A CSMP TECHNIQUE FOR SURGE TANK PRESSURE PREDICTION DURING INSURGE, OUTSURGE AND MULTIPLE SURGE

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SYNOPSIS

An increase in the unit size of nuclear reactor power stations and marine propulsion installations necessarily implies an increase in the pressurizer volume. The dimensions of this vessel are such that a procedure for minimization of this volume is warranted. For heavy water reactors this will mean a smaller requirement of heavy water inventory. The pressure transients produced in the system during the operation of plant may cause unacceptable piping stresses and seriously affect the reactivity. Hence the pressurizer is employed in the system for cushioning the volume transients and maintaining the coolant pressure within the prescribed limits.

The purpose of this thesis is predicting pressure transients of a pressurizer during prescribed rates of level changes. Included is a documented S360/CSMP program which is based upon the theoretical models described in this thesis and can be used to predict pressure transients of any steam surge tank.

There are three distinct modes of surges, namely, outsurge, insurge and multi-surges,the latter being a combination of series of insurges and outsurges.

Experimental data obtained from numerous tests on actual surge tank for both outsurge and insurge are included. Good agreement between pressure predicted by the previous investigators and the experiments during the outsurge was obtained. It is found that the isentropic model for insurge is highly inappropriate and very good agreement between the the experiments and predicted pressure transients is obtained from the present study. Theoretical analysis for the multiple surges is almost

non-existent in the known available literature and an attempt is made to study the multiple surges by S 360/CSMP.

The outsurge, insurge and multiple surge considered in this thesis are the basic cases and they can be extended to solve any other surges or the combination of surges with a little modification. Although the program is written for the particular NPD (Nuclear Power Demonstration) pressurizer, there will not be any difficulty in using it for any other pressurizer of different geometry and dimensions.

Summarizing the results, it may be stated that the information presented here is believed to yield better insight in the different surge phenomena, and should consequently be of considerable assistance in developing a less conservative design procedure.

CONTENTS

LIST OF SYMBOLS

A	Vapor surface area	m^2
a	Initial liquid volume	m^3
b	Cross sectional area of the tank	m^3/cm
C_p	Speciffic heat of the wall	J/g^oC
h	Speciffic enthalpy	J/g
J	Mechanical equivalent of heat	
k	Thermal conductivity of the wall	$J/m\ sec\ ^oC$
K	Isentropic expansion coefficient	--
ℓ	Thickness of the wall	m
L	Level of the liquid	cm
M	Mass in the vessel	kg
P	Pressure	bar
Q	Heat transferred	J
R	Modified Universal gas constant	
T	Temperature	^oR
t	Time	sec
v	Specific volume	m^3/kg
V	Volume	m^3
W	Mass flow	kg/sec

Greek Symbols.

ρ	Density of the wall material	m^3/kg
ζ	Dummy time variable	
ϕ	Saturation temperature history	^oR
α	Thermal diffusivity of the wall	

LIST OF SUBSCRIPTS

1	Steam from the risers entering the mixing belt
2	Steam leaving the drum
3	Water from the risers entering the mixing belt
6	Feedwater entering the mixing belt
7	Water from the drum entering the downcomer
amb	ambient
c	Condensate
d	Drum
fd	Water leaving the mixing belt
gd	Steam leaving the mixing belt
s	Steam
sas	Saturated steam
sus	Superheated steam
saw	Saturated Water
suw	Sub-cooled water
w	Water

LIST OF ACRONYMS

ASME	American Society of Mechanical Engineers
BLW	Boiling Light Water
CSMP	Continuous System Modeling Program
EAI	Electronics Associates Inc.
IBM	International Business Machines
IFC	International Formulation Committee
NASA	National Aeronautics and Space Administration
NPD	Nuclear Power Demonstration
PWR	Pressurized Water Reactors
USAEC	United States Atomic Energy Commission

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CHAPTER 1

INTRODUCTION

1.1 General

Any closed liquid system subject to change in temperature and hence volume must be protected against excessive pressure variations. One possible way of doing it is an open connection to a gas or vapor filled chamber capable of cushioning the volume transients of the system. The lower portion of such chamber is filled with the liquid, while the upper portion contains vapor or gas. The outer surface of this tank is insulated so as to minimize heat losses to the surroundings. Fig. 1.1 shows the primary and the secondary cooling circuits of a closed cycle nuclear power plant, using the pressurized water as the coolant. The primary coolant may be either heavy or light water, and for the present study we use the heavy water as a primary coolant.

Liquid is heated in the reactor core and flows through a steam generator, where steam is generated for the turbine system, and the primary coolant returns to the reactor vessel through a circulation pump. The primary coolant temperature tends to shift with the changing turbine load, which in turn depends upon the various factors like load factor etc. This temperature shift causes the volume transients. The chamber providing the required expansion volume is called a pressurizer because of its additional function of maintaining the coolant pressure within the prescribed limits.

Analysis are required for three distinct processes, the insurge, the outsurge, and the multiple surges.

It is evident that the capability of predicting pressure transients during insurge is a must if one has to make provisions for associated

