 and 53, rewriting them:

$$\frac{dp}{dt} + a1m_F + a2m_G + (0)\frac{dz}{dt} = -[a3M_{SU} + a4M_{SP} + a5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + a6Q_h - a7m_{rvsv}]$$
(54)

$$-b8\frac{dP}{dt} - b1m_F + \frac{dz}{dt} = [b2(0) + M_{SU}b3 + M_{SP}b4 + b5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + b6Q_h$$
(55)

$$\frac{dh_G}{dt} = c1(-m_{RO} - m_{CW} - m_{CS} + m_{FL} - m_G) + c2\frac{dp}{dt} \quad (56)$$

$$\frac{dh_F}{dt} = d1M_{SU} + d2M_{SP} + d3(m_{RO} + m_{CW} + m_{CS} - m_{FL} - m_F) + d4\frac{dP}{dt} + d5Q_h \quad (57)$$

$$\frac{dM_G}{dt} = [-m_{rvsv} - m_{RO} - m_{CW} - m_{CS} + m_{FL}] \quad (58)$$

$$\frac{dM_F}{dt} = [M_{SU} + M_{SP} + m_{RO} + m_{CW} + m_{CS} - m_{FL}] \quad (59)$$

F. State Space Representation of the Modeling of the Pressurizer In-surge Transient

Refer to equations 54 through 59 writing them in the state space form of:

$$H\dot{Y} = AY + B$$

Define

$$Y = [P, Z, M_F, M_G, H_F, H_G] \quad (48) \qquad \dot{Y} = [\dot{P}, \dot{Z}, \dot{M}_F, \dot{M}_G, \dot{H}_F, \dot{H}_G] \quad (49)$$

$[a3M_{SU} + a4M_{SP} + a5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + a6Q_h - a7m_{rvsv}] = B1$ = B1

$[b2(0) + M_{SU}b3 + M_{SP}b4 + b5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + b6Q_h[b2(0) + M_{SU}b3 + M_{SP}b4 + b5(m_{RO} + m_{CW} + m_{CS} - m_{FL}) + b6Q_h]$ = B2

$c1(-m_{RO} - m_{CW} - m_{CS} + m_{FL} - m_G) + c2\frac{dp}{dt}$ = B3

$d1M_{SU} + d2M_{SP} + d3(m_{RO} + m_{CW} + m_{CS} - m_{FL} - m_F) + d4\frac{dP}{dt} + d5Q_h\ [-m_{rvsv} - m_{RO} - m_{CW} - m_{CS} + m_{FL}] = B4 =$ B4

$[M_{SU} + M_{SP} + m_{RO} + m_{CW} + m_{CS} - m_{FL}] = B3 =$ B5

Then equations 54 - 59 will reduce to:

$$\dot{P} + a1\,\dot{M}_F + a2\dot{M}_G = -B1 \quad (60) \qquad -b\dot{8}P - a1\,\dot{M}_F + \dot{Z} = B2 \quad (61) \qquad -c2\dot{P} + \dot{H}_G + c1\dot{M}_G = -B3 \quad (62)$$

$$-d3\dot{P} + \dot{H}_F + d2\dot{M}_F = B4 \quad (63) \qquad \dot{M}_G = -B5 \quad (64) \qquad \dot{M}_F = -B6 \quad (65)$$

Then system of eqs.60-65, may be written in the compact form where $H\dot{Y} = B$, matrices H (6×6), and B (6×6), as follow:

$$H = \begin{bmatrix} 1 & 0 & 0 & 0 & a2 & a1 \\ -b8 & 1 & 0 & 0 & 0 & -b1 \\ -c2 & 0 & 1 & 0 & c1 & 0 \\ -d3 & 0 & 0 & 1 & 0 & d2 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \quad \dot{Y} = \begin{bmatrix} \dot{p} \\ \dot{z} \\ \dot{H}_G \\ \dot{H}_F \\ \dot{M}_G \\ \dot{M}_F \end{bmatrix} = \mathrm{B} \begin{bmatrix} B1 \\ B2 \\ B3 \\ B4 \\ B5 \\ B6 \end{bmatrix}$$

One can take several cases to cover different possible situation that arise during in-surge action as follows:

a) In-surge with heater and no spray
b) In-surge with no heater
c) In-surge with no spray and heater

V. RESULTS AND DISCUSSION

The equations 60-65 were solved by POLYMATH computational system based on the calculation method styled in the previous sector, the first test is the MIT pressurizer main tests. The experiment of in-surge action taken for MIT pressurizer from foreign pressurizer because there is no pressurizer in Iraq or any nuclear power plant in-surge test. During the in-surge condition, the cold water pushes the saturated water inside the tank like a piston and causes the steam to be compressed. However, condensation on the wall works against this process and decreases the pressure, fig. 6. Shows that the calculated rate of pressure rise is close to the measured value of the MIT pressurizer test from ref [32]. As the cold water was injected into the pressurizer, the pressure increased due to compression of the steam volume. As the pressure increased the saturation temperature also increased. Energy transferred from the vapor to the wall and condensation at the liquid/vapor interface mitigated the pressure rise. The calculated pressure rise was slightly under-predicted. An atypical decrease in the calculated pressure because in our model taking into account all condensation occur in transient in-surge and we don't taking into account the effect of non-condensing gas calculation at about 40 seconds was a result of rapid condensation as subcooled liquid droplets entered a saturated steam environment near the top of the pressurizer. It is supposed the difficult is related with the interfacial heat transfer model. In Fig. 7, the pressure behavior of the MIT pressurizer is shown for the test of MIT pressurizer tests have been conducted in low pressures and densities compare to other tests addressed in [29] study.

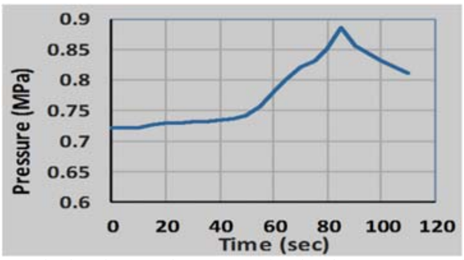

Fig. 7.Calculated Rate of Pressure Rise for the MIT Pressurizer.

Therefore, even a little condensation and evaporation cause a rapid pressure change. This aspect makes it necessary to model these processes accurately. In the in-surge condition, the pressure response is fairly complex to small fluctuations in the condensation procedure. Good covenant gotten in this condition shows the measured classic condensation model (the Nusselt model) is acceptable. After the in-surge stops, the heat loss to the environment from the tank causes the pressure to decrease (Fig. 7). Based on the MIT experiments, a total heat loss of 1.1 kW can be considered for all the tests [33]. Nevertheless, this amount of heat loss is not constant during the experiments. Since heat loss to environment is the most important mechanism after the in-surge stops, pressure is changed based on the measured value of it. If this heat loss

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is not taken into account, the pressure will not change after in-surge flow stops.

VI. CONCLUSION

In this research, two-region non-equilibrium model was realistic to define the pressurizer behavior of a nuclear power plant. Nevertheless, the system is very complex so it is obligatory to use a numerical technique. Pressurizer behavior can be best estimated by using balances laws in conjunction with accurate constitutive models and proper simplifications. As shown in this study, an appropriate numerical scheme led to predicting the pressure behavior of a pressurizer accurately. This model included all the important phenomena that occur in the pressurizer as well as proper simplification assumptions. All of these aspects enabled us to develop a model, which was capable of estimating pressurizer dynamic behavior in different scales and conditions. To show this feature, a preliminary validation and verification using well-known experimental tests facilities in different scales were applied. In selecting these cases it was tried to considered cases that covers extended ranges of transients either separated or combination of in-surge-out-surge flows. The result was compared with two foreign studies with their models of the in-surge transients and it is showing realistic approximately between the results can be considered as an important step toward the pressurizer component verification of the power plants. Although the most important phenomena occurred in a pressurizer are verified well, it is very significant to include these occurrences in an appropriate calculation model. When the analyzing of a pressurizer behavior, done it was established that since the physical phenomena play roles further important in a small low-pressure system, the developed models are applicable to large high-pressure systems too. In this article, two-region non-equilibrium model was applied to describe the pressurizer behavior of a nuclear power plant. However, the complexity of the system made it necessary to use a numerical technique. For future work one can derive the model of out surge transients and model it in state space representation. And take several cases to cover different possible situations that arise during in-surge actions as follows:

a) In-surge with heater and no spray
b) In-surge with no heater
c) In-surge with no spray and heater

See also reference [29].

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APPENDICES

Table of Abbreviations and Acronyms

D	Pressurizer diameter in m
Vt	Pressurizer total volume m^3
VF	Volume of liquid phase m^3
VG	Volume of steam phase m^3
MF	Mass of liquid phase in Kg
MG	Mass of steam phase in Kg
$\boldsymbol{h_F}$	Subcooled water enthalpy in KJ
$\boldsymbol{h_G}$	Superheated steam enthalpy in KJ
hf	Saturated water enthalpy in KJ
hg	Saturated steam enthalpy in KJ
hfg	Latent heat =hg-hf in KJ
hsp	Spray water enthalpy (cold leg) in KJ
hsu	Surge water enthalpy (hot leg) in KJ
V_F	Subcooled water specific volume in m^3/Kg
V_G	Superheated steam specific volume in m^3/Kg
V_f	Saturated water specific volume in m^3/Kg
V_g	Saturated steam specific volume in m^3/Kg
Z	Water level in the pressurizer in m
m_F	dm_F/dt Change of Mass rate of Water phase Kg/sec
m_G	dm_G/dt Change of Mass rate of steam phase Kg/sec
m_{sp}	Spray mass flow rate Kg/sec
m_{suri}	Surge in mass flow rate Kg/sec
m_{suro}	Surge out mass flow rate Kg/sec
$m_{spr(max)}$	Maximum spray mass rate Kg/sec
ΔU	Change in internal energy
Q	Heat added to the system
W	Work done by the system
P	Pressurizer pressure in Pascal
P1	Spray valve low set point pressure Pascal
P2	Spray valve high set point pressure Pascal
Ph	Heater set point KJ
Qh	Hater input power in KJ
$\boldsymbol{q_F}$	Net heater rate added to water phase in KJ
$\boldsymbol{q_G}$	Net heater rate added to steam phase in KJ
Qh	Heater output power in KJ
t	Time in sec
T	Temperature in degree
V_{fg}	V_g -V_f in m^3/Kg
X_F	Steam quality in water phase
X_G	Steam quality in steam phase
$\boldsymbol{\dot{U}_G}$	Steam Internal energy rate dU/dt in KJ/sec
U_G	Steam Internal energy in KJ
U_F	liquid internal energy in KJ
$\boldsymbol{\dot{U}_F}$	liquid Internal energy rate dU/dt in KJ/sec
$\boldsymbol{v_b}$	Velocity of the bubble rise
$\boldsymbol{v_f}$	Velocity of the falling droplets ,taken as nominal
$\boldsymbol{A}$	Cross- section area of the pressurizer
$\boldsymbol{\alpha}$	Void fraction from a condensation of quality and specific volume
$\boldsymbol{A\,\alpha}$	Area occupied by the steam
$\boldsymbol{\rho_g}$	Density expressed in the model in terms of specific volume
$\boldsymbol{(1-\alpha)}$	Amount of liquid
$\boldsymbol{A\,(1-\alpha)}$	Area occupied by the liquid
$\boldsymbol{\rho_f}$	Density expressed in the model in terms of specific volume
$\boldsymbol{K_P}$	is heat transfer coefficient,
$\boldsymbol{A_P}$	is exchange surface area
$\boldsymbol{T_{sat}}$	is saturation temperature
$\boldsymbol{m_{rvsv}}$	Steam flow rate from relief and safety valve

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Appendix A1: The General Mass and Energy Conservation Equation of Two –Fluid Pressurizer Model

Mass balance equation: The instantaneous rates of energy addition to the system must equal the rate of change of stored energy and the rate of doing expansion work. The summation of convected energy terms is over the number of streams of material entering or leaving the system. The summation of stored energy is over the phases of material identified within the system.

$$\sum Q^{\cdot} + \sum \dot{K}\, W + \sum h m^{\cdot} = \sum (M\dot{u}) \; + P\dot{V}$$

$$\Delta U = Q - W \,, \;\; H = U + pV$$

$$dH = \; dU \; + \; pdV \; + \; Vdp$$

$$U = Q - W = H - PV \;\; , dU = dH + \; pdV \; + \; Vdp$$

Whenever the pressure is uniform throughout an apparatus, the general energy balance may be more readily applied through application of the definition of enthalpy.

$$\sum Q^{\cdot} + \sum \dot{K}\, W + \sum h \; \dot{m} = \sum \dot{M}\, h - V\dot{P}$$

1) *Steam:*

$$M_G \;\; = -M_{re} - M_{sv} - \; M_{RO} - M_{CS} - M_{CW} + M_{FL} \qquad (1)$$

2) *Water:*

$$M_F = M_{SU} + M_{SP} + M_{CS} + M_{RO} + M_{CW} - M_{FL} \qquad (2)$$

3) Energy balance equation: *The equation of energy conservation for either the liquid or vapor volumes can be derived using tile three dimensional form of the fluid energy equation (kinetic and potential energy are neglected) as:*

Steam

$$h_G = \frac{U_G}{M_G} + P\frac{V_G}{M_G}$$

$$h_G M_G = U_G + PV_G$$

$$\frac{\partial (M_G h_G)}{dt} = q_G + V_G \frac{dP}{dt} \qquad (3)$$

$$q_G = (m_{FL} - \; m_{re} - m_{sv} - m_{CS} - m_{CW})h_G - m_{RO}h_F \ldots (3a)$$

Water

$$h_F = \frac{U_F}{M_F} + P\frac{V_F}{M_F}$$

$$M_F h_F = U_F + PV_F$$

$$\frac{\partial (M_F h_F)}{dt} = q_F + V_F \frac{dP}{dt} \qquad (4)$$

$$q_F = m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} - m_{FL})h_F + m_{RO}h_F + Q_h \ldots \ldots (4a)$$

Volume balance equation:

$$h_G = \frac{U_G}{M_G} + P\frac{V_G}{M_G} \qquad (5)$$

$h_F = \frac{U_F}{M_F} + P\frac{V_F}{M_F}$…… (5a)

Steam: $V_G = M_G \vartheta_G$ (6)

Water: $V_F = M_F \vartheta_F$ (7)

$Pressurizer$: $V_T = V_G + V_F$ (8)

Equation of state for the two region model with in-surge transient and the upper region is superheated and the lower region is subcooled:-

$\vartheta_G = \vartheta_G(p, h_G)$ (9)

$\vartheta_F = \vartheta_F(p, h_F)$ (10)

Differentiating: eqs (6, 7, 8, 9 and 10) with respect to time, we get:

$$\frac{dV_G}{dt} = m_G \vartheta_G + M_G \frac{d\vartheta_G}{dt} \quad (11)$$

$$\frac{dV_F}{dt} = m_F \vartheta_F + M_F \frac{d\vartheta_F}{dt} \quad (12)$$

$$0 = \frac{dV_G}{dt} + \frac{dV_F}{dt} \quad (13)$$

Taking the partial derivative of Eqs. (11) And (12), we get:

$$\frac{d\vartheta_G}{dt} = \left(\frac{d\vartheta_G}{dp}\right)_{h_G} \frac{dp}{dt} + \left(\frac{d\vartheta_G}{dh_G}\right)_P \frac{dh_G}{dt} \quad (14)$$

$$\frac{d\vartheta_F}{dt} = \left(\frac{d\vartheta_F}{dp}\right)_{h_F} \frac{dp}{dt} + \left(\frac{d\vartheta_F}{dh_F}\right)_P \frac{dh_F}{dt} \quad (15)$$

substituting eqs. 14 and 15 into eqs. 11&12, we get:

$$\frac{dV_G}{dt} = M_G \vartheta_G + M_G \left[\frac{d\vartheta_G}{dt} = \left(\frac{d\vartheta_G}{dp}\right)_{h_G} \frac{dp}{dt} + \left(\frac{d\vartheta_G}{dh_G}\right)_P \frac{dh_G}{dt}\right] \quad (16)$$

$$\frac{dV_F}{dt} = M_F \vartheta_F + M_F \left[\left(\frac{d\vartheta_F}{dp}\right)_{h_F} \frac{dp}{dt} + \left(\frac{d\vartheta_F}{dh_F}\right)_P \frac{dh_F}{dt}\right] \quad (17)$$

Substitute eqs. 16&17 into eq. 13, we get:-

$$M_G \vartheta_G + M_G \left[\frac{d\vartheta_G}{dt} = \left(\frac{d\vartheta_G}{dp}\right)_{h_G} \frac{dp}{dt} + \left(\frac{d\vartheta_G}{dh_G}\right)_P \frac{dh_G}{dt}\right] + M_F \vartheta_F + M_F \left[\left(\frac{d\vartheta_F}{dp}\right)_{h_F} \frac{dp}{dt} + \left(\frac{d\vartheta_F}{dh_F}\right)_P \frac{dh_F}{dt}\right] = 0$$

Rearranging:-

$$\frac{dp}{dt}\left[M_G \left(\frac{d\vartheta_G}{dp}\right)_{h_G} + M_F \left(\frac{d\vartheta_F}{dp}\right)_{h_F}\right] = -\left[m_G \vartheta_G + m \vartheta_F + M_G \left(\frac{d\vartheta_G}{dh_G}\right)_P \frac{dh_G}{dt} + M_F \left(\frac{d\vartheta_F}{dh_F}\right)_P \frac{dh_F}{dt}\right]. \quad (18)$$

Then differentiate eqs. 3 and 4 and, we get,

$$m_G h_G + M_G \frac{dh_G}{dt} = qG + V_G \frac{dp}{dt}$$

$$m_F h_F + M_F \frac{dh_F}{dt} = qF + V_F \frac{dp}{dt}$$

Rearranging above equations to establish the from $\frac{\mathrm{dhG}}{\mathrm{dt}}$ and $\frac{\mathrm{dhf}}{\mathrm{dt}}$ respectively

$$\frac{dh_G}{dt} = \frac{1}{M_G}\left[qG + V_G\frac{dp}{dt} - m_G h_G\right] \quad (19)$$

$$\frac{dhf}{dt} = \frac{1}{M_F}\left[qF + V_F\frac{dp}{dt} - m_F h_F\right] \quad (20)$$

Substituting equations 3a and 4a respectively into eqs 19&20, we get:

$$\frac{dh_G}{dt} = \frac{1}{M_G}\left[- m_{re}h_G - m_{sv}h_G - (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_G + V_G\frac{dp}{dt} - m_G h_G\right] \quad (21)$$

$$\frac{dh_F}{dt} = \frac{1}{M_F}\left[m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh + V_F\frac{dp}{dt} - m_F h_F\right] \quad (22)$$

then substitute eqs. 19 and 20 into eq. 18 and simplifying:

$$\frac{dp}{dt}\left[M_G\left(\frac{d\vartheta_G}{dp}\right)_{h_G} + M_F\left(\frac{d\vartheta_F}{dp}\right)_{h_F}\right]$$
$$= -[m_G\vartheta_G + m_F\vartheta_F + M_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P] * \frac{1}{M_G}\ [(- m_{re}h_G - m_{sv}h_G - m_{CS}h_G - m_{CW}h_G + m_{FL}h_G - m_{RO}h_G) + V_G\frac{dp}{dt} - m_G h_G]$$
$$+ M_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P * \frac{1}{M_F}\left[m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh + V_F\frac{dp}{dt} - m_F h_F\right]$$

$$\frac{dp}{dt}\left[\!\left[M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P\right]\!\right] = -\left[m_G\vartheta_G + m_F\vartheta_F + \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P[- m_{re}h_G - m_{sv}h_G - m_{CS}h_G - m_{CW}h_G - m_{RO}h_G + m_{FL}h_G - m_G h_G] + \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P * [m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh - m_F h_F\right] \quad (23)$$

simplifying eq. 23 to get:

$$\frac{dp}{dt} = -\frac{\left[m_G\vartheta_G + m_F\vartheta_F + \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P(- m_{re}h_G - m_{sv}h_G - m_{CS}h_G - m_{CW}h_G - m_{RO}h_G + m_{FL}h_G - m_G h_G) + \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P(m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh - m_F h_F)\right]}{M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P} \quad (24)$$

$$\frac{dp}{dt} = -\frac{\left[m_F\vartheta_F + m_G\vartheta_G + \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P[m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh - m_F h_F] + \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P[- m_{re}h_G - m_{sv}h_G - m_{CS}h_G - m_{CW}h_G - m_{RO}h_G + m_{FL}h_G - m_G h_G]\right]}{M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P} \quad (25)$$

Rearranging eq. 25 and simplifying. One gets:

$$\frac{dp}{dt}\left[M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P\right] = -\left[\!\left[\vartheta_F\frac{dM_F}{dt} + \vartheta_G\frac{dM_G}{dt} + \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P\left(\frac{dM_{SU}}{dt}h_{SU} + \frac{dM_{SP}}{dt}h_{SP} + \frac{dM_{CS}}{dt}h_F + \frac{dM_{RO}}{dt}h_F + \frac{dM_{CW}}{dt}h_F - \frac{dM_{FL}}{dt}h_F + \frac{dQh}{dt} - \frac{dM_F}{dt}h_F\right) - \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P\left(\frac{dM_{CS}}{dt}h_G + \frac{dM_{re}}{dt}h_G + \frac{dM_{SV}}{dt}h_G + \frac{dM_{cw}}{dt}h_G + \frac{dM_{RO}}{dt}h_G - \frac{dM_{FL}}{dt}h_G + \frac{dM_G}{dt}h_G\right)\right]\!\right] \quad (26)$$

Collect the same derivatives:

$$\frac{dp}{dt}\left[M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P\right] = -\left[\frac{dM_F}{dt}\left(\vartheta_F - \left(\frac{\partial\vartheta_F}{dh_F}\right)_P h_F\right) + \frac{dM_G}{dt}\left(\vartheta_G - \left(\frac{\partial\vartheta_G}{dh_G}\right)_P h_G\right) + \frac{dM_{SU}}{dt}\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_{SU} + \frac{dM_{SP}}{dt}\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_{SP} + \left(\frac{dM_{RO}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{CS}}{dt} - \frac{dM_{FL}}{dt}\right)\left[\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_F - \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P h_G\right] + \frac{dQh}{dt}\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P - \left(\frac{dM_{re}}{dt} + \frac{dM_{SV}}{dt}\right)\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P h_G\right] \quad (27)$$

Defining:-

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$K = M_G\left(\frac{\partial\vartheta_G}{\partial p}\right)_{h_G} + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F} + V_G\left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P + V_F\left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P$, $K1 = \vartheta_F - \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_F$, $K2 = \vartheta_G - \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P h_G$,$K3 = \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P\ h_{SUr}$,$K4 = \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_{SP}$,$K5 = \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P h_F - \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P h_G$ $K6 = \left(\frac{\partial\vartheta_F}{\partial h_F}\right)_P$,

$K7 = \left(\frac{\partial\vartheta_G}{\partial h_G}\right)_P h_G$

Then equation (27) will be reduced to:-

$$\frac{dp}{dt}[K] = -\left[\frac{dM_F}{dt}\ K1 + \frac{dM_G}{dt}K2 + \frac{dM_{Su}}{dt}K3 + \frac{dM_{SP}}{dt}K4 + \left(\frac{dM_{RO}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{CS}}{dt} - \frac{dM_{FL}}{dt}\right)k5 + \frac{dQh}{dt}K6 - K7\left(\frac{dM_{re}}{dt} + \frac{dM_{SV}}{dt}\right)\right] \tag{28}$$

Then divided above equation by K and let:-

$\frac{K1}{K} = a1, \frac{K2}{K} = a2, \frac{K3}{K} = a3, \frac{K4}{K} = a4, \frac{K5}{K} = a5\ , \frac{K6}{K} = a6\ , \frac{K7}{K} = a77, one\ gets: -$

$$\frac{dp}{dt} = -\left[a1\frac{dM_F}{dt} + a2\frac{dM_G}{dt} + a3\frac{dM_{SU}}{dt} + a4\frac{dM_{SP}}{dt} + a5\left(\frac{dM_{RO}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{CS}}{dt} - \frac{dM_{FL}}{dt}\right) + a6\frac{dQh}{dt} - a7\left(\frac{dM_{re}}{dt} + \frac{dM_{SV}}{dt}\right)\right] \tag{29}$$

Then to find the relation between the liquid levels inside the pressurizer since it is a function of the rate of fluid flow between the pressurizer and the primary coolant loop circuit. Therefore since assumed the pressurizer vessel of a cylindrical shape then the volume of water will be:-

$$V_F = \frac{\pi D^2}{4} * Z \tag{30}$$

$$\frac{dV_F}{dt} = \frac{dZ}{dt}.\frac{\pi D^2}{4} \tag{30a}$$

then the rate of change of water level as function of the rate of change of water volume

is given by:-

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\frac{dV_F}{dt} \tag{31}$$

Then substitute for $\frac{dV_F}{dt}$ from eq. (12) to get:-

$$\frac{dz}{dt} = \frac{4}{\pi D^2}[m_F\vartheta_F + M_F\frac{\vartheta_F}{dt}]\ ] \tag{32}$$

Substitute eq. (15) into above equation to get:-

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F\vartheta_F + M_F\left\{\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F}\frac{dp}{dt} + \left(\frac{\partial\vartheta_F}{\partial h}\right)_p\frac{\partial h_F}{dt}\right\}\right] \tag{33}$$

simplifying eq. (33), we get:-

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F\vartheta_F + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F}\frac{\partial p}{dt} + M_F\left(\frac{\partial\vartheta_F}{\partial h}\right)_p\frac{\partial h_F}{dt}\right] \tag{34}$$

Substitute eqs. (20) & (6) into eq. (34) and simplifying:

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F\vartheta_F + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F}\frac{dp}{dt} + M_F\left(\frac{\partial\vartheta_F}{\partial h}\right)_p.\frac{1}{M_F}\left(q_F + V_F\frac{dp}{dt} - m_F h_F\right)\right]$$

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F\vartheta_F + M_F\left(\frac{\partial\vartheta_F}{\partial p}\right)_{h_F}\frac{dp}{dt} + \left(\frac{\partial\vartheta_F}{\partial h}\right)_p\left(m_{SU}h_{SU} + m_{SP}h_{SP} + (m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + Qh\ + V_F\frac{dp}{dt} - m_F h_F\right)\right]$$

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\,[m_F \vartheta_F + M_F\left(\frac{\partial \vartheta_F}{\partial p}\right)_{h_F}\frac{\partial p}{dt} + \left(\frac{\partial \vartheta_F}{dh}\right)_p \ m_{SU}h_{SU} + \left(\frac{\partial \vartheta_F}{dh}\right)_p (m_{SP} + m_{CS} + m_{CW} + m_{RO} - m_{FL})h_F + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p Qh \ + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p V_F\frac{dp}{dt} - \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_F h_F]$$

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F\left((\vartheta_F - \left(\frac{\partial \vartheta_F}{\partial h}\right)_p h_F\right) + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{SU}h_{SU} + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{SP}h_{SP} + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{CS}h_F + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{CW}h_F + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{RO}h_F - \left(\frac{\partial \vartheta_F}{\partial h}\right)_p m_{FL}h_F + \left(\frac{\partial \vartheta_F}{\partial h}\right)_p Qh + \frac{dp}{dt}\{V_F\left(\frac{\partial \vartheta_F}{\partial h}\right)_p + M_F\left(\frac{\partial \vartheta_F}{\partial h}\right)_p\}\right] \quad (35)$$

Defining:

$$\left(V_F\left(\frac{\partial \vartheta_F}{\partial h}\right)_p + \ M_F\left(\frac{d\vartheta_F}{dp}\right)_{h_F}\right) = K8,$$

Substitute for K1, K2, K3, K4, K5, K6, K7, K8 into eq. (35), then eq. (35) will be as below:

$$\frac{dz}{dt} = \frac{4}{\pi D^2}\left[m_F K1 + +(0)K2 + m_{su}K3 + m_{SP}K4 + (m_{CS} + m_{CW} + m_{RO} - m_{FL})K5 + QhK6 + (0)K7 + \frac{dp}{dt}K8\right] \quad (36)$$

Let $\frac{4\,K1}{\pi D^2}$=$b1$, $\frac{4\,K2}{\pi D^2}$=$b2$=0, $\frac{4\,K3}{\pi D^2} = b3$, $\frac{4\,K4}{\pi D^2} = b4$, $\frac{4\,K5}{\pi D^2} = b5$ $\frac{4K6}{\pi D^2}$=$b6$, $\frac{4\,K7}{\pi D^2}$=$b7$, $\frac{4\,K8}{\pi D^2}$=$b8$,

Then equation (36) will be reduce to:

$$\frac{dz}{dt} = [\![b1\frac{dM_F}{dt} + (0)b2 + \frac{dM_{SU}}{dt}b3 + \frac{dM_{SP}}{dt}b4 + (\frac{dM_{CS}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{RO}}{dt} - \frac{dM_{FL}}{dt})b5 + \frac{dQ_h}{dt}b6 + (0)b7 + \frac{dP}{dt}b8]\!] \quad (37)$$

Rearrange eqs. 1, 2, 21 &22:

$$\frac{dM_G}{dt} = +\frac{dM_{FL}}{dt} - \frac{dMre}{dt} - \frac{dM_{sv}}{dt} - \frac{dM_{CS}}{dt} - \frac{dM_{CW}}{dt} - \frac{dM_{RO}}{dt} \quad (38)$$

$$\frac{dM_F}{dt} = \frac{dM_{SU}}{dt} + \frac{dM_{SP}}{dt} + \frac{dM_{CS}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{RO}}{dt} - \frac{dM_{FL}}{dt} \quad (39)$$

$$\frac{dh_F}{dt} = \left[\frac{h_{SU}}{M_F}\frac{dM_{SU}}{dt} + \frac{h_{SP}}{M_F}\frac{dM_{SP}}{dt} + \frac{h_F}{M_F}(\frac{dM_{CS}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{RO}}{dt} - \frac{dM_{FL}}{dt} - \frac{dM_F}{dt}) + \frac{V_F}{M_F}\frac{dP}{dt} + \frac{dQ_h}{M_F dt}\right] \quad (40)$$

$$\frac{dh_G}{dt} = \frac{h_G}{M_G}\left(-\frac{dM_{CS}}{dt} - \frac{dM_{CW}}{dt} - \frac{dM_{RO}}{dt} + \frac{dM_{FL}}{dt} - \frac{dM_G}{dt}\right) + \frac{V_G}{M_G}\frac{dp}{dt} \quad (40a)$$

Now let : $d1$=$\frac{h_{SU}}{M_F}$, $d2$=$\frac{h_{SP}}{M_F}$, $d2$=$\frac{h_F}{M_F}$, $d4$=$\frac{V_F}{M_F}\vartheta_F$, $d5$ =$\frac{1}{M_F}$, $c1 = \frac{h_G}{M_G}$, $c2 = \frac{V_G}{M_G} = \vartheta_G$,

Substitute C1, C2, d1, d2, d3 and d4 into above equation,(40) and (40a) one can gets:-

$$\frac{dh_G}{dt} = c1\left(-\frac{dM_{CS}}{dt} - \frac{dM_{CW}}{dt} - \frac{dM_{RO}}{dt} + \frac{dM_{FL}}{dt} - \frac{dM_G}{dt}\right) + c2\frac{dp}{dt} \quad (41)$$

$$\frac{dh_F}{dt} = d1\frac{dM_{SU}}{dt} + d2\frac{dM_{SP}}{dt} + d3(\frac{dM_{CS}}{dt} + \frac{dM_{CW}}{dt} + \frac{dM_{RO}}{dt} - \frac{dM_{FL}}{dt} - \frac{dM_F}{dt}) + d4\frac{dP}{dt} + d5\frac{dQ_h}{dt} \quad (42)$$

Now rewrite the six equations of the pressurizer transients to insurge actions, the general form of response to insurge action is: - these equations are 29, 37,38,39,41 and 42 which are listed below:-

For transient process in the pressurizer at in-surge action, we can conclude that:
m_{su}, h_{su} are input variables
m_{sp}, h_{sp} are input variables
m_{cs} depends on m_{sp}, so it is an input variable,
Q_h is an input variable
Therefore m_{su}, m_{sp}, m_{cs} and Q_h are not state variables i.e. they are not time dependent.

$$\dot{P} \ = \left[\vartheta(P,h), h(P,t), \dot{M}(t), \frac{\partial \vartheta}{\partial t}(P,h), \frac{\partial \vartheta}{\partial P}(P,h)\right]$$

Which contracts to an initial value problem: $\dot{P} = f(p,t); p(0) = p_o$