AN EXPERIMENTAL AND ANALYTICAL MODEL OF A PWR PRESSURIZER DURING TRANSIENTS.

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Abstract

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 surges to a partially-full tank, outsurges, surges to a tank with hot walls, empty tank surges, a combined insurges and outsurges, the effect of noncondensable gases, and free surface heat transfer. The experiments provide a data base, from which recommendations are made for calculating such phenomena, as: i) Stratification of the hot water and incoming cold water, ii) wall condensation, iii) flashing, iv) rainout, v) suppression of flashing, iv) wall boiling, vii) the effect of noncondensable gases on the wall heat transfer and, viii) free surface heat transfer.

A general model of a PWR pressurizer has been developed to predict the pressure behavior of PWR pressurizer during transients. The model was benchmarked against the low pressure experiments of this study and full scale pressurizer transient data.

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A  The Sum of Mass Transfer Associated with the Vapor Phase.
B  The Sum of Mass Transfer Associated with the Noncondensable Gases.
C  The Sum of Mass Transfer Associated with the Hot Liquid.
D  The Sum of Mass Transfer Associated with the Cold Liquid.

\( C_p \) Specific Heat.

A(t) The Wall Area Exposed to Vapor Phase.

AA The Sum of Heat Transfer Associated with the Vapor Phase.
BB The Sum of Heat Transfer Associated with the Noncondensable Gases.

CC The Sum of Heat Transfer Associated with the Hot Liquid.
DD The Sum of Heat Transfer Associated with the Cold Liquid.

e  Specific Internal Energy.

g  Gravitational Acceleration.

h  Specific Enthalpy.

k  Conductivity.

L  Length.

n  Normal Vector.

M  Mass.

p  Pressure.

Q  Rate of Heat Transfer.

R( ) Rate of Temperature Change at the Inside Wall.

S,dS Surface or Surface Element.

t  Time.

T  Temperature.
v  Specific Volume.
V  Velocity Vector.
V,dV Volume or Volume Element.
W  Work Rate.
X  Distance.

SUBSCRIPTS.

b  Boiling.
c  Cold Region.
c_l  Cold Liquid.
c_li  Incoming Cold Liquid.
c_{D}  Penetration Depth.
eg  Equivalent.
fg  Difference of Liquid and Vapor.
h  Hot Region.
h_l  Hot Liquid.
h_li  Incoming Hot Liquid.
h_lo  Outgoing Hot Liquid.
htr  Heater.
g  Gas Phase.
g_i  Gas Phase Incoming
go  Gas Phase Outgoing.
\text{ic}  Interface Condensation.
in  Initial Properties.
iw  Inside Wall.
mx  Mixing.
nc  Noncondensible Gas.
nci Noncondensible Incoming.
ncoco Noncondensible Outgoing.
ro  Rainout.
sat Saturation.
sc  Spray Condensation.
sp  Spray.
srg Surge.
st  Stripping.
wil Wall
wco Wall Condensation.

GREEK.
α  Heat Transfer Coefficient.
ρ  Density.
δ  Thickness.
μ  Viscosity.
τ  Time Constant of Heater.
CHAPTER 1

INTRODUCTION.

1.1 INTRODUCTION.

During the life of a nuclear power plant, the reactor system will be subject to a number of abnormal situations. In these situations, to operate the reactor safely and decide what actions can be taken to prevent an accident or mitigate the consequence of an accident, it is important and desirable to predict the state variables of the system (pressure, temperature, density, enthalpy, etc.) precisely. For a pressurized water reactor (PWR), in particular the prediction of pressure in the system is essential since the pressure in the system is a key control parameter of the system.

Before the Three Mile Island accident many people were not interested in small break LOCA's (Loss Of Coolant Accidents) since they were considered less severe than large break LOCA's. But the small break LOCA's are much more likely and complicated than the large break LOCA's. In such an accident, the information about pressure behavior is critical in order to take preventive action or optimize
the operation when trying to attain a safe shutdown. Similarly in an overcooling transient of primary system (which can result from the streamline break in the secondary system), the rapid depressurization of the primary system sometimes can cause the actuation of Safety Injection System (SIS). This actuation can, apparently, sometimes lead to pressurized thermal shock.

The pressure prediction, however, is generally not satisfactory. These predictions have been done by computer codes like RETRAN, RELAP, TRAC, THERMIT, etc.. As shown in Fig 1.1 and Fig 1.2, the predictions of the advanced computer code (by RETRAN) for LOFT transients and Connecticut Yankee nuclear power plant transients does not adequately represent the phenomena (1),(2).

The models for pressurizers, mostly developed in the early 1960's, were applied in many computer codes and design problems without raising serious questions. These models predict the pressure during normal operation even though some fundamental physical phenomena are missing and unreasonable assumptions have been made. The reason is that in the normal operation, the deviation from steady state operating conditions is quite so that a rough prediction satisfies most users.

After the TMI accident many people realized the importance of small break LOCA's and the necessity for reliable physical models for all of the components in the loop. To meet this need, the present research work has been focused on the experiments simulating pressurizer transients, and the development of simple physical model based on the experimental results. The new model could also be checked against experimental data and the Connecticut Yankee nuclear power plant transient data.
Figure 1.1 Prediction of Connecticut Yankee Power Plant Transient by RETRAN Computer Code (Sept., 1980)
Figure 1.2 Prediction of LOFT Experiment by RETRAN Computer Code.
1.2 OBJECTIVES OF THE PRESENT RESEARCH.

Most of the physical phenomena happening in the pressurizer are, in principle, reasonably well understood. It remains to put them together into a coherent calculation model however. These phenomena include follows.

1. Heat transfer at the pressurizer wall by condensation.

2. Heat transfer at the pressurizer wall by boiling.

3. Heat transfer between the vapor and wall by natural convection.

4. Heat transfer between the liquid and wall by natural convection.

5. Heat transfer at the liquid/vapor interface.

6. Heat transfer through the pressurizer wall.

7. Flashing from the bulk of the hot liquid in a rapid outsurge.

8. Heating by the electrical heaters.

9. Stripping of non-condensable gases and suppression of condensation by the gases.

10. Condensation on the sprays.

11. Evaporation of spray water hitting the pressurizer wall.
12. Relief and/or safety valve opening.

13. Transient heating or cooling of the walls.

Since the reactor system analysis computer codes have to consider many other important components as well as the pressurizer, one can only allocate a limited amount of computing time, the memory, and the number of meshes to the calculation of the pressurizer. Therefore to incorporate all of these phenomena into a model would lead to such complications that the code would not be useful. It is necessary to identify the processes which are really important and show how much they affect the pressurizer behavior during the important transients.

In particular the key questions for this investigation are the following.

1. How can the condensation and boiling rate be predicted at the pressurizer wall in an insurge or an outsurge?

2. How can the heat transfer rate be predicted at the interface during an insurge?

3. How significant is the mixing rate of cold, incoming jet with the hot water already present in the pressurizer?

4. What is the heat transfer rate between the wall and the liquid inside the pressurizer?

5. What are the heat losses through the wall?

6. How can the bulk flashing rate be predicted?
7. What is the effect of non-condensable gases?

To get answers to these questions, an experimental apparatus was built to run different sets of transient experiments simulating actual pressurizer transient operations. Based on these discoveries, a simplified model has been made and checked against the present experiment data and the Connecticut Yankee nuclear power plant transient data.
CHAPTER 2

THE OPERATION OF THE PRESSURIZER.

Before discussing the experimental work, it is helpful to degress a bit to overview the operation of the pressurizer.

In a Pressurized Water Reactor (PWR), the volume and pressure of the coolant can fluctuate due to the temperature variation and/or water inventory variation in Reactor Coolant System (RCS) during normal operation, transients, and accidents. To keep the system pressure within safe limits and to accommodate the rapid coolant volume changes, these systems have a device called a "Pressurizer". It consists of a thick, carbon steel vessel lined with stainless steel (which protects the wall from the corrosive boric acid), heaters, a spray nozzle, relief and safety valves, a sparger, manhole, and instrumentation. There is an insurge line from the hot leg which is connected to the bottom of the pressurizer, while the spray line from the cold leg is connected to the top.

The pressurizer controls the system pressure by turning on and off the spray, the heaters, and the relief and/or safety valves, or by combinations of those operations. Let's discuss the details of each
Figure 2.1 Schematic Diagram of PWR Pressurizer.
of these operations with the related transients.

In the case of an insurge transient, which usually results from loss of heat sink, or a reactor power increase, the excess coolant volume rushes to the pressurizer. Consequently the system pressure rises. At the same time, the spray of cold water from the top of pressurizer reduces the pressure by condensing the steam in the upper part of the pressurizer. If the pressure suppression by the spray is not sufficient, the relief and/or safety valves are opened sequentially to keep the pressure below safety limits. Even in the insurge case, the electrical heaters turn on to heat the subcooled insurge water up to saturated state so that it can quickly respond to next perturbation.

In the outsurge case (due to the coolant volume shrinkage or loss of coolant), the heaters turn on and generate saturated steam so that pressure can be raised. When the heating rate is not enough to compensate the rapid outsurge or heaters are not available, quick flashing of hot liquid present already in the pressurizer reduces the rate of drop in the system pressure significantly. Also the boiling of water at the hot wall contributes a little to the reduction of pressure drop.

The function of sparger is to prevent the mixing of the cold (subcooled) water with the hot (almost saturated) water present and thus improve the maneuverability and energy economy of pressurizer.

To prevent thermal shock to the spray nozzle and connecting piping and to uniformly mix the boric acid in the primary system and pressurizer, a constant flow of water dribbles from the spray nozzle into the steam in the top of the pressurizer. To compensate the heat loss from the pressurizer and heat the dribbling water up to the saturated state, a single bank of heaters out of the five present is usually on.

During normal operation, non-condensable gases, mainly hydrogen, can be stripped from the spray and dribbling water. The hydrogen gas is purposely dissolved in the coolant to protect the system from the corrosion and help to eliminate the positive ions in the coolant. During transients, nitrogen gas can flow into the pressurizer from
accumulator or/and Emergency Core Cooling System (ECCS). In addition, if the cladding temperature is higher than 180 F, the Zirconium in the cladding can react with the water and yield the hydrogen gas.

In the pressurizer, there are usually five banks of direct immersion, straight tubular heaters, installed vertically from the bottom. They are grouped into two categories, the back-up heaters and the control heaters. The back-up heaters are usually on and off completely depending on the water level and the pressure, while the control heater ramps between full on and full off.
CHAPTER 3

EXPERIMENTAL WORK.

3.1 THE OUTLINE OF EXPERIMENTAL WORK.

To find out the important physical processes happening in the pressurizer, an experimental apparatus has been built. The apparatus is same as the one that Saedi (3) and Leonard (9) used.

The temperature profile at the centerline and on the outside of tank wall is detected by 21 thermocouples (13 at the centerline and 8 on the wall). The pressure behavior is traced by a pressure transducer. These data are utilized to make and check a new model.

The experiments were run as follows.

1. Preliminary test experiments.

2. Tank heat loss to the environment experiments.
3. Part full insurge experiments.

4. Outsurge experiments.

5. Insurge to hot tank experiments.

6. Outsurge after insurge experiments.

7. Empty tank insurge experiments.

3.2 EXPERIMENTAL APPARATUS.

The apparatus consists of two cylindrical stainless steel pressure tanks called the "main tank" and the "drive tank", a PERKIN-ELMER micro-computer, a non-condensable gas injection rig, and connecting piping and instrumentation (See Fig. 3.1.).

The main tank (45 inches in height and 8 inches in inside diameter) consists of stainless steel tank (1/2 inch thickness), a sight glass (43 inches tall) for measuring water level, six cylindrical windows (2-1/2 inches ID and 2-1/2 inches long), a pressure gage, a pressure transducer, a vent hole, a sparger, and piping (See Fig. 3.2.).

The side windows are to look at the phenomena occurring in the tank. The pressure gage is to measure steady state pressure, while the pressure transducer was used to measure the transient pressure. The vent hole and quick opening valve was used to vent steam to eliminate the noncondensable gases in the system. The sparger was geometrically similar to the real one and was used to dissipate the momentum of the incoming water.

The heaters are connected to an amp meter. The current was controlled by six variable transformers. The individual power can be calculated by the I²R. The resistance of the six heaters is 21.25, 27.54 X 2, 28.1 X 2, and 28.85 ohms, respectively.
Figure 3.1 Schematic Diagram of the Experimental Apparatus.
Figure 3.2 Schematic Diagram of Main Tank.
In the preliminary test experiments, the thermocouples were positioned at four different radial locations, at the center, at 3" off the center, 1/4" inside of the tank wall, and on the wall outside of the tank, respectively. These locations were chosen to detect the radial temperature gradient in the tank. The positions of each thermocouple can be referred to Fig. 3.3 and Fig. 3.4.

Since the preliminary experimental results showed the radial temperature gradient was very small, the position of the thermocouples was changed. Therefore there were more thermocouples available to measure the axial temperature variation at the centerline and on the wall (See Sec. 3.3.1 and Sec. 3.4.1).

The cold water from city water system was pumped into the main tank and heated up to the saturated state by the electrical heaters. During heating, the noncondensable gas stripped from the water and the air already present was vented four or five times by opening the vent valve located 5" from the top.

The cold surge water from the surge tank came into the main tank through the orifices, two quick opening valves, and a regulating valve. The differential pressure across the orifice was converted into an electrical voltage signal and recorded in floppy disk along with all the temperatures.

3.3 INSTRUMENTATION.

3.3.1 Thermocouples.

The thermocouples used in this experiment are made of the copper-constantan wire sheathed with 304 stainless steel and insulated by a ceramic material (Product of OMEGA CO.; Cat. No 304 - T - MO - 1.5mm). The diameter of the wire with sheath is 0.059 in and the diameter of bare thermocouple wire is 0.010 in. To improve the time constant of the thermocouples, the bare beads were used as hot junctions (See Fig. 3.5). The average diameter of the hot junction
Figure 3.3 The Arrangement of Thermocouple sets.
Figure 3.4 The Positions and Channel Numbers of the Thermocouples.
a) Thermocouple Bead Dimensions

b) Orifice Dimensions

Figure 3.5 Dimensions of the Orifice and Thermocouple Bead
bead is 0.03".

Instead of an ice junction, a RTD built into the microcomputer was used. The off-set voltages of the terminal were measured (with the computer air conditioner on) for the DAS in operation and compensated.

Thirteen thermocouples are installed at the center of the tank and eight are on the outside wall of the tank. The position of the thermocouples measured from the tank bottom is as follows:


On the wall; 4", 10", 16", 22", 28", 31", 34", 37".

3.3.2 Orifice

The instrumentation for surge flow rate consists of an ASME standard orifice (1/16" thickness, 0.218" dia.) and a pressure transducer. The calibration of this was performed by comparison of flow rate from the microcomputer output with the actual water level of the tank (3).

3.3.3 Pressure Transducers.

Two pressure transducers were used; one for the orifice and the other for tank pressure. Both transducers were made by Validyne (Model No; DP 15-54 and DP 15). The orifice for tank pressure was calibrated by a dead weight tester. The voltage signals from the transducers are modulated by CD-15 modulator from Validyne.
3.3.4 Microcomputer.

The PERKIN-ELMER microcomputer consists of a main processor, a real time data acquisition system, a console, and a printer. The software for the data acquisition system requests to input File Name (NA), Maximum number of Sampling (MS), Sampling Time interval (TS), Number of Channel used (NC), Amplification (GA) from 5 mV to 1V. The signal from the terminal was digitized and recorded on a floppy disc.

The microcomputer has 24 channels. The first is allocated to the reference junction, the second to fourteenth to thermocouples at the center, and the fifteenth to twenty-second to the outside wall thermocouples. The twenty-third and twenty-fourth are assigned to the orifice and the tank pressure transducer, respectively.

The data were read into the VAX machine from the floppy disc and converted to the temperatures, flow rate, and tank pressure, respectively. These data were also plotted by the VAX computer.

3.4 EXPERIMENTAL PROCEDURES.

The experiments were divided into two kinds; the preliminary ones and the main ones. In the preliminary experiments, the calibration, the checking of the instruments, the set-up of the software, and heat loss experiments were completed. A couple of insurge experiments were also executed to investigate the magnitude of the radial temperature gradient and the mixing of the cold and the hot water. In the main experimental stage, five different transient experiments were executed. These are the partially full insurge, the outsurge, the insurge to a hot tank, the insurge after outsurge, and the empty tank insurge. Each of these experiments will be discussed.
3.4.1 Experimental Measurement Of The Heat Loss.

To measure the heat loss to the environment, several experiments were made. Some with heaters on and the others with live steam.

a). The procedures with heaters on are as follows.

1. Fill the tank with water to 1/3 or 2/3 of its height to check the dependence of the heat loss on the water level.

2. Turn the heaters on.

3. When the pressure reaches around 50 psia, open the vent valve four or five times for about 30 seconds each time to remove the non-condensable gas.

4. When the pressure reaches 90 psia, reduce the heater power and keep a steady state for a minute.

5. Read the amp meter and calculate the input power. That power is the heat loss to the environment.

6. Repeat the same experiment for a different water level and pressure.

b). The procedure with the live steam is as follows.

1. Fill the tank with water.

2. Open the steam line valve and leave it on until the water becomes saturated.
3. Put the microcomputer in the standby state ready to take data by hitting the return key.

4. Vent the non-condensable gas four or five times for 30 sec at each venting.

5. Keep the steady state for five minutes and close the streamline valve.

6. Wait a few seconds then hit the return key of microcomputer to take data.

7. From an energy balance equation on the closed system, the heat loss can be calculated (there is no flow, only a heat loss from the tank).

3.4.2 Partially Full Insurge Experiment

These experiments were intended to simulate insurge transients without heater and spray operation from an initially equilibrium state. The tank was filled with saturated water (at about 100 psia) to about 15 inches height (1/3 of the tank height). Then the cold water was insurged from the bottom until the water level reached about 30 inches (2/3 of the tank height). The pressure change was 25 psia (from 100 to 125 psia). The insurge rate was about 0.6 lbm/sec.

The results of these experiments showed the axial temperature profile at the centerline and on the wall and were expected to provide useful information about the condensation at the vapor/wall interface and the stratification between the hot water and the cold water. The information about the heat transfer between the hot and cold water in the lower part of the liquid region was also obtained.

The procedure for running these experiments is as follows.
1. Fill the main tank with the water.

2. Boil the water with electrical heaters while venting the non-condensable gas.

3. While the water is boiling, make the computer ready to take data by hitting the return key.

4. After the computer is ready, fill the drive tank with the cold water and pressurize it up to 150 psia.

5. When the desired pressure is obtained, reduce the heat input and attain a steady state.

6. After turning heaters off completely, wait 30 sec then hit the return key to take data.

7. If a message of "First Sample" is delivered from the computer, open the two quick acting valves so that the cold water can be pushed into the main tank through the orifice and regulating valve (which is already open).

8. When the water level reaches the desired value, close the two quick acting valves and complete the experiment.

9. By varying the water level and the driving pressure, many different experiments can be executed.

10. Before opening the quick acting valves, open the nitrogen gas line to the drive tank to keep the insurge rate approximately uniform.
3.4.3 Outsurge Experiment.

The purpose of this experiment is a simulation of the outsurge transients from an initially equilibrium state. Experiments were executed by discharging saturated liquid from the tank bottom. The initial water level and pressure were about 30 inches (2/3 of the tank height) and 120 psia, respectively.

During the outsurge transient, the important phenomena were expected to be steam expansion, rainout, flashing, and wall boiling. The temperature profiles at the centerline and on the outside wall and the pressure behavior would help us to construct a model of these phenomena.

All the procedures from 1) to 9) are the same as the part full insurge except 4) and 7). Those steps become.

4. Fill the drive tank with some cold water and keep its pressure at atmosphere so it can accept hot water drained from the main tank and cool it.

7. If a message of "First Sample" is delivered from the computer, open the two quick opening valves so that hot water can be drained from the tank.

3.4.4 Insurge To A Hot Tank Experiment.

The purpose of these experiments is to simulate an insurge transient during the continuous operation of pressurizer which may not be in equilibrium state for some real transients. In the some transients, there are many sequential operations like outsurge, insurge, safety or relief valve opening, spray on, or heaters on. As a result of these maneuvers, the temperature of the wall may be different from the saturation temperature for the system pressure.
During an insurge transient when the initial wall temperature is colder than the saturation temperature for the system pressure, condensation will occur much sooner than for the case in which the initial wall temperature is same as the saturation temperature for the system pressure. On the other hand, when the initial wall temperature is higher than the saturation temperature for the system pressure, wall condensation will not occur until the saturation temperature of the system is higher than the wall temperature.

The experiments were done by admitting the cold water into the hot tank. By hot tank we mean that the initial wall temperature is higher than the saturation temperature for the system pressure (the hot wall can be made by venting the saturated steam).

The procedure from 1). to 6). is same as the part full insurge. The seventh step is:

7. If a message of "First Sample" is delivered from the microcomputer, open the vent valve to make the pressure drop to the wanted value (usually about 20 psia less than initial one). Then close the vent valve to make the system pressure drop but let the wall temperature stay same as the initial one.

8. Wait a few seconds for the system to almost reach the equilibrium state, then open the two quick opening valves to allow a rush of cold water into the main tank from the drive tank.

9. The following procedures are the same as for the "partially full insurge experiment".
3.4.5 Outsurge After Insurge Experiment.

These experiments were intended to simulate transient operation of a pressurizer whose liquid temperature is not uniform. The nonuniformity could be caused by the sequence of insurge and outsurge. The experiments were executed by discharging the cold water (which entered during an insurge) from the bottom of the tank. The difference between this experiment and the simple insurge experiment is a steam expansion without flashing, and the suppression of flashing.

The procedure from 1) to 7) is same as for the partially full insurge experiment. Items 9 to 10 differ as follows.

9. When the water level reaches the desired value, close the nitrogen line valve and depressurize the drive tank suddenly so that the flow reverses.

10. When the water level reaches about seven inches, close the valves and stop the experiment.

3.4.6 Empty Tank Insurge Experiment.

Usually the water level of the pressurizer is deep enough to neglect the interface heat transfer. However, sometimes the water would be so depleted that the momentum of the insurge jet can disrupt the vapor/liquid interface and improve the interface heat transfer significantly. Due to this heat transfer, the pressure of the system might drop to the safety injection system actuation pressure. This could then cause pressurized thermal shock. This experiment was run to see how significant this drop in pressure is. The pressure drop in this experiment will be much more dramatic than that in full-scale high pressure system. The reasons are as follows.
First, in the full-scale high pressure system, the pressure in the pressurizer is controlled by the compressibility of the water in the reactor coolant system. That is, since the water in high pressure system is more compressible than that in low pressure system, and the volume in the RCS is huge, the volume increase due to depressurization is quite significant. Therefore a drastic drop in system pressure does not occur.

Secondly, the subcooling enthalpy of insuring cold liquid has much more drastic effect on the system pressure than it does at high pressure because the volume enthalpy for high pressure steam is much greater than for low pressure steam. The same amount of subcooling condenses a smaller volume of steam.

The procedure is as follows.

1. Fill the main tank with cold water up to the 10" elevation (high enough to submerge the heaters).

2. Boil the water with the electrical heaters while the non-condensable gas is venting.

3. While the water is boiling, set up the computer ready to take data by hitting the return key.

4. Fill the drive tank with cold water and presurize it about 20 psia higher than starting pressure of the main tank.

5. When the desired pressure is obtained approximately 100 psia, reduce the heat input and make a steady state.

6. Turn the heaters off and drain off all of the water (one should be careful not to drain the steam unnecessarily).

7. Wait a few seconds and hit the return key.
8. When a message of "First Sample" is delivered from the microcomputer, open the valves so that the cold water can be pushed into the main tank.

9. When the water level reaches the desired value, close the valves, and finish the experiment.

3.5 RESULTS.

Detailed experimental results for all these experiment are reported in Appendices A through F so that any new pressurizer model can be checked against this data.

3.5.1 Preliminary Experiments.

The preliminary experiments showed the following(3).

1. The functional check of system and instruments (the piping, valves, pressure gage, pressure transducer, thermocouples, etc.) was satisfactory.

   The computer software (that is the computer program needed to convert the binary signals to corresponding physical quantities) was satisfactory.

2. There was almost no mixing (stratification of the cold and hot water occurred). The insurged cold water tends to stratify due to its negative buoyancy and the design of the inlet sparger (Sec. 4.3.2.).
3. The radial temperature gradient found to be so small and local that it does not matter.

4. The water temperature profile near the inlet sparger shows some disturbances (Sec.3.5.2.3).

5. The heat loss from the tank is about 1 BTU/SEC, equivalent to 0.127 BTU/FT²/SEC.

3.5.2 Partially Full Insurge Experiment (Appendix A).

The partially full insurge experiments showed the following.

1. The incoming cold water stratifies below the hot water. This phenomena is described using a Froude Number (Sec.4.3.2.)

2. The gas phase is slightly superheated. The superheat is weakly dependent on the surge flow rate. This indirectly verifies that there is wall condensation.

3. The temperature profile in the liquid shows some wake effects due to the sparger. The radial temperature distribution near the sparger elevation shows that the temperature at the center of the tank is higher than that near the wall. Also the axial profile above the sparger shows that the temperature just above the sparger is higher than that of a few inches above the sparger. This explanation can be seen by referring to Fig.A.3. The figure shows that the insurge water hits the sparger roof, changes its path from the axial to the radial direction, then hits the wall to change its course again to the inward radial direction. During these flow processes, the insurged water picks up the heat from the
hot water in the vicinity of the sparger.

4. The temperature measured at the outside of the wall shows that the wall receives a significant amount of heat from the vapor (in the upper part) and also discharges heat to the cold insurge water (in the lower part). The absolute amount of heat in these two parts is comparable. But the relative amount compared to the total energy of corresponding phase in the tank is quite different. The heat transfer in the vapor phase has much more significance to the pressure behavior. Wall condensation can be observed through the tank windows during this experiment.

3.5.3 Outsurge Experiment (Appendix B).

The following things have been discovered in this experiment.

1. In the simple outsurge experiment transient (with hot water only), the pressure drops very slowly. The initial vapor volume height is almost 10" and the final 30". The vapor volume increases by a factor of three, while the pressure drops from 125 psia to 118 psia only. Flashing serves to maintain system pressure very effectively.

2. The flashing and rainout (fog) can be observed through the tank windows. The violent flashing and rainout can explain the very slow pressure drop. Vapor is evolved which maintains the pressure.

3. The wall temperature profile at a level which is originally submerged and then dried as the outsurge proceeds, yields some useful information also. When it is submerged the wall
temperature drops as rapidly as the saturation temperature. But after it is dried out, the temperature drops very slowly. It can be explained as following. When the tank is submerged boiling increases the heat transfer so the wall temperature drops quickly. However, if the wall is dried the heat transfer at the wall by natural convection is very poor, so the wall temperature drops slowly.

4. The liquid needs some degree of superheat (.5 to 1 F) for flashing to occur. This superheat energy can be recovered after the outburst stops.

5. The temperature profile in the liquid shows that before the outburst starts, the temperature of the upper part of the liquid is slightly higher than that of the lower part (≈ 1.0 F), but as the outburst goes on, the temperature become flat and finally the temperature of the lower part liquid is higher (≈ 2.0 F). This phenomena can be explained by the boiling agitation as well as the hydrostatic pressure difference between top and bottom.

3.5.4 Insurge To A Hot Tank (Appendix C).

In this experiment we discovered the following facts.

1. The natural convection heat transfer between the wall and vapor is negligible. When the wall is dry and above the saturation temperature, the only possible heat transfer mechanism is natural convection in the vapor (except the heat loss to the environment).
2. The tank pressure rise approaches that of an isentropic process. With the tank wall hot, the heat loss from vapor phase is very small indeed.

3. The wall condensation starts when the pressure is slightly lower than the saturation pressure of the initial wall temperature. (The deviation of the condensation temperature from the initial temperature may result from the heat loss of the wall to the environment and/or heat transfer with the vented steam when the steam is vented to make the wall hot).

4. The superheat of the vapor phase increases linearly until condensation starts. After condensation starts, the temperature rises more slowly as the pressure rises.

5. Superheat in the vapor phase affects the pressure behavior.

3.5.5 Outsurge After Insurge Experiment (Appendix D).

1. For the insurge, all the phenomena is same as described in section 4.4.

2. In this outsurge, the pressure behavior is quite different from that of the simple outsurge transient experiment.

3. When the outsurge starts from the initial saturated state, all the water in the tank is available for flashing as soon as pressure drops. However, in the insurge-oulsurge transient, only the hot water can contribute to flashing. Furthermore the hot water cannot flash until its pressure drops to the saturation pressure of the initial temperature.
Therefore in the former case the steam is almost saturated, but in the latter case the steam can deviate from the saturation. It is subcooled.

4. After the pressure reaches the saturation pressure for the initial temperature, the heat transfer to the cold wall reduces the flashing. Due to this, the pressure drops faster than that of the simple outsurge transient. It can be called "suppression of flashing".

3.5.6 Empty Tank Insurge Experiment (Appendix E).

1. The pressure response of the empty tank insurge is very sensitive to the initial hot water depth. The hot water not only damps the insurge momentum but also raises the water temperature so that it reduces the interface heat transfer rate.

2. The pressure behavior shows that when the water depth is very shallow there is lot of heat transfer at the interface so pressure drops sharply. Then as the depth is raised, the drop becomes mild until finally the compression of the steam governs the pressure.

3. The penetration depth for the sparger used in this experiment is about 7" for 1 lbm/sec flow rate.

4. When the pressure drops, the temperature stay close to the saturated value. However, when the pressure rises, the vapor becomes quite superheated. The reason is thought to be that the wall temperature is almost same as initial one for that transient so that the vapor is compressed adiabatically.
Figure 4.1 Noncondensable Gas Control Volume Scheme.
CHAPTER 4

GOVERNING EQUATIONS AND MODELLING.

4.1 INTRODUCTION.

As was mentioned in the Sec.1.2., most of the physical phenomena happening in the pressurizer are understood well enough so that if all of the phenomena could be incorporated into a computer code, a reasonable prediction can be obtained. However, reactor analysis computer codes have to calculate many other components as well as the pressurizer. Therefore there are very tough restrictions in the number of meshes, the computing time, the memory size, and etc.. The allocation of these variables should be decided based on the purposes of the calculation, the sensitivity of the system to the overall analysis, the structure of the computer code, and so forth. These questions will not be discussed any further since these problems are beyond the scope of the present research. Rather we shall make recommendations on the method of calculating each quantity and leave
it up to the code developer to decide on the basis of a sensitivity study which is to be made or the system of concern whether a term should be left in or not.

In this section, a pressurizer model based on the results of the present experiments simulating different transients will be presented. The model should assure reasonable accuracy as well as the usefulness. The optimal number of meshes (control volumes) in the pressurizer was chosen as seven. This is not absolute.

As shown in Fig 6.1, the pressurizer can be divided into seven regions; cold liquid, hot liquid, vapor, non-condensable gas (if any), and three wall regions. Let us discuss these seven regions.

The cold water region is occupied by the insurger cold water from the bottom and hot liquid region by the already present hot water (almost saturated state), rain-out, and condensate.

The vapor region is defined as a control volume on the top of the liquid phase. The non-uniformity of temperature distribution in the vapor phase depends on the transients, but its effect is not significant to the pressure response because its deviation from saturation is negligible in the Rankin Scale. That is the error evaluated from equation (1) is very small.

\[
\text{error} = \left( \frac{T_{\text{sat}} - T_{\text{sat}}}{T_{\text{sat}}} \right) \times 100 \%
\]  

The region set aside for the noncondensable gas may be stratified on the top of the vapor phase (for a lighter gas), or stratified on the bottom of the vapor phase (for a heavier gas), or concentrate at the walls impeding the condensation heat transfer (medium gas) (See Fig. 4.1.). When determining the molecular weight of the gas it is recommended that the average for the mixed noncondensable species be used.

The wall can be divided into three vertical control volumes; the cold wall, the hot wall, the vapor phase wall. The heat reaction of the each wall is only related with the corresponding control volume.

The following can reasonably be assumed.
1. All of each region is modeled as having a single value for each of the state variables describing it. That is temperature and pressure. The variable for a wall slab is temperature.

2. The state of the vapor phase is always saturated(3).

3. All the condensate on the wall and rainout is assumed to be deposited on the top of the hot liquid immediately.

4. The condensing interface is saturated.

5. All the spray is assumed to be saturated as soon as it enters pressurizer (4),(5).

5. When the steam is expanding in an outsurge transient, the state of steam is always saturated (not subcooled).

4.2 GOVERNING EQUATIONS.

In this section, the equations governing the pressure in the pressurizer will be derived as thoroughly as possible for various kinds of transients. It will then be simplified for predicting some of our experiments.

1) Mass Conservation Equation for an Open System.

\[
\frac{d}{dt} \int_{c.v} \rho \, dV + \int_{c.s} \rho \, \mathbf{V}_r \cdot \mathbf{n} \, dS = 0
\]  

(2)
2) Energy Equation.
(Deformable Control Volume for an Open System)

\[ Q - W = \frac{d}{dt} \int_{c,v} \rho \ e \ dV + \int_{c,s} \rho \ h \ \vec{V}_r \cdot \hat{n} \ dS + \int_{c,s} p \ \vec{V}_b \cdot \hat{n} \ dS \]  \hspace{1cm} (3)

3) Equation of State.

\[ h = e + p \ v \]  \hspace{1cm} (4)

\[ V = M \ v = \text{Const.} \]  \hspace{1cm} (5)

\[ v = v(p,h) \]  \hspace{1cm} (6)

Generally, if the heat and mass transfer rate for a control volume is known for a time interval from \( t \) to \( t + dt \), the new specific volume (\( v \)) and internal energy (\( e \)) or specific enthalpy (\( h \)) can be calculated by using the mass and energy conservation equations only (First Law). If these two variables (\( v \) and \( e \)) are known, then all of the other variables (\( p \), \( t \), and etc.) can be obtained easily from the equation of state. However, since the heat and mass transfer rate are not simple but complicated functions of these variable, one needs an analytical and numerical technique to solve the problem. Also to make this problem simple, some additional assumptions are sometimes necessary too, like equilibrium, or the existence of an isentropic process.

After some manipulations, the mass and energy conservation equations for each control volume become follows.
\begin{align}
\frac{d}{dt}(M_g) &= M_f + M_b + M_{gi} - M_{sc} - M_{we} - M_{ic} - M_{ro} - M_{go} \tag{7} \\
\frac{d}{dt}(M_{ac}) &= M_{st} + M_{ai} + M_{ao} \\
\frac{d}{dt}(M_{hl}) &= M_{sp} + M_{se} + M_{we} + M_{ic} + M_{ro} + M_{hli} - M_r - M_b - M_{hlo} \tag{9} \\
\frac{d}{dt}(M_{cl}) &= M_{cli} + M_{mx} - M_{clo} \tag{10} \\
\frac{dv}{dt} &= \left[ Q_g + (M_f + M_b - M_{sc} - M_{we} - M_{ic} - M_{ro}) (h_{sat} - h_g) \\
&+ M_{gi} (h_{gi} - h_g) - M_{go} (h_{go} - h_g) + V_g \frac{dp}{dt} - M_g \frac{\partial h_g}{\partial p} \frac{dp}{dt} \right] / M_g \frac{\partial h_g}{\partial v} \tag{11} \\
\frac{dv}{dt} &= \left[ Q_{nc} + M_{st} (h_{st} - h_{nc}) + M_{aci} (h_{aci} - h_{ac}) - M_{aco} (h_{aco} - h_{ne}) \\
&- M_{aco} (h_{aco} - h_{ne}) + V_{ne} \frac{dp}{dt} - M_{nc} \frac{\partial h_{nc}}{\partial p} \frac{dp}{dt} \right] / M_{ac} \frac{\partial h_{nc}}{\partial v} \tag{12} \\
\frac{dv}{dt} &= \left[ Q_{hl} + (M_{sp} + M_{se} + M_{we} + M_{ic} + M_r - M_t - M_b) (h_{sat} - h_{hl}) \\
+ M_{hli} (h_{hli} - h_{hl}) - M_{hlo} (h_{hlo} - h_{hl}) - M_{htr} (h_{htr} - h_{hl}) - M_x (h_{mx} - h_{hl}) \\
+ V_{hl} \frac{dp}{dt} + M_{hl} \frac{\partial h_{hl}}{\partial p} \frac{dp}{dt} \right] / M_{hl} \frac{\partial h_{hl}}{\partial v} \tag{13} \\
\frac{dv}{dt} &= \left[ Q_{cl} + M_{cli} (h_{cli} - h_{cl}) + M_x (h_x - h_{cl}) - M_{clo} (h_{clo} - h_{cl}) + V_{cl} \frac{dp}{dt} \\
- M_{cl} \frac{\partial h_{cl}}{\partial p} \frac{dp}{dt} \right] / M_{cl} \frac{\partial h_{cl}}{\partial v} \tag{14} \end{align}
From Eq. (4)

\[
\frac{dV}{dt} = v_s \frac{dM_s}{dt} + v_{ac} \frac{dM_{ac}}{dt} + v_{hl} \frac{dM_{hl}}{dt} + v_{cl} \frac{dM_{cl}}{dt} + M_s \frac{dv}{dt} + M_{ac} \frac{dv_{ac}}{dt} + M_{hl} \frac{dv_{hl}}{dt} + M_{cl} \frac{dv_{cl}}{dt} = 0
\]

(15)

Substitute Eq. (7), (8), (9), (10) and (11), (12), (13), (14) into Eq. (15), then

\[
\frac{dp}{dt} = \left[ A v_s + B v_{ac} + C v_{hl} + D v_{cl} + AA \frac{\partial v_s}{\partial h_s} + BB \frac{\partial v_{ac}}{\partial h_{ac}} + CC \frac{\partial v_{hl}}{\partial h_{hl}} + DD \frac{\partial v_{cl}}{\partial h_{cl}} \right] \div \left[ M_s \left( v_s - \frac{\partial v_s}{\partial p} \right) \frac{\partial v_s}{\partial h_s} \right]
\]

\[+ M_{ac} \left( v_{ac} - \frac{\partial v_{ac}}{\partial p} \right) \frac{\partial v_{ac}}{\partial h_{ac}} + M_{hl} \left( v_{hl} - \frac{\partial v_{hl}}{\partial p} \right) \frac{\partial v_{hl}}{\partial h_{hl}} + M_{cl} \left( v_{cl} - \frac{\partial v_{cl}}{\partial p} \right) \frac{\partial v_{cl}}{\partial h_{cl}} \]

(16)

And the equation for the walls are,

\[
\frac{dT_{wall}}{dt} = \frac{\alpha \Delta T}{\delta \rho C_p}
\]

(17)
where;

A is the sum of the mass transfer associated with the vapor phase, that is,

\[ A = M_f + M_b + M_{gi} - M_{sc} - M_{wc} - M_{l_c} - M_{ro} - M_{go} \]

B is the sum of the mass transfer associated with the noncondensable gases, that is,

\[ B = M_{st} + M_{gi} + M_{so} \]

C is the sum of the mass transfer associated with the hot liquid, that is,

\[ C = M_{sp} + M_{sc} + M_{wc} + M_{ic} + M_{ro} + M_{hli} - M_f - M_b - M_{hlo} \]

D is the sum of the mass transfer associated with the cold liquid, that is,

\[ D = M_{cli} + M_{mx} - M_{cio} \]

AA is the sum of the heat transfer associated with the vapor, that is,

\[ AA = Q_g + ( M_f + M_b - M_{sc} - M_{wc} - M_{ic} - M_{ro} ) ( h_{sat} - h_g ) + M_{gi} ( h_{gi} - h_g ) - M_{go} ( h_{go} - h_g ) \]

BB is the sum of the heat transfer associated with the noncondensable gases, that is,

\[ BB = Q_{nc} + M_{st} ( h_{st} - h_{nc} ) + M_{nci} ( h_{nci} - h_{nc} ) - M_{nco} ( h_{nco} - h_{nc} ) \]

CC is the sum of the heat transfer associated with the cold liquid, that is,

\[ CC = Q_{hl} + ( M_{sp} + M_{sc} + M_{wc} + M_{ic} + M_f - M_f - M_b ) ( h_{sat} - h_{hl} ) + M_{hli} ( h_{hli} - h_{hl} ) - M_{hlo} ( h_{hlo} - h_{hl} ) - M_{htr} ( h_{htr} - h_{hl} ) - M_{x} ( h_{mx} - h_{hl} ) \]
DD is the sum of the heat transfer associated with the hot liquid, that is,

\[ DD = Q_{cl} + M_{cl} ( h_{cli} - h_{cl} ) + M_x ( h_x - h_{cl} ) - M_{clo} ( h_{clo} - h_{cl} ) \]

From the Eq.(15) pressure can be calculated if terms A, B, C, D, AA, BB, CC, DD are known.

In the prediction of these experiments (Chap. 5), some terms of this equation can be eliminated because of their relative unimportance.

4.3 MODELLING.

4.3.1 Wall Condensation And Heat Transfer.

The condensation rate at the tank wall can be calculated from the energy balance at the boundary between the vapor phase and the tank wall. The heat transfer from the vapor phase to the tank wall is due to condensation or/and the natural convection. However, the heat transfer by the natural convection is quite negligible compared to the condensation, so the following equation can be written,

\[ M_{wc} = \frac{Q_{wc}}{h_f} \]  

(18)

The heat transfer rate to the tank wall can be calculated by trial and error method utilizing heat conduction equation and Nusselt Theory. The logic is as follows.

1. Solve the heat conduction at the tank wall with adequate boundary and initial conditions.

\[ \frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2} \]  

(19)
1) Boundary conditions.

\[ T(0,t) = T_{iw}(t) \quad (20) \]

\[ \frac{\partial T(X,t)}{\partial X} \bigg|_{x=b} = 0 \quad (21) \]

ii) Initial condition.

\[ T(X,0) = T_{in} \quad (22) \]

Then,

\[ T(X,t) - T_{in} = T_{iw} + \sum_{n=0}^{(-1)^{n+1}} \frac{4}{(2n+1)\pi} \cos \left( \frac{(2n+1)\pi X}{2\lambda} \right) \times \exp\left( -\frac{(2n+1)^2 t}{\lambda} \right) \times \int_0^t \exp\left( \frac{(2n+1)^2 \tau}{\lambda} \right) R(\tau) \, d\tau \quad (23) \]

2. From the solution the heat transfer rate can be calculated.

\[ dQ = A(t) \rho \, C_p \, T(X,t) \, dX \quad (24) \]

\[ Q(t) = \rho \, C_p \, A(t) \left[ T_{iw}(t) \, L - \sum_{n=0}^{(2n+1)^2 \pi^2} \frac{8L}{(2n+1)^2 \pi^2} \exp\left( -\frac{(2n+1)^2 X}{\lambda} \right) \times \int_0^t \exp\left( \frac{(2n+1)^2 \tau}{\lambda} \right) R(\lambda) \, d\lambda \right] \quad (25) \]

\[ \frac{dQ}{dt} \sim \frac{Q(t + \Delta t) - Q(t)}{\Delta t} \quad (26) \]
3. From the Nusselt Theory,

\[ Q = h \Delta T = 0.943 \sqrt{\frac{g \rho \Delta p k^3 h_{fg}}{L \mu}} \Delta T^3 \]  \hspace{1cm} (27)

4. Guess the inside wall temperature, then \( \Delta T \) is known because the inside film temperature can reasonably be assumed to be saturated.

5. Calculate the heat transfer rate based on Eq.2.26.

6. Calculate the heat transfer rate based on Eq.2.27.

7. Repeat 4., 5., and 6. until same heat transfer rate is obtained from 5. and 6. method. This is the answer.

In this analytical model, we neglected the temperature drop across the condensate film, that is, the inside wall temperature is assumed to be saturated. The heat transfer rate can then be calculated using (5) only. It may overpredict the heat transfer rate and result in underprediction of pressure within a few psia. This assertion can be verified by an order of magnitude calculation. The justification for this simplification is as follows. The stainless steel thickness that has a resistance equivalent to that of the condensate is 1/25 inch. Condensate film whose equivalent thickness of carbon steel is 1/4". Therefore the condensate film offers a negligible resistance in a 4" thick pressurizer.
4.3.2 Interface Condensation.

The heat transfer at the free surface can be accomplished by two mechanisms; one is by the conduction without fluid motion (conduction only) and the other is by convection due to fluid motion. The part full insurge is largely the former case and the empty insurge is the latter case.

When the pressurizer is filled with cold water from the bottom, the momentum of the cold insurge water cannot overcome the negative buoyancy force by the hot water present already so that the cold water stays at the bottom. The liquid is stratified. Heat transfer is quite negligible.

The mixing depth of cold water and hot water depends on the momentum of insurge water and negative buoyancy force of hot water. The depth can be calculated by the ratio of inertia force to negative buoyancy force. In our experiment and actual pressurizer, the penetration depth would be about 70", and 15", respectively[6]. This length is not negligible compared with the whole pressurizer length. However, the inlet of actual pressurizer as well as our experiment has an inlet distributor which dissipates the insurge momentum so that the penetration depth is much smaller than that calculated from a direct jet (See Fig. 4.2). The stratification in the liquid region is the rule rather than the exception.

In an actual PWR pressurizer as mentioned in the Sec. 2.1, there is a distributor (See Fig. 4.3) to reduce the incoming momentum so that the jet penetration is very small (about 6""). Due to this, the
Figure 4.2 Pressure vs Initial Hot Water Depth
cold water and hot water are just like a solid piston which has a thermal conductivity of water. In this case the heat transfer can analytically be calculated by solving the conduction equation as following.

1. Solve the conduction equation with boundary and initial conditions.

\[
\frac{\partial T(x,t)}{\partial t} = \alpha^2 \frac{\partial^2 T}{\partial x^2} \tag{28}
\]

i) Boundary conditions.

\[
T(0,t) = T_{\text{sat}} \tag{29}
\]

\[
T(\infty,t) = T_{\text{in}} \tag{30}
\]

ii) Initial condition.

\[
T(x,0) = T_{\text{in}} \tag{31}
\]

Then,

\[
T(x,t) = \frac{x}{\sqrt{4 \pi \alpha}} \int_0^t T_{\text{sat}}(\lambda) \frac{\exp\left(-\frac{x^2}{4\alpha(t-\lambda)}\right)}{(t-\lambda)^{1.5}} d\lambda \tag{32}
\]

2. From the equation (32) calculate the heat transfer rate.

\[
Q = 2\rho C_p A \sqrt{\frac{\alpha}{\pi}} \int_0^t \sqrt{t-\lambda} T_{\text{sat}}(\lambda) d\lambda \tag{33}
\]

\[
\frac{dQ}{dt} \sim \frac{Q(t + \Delta t) - Q(t)}{\Delta t} \tag{34}
\]
a). Yankee Power Plant

b). Millstone 2 Power Plant

c). B & W and M.I.T. Experiment.

Figure 4.3 Different Sparger Geometries.
Figure 4.4 Free Surface Heat Transfer for Empty Insurge.
In this experiment the heat transfer rate at the interface is negligible if the hot water level is higher than the "Effective Depth". The depth is dependent on the insurge water velocity and the sparger geometry. The interface heat transfer rate is only 2-3 percent of that by the wall condensation when the water level is higher than the effective depth (> 10 inches).

On the other hand, when the water level is below the effective depth, the insurge momentum can penetrate the hot liquid or disturb the interface. It increases the heat transfer tremendously(7) (See Fig. 4.4).

4.3.3 Flashing.

During the outsurge transient, if there is saturated water in pressurizer, the pressure is dominated by flashing. The amount of energy available to the flashing is equal to the superheated energy due to the depressurization.

\[ \frac{dE}{dt} = M_{hl} C_p \frac{dT}{dt} = dM_f h_{fg} \]  

\[ \frac{dM_f}{dt} = \frac{M_{hl} C_p}{h_{fg}} \frac{dT}{dt} = A(cs) L_{hl} \rho C_p T_{sat} v_{fg} / h_{fg}^2 \frac{dp}{dt} \]  

4.3.4 Rainout.

From the mass conservation equation, we obtained the follows,

\[ \frac{d}{dt} \left( M_z \right) = \frac{d}{dt} \left( \frac{V_g}{v_g} \right) = -1 / v_g \{ V_{srg} + M_g \left( \frac{\partial v_g}{\partial p} \right)_T \frac{dp}{dt} + \frac{\partial v_g}{\partial T} \frac{dT}{dt} \} \]
If the process occurs along the saturation line then, from Clapeyron Equation,

\[
\frac{dT}{dp} = \frac{T_{\text{sat}} v_{fg}}{h_{fg}}
\]  

(37)

and finally,

\[
\frac{M_{ro}}{M_g} = \frac{V_{srg}}{V_g} + \frac{1}{v_g} \left( \frac{\partial v_g}{\partial p} |_T + \frac{T_{\text{sat}} v_{fg} \partial v_g}{h_{fg} \partial T |_p} \right) \frac{dp}{dt}
\]  

(38)

If flashing or some other process are involved, this equation can be easily modified as follows,

\[
\frac{M_{ro}}{M_g} = \frac{M_b + M_{fg} \ldots}{m_g} + \frac{V_{srg}}{V_g} + \frac{1}{v_g} \left( \frac{\partial v_g}{\partial p} |_T + \frac{T_{\text{sat}} v_{fg} \partial v_g}{h_{fg} \partial T |_p} \right) \frac{dp}{dt}
\]  

(39)

4.3.5 Condensation On The Spray Drops.

From many other experiments, it was found that the spray drops get saturated very quickly by the direct contact condensation when the water enters the pressurizer(4), (5). Experimentally it has been found that almost 100% of thermal equilibrium is obtained within a few feet from the nozzle. It has also been found that the droplet diameter was the most important parameter affecting the degree of thermal equilibrium attained.

From those observations, it can reasonably be assumed that the spray droplets become saturated soon after they enter the pressurizer.

The energy balance equation at the each spray droplet can be lumped together into one simple equation. The equation is following.

\[
M_{sp} = \frac{\left( h_{\text{sat}} - h_{sp} \right) \dot{M}_{sp}}{h_{fg}}
\]  

(40)
4.3.6 Boiling.

Boiling in the pressurizer can occur at the wall and/or the heaters. The former is called "Wall Boiling" and the latter "Heater Boiling". The wall boiling can occur during the outsurge transient because the temperature at the wall may be higher than saturated state during an outsurge depressurization. The superheated energy of the wall is

\[
dE = (\rho C_p V)_{wall} \Delta T
\]

(41)

Therefore the boiling mass

\[
M_b = \frac{1}{h_{fg}} \frac{dE}{dt} = \frac{\dot{P}}{h_{fg}} \left(\rho C_p V\right)_{wall} T_{sat} v_{fg}
\]

(42)

The heater boiling can be easily be modelled(δ).

\[
M_{htr} = \frac{Q_{htr}}{h_{fg}} \left(1 - e^{-rt}\right)
\]

(43)

4.3.7 The Non-condensable Gases.

It is well known that even in normal operation, there may be a significant amount of noncondensable gases in pressurizer. This can affect the pressure response drastically. The heating of the insurge coolant, spray, or dribbling to the saturation state can cause the noncondensible gas desolved in the coolant to be stripped and accumulate. These gases are composed of hydrogen and some fission products. In accidents, nitrogen gas can be discharged into the reactor system from the accumulator after the water inventory depleted.

Even small amounts of noncondensables can sometimes affect heat transfer so that we must look into the effect of noncondensables in
this project.
A noncondensable gas model has been proposed based on the Leonard's experiments (9). The experiments show that the pressure response is dependent on the amount of the noncondensable gas as well as its molecular weight. The dependency on molecular weight can be explained as it affects the amount and effect of stratification of the gas. That is, the lighter gas tends to accumulate at the top of the vessel while the heavier gas tends to accumulate at the water surface. Therefore the lighter gas control volume is on the top of the vapor region while the heavy one is on the bottom.

The p-v relation for the gas phase is not significantly affected by the noncondensibles until they exceed several percent.

On the other hand, the heavier gas which occupies the bottom of the vapor region blocks the interface. So only a few percent of heavy gas can eliminate all the interface condensation even for an empty tank insurge.

For the intermediate molecular weight gas (like N₂), which mixes well with vapor, it occupies its volume as well as blocking wall and interface condensation.

The noncondensable gases can be modelled as an ideal gas without any serious problem. The main question in modelling of noncondensable gases is that how can we make a quantitative correlation between the amount of the noncondensable gases in the pressurizer and heat transfer rate. For this problem, refer to Leonard's thesis(9).

4.3.8 Heat Transfer Between The Liquid And The Wall.

There are four different kinds of heat interaction between the liquid and the wall. i) The heat interaction between the cold insurge water and the hot tank wall (in the lower part of pressurizer during insurge transients). ii) The heat interaction between the subcooled hot water and the saturated hot wall (near the boundary region of the vapor and the hot liquid during insurge transients). iii) The heat
interaction between the saturated hot water and the cold wall (near
the boundary region of hot water and cold water during outsurge
transients). iv) The heat interaction between the saturated hot water
and the superheated tank wall (in the hot water region during outsurge
transients with wall boiling).

For the case iv), it was already mentioned in the wall boiling (Sec
3.3.6). And for the case ii), not only the amount of heat is small
but the effect on the pressure is negligible. However, even though
the heat interaction of case i) has no effect on the pressure directly
since the volume change of the liquid due to enthalpy increase, later
on when the outsurge starts with flashing the degree of the wall
cooling by the case i) is quite sensitive to the heat interaction of
case iii).

Let us discuss the details of this phenomena further. When the
cold water comes from the bottom during insurge, the hot wall loses
heat to the cold water. After that when outsurge starts, the hot
water comes down and makes contact with the cold wall cooled already.
When the system pressure is higher than that of saturation temperature
of hot water, the hot water just loses heat without any significant
pressure effects. However when the flashing starts, the heat loses to
the cold wall suppresses the flashing. This will be called "
Suppression of Flashing". For the case i) a forced convection heat
transfer coefficient can be used. For the case iii), on the other
hand, the selection of heat transfer coefficient quite ambiguous. But
by the observation of the tank wall temperature it is around 1000
BTU/FT²hr F. The reason is thought to be there is a lot of
agitation due to flashing in the bulk of the liquid.
4.3.9 Axial Heat Transfer Through The Wall.

There may be axial heat transfer down the wall when the cold insurge stratifies under the hot water or the steam on the top of the water is hotter than the water. This gives a significant temperature difference across short distance. The necessary condition for which this heat transfer to be significant is that the heat transfer coefficients along the inside of the wall, in both the hot and cold region, should be order of thousand(STU/FT\(^2\)hr F). In the actual pressurizer, this situation is impossible. Therefore the amount of heat transfer in this way is negligible compared to that by the other ones. This assertion can easily be verified by the following. The heat transfer from the hot region to the interface

\[
q = \sqrt{h_h \ p \ k \ s \ \Theta_{bh} \ tanhB_1L_1}
\]  

(44)

The heat transfer from the interface to the cold region

\[
q = \sqrt{h_c \ p \ k \ s \ \Theta_{bc} \ tanhB_2L_2}
\]  

(45)

At best, we can assume that \(tanB_1L_1\) and \(tanB_2L_2\) are 1. then

\[
\Theta_{tb} = q \left( 1. /\sqrt{h_c \ p \ k \ s} + 1. /\sqrt{h_h \ p \ k \ s} \right)
\]  

(46)

\[
q \sim \Theta_{tb} \sqrt{p \ k \ s} \ Min \left( \sqrt{h_c}, \sqrt{h_h} \right)
\]  

(46)
For Example, for the case of follows;

\[ P = 80.0'' = 20.9' \]
\[ k = 23 \text{ Btu/hr ft F} \]
\[ s = 5.0'' \times P = 10.5 \text{ ft}^2 \]
\[ h = 400 \text{ Btu/hr ft}^2 \text{ F} \]
\[ \Theta_{tb} = 100 \text{ F} \]

then approximately we get 30,000 btu/hr. This magnitude is of the same order of the heat loss of pressurizer to the environment. For a very slow transient, consideration of this heat transfer mode may be desirable. It may also be important to have a model that reduces to a uniform temperature for large times and this is probably the most significant heat transfer to a stratified pool.
CHAPTER 5

COMPARISON AND DISCUSSION.

5.1 THE METHOD OF PREDICTION.

After the elimination of the irrelvant and negligible terms in Eq. 4.1, and some manipulations, the governing equation can be simplified to the following,

\[
\frac{dP}{dt} = \\
\frac{\sum Q_g / M_g + h_{fg} V_{srg} / V_g + (- M_{we} - M_{spc} - M_{ro} + M_b + M_f) h_{fg} / M_g}{(1 - \frac{h_{fg} \partial v}{v_g \partial h}) \left( \frac{C_p T_{sat} v_{fg}}{l_{fg}} + \frac{\partial h}{\partial p} \right) - (v_g + \frac{h_{fg} \partial v}{v_g \partial p})}
\]

(42)
In the above equation, $\Sigma \hat{Q}$ and $\Sigma \hat{N}$ are the implicit function of $\hat{p}$ (Sec. 4.2.). To find a solution of above equation, the half-interval root finding technique was used (other methods are also usable).

5.2 DISCUSSION.

As shown in the Fig. 5.1., Fig. 5.2., Fig. 5.3., Fig. 5.4., Fig. 5.5., Fig. 5.6., and Fig. 5.7., our experiment data and some PWR power plant transient cases were compared with our predictions. The predictions of the part full insurge experiment (Fig. 5.1), simple outsurge experiment (Fig. 5.2.), insurge to the hot wall experiment (Fig. 5.3.) show that our model is quite accurate and reliable. However, the prediction of the insurge after outsurge experiment (Fig. 5.4.), empty tank insurge experiment (Fig. 5.5.), and outsurge after insurge transient of PWR power plant (Fig. 5.6 and Fig. 5.7.) show some deviation from the expermental data. The deviations come from the uncertainty in the state of the steam. In the expansion process of steam, the final steam of state can be determined assuming either an adiabatic process or an isentropic process. The actual process is between this. According to other study (10), the actual process yields thermodynamic states which are 30% close to that predicted by the adiabatic process than by the isentropic process.

The deviation of the actual state from the mean of the adiabatic and isentropic states was defined as follows:

$$\frac{\Delta T_{is} - \Delta T_{sat}}{\Delta T_{sat}} = \text{Deviation}$$

(49)

Therefore, the actual state can be found as follows:

$$T = \frac{T_{is} + 1.3 \ T_{sat}}{2.3}$$

(50)
Figure 5.1 Prediction of the Pressure of Partially Full Tank Insurge Experiment.
Figure 5.2: Prediction of the Pressure for the Outsurge Experiment.
Figure 5.3 Prediction of the Pressure for the Insurge to a Hot Tank Insurge.
Figure 5.4 Prediction of the Pressure for an Insurge after Outsurge Experiment
Figure 5.6 Prediction of the Pressure for the Connecticut Yankee Nuclear Power Plant Transient (Sept., 1980)
Leonard's Experiments show that the N.C gases tend to stratify depending on the molecular weight. Compared with the steam molecular weight, light gases tend to stay on the top of the steam, heavy N.C gases tend to stay on the top of the liquid, and medium gases stay mixed with the steam.

However, thermodynamically the stratification of noncondensable gases by a gravitational field in short distance like pressurizer is almost impossible.

In spite of this, the stratification is quite possible because in the pressurizer there is a lot of flashing, boiling, condensation, and insurge and outsurge. All these processes involve heat transfer and enhance the separation. This disturbance can enhance stratification. For more information on this question, refer to Leonard's thesis(9).

5.3 SCALE EFFECTS COMPARING LARGE, HIGH PRESSURE AND SMALL, LOW PRESSURE SYSTEMS.

The goal of this experiment is to identify the important phenomena and quantify the mass and heat transfer rates associated with these phenomena from experiments performed on a small, low pressure system. The small, low pressure experiment has the advantages of low cost, good control of boundary conditions, maneuverability, and exaggeration of some effects. Because of this exaggeration, it is necessary to look at the results of these experiments in dimensionless terms.

5.3.1 Mass And Heat Transfer Associated With The Wall.

The effect of scale on the mass and heat transfer associated with the wall can be analyzed by comparison of the heat capacity of the wall to the latent heat capacity of the vapor for the large, high pressure system and small, low pressure system.
The heat capacity of the wall,

$$2\pi RH(\delta \rho C_p)_{wall} \Delta T$$

The heat capacity of the vapor phase,

$$\pi R^2 H(\rho u_g)_{stm}$$

The ratio of heat capacity (R.H.C) of the wall to that of the vapor,

$$\text{(R.H.C)} = \frac{2\delta(\rho C_p)_{wall}\Delta T}{(\rho u_g)_{stm}}$$

R.H.C for high pressure is 0.066

R.H.C for low pressure is 0.55

Therefore it can be concluded that the effect of wall condensation in a high pressure full scale system is about one tenth of that in the low pressure system. This can easily be checked by comparing Fig. 5.1 and Fig. 5.6.

In the same way, the R.H.C of the wall in the liquid region (this region is associated with the wall boiling and the suppression of flashing) shows a similar scale effect.

5.3.2 Mass And Heat Transfer Associated With The Interface Condensation.

The importance of the mass and heat transfer associated with the interface can be analyzed by comparing the total amount of heat transferred to the liquid from the vapor to the total energy stored in the vapor phase for small, low pressure system and full, high pressure system.

However, since the interface heat transfer is significant only when
the water level is lower than the sparger height, this scale effect analysis is only applicable to the empty tank insurge transient. When the water level is higher than the sparger height, the heat transfer at the interface is negligible.

An approximation for the total amount of heat transfer to the liquid is

\[ Q = \int_{0}^{t} h(t) A \Delta T(t) \, dt = \pi R^2 \frac{L_{pen}}{v} h_a \Delta T \]

The heat transfer coefficient are dependent on the transient characteristics (such as the insurge rate, the state variable) and the pressurizer geometry (in particular, sparger geometry).

For the B&W sparger geometry at the typical insurge flow rate of a PWR,

\[ St = 0.02 \]

\[ G = 38,000 \text{ lbm/ft hr (at the sparger outlet).} \]

\[ \Delta T = 50 \text{ F} \]

\[ t = L_p/v = 18''/0.6 \text{ft/sec} = 30 \text{ sec} \]

\[ Q = h_a A \Delta T t = 625 \text{ Btu} \]

\[ E_t = A h_a u_g \rho \]

\[ \frac{Q}{E_t} = \frac{h_a \Delta T L_p/v}{H \rho u_g} / 3600 \]

\[ (Q/E_t)_h = 0.003 \]

\[ (Q/E_t)_l = 0.2 \]
Therefore we can conclude that the pressure drop in the high pressure full-scale system due to the interface heat transfer will be only one senventieth of that of low pressure system.

5.3.3 Penetration Depth.

The scale effect on the penetration depth can be analyzed by comparing the Froude Number for the high pressure system and the low pressure system.

The Froude Number for the low pressure system is,

\[ Fr = \frac{V_0^2 \rho}{g L \beta \Delta T \rho} \sim 6 \]

The Froude Number for the high pressure system is,

\[ Fr = \frac{4^2 \times 40}{32 \times 5 \times L \rho} \sim 6 \]

Therefore the depth at which stratification is important in a high, full scale system is 17".

5.4 RELATIVE IMPORTANCE OF THE INDIVIDUAL PHENOMENA.

The purpose of this section is to analyze the relative importance of the individual phenomena in order to guide future research efforts. The energy stored in the vapor phase was chosen for the reference of the comparison since that energy mostly governs the pressure behavior in the pressurizer. In addition, the duration of the transient was selected as 100 sec since that period is typical for a PWR transient.
5.4.1 Wall Condensation.

The heat capacity of the pressurizer wall is,

\[ 2\pi RH\delta (\rho C_p)\Delta T \]

\[ \Delta T = T(x) \, dx/L \]

\[ \Delta T = 10 F \]

The heat capacity of the steam is,

\[ \pi R^2 H (\rho C_p u_g) \]

Relative Importance (R.I) \( w_c \) is,

\[ \frac{2\pi RH \delta (\rho C_p)_w \Delta T}{\pi R^2 H (\rho u_g)_{stm}} = \frac{2\delta C_p \Delta T}{R(\rho u_g)_{stm}} = 0.024 \]

5.4.2 Interface Heat Transfer (Empty Tank Only)

The total heat transferred from the vapor to the liquid for time which the water level reaches the effective depth is

\[ Q_{in} = \int_0^\tau A \Delta T(t) \, h(t) \, dt \]

where; \( \tau \) is the duration time of empty insurge, that is, it is the time it takes the water level to reach the effective depth.

\[ \Delta T(t) \sim \text{const.} \]

\[ \int_0^\tau h(t) \, dt = h_a \tau = \frac{L_p}{v} h_a \]

where; \( v \) is the velocity of the water front in the pressurizer.
\[ h_a = 1500 \text{ Btu/ft hr F} \]

Therefore \((R.I)_{ic} = \frac{A \Delta T L_p/v h_a}{A H(\rho u)_{stn}} = \frac{\Delta T L_p/v h_a}{H(\rho u)_{stn}} = 0.006 \]

5.4.3 Heater

The total power of heater is 1300 kw.

\[(R.I)_h = \frac{1300/1.055}{\pi R^2 H \rho u_g} \times 100 = 0.038 \]

5.4.4 Condensation On The Spray Drops.

The total heat transferred to the liquid from the vapor for the transient by the condensation on the spray drops is,

\[ Q_{spc} = m_{sp} (H_{sat} - H_{sp}) \]

The importance of the condensation on the spray drops is,

\[(R.I)_{sp} = 0.076 \]

5.4.5 Wall Boiling And The Suppression Of Flashing

As shown in the section on wall condensation, the wall heat transfer for boiling and single phase has same order of importance as it does for the wall condensation.
5.4.6 Flashing.

The relative importance of flashing can be calculated by the ratio of the total energy or mass transferred to the hot liquid from the vapor phase to the total mass or energy in the vapor phase. The former is equal to the total superheated energy in the hot liquid due to the depressurization during the outsurge transient period.

\[
(R.I)_{fl} = \frac{\pi R^2 (H \rho c_p \Delta T)_{hl}}{\pi R^2 H (\rho u)_{stm}} = \frac{\rho_{hl} c_p \Delta T_{hl}}{\rho_g u_g} = 0.216
\]

5.4.7 Rainout.

The mass of rainout for the period of transient can be approximated as follows:

For a typical PWR transient the initial condition of the pressurizer before start of rainout is about 50% water (2200 psia) and final condition is about 40% water (1850 psia). So rainout is,

\[
M_{ro} = V_2 / v_2 - V_1 / v_1
\]

\[
(R.I)_{ro} = \frac{M_{ro} h_{fg}}{\pi R^2 H \rho u_g} = 0.06
\]

Therefore if we normalize the relative importance of the flashing as 1., the relative importance of the various pressurizer processes is as follows;

<table>
<thead>
<tr>
<th>Process</th>
<th>Normalized Relative Importance</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flashing</td>
<td>1.0</td>
</tr>
<tr>
<td>Spray Condensation</td>
<td>0.3</td>
</tr>
<tr>
<td>Rainout</td>
<td>0.3</td>
</tr>
<tr>
<td>Description</td>
<td>Value</td>
</tr>
<tr>
<td>------------------------------</td>
<td>--------</td>
</tr>
<tr>
<td>Heater Input</td>
<td>0.15</td>
</tr>
<tr>
<td>Wall Condensation</td>
<td>0.1</td>
</tr>
<tr>
<td>Wall Boiling</td>
<td>0.1</td>
</tr>
<tr>
<td>Suppression of Flashing</td>
<td>0.1</td>
</tr>
<tr>
<td>Interface Condensation</td>
<td>0.025</td>
</tr>
</tbody>
</table>
CHAPTER 6

SUMMARY.

6.1 SUMMARY

Based on our simple model, the prediction of the pressure behavior during the transient is quite reliable and accurate. Our model are summarized in Fig. 6.1. and Table 6.1.

6.2 THE EXPLANATION OF HEAT TRANSFER MATRIX.
Figure 6.1 Mass and Heat Transfer in the Pressurizer.
Table 6.1 Mass and Heat Transfer Matrix in the Pressurizer.

<table>
<thead>
<tr>
<th>PHENOMENA</th>
<th>CONDITION</th>
<th>HEAT TRANSFER</th>
<th>MASS TRANSFER</th>
<th>HEAT TRANSFER COEFF</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>T_w &gt; T_sat</td>
<td>Q_{ncg} = H_{ncg}(T_w - T_g)</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td>HEAT TRANSFER</td>
<td>L_w &lt; L</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TO THE WALL</td>
<td>T_w &lt; T_sat</td>
<td>Q_{wcd} = H_{wcd}(T_g - T_w)</td>
<td>M_{wcd} = \frac{U_{wcd}}{H_{fg}}</td>
<td></td>
</tr>
<tr>
<td></td>
<td>T_w &gt; T_sat</td>
<td>Q_b = H_b (T_w - T_b)</td>
<td>M_b = \frac{Q_b}{H_{fg}}</td>
<td></td>
</tr>
<tr>
<td></td>
<td>L_w &gt; L T_k &lt; T_w &lt; T_sat</td>
<td>Q_{tc} = H_{tc} (T_w - T_k)</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td></td>
<td>T_k &gt; T_w</td>
<td>Q_{tc} = H_{tc} (T_k - T_w)</td>
<td>0</td>
<td></td>
</tr>
<tr>
<td></td>
<td>L_w &gt; L_eff</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>HEAT TRANSFER</td>
<td>L_{pen} &lt; L &lt; L_eff</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>AT INTERFACE</td>
<td>L_{spa} &lt; L &lt; L_{pen}</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>L_w &lt; L_{spa}</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>CONDENSATION</td>
<td>T_{sp} &lt; T_sat</td>
<td>Q_{sp} = M_{sp}(T_{sat} - T_{sp})C_p</td>
<td>M_{esp} = \frac{Q_{sp}}{H_{fg}}</td>
<td></td>
</tr>
<tr>
<td>AT SPRAY DROP</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>FLASHING</td>
<td>T_{so} &gt; T_sat +1</td>
<td>Q_{so} = M_{so} H_{fg}</td>
<td></td>
<td></td>
</tr>
<tr>
<td>HEATER</td>
<td></td>
<td>Q_{htx} = Q_{ele}(T - \tau)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

\[ St = St \left( \frac{L}{2} \right) = \frac{m}{m} \left( \frac{L}{2} - 1.6 \right) + 0.0115 \]
\[ where \ -0.0048 \leq m \leq -0.0022 \]
\[ St = St \left( \frac{L}{2} \right) = -0.044 \frac{L}{2} + 0.082 \]
\[ St = St \left( \frac{L}{2} \right) = -0.015 \frac{L}{2} + 0.053 \]
\[ H_{sp} = \infty \]
\[ A_{n_l} \frac{L_l \rho C_p T_{so} v_g}{b_{la} \frac{dp}{dt}} \]
6.2.1 Heat And Mass Transfer On The Wall.

There may be five different kinds of heat transfer associated with the tank wall. i) Natural convection in the vapor/wall interface. ii) Wall condensation in the vapor/wall interface. iii) Wall boiling in the liquid region during outsurge. iv) Forced convection heat transfer (wall cooling) in the lower part of the cold liquid region during the cold water insurge. v) Forced convection in the lower part of hot hot liquid region during the cold liquid outsurge.

For the case i), it usually occurs when the vapor temperature is lower than the wall temperature during the outsurge or hot tank insurge. Since the heat transfer rate by natural convection is much less than that of the tank to the environment, the heat transfer rate is negligible.

For case ii), it happens when the vapor temperature is higher than the wall temperature (more exactly the saturation temperature corresponding to partial vapor pressure should be higher than the wall temperature) during insurge transients.

The condensation heat transfer rate based on Nusselt's theory may cause some trouble. First, the theory assume a steady state, however, the processes in the pressurizer are not steady. Second, to calculate the heat transfer rate, the wall inside temperature should be known. Even if the temperature can be calculated by solving conduction equation in the wall, the calculation itself needs long computing time. So it is recommended that the wall inside temperature be assumed as the saturated one then, when solving the conduction equation in the tank wall. From this solution the heat transfer rate can be calculated. This method is more easy and fast (Sec. 4.3.1).

If the noncondensible gas fraction is large enough to affect the wall condensation, the assumption that the wall inside temperature is saturated is not valid. Therefore this calculation scheme is not useful. In this case, Nusselt's theory considering the existence of noncondensables should be used.

For the case iii), the wall boiling can occur when the wall temperature in the liquid region is higher than saturation temperature
of the system during the outsurge transients. For the same reason as in the case ii), the inside wall temperature should be assumed as saturated. The error due to this assumption is negligible (See Sec. 4.3.1). Also in the same way, the heat transfer rate can be calculated.

For the case iv), this heat transfer can occur when the cold water rushes into hot tank during insurex transient. The insurex cold water accompanies some turbulent characteristics generated during the passage of the surge pipe. Also Reynolds number in the pressurizer belongs to turbulent regime. Therefore heat transfer coefficient of the forced convection in tube was used. However that is questionable.

For the case v), this heat transfer can occur when the saturated hot water contacts the cold wall cooled already by the heat transfer of case iv) during the outsurge after cold water insurex. The heat transfer coefficient of turbulent flow over flat plate was used because the diameter of the tank is large enough and Reynolds number in the tank belongs to the turbulent regime.

6.2.2 Heat Transfer At Interface(7).

Heat Transfer at the interface is governed by the nature of the incoming turbulent jet, the water depth, and the sparger geometry. According to these facts, the heat transfer coefficient can be divided four regions. i) Heat transfer by conduction only (region 4). ii) Heat transfer by the turbulent jet plume (region 3). iii) Heat transfer by direct turbulent jet when the water level is same as sparger height (region 2). iv) Heat transfer by direct turbulent jet when the water level is below the sparger height (region 1).

The condition for the region 4 is that the water level should be higher than the level at which incoming turbulent jet plume could not affect the interface heat transfer. It would be called "Effective Depth". In this situation, the water is like a solid piston. So the heat can be transferred by the conduction only (See Sec.4.3.2).
Since the water conductivity is small, the heat transfer of this region is almost zero. All of the pressurizer operation during the normal condition belongs to this region.

The condition for the region 3 is that the water level should be higher than the penetration depth but lower than the effective depth. In this situation, the incoming turbulent jet is stratified from the hot water at the interface, but still the incoming turbulent jet affects the interface heat transfer. In this region, the heat transfer coefficient is function of dimensionless water depth, and the Froude Number or the Richardson Number.

The condition for region 2 is that the water level is higher than the sparger height but lower than the penetration depth. When the water level is little higher than the sparger height, the incoming jet momentum can penetrate thin hot layer at the interface. So there is no stratification. In this case, the heat transfer coefficient is function of water depth and flow rate only, not a function of Froude Number.

The condition for region 1 is that the water level is below the sparger height. In this case, the heat transfer coefficient has same trend as that of the region 2 but the magnitude of the region 1 is larger than that of the region 2.

6.2.3 Condensation On The Spray Drops.

As mentioned in the Sec.4.3.5, the assumption that the spray water immediately gets saturated as it enters the vapor region of pressurizer is very plausible.
6.2.4 Rainout.

The amount of rainout was derived from the assumption that the vapor phase expands adiabatically.

6.2.5 Flashing.

The amount of the flashing was derived based on the assumption that all the superheated energy of the water due to the depressurization contributes to the flashing.

6.2.6 Heater.

The heaters have a time constant. So the heat input form the heaters is expontially increasing or decreasing even if their power is on and off as step function. Therefore the power from the heater can roughly be approximated step function lagged by heater time constant. The typical time constant is 1 to 3 minutes.

6.2.7 Heat Transfer Axially Through The Wall.

The heat transfer through the wall from the hot region to the cold region can be approximated as a pin problem, that is one side of the pin absorbs heat (hot side) and the other side discharges heat (cold side). The amount of heat transfer through the wall is same order of the wall heat loss to the environment. Therefore in slow transients, it has some signifincance, but in fast transients it does not.
6.3 CONCLUSIONS.

6.3.1 Part Full Insurge.

1. The wall condensation has some effect on the pressure.

2. Heat transfer at the interface is negligible if the water depth is enough.

3. Heat transfer axially through the wall is negligible.

4. There is a stratification between the hot water and the cold insurge water.

6.3.2 Outsurge

1. Flashing of the saturated liquid as well as rainout control the pressure response.

2. The vapor phase is always at the saturated state.

3. Heat input from the wetted wall is important.

4. The liquid is only very slightly superheated (.5 to 1 F).

5. At first, the temperature of the top liquid is higher than that of at the bottom, but as the outsurge goes on, the temperature profile reverses. All these temperature differences are quite small, however, and can be ignored.
6.3.3 Outsurge After Insurge.

1. When the steam expands without flashing, the deviation from the saturated state is significant. The actual state can be found as follows:

\[ T = \frac{T_{1/2} + (1 + \text{Deviation}) T_{\text{sat.}}}{2 + \text{Deviation}} \]

2. Suppression of flashing by the heat transfer from the hot water to the cold wall (cooled down by cold insurge water) is important.

3. The non-uniformity of temperature in the vapor phase is not important.

6.3.4 Empty Tank Insurge.

1. An appropriate Stanton Number can be used to calculate the free surface heat transfer coefficient.
2. Geometry of the sparger at the pressurizer inlet is very important.

3. The depth of the water is important in that it affects the Stanton Number.

4. No additional water temperature effects are evident.

5. When the pressure tank rises again after the sharp drop, the vapor compression process appears to be adiabatic.

6. Noncondensable gases can practically stop free surface heat transfer if the noncondensable gases have molecular weight much greater than 18.

6.3.5 Non-condensable Gas Effect.

1. The effects of the non-condensable gases can be significant. In particular, interface and wall condensation is quite sensitive to the existence of noncondensables. Light noncondensable gases congregate at the top of the pressurizer and have very little effect. Moderate weight noncondensable gases (mw-18) impede the wall heat transfer. Heavy noncondensable gases congregate on the free surface and practically stop any free surface heat transfer even for an insurge into an empty tank.

2. The ratio of the molecular weight of N.C to that of the water has a significant effect on the pressure response of pressurizer.
6.4 RECOMMENDATIONS.

The followings are required further investigation.

1. Further experiments to identify the cause of the error in the steam expansion without flashing.

2. Specification of the heat transfer coefficient between the hot water and the cold wall in the outsurge after insurge transients.

3. The correlation of the wall heat transfer coefficient depending on the molecular weight of noncondensable gases.

4. Incorporation of our model into an advanced reactor analysis code and systematic refinement of the model.
References.


10. Melker P.de and Latzko D.G.H.,"Digital Analysis of Pressurizer Transients and Comparison with Experimental Results",Technological Univ. of Delft, the Netherlands.
Appendicies

Appendicies A through E are a complete reporting of the data taken for a number of different transients. This data has been compared to the recommended calculations in the body of the thesis. It is presented here in completed detail so that one can compare their pressurizer model to the data taken in this program.

The experiments are as follows.

<table>
<thead>
<tr>
<th>Transient</th>
<th>Appendicies</th>
</tr>
</thead>
<tbody>
<tr>
<td>Partially Full Insurge Experiment</td>
<td>A</td>
</tr>
<tr>
<td>Outsurge Experiment</td>
<td>B</td>
</tr>
<tr>
<td>Insurge to a Hot Tank Experiment</td>
<td>C</td>
</tr>
<tr>
<td>Outsurge after Insurge Experiment</td>
<td>D</td>
</tr>
<tr>
<td>Empty Tank Insurge Experiment</td>
<td>E</td>
</tr>
</tbody>
</table>
APPENDIX A.

PART FULL INSURGE EXPERIMENT.

Initial Pressure 101.0 psia
Initial Water Level 13.9"
Final Water Level 34.0"
Insurge Water Temp. 75.0 F

Figure A.1 Flow Rate vs Time ( \dot{M} - t ).
Figure A.2 Pressure vs Time ( P - t ).
Figure A.3 Temperature vs Axial Position of the Centerline ( T - z ).
Figure A.4 Temperature vs Axial Position of the Wall Outside ( T - z ).
Figure A.5 - A.17 Temperature vs Time at the Center ( T - t ).
Figure A.18 - A.25 Temperature vs Time on the Wall ( T - t ).
*Ignore the double spikes, they were due to an uncorrected error in the DAS.
Figure A.3 Axial Tank Temperature.
Figure A.9 Temperature vs Time at the Centerline (16"")
Figure A.10 Temperature vs Time at the Centerline (19")
Figure A.11 Temperature vs Time at the Centerline (22")
Figure A.14 Temperature vs Time at the Centerline (31"")
Figure A.15 Temperature vs Time at the Centerline (34")
Figure A.16 Temperature vs Time at the Centerline (37")
Figure A.19 Temperature vs Time on the Wall (10"")
Figure A.21 Temperature vs Time on the Wall (22")
Figure A.22 Temperature vs Time on the Wall (28")
Figure A.23 Temperature vs Time on the Wall (31"")
Figure A.24 Temperature vs Time on the Wall (34")
APPENDIX B
OUTSURGE EXPERIMENT.

<table>
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<tr>
<th>Parameter</th>
<th>Value</th>
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<tbody>
<tr>
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<tr>
<td>Initial Water Level</td>
<td>29.5&quot;</td>
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<tr>
<td>Final Water Level</td>
<td>9.6&quot;</td>
</tr>
<tr>
<td>Final Pressure</td>
<td>118.5 psia</td>
</tr>
<tr>
<td>Insurge Water Temperature</td>
<td>73.0 F</td>
</tr>
</tbody>
</table>

Figure B.1 Flow Rate vs Time (M - t).
Figure B.2 Pressure vs Time (P - t).
Figure B.3 Temperature vs Axial Position of the Centerline (T - z).
Figure B.4 Temperature vs Axial Position of the Wall Outside (T - z).
Figure B.5 - B.17 Temperature vs Time at the Center (T - t).
Figure B.18 - B.25 Temperature vs Time on the Wall (T - t).

*Ignore the double spikes, they were due to an uncorrected error in the DAS.
Figure B.1 Flow Rate vs Time.
Figure B.2 Pressure vs Time
Figure B.3 Temperature vs Axial Position.
Figure B.5 Temperature vs Time at the Centerline (4")
Figure B.6 Temperature vs Time at the Centerline (7")
Figure B.8 Temperature vs Time at the Centerline (13"")
Figure B.11 Temperature vs Time at the Centerline (22")
Figure B.12 Temperature vs Time at the Centerline (25"")
Figure B.14 Temperature vs Time at the Centerline (31"")
Figure B.15 Temperature vs Time at the Centerline (34"")
Figure B.16 Temperature vs Time at the Centerline (37")
Figure B.17 Temperature vs Time at the Centerline (40"")
Figure B.25 Temperature vs. Time on the Wall (37°)
APPENDIX C.
INSURGE TO THE HOT WALL EXPERIMENT.

<table>
<thead>
<tr>
<th>Description</th>
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<tbody>
<tr>
<td>Initial Pressure</td>
<td>113.7 psia</td>
</tr>
<tr>
<td>Initial Water Level</td>
<td>11.0 &quot;</td>
</tr>
<tr>
<td>Pressure before Insurge</td>
<td>86.4 psia</td>
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<tr>
<td>Water Level before Insurge</td>
<td>10.4 &quot;</td>
</tr>
<tr>
<td>Insurge Water Temp.</td>
<td>78.0 F</td>
</tr>
<tr>
<td>Final Water Level</td>
<td>35 &quot;</td>
</tr>
<tr>
<td>Final Pressure</td>
<td>123.5 psia</td>
</tr>
</tbody>
</table>

Figure C.1 Flow Rate vs Time ( M - t ).
Figure C.2 Pressure vs Time ( P - t ).
Figure C.3 Temperature vs Axial Position of the Centerline ( T - z ).
Figure C.4 Temperature vs Axial Position of the Wall Outside ( T - z ).
Figure C.5 - C.17 Temperature vs Time at The Center(T-t ).
Figure C.18-C.25 Temperature vs Time on the Wall ( T - t ).

*Ignore the double spikes, they were due to an uncorrected error in the DAS.
Figure C.3 Temperature vs Position at the Center.
Figure C.4 Temperature vs Axial Position on the Wall.
Figure C.8 Temperature vs Time at the Centerline (13")
Figure C.12 Temperature vs Time at the Centerline (25")
Figure C.17 Temperature vs Time at the Centerline (40")
Figure C.18 Temperature vs Time on the Wall (4")
Figure C.19  Temperature vs Time on the Wall (10")
APPENDIX D
OUTSURGE AFTER INSURGE EXPERIMENT

Initial Pressure 101.3 psia
Initial Water Level 20.5"
Final Water Level 7.7"
Insurge Water Temperature 78 F

Figure D.1 Flow Rate vs Time (M - t).
Figure D.2 Pressure vs Time (P - t).
Figure D.3 Temperature vs Axial Position of the Centerline (T - z).
Figure D.4 Temperature vs Axial Position of the Wall Outside (T - z).
Figure D.5 - D.17 Temperature vs Time at the Center (T - t).
Figure D.18 - D.25 Temperature vs Time on the Wall (T - t).
*Ignore the double spikes, they were due to an uncorrected error in the DAS.
Figure D.4 Temperature vs Axial Position on the Wall
Figure D.7 Temperature vs Time at the Centerline (10"")
Figure D.8 Temperature vs Time at the Centerline (13")
Figure D.14 Temperature vs Time at the Centerline (31")
Figure D.16 Temperature vs Time at the Centerline (37")
Figure D.17 Temperature vs Time at the Centerline (40")
Figure D.21 Temperature vs Time on the Wall (22")
Figure D.23 Temperature vs Time on the Wall (31")
Figure D.24 Temperature vs Time on the Wall (34")
APPENDIX E

EMPTY TANK INSURGE EXPERIMENT

<table>
<thead>
<tr>
<th>Initial Water Level</th>
<th>9.3&quot;</th>
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</thead>
<tbody>
<tr>
<td>Initial Pressure</td>
<td>126 psia</td>
</tr>
<tr>
<td>Empty tank Water level</td>
<td>0.0&quot;</td>
</tr>
<tr>
<td>Empty Tank Pressure</td>
<td>112.5 psia</td>
</tr>
<tr>
<td>Insurge Water Temp.</td>
<td>75 F</td>
</tr>
<tr>
<td>Insurge Tank Pressure</td>
<td>130 Psia</td>
</tr>
<tr>
<td>Final Water Level</td>
<td>27.8&quot;</td>
</tr>
<tr>
<td>Final Pressure</td>
<td>54.5 psia</td>
</tr>
</tbody>
</table>

Figure E.1  Flow Rate vs Time (M - t)
Figure E.2  Pressure vs Time (P - t).
Figure E.3  Temperature vs Axial Position of the Centerline (T - z).
Figure E.4  Temperature vs Axial Position of the Wall Outside (T - z).
Figure E.5 - E.17  Temperature vs Time at the Center (T - t).
Figure E.18 - E.25  Temperature vs Time on the Wall (T - t).

*Ignore the double spikes, they were due to an uncorrected error in the DAS.
Figure E.3 Temperature vs Axial Position.
Figure E.4 Temperature vs Axial Position on the Wall.
Figure B.5 Temperature vs Time at the Centerline (4"
Figure E.6 Temperature vs Time at the Centerline (7"")
Figure E.7 Temperature vs Time at the Centerline (10")
Figure E.9 Temperature vs Time at the Centerline (16")
Figure E.11 Temperature vs Time at the Centerline (22")
Figure E.12 Temperature vs Time at the Centerline (25")
Figure E.13 Temperature vs Time at the Centerline (28")
Figure E.14 Temperature vs Time at the Centerline (31"")
Figure E.15 Temperature vs Time at the Centerline (34"")
Figure E.16 Temperature vs Time at the Centerline (37"")
Figure E.17 Temperature vs Time at the Centerline (40")
Figure E.19 Temperature vs Time on the Wall (10")
Figure E.20 Temperature vs Time on the Wall (16"")
Figure E.21 Temperature vs Time on the Wall (22")
Figure E.22 Temperature vs Time on the Wall (28")
Figure B. 23 temperature vs time on the wall (31")
Figure E.24 Temperature vs Time on the Wall (34")