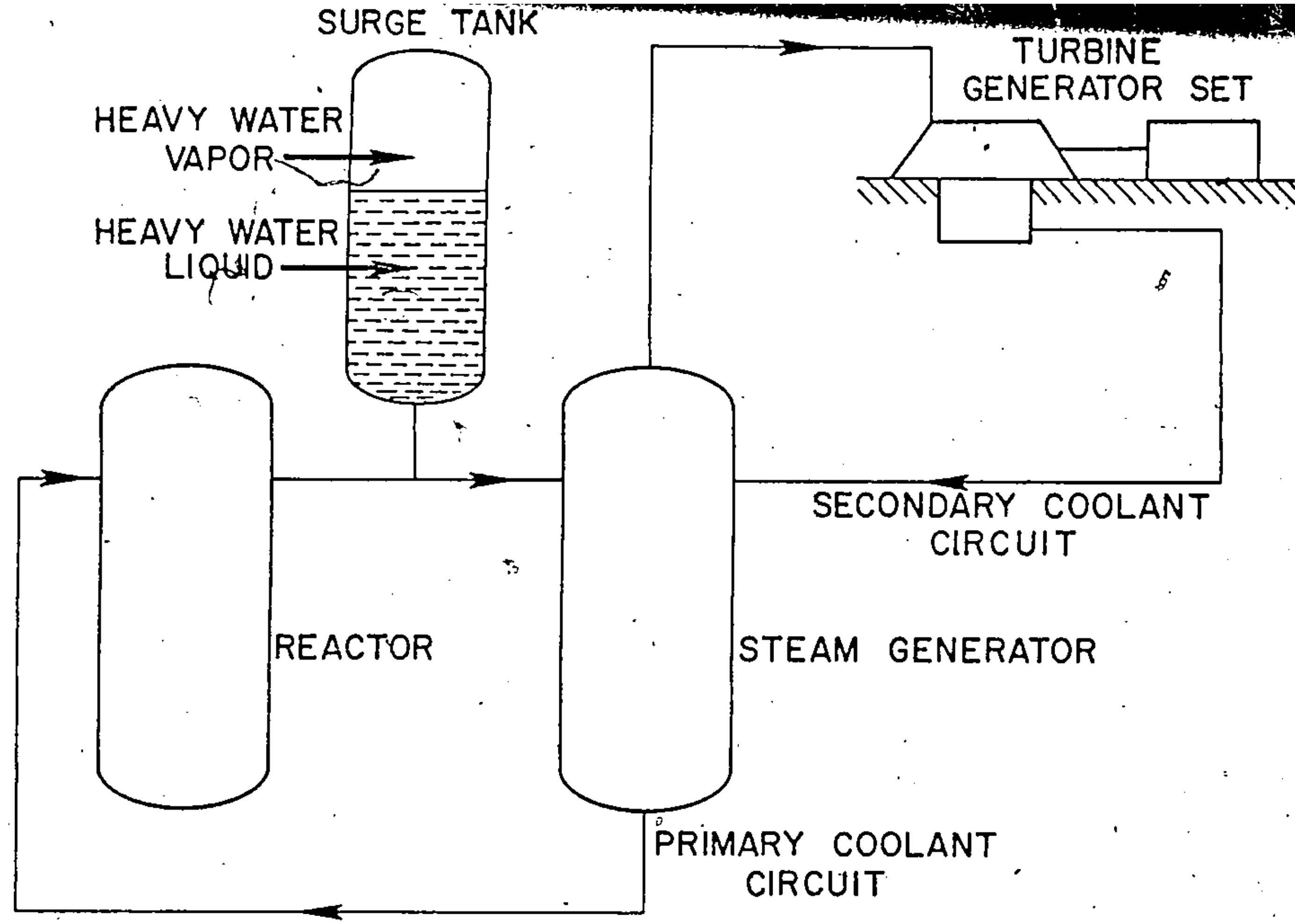

FIG. 1.1 SIMPLIFIED SKETCH OF A REACTOR

increase in system piping stresses and changes in reactivity. The reactivity is affected by changes in coolant density. Hence large changes in the coolant pressure can have very significant effects on the reactivity.

Also the pressure prediction during an outsurge is essential information so that the designer can make sure that the flashing of liquid does not occur in the system and the reactivity effects can be predicted for the system.

Because of the considerable reduction in the nuclear power cost, it is a general practice to use the nuclear power plants as the base load power plants. The multiple surges will occur when a nuclear power plant experiences large fluctuations in the power demand. This kind of rapid sequence of several in-outsurge cycles may be expected from a ship propulsion plant during manoevers. This essentially requires the means to predict the pressure transients during the multiple surges.

The conventional type of pressurizer is a seperate vessel containing liquid and vapor. In the steady state, the vapor and all part of the liquid is maintained in a saturation condition corresponding to the steady state pressure. A number of heating ellements submerged in the liquid portion of the pressurizer, supply sufficient heat to counterbalance the steady heat losses from the pressurizer wall to the surrounding through the outer insulation. The vapor pressure and hence the system pressure can also be adjusted to desired level for the steady state operation by means of these heaters.

During the steady state operation of a reactor plant the amount of energy supplied by the reactor equals the amount of energy extracted from the steam generator by the secondary system, so that the total energy content of the primary coolant and hence its volume remains constant. A decrease of turbine load causes the rise in average temperature of the primary coolant and, consequently, an increase of its volume. Liquid surges into the pressurizer and the vapor present in it is compressed. As a result, the pressure increases, but in order to keep the pressure below the permissible limit, some of the vapor is condensed by spraying relatively sub-cooled liquid into the pressurizer. This phenomenon is termed as spray INSURGE.

An increase in turbine load results in a drop in the average temperature of the primary coolant and a decrease in its volume. To make up for this loss of volume, some of the liquid from the pressurizer is extracted and a pressure drop generally results. This phenomenon is termed OUTSURGE. Liquid surges out of the pressurizer and the pressure decreases, causing some of the liquid in the pressurizer to flash into the vapor. This effect, coupled with the vapor production resulting from the switching on of extra heater elements limit the pressure fall. The pressure rise upon the insurge can be readily controlled by spraying sub-cooled liquid into the vapor phase, but the pressure drop upon the outsurge is far more difficult to control.

The surge tank thus serves a dual function. It provides a means to control the system pressure, as well as takes care of the liquid thermal expansion, by admitting more liquid.

1.2 Current Trends in the PWR Developments

Once the light water reactors began to be accepted as conventional power plants in the early sixties, a remarkably large number of power plants were ordered. Towards the end of 1970, more than 100 power plants were either operating , or under construction or ordered in the United States alone. Currently, unit ratings of more than 1000 Mega-watts are acceptable . The largest power plant under construction in Europe is the " KERNKRAFTWERK BIBLIS " having a capacity of 1,150 Mega-watts.

Table 1.1 shows some of the thermodynamic parameters for a large third generation power plant, the "Diablo Canyon Plant". Special attention should be paid to the pressurizer dimensions : the internal volume is 5 cubic meters, diameter is 2.3 meters and height as 13 meters weighing in the order of 40 tons. In other words, the pressurizer is seen to be a relatively large component, typically in the order of one third of the reactor vessel by volume.

From a USAEC study, the approximate PWR capacity increase rate can be taken to be 15000 Mega-watts per year on the average over the period of 1970 to 1980. The capital cost can be generally set at the rate of $240.00 per kilo-watt with a very little chance that this value will decrease in the near future. Consequently, the total sale volume for the pressurized water reactors will amount to a sizeable figure of about 3.6 billion dollars per year. The pressurizer cost may be assumed to be 0.5 % of the capital cost of the plant, thus equalling 18 million dollars per year.

During the early days of development, the attention of the designers was primarily and understandably focussed on the reactor itself, so an oversized pressurizer always worked well.

A further reason for continued and increasing attention to the pressurizer design and performance stems from the anticipated change in the mode of operation of nuclear power plants. So far, these plants have been designed for and operated in base load only. With the increasing percentage of the nuclear power, it will be necessary for the nuclear power plants to operate on the two shift basis, requiring special control concepts in order to ensure uniform fuel burn up in the

TABLE 1.1

DESIGN DATA FOR THE PWR PLANT LOCATED AT DIABLO CANYON

Thermal Output	3250 Mega-watts.
Electrical Output	1060 Mega-watts.
Coolant Flow Rate	17000 Kg./Sec
Coolant Temp.(Inlet)	282° C
Coolant Temp.(Outlet)	317° C
Total Coolant Volume	351 cubic meters
Pressurizer Volume	51 cubic meters
Pressurizer Int. Diam	2.3 meters
Reactor Vessel Diameter	4.4 meters
Primary Coolant Pressure	155 Bars
Secondary Steam Pressure	49.6 meters

system.

1.3 Scope and Motivation of the Project

Conclusions from the foregoing paragraphs may be summarized as follows in order to describe the motivation of this project. For the land based power plants, the large dimensions of the pressurizer vessel make it worth while to search for a method to minimize the vessel volume. Whereas in case of the naval plants, space and weight considerations make it desirable to design the minimum pressurizer volumes. Manoeuvrability requirements for the ship will result in severe performance criteria for the pressurizer in terms of the pressure limits to be maintained upon the consecutive in-and out-surges. Because of the reduction in pressurizer volume and the other dimensions, not only the fabrication cost will be affected, but also the costs for providing the additional containment space. A reduction in the pressurizer cost of 10 % will mean the saving of 1.8 million dollars per year.

CHAPTER 2

SURVEY OF THE LITERATURE

2.1 General

A number of theoretical models have been proposed for predicting the pressure transients during insurge and outsurge. Yet the important flashing phenomenon associated with the outsurge and the condensation phenomenon associated with the insurge is not adequately investigated. The theoretical analysis and the experimental information for the multiple surges is almost non-existent in the available literature.

2.2 Steam Compression

When the vapor-liquid interface rises due to an insurge of water into the pressurizer, the vapor phase is naturally compressed. The initially saturated steam becomes superheated . Initially, all the contents of the pressurizer are at the saturation temperature corresponding to the steady state pressure. Once the vapor is compressed, heat transfer from the vapor to the vessel wall starts. Vapor starts condensing on the cooler boundaries. i.e. the wall and the liquid. Condensate surface will necessarily follow the saturation temperature. Hence the temperature of the outside boundary of the vapor volume will also have the saturation temperature corresponding to the pressure at any particular instant.

The conclusion is that the vapor phase is nonhomogeneous during insurge and is composed of inner volume of the superheated vapor and a surrounding volume of vapor with a temperature going to the saturation temperature at the boundary. The thickness of this 'Boundary Layer' depends upon the thermal diffusivity of vapor and is generally small.Hence in most of the cases, it is, therefore, allowable to approximate this nonhomogeneous vapor volume by a homogeneous one.

All the previous investigators agree at one point that the isentropic model for the insurge is highly inappropriate and the pressure predicted on the basis of this model is excessively high compared to those measured during experiments. This discrepency is related to the fact that an additional amount of energy from the steam will be removed due to the heat transfer from the vapor to the pressurizer wall, which is at a relatively low temperature.

Drucker and Tong(1) conducted a series of experiments on a low pressure laboratory model of a surge tank and tried to obtain a mathematical correlation between the level and pressure. They used a step by step time increment and conducted the energy balance at the end of each time increment. The heat flow from the vapor is calculated from an empirical relation and heat sink constant for the vessel walls and liquid, which was obtained experimentally

Drucker and Gorman (2) described an iterative procedure for the pressure prediction which was similar to the described by Drucker and Tong (1), except that no empirical constants were reqiored. Good agreement between the experimental and predicted values was obtained, but it was far outside the pressure range encountered in the general practical applications. Gupta and Gorman (3) extended the work of Drucker and Gorman for higher and practical pressure ranges. Excellent agreement was obtained between the experimental and analytical pressure prediction.

van den Honert (4) described the results of experiments carried out on a 1 cubic meter water steam pressurizer at 125 atm. and studied the effects of outsurge, insurge, electrical heating, and use of a spray injection system.

Kendall (5) used the continuity and energy approach and studied the surges of a surge tank in the Gentilly Reactor Power Plant. Moeck (6) continued his work and studied both the insurge and outsurge. However, the adiabatic model described by him for the insurge leads to somewhat high pressure prediction.

2.3 Steam Expansion

In the event, that the liquid surges out of the pressurizer, the vapor volume increases and expansion takes place. The saturation temperature drops with the pressure , but the phases remain in thermodynamic equillibrium. Condensation process, if present, stops and the evaporation of initially saturated liquid starts. Very little work has been done on the pressure transients during outsurge.

van den Honert (4) has described a complex model involving the sub-cooling of the vapor. A relatively simpler model has been utilized by Gorman (7) to predict the pressure transients during an outsurge from an actual reactor surge tank at the usual operating pressures and the known surge rates. He has clearly demonstrated the inadequacy of the closed isentropic expansion model.

Gupta and Gorman (3) have used the 'stepped open system' for the pressure prediction . In their words," The outsurge is considered to be broken up into small increments, each increment corresponding to a drop in tank level, The system consisting of the initial tank content is considered to undergo a closed isentropic expansion associated with the first increment drop in the level. A solution for the pressure at the end of this expansion is obtained. Now the new system consisting of the liquid and vapor in the tank is considered at the begining of the second increment. Again this system is considered to undergo a closed isentropic expansion and a solution for the pressure at the end of expansion is obtained. This analytical procedure is repeated until the pressure associated with the entire level history during outsurge is obtained." Good agreement between the experimental and the analyticaly predicted values is obtained.

Moeck (6) used the same energy and continuity equations and obtained an expression relating the pressure and level history. Very good agreement between the experimental pressure history and predicted pressure history was observed.

CHAPTER 3

FORMULATION OF THE PROBLEMS

3.1 Pure Outsurge

As a result of the outsurge, the pressure drops, which in turn causes flashing of liquid in the pressurizer. But it is reasonable to assume that the outsurge process is adiabatic one. This is because of the fact that the wall of the pressurizer vessel will be completely dry and the heat will enter the vapor phase through the sensible heat transfer only Because of this relatively poor mechanism of heat transfer and the short duration of outsurge process, its effect is completely neglected and the process is assumed to be adiabatic.

Starting with the continuity and the energy equations, the outsurge model for a steam drum, a general case of pressurizer is developed. The analysis given below is exactly similar to that given in Ref.6.

The conservation of mass for the system illustrated in Fig. 3.1 is:-

$$\frac{d}{dt}(M_w + M_s) = W_{fd} + W_{gd} - W_7 - W_2 \qquad (3.1.1)$$

The energy equation for the system is:-

$$\frac{d}{dt}(M_w h_{saw} + M_s h_{sas}) = \dot{P}(v_{saw} M_w + v_{sas} M_s) + (W_{fd} - W_7)h_{saw} + (W_{gd} - W_2)h_{sas} \qquad (3.1.2)$$

Where the changes in kinetic and the potential energies are assumed to be negligibly small. The drum volume equation leads:-

$$\frac{dV_d}{dt} = \frac{d}{dt}(V_s + V_w) = 0 \qquad (3.1.3)$$

The equations presented above constitute a general dynamic model

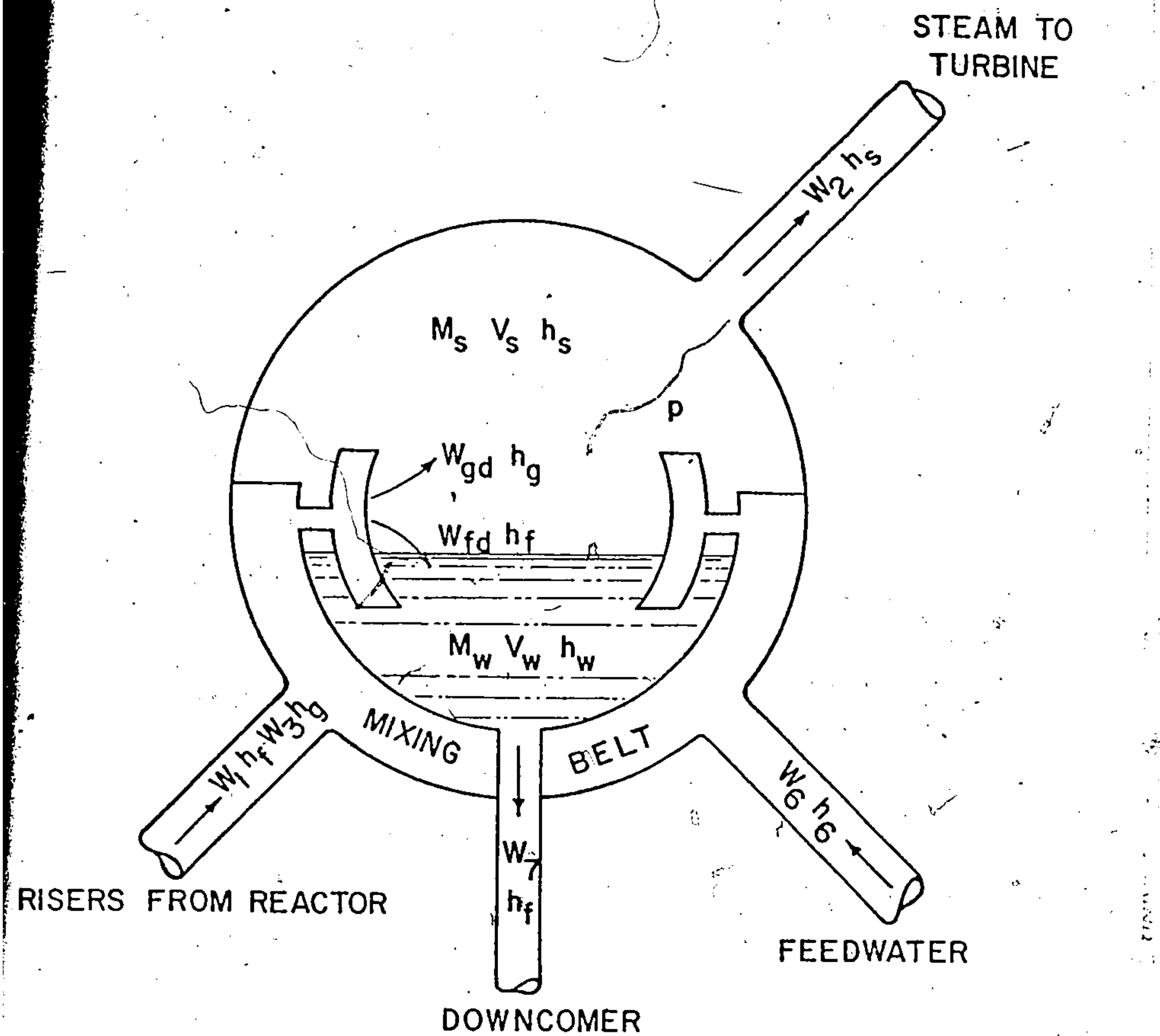

FIG. 3.1 STEAM DRUM SCHEMATIC

of a steam drum. Those can be adopted to the special case of pressurizer having:-

$$W_7 \neq 0 \quad \text{and} \quad W_{fd}=W_{gd}=W_1=W_2=W_3=W_6=0 \qquad (3.1.4)$$

By rearranging Eqs. (3.1.1) to (3.1.4), the relationship between the liquid level history drop in a surge tank and the corresponding vapor pressure drop can be obtained as follows.

$$\frac{dP}{dt} = \frac{1}{\frac{1}{b}\left(F1\ a + V_d\ F2\right) + F1\ L} \frac{dL}{dt} \qquad (3.1.5)$$

where F1 and F2 are the complicated functions of vapor and the liquid properties and are given by :-

$$F1 = \left[\frac{v_{sas}}{v_{saw}} \frac{dh_{sas}}{dP} - \frac{dh_{sas}}{dP}\right] \frac{1}{h_{sas} - h_{saw}} + \frac{1}{v_{sas}} \frac{dv_{sas}}{dP}$$

$$F2 = \left[\frac{dh_{sas}}{dP} - \frac{v_{sas}}{J}\right] \frac{1}{h_{sas} - h_{saw}} - \frac{1}{v_{sas}} \frac{dv_{sas}}{dP}$$

This special case investigated in this study and the behavior predicted by Eq.(3.1.5) can be checked againest the experimental data available from the NPD pressurizer, the only source of experimental results.

3.2 Pure Insurge

When an insurge to the pressurizer occurs, the vapor temperature and pressure (therefore the system pressure) begin to rise. As a result, the saturation temperature increases and the vapor condenses on the cooler boundary surfaces and heat is transferred to the vessel wall and the liquid present in the pressurizer. It is assumed that the whole tank and its contents are initially at a uniform saturation temperature corresponding to the initial vapor pressure and that the temperature of the incoming liquid does not affect the process, i.e. no significant mixing occurs. It is shown by Gupta (8) that the heat transfer between the phases is negligibly small because of the relatively low thermal diffusivity of the liquid and tank geometry, and hence is completely neglected.

The surge tank which contains both vapor and liquid, is considered as an open thermodynamic system. The solution for insurge employes the concept of conservation of energy and mass as suggested by Kendall (5) and later applied by Moeck (6). The analysis presented here follows in part the work done by Moeck (6).

Equations of continuity for the vapor and the liquid phases are:-

$$\frac{d}{dt} M_w = W_{fd} - W_7 \tag{3.2.1}$$

$$\frac{d}{dt} M_s = W_{gd} - W_2 \tag{3.2.2}$$

Energy equations for both the phases are:-

$$\frac{d(M_w h_{suw})}{dt} = \frac{M_w v_{suw}}{J} \frac{dP}{dt} + W_{fd} h_{saw} - W_7 h_{suw} \tag{3.2.3}$$

$$\frac{d(M_s h_{sus})}{dt} = \frac{M_s v_{sus}}{J} \frac{dP}{dt} + W_{gd} h_{sas} - W_2 h_{sus} + \frac{dQ}{dt} \tag{3.2.4}$$

Starting with these four equations, the general equation describing the relationship between the drum pressure and the liquid level history was derived. Similar to the previous pure outsurge case, a special case of pressurizer was considered by setting:-

$$W_7 \neq 0 \text{ and } W_{fd} = W_{gd} = W_1 = W_2 = W_3 = W_6 = 0 \tag{3.2.5}$$

The final form of the insurge equation becomes:-

$$\frac{1}{P\,K}\,(V_d - a - b\,L)\,\frac{dP}{dt} = b\,\frac{dL}{dt} + \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{dQ}{dt} \tag{3.2.6}$$

or in a more workable form:-

$$\frac{dP}{dt} = \frac{P\,K\,b\,\dot{L} + P\,K\,(\partial v_{sus}/\partial h_{sus})_P\,(\dot{dQ}/dt)}{(V_d - a - b\,L)} \tag{3.2.7}$$

The complete derivation of this equation is given in Appendix 1. The heat transfer to the wall is considered to be negative.

3.2.1 Heat Transfer to the Pressurizer Wall

The inner surface of the vessel wall in contact with the vapor is assumed to follow the vapor saturation temperature history at all times during the insurge.

The heat is transfered from the superheated steam to the vessel wall and the liquid in the pressurizer. However, as shown by Gupta (8) due to the low thermal diffusivity of the liquid, the heat from the vapor will penetrate a short distance below the liquid-vapor interface, so that the heat transferred from the vapor to the liquid is comparably negligible. Hence for all practical purposes, the heat flow to the liquid could be neglected for the geometries and pressures in the range under study here. The expression for the rate of the heat transfer from the system to the wall is shown as follows. The complete derivation of this expression is given in Appendix 2.

$$\frac{dQ}{dt} = \frac{2\,K}{\ell}\left[\sum_{n=0}^{\infty} e^{-a_n t}\left[\frac{dA}{dt} - A\,a_n\right]\int_0^t e^{a_n \zeta}\,\phi(\zeta) + A\,\phi(t)\right]$$

where
$$a_n = \frac{(2n+1)^2\,\pi^2\,\alpha}{4\,\ell^2} \tag{3.2.8}$$

However, the above expression is difficult to use in the S360/CSMP for the reasons mentioned below.

1) The exponential arguement increases with time, and since the computer cannot handle the exponential arguements beyond + or - 175, the expression cannot be used for longer insurges, and in practice the insurge process is longer. Similarly, as it is a summation of infinite terms, it has to be truncated after some number of finite terms, and it is seen that because of the computer limitations no more than the first five terms can be included.

2) Further difficulty arises due to the fact that multiplication of a very large number and a very small number makes a problem inherently unstable in nature. In Eq.(3.2.8) a multiplication involves a very small number, $e^{-a_n t}$, and a very large intgrl , $\int e^{a_n \zeta} \phi(\zeta)$.

However, incorporating first five terms, with a very small step size for integration and for a short insurge, the above expression was used to calculate the heat transfer rate. The pressure predicted by using this expression for heat transfer rate, is in fairly good agreement with the experiments.

In order to deal with the more general case, an attempt has been made to obtain the heat transfer rate by solving the Fouriers differential equation for one dimensional, transient heat conduction by S360/CSMP. The details of it are given in Appendix 3.

3.3 Multiple Surges

These are the combination of insurges and outsurges of the pressurizer. During multiple surges, an accurate knowledge of the outsurge process is very important, relative to other processes, because outsurge limits the power plant manoeuvrability and thus mainly determines the dimensions of the pressurizer vessel. This is of acute importance when the several consecutive insurge-outsurge cycles are to be accomodated within the tolerated circuit pressure limit.

3.3.1 Case 1, Outsurge Followed by Insurge

This is a combination of one outsurge and one insurge, in which expansion of vapor is followed by compression.

3.3.1.1 Vapor Expansion

During the outsurge, there is always a thermodynamic equilibrium between the phases. Both the vapor and the liquid remain in saturated condition during the entire outsurge and there is no subcooling of the vapor. This necessarily means that at the end of the outsurge the same thermodynamic equilibrium will exist and both the vapor and liquid will be in saturated condition.

During the outsurge, there is no condensation of the vapor and heat transfer between the vapor and wall is merely the sensible heat transfer which is quite insignificant as shown by the other investigators and hence is neglected. Since there is no heat transfer from the wall, the wall will remain at the uniform temperature, which is equal to the initial steady state wall temperature.

The discussion in the foregoing paragraphs reveals that the first half part of the cycle, i.e. the outsurge can be considered as a pure outsurge and the relationship between the liquid level history and the corresponding vapor pressure drop can be expressed by the same Eq.(3.1.5) From this the pressure drop during the first half cycle for any level history can be predicted.

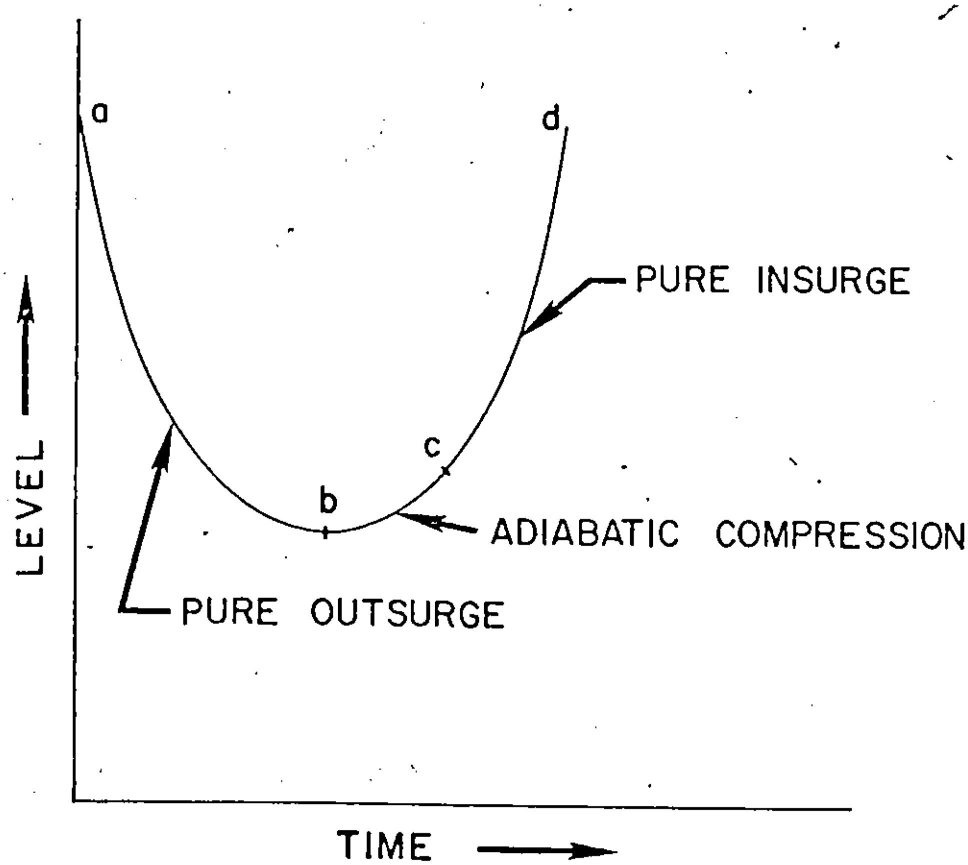

FIG. 3.2 LEVEL HISTORY FOR CASE I, OUTSURGE FOLLOWED BY INSURGE

3.3.1.2 Vapor Compression

Expansion of the vapor is followed by the compression. At the start of the insurge, i.e. at the end of the outsurge, as mentioned above, there is a thermodynamic equillibrium between the phases. But the wall of the vessel is at a higher but uniform temperature than the system at start and is equal to the initial steady state temperature. Hence this compression resembles the pure insurge case except that the heat transfer from the vapor to the vessel wall will only start after the vapor reaches the wall temperature. After this point the vapor condensation on the cooler boundaries will start and before this, the wall, being at a higher temperature than the vapor, will naturally be dry, and there will be very little sensible heat transfer from the wall to the vapor because of the temperature difference.

Thus the insurge is composed of two types, the first being the simple adiabatic compression till the start of condensation and the second being the pure insurge involving superheated steam, subcooled liquid and the cooler vessel wall. Therefore the same Eq.(3.2.7) can be used to describe the latter process.

3.3.2 Case 2, Insurge Followed by Outsurge

This is a combination of one insurge and one outsurge, in an exact reverse order to the previous case.

3.3.2.1 Vapor Compression

In this case, the vapor compression is exactly similar to the pure insurge. The analysis warrants keeping track of the degree of superheat of the vapor during this process, hence an attempt to compute the temperature of the superheated vapor is made at every instant by using the characteristic gas equation with the concept of modified characteristic gas constant.

$$P V = M R(P,T) T \tag{3.3.1}$$

As the vapor does not behave as a perfect gas in the region of interest, the assumption R as constant can lead to serious error in the temperature predictions. The same method and the expression as given in Jadwani's work (11)are used to compute the modified characteristic gas constant R, as follows.

$$R = R_o + \frac{1}{T} \sum_{n=1}^{5} C_n \left(\frac{P}{T}\right)^n \tag{3.3.2}$$

Where for the pressure range from 600 to 1000psi, and the degree of superheat from 0 to 50^{o}F, the values of constants as derived by Jadwani (11) are:-

C_1 = - 192.6099

C_2 = -3097.1170

C_3 = 3961.1560

C_4 = 5729.5350

C_5 = -8628.5430

R_o = .81847790

Volume of the vapor at any particular instant can be very easily calculated as:- Volume of vapor = Volume of vessel - Volume of the liquid present. Since the amount of vapor condensed during the insurge is very small, it is fairly accurate to assume that the level change of the liquid because of the addition of the condensate is negligible. Hence,

$$\text{Volume of vapor} = V_d - a - b\, L(t)$$

The mass of vapor is constantly changing because of the condensation on the wall. The rate of condensation can be found out as follows(8).

$$-\frac{dQ}{dt} = A K (T_{amb} - T_{sat}) - (h_{sas} - h_{saw}) \frac{dM_c}{dt} \tag{3.3.3}$$

Where K is the sensible heat transfer coefficient between the vapor and the pressurizer wall. The value of K will be critical only if the degree of superheat attained by the vapor is significant. The heat transfer coefficient, K, for the steady state free convection (12) can be expressed as

$$K = \frac{C\,K_f}{L}\,(\,Gr\;Pr\,)^m$$

where K_f = vapor conductivity

L = length of the rectangular wall

Gr = Grashoff number

Pr = Prandtl number

and C and m are the constants whose values depend upon the nature of the system. By using K = 0.25 Btu/hr ft^2 sec ^oF, Gupta (8) has shown that the sensible heat transfer is negligible. As the degree of superheat attained by the vapor is not more than 10^oF , the sensible heat transfer during the compression can be neglected, and the Eq.(3.3.3). reduces to:-

$$\frac{dQ}{dt} = (h_{sas} - h_{saw})\,\frac{dM_c}{dt} \qquad (3.3.4)$$

Thus from the Eq.(3.3.4) , by integrating that expression with respect to time, the value of M_c, the mass of condensate can be found out, and then

Mass of vapor = Original mass of vapor - Mass of condensate.

Once all the other parameters in Eq. (3.3.1) , that is, pressure, mass of the vapor, and the volume of the vapor are known, the only unknown in the expression is temperature. So solving Eq. (3.3.1), the value of T can be computed.

3.3.2.2 Vapor Expansion

At the end of the insurge, at point b in Fig. 3.3, the liquid is subcooled and the vapor is superheated and the pressurizer vessel wall has some temperature distribution. During the outsurge, vapor and liquid

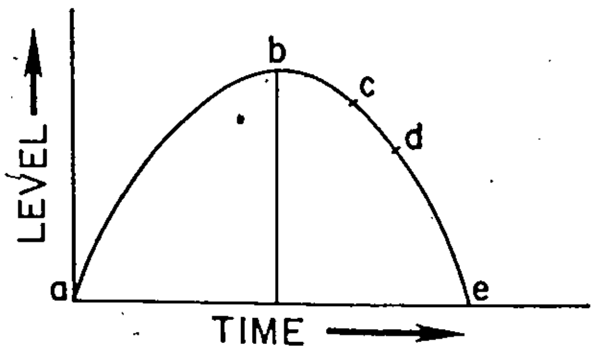

FIG. 3.3 LEVEL HISTORY FOR CASE 2, INSURGE FOLLOWED BY OUTSURGE

tend to become saturated, but, naturally, vapor and liquid do not reach the saturaion condition at the same instant. It was observed that, the vapor reaches the saturation condition before the liquid and hence this outsurge can be devided into three stages.

1) Outsurge during which the vapor is superheated and the liquid is subcooled, corresponding to the region bc as shown in Fig.3.3.

2) Outsurge during which the vapor is saturated and the liquid is still subcooled, corresponding to region cd in Fig.3.3.

3) Outsurge during which both the liquid and the vapor are in saturated condition corresponding to the region de in Fig.3.3.

For the first stage of the outsurge, starting with the same equations of continuity and energy, for the superheated vapor and the subcooled liquid, the expression relating the pressure transients and the level history can be derived. This case resembles the reverse pure insurge & so the same Eq.(3.2.7) can be used to describe the process with a slight modification in the heat transfer mechanism.

As the pressurizer wall follows the saturation temperature history, and as the vapor is superheated, the wall is at a lower temperature than the vapor and vapor is bound to condense on the cooler boundaries of the system for some period, until, the wall temperature reaches the vapor temperature. During this part of the outsurge, the heat transfer due to the condensation of the vapor continues from the system to the wall, until, the rate of heat transfer becomes zero.

Part of the condensate formed during the entire process is amassed on the pressurizer wall and the remaining part drains out and mixes with the main liquid volume. The condensate adhering to the wall gets evaporated during the subsequent outsurge process. As the pressure of the system decreases, the condensate which is at the saturation temperature evaporates and mixes with the vapor volume. If the mass of the liquid evaporated during the outsurge is significant, it plays an important role in controlling the drop of pressure. Hence the thickness of the condensate deposit on the pressurizer wall must be computed.

For further understanding of the above mentioned condensation phenomenon, the average heat transfer coefficient k_{cave} and the condensate layer thickness d_c are required. These can be derived from the Nusselt equation (9) which is valid for the laminar flow caused by the condensation of pure vapor on a vertical surface of a given height.

$$k_{cave} = 0.94 \left[\frac{\Gamma \, v_L \, (v_L - v_S) \, h^3 \, g}{H \, \lambda_L \, \Delta T_c} \right]^{0.25}$$

$$q_{wall} = k_{cave} \, \Delta T_c$$

$$k_{cave} = H / d_c$$

$$d_c = 1.08 \left[\frac{\Gamma v_L (v_L - v_s) g}{H \lambda_L q_{wall}} \right]^{-0.33}$$

where h = Coefficient of thermal conductivity

g = Constant of gravity

H = Height of the wall

λ = Dynamic viscocity

Γ = Mass flow rate per unit width

In the above expression, the quantity Γ is unknown and is difficult to determine without any experimental knowledge, hence the two extreme cases of condensate formation can be considered.

1) All the condensate formed during the process is drained and mixed with the main water volume and the instant at which the condensation stops, the wall of the pressurizer is completely dry and having no condensate deposit.

2) All the condensate formed during the process is on the wall in the form of a thin film and no condensate is mixed with the main water volume and as the wall follows the saturation temperature history, all the condensate on the pressurizer wall is at the saturation temperature corresponding to the pressure at any particular instant.

When the mass of the condensate formed during the entire process was computed, it was found that the total mass of condensate is less than 1.5% of the total mass of the vapor in the pressurizer. This negligibly small mass of the condensate does not affect the pressure and temperature of the system significantly. By using experimental information Honert(9) has calculated the average thickness of the condensate film as 27 micro meter. Furthermore, this thin film of the deposit dries out very rapidly during the subsequent process. Heat required for the evaporation of t is deposited film does not come from the pressurizer wall and hence the energy required for the evaporation is simply an internal heat transfer as in case of pure outsurge.

The discussion in the foregoing paragraphs shows that, after the condensation is stopped, there is no heat transfer to or from the pressurizer wall. Hence during the second and the third stage of outsurge, the heat content of the wall remains constant.

It was observed during the simulation that the degree of subcooling of the liquid, when the vapor became saturated, was very small. Moreover, for the same pressure , change in the specific enthalpy of the saturated liquid due to 5° F of subcooling is less than 3 %, and the change in specific volume is less than 2%. Hence it is fairly correct approximation to assume that the liquid is in saturated condition when the vapor reaches the saturated condition. Thus the second stage of the outsurge was eliminated and thermodynamic equillibrium in both liquid and vapor phases was assumed.

The third stage of the outsurge involves the phases having thermodynamic equillibrium and resembles to the pure outsurge case, which can be described by the same Eq.(3.1.5).

3.3.3. Extension to Case 2

Case 2, described in the previous section can be extended for the surge combination as illustrated in Fig. 3.4 There the region abcde remains same as in case 2.From point e onwards, upto f the pure outsurge is continued,during which the temperature distribution inside the wall is constantly evolving and tends to reach equillibrium.

At point f, the vapor temperature is less than the inner temperature of the pressurizer wall, so condensation is abscent during first few seconds of this insurge. Since there is no condensation, this corresponds to

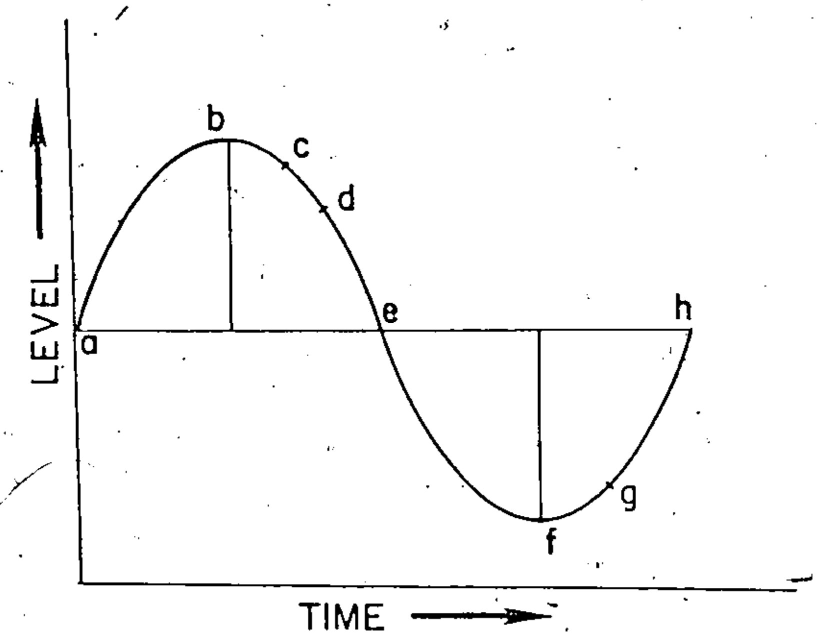

FIG. 3.4 LEVEL HISTORY FOR EXTENSION TO CASE 2

a pure adiabatic compression. The vapor temperature reaches the wall temperature at some point, which is denoted in Fig.3.4 as g. From this point onwards the condensation and hence the heat transfer starts. The process gh resembles the pure insurge case.

CHAPTER 4

COMPUTATIONAL TECHNIQUES

At this stage, the complete formulation of the problems, based upon the models described in the previous chapters, is accomplished. Before proceeding to obtain the actual solution of the problem, it is necessary to study the different computing facilities and techniques available and proper selection can thus be made. At present, three types of computing facilities are available.

1) Analog computer

2) Digital computer

3) Hybrid computer, the combination of the first two.

As the analog computer normally has limited capacity and logic devices, the digital or hybrid computer is more suitable for the present simulation.

4.1 Hybrid Computer

Basically, it consists of an analog computer which can simulate through electronic circuits an analogous real life dynamic system. In the total hybrid computer system, separate analog and digital computer with the appropriate interface hardware are incorporated so that the information can be exchanged between the two parts. Thus the two sections operate in unison. Hybrid computer is one of the best tools for the continuous simulation of the dynamic systems.

An attempt was made to simulate the pressurizer surges on the EAI 590 Hybrid Computer System available at the University of Ottawa, but as the capacity of the computer was not enough to handle this simulation problem, the other tool of simulation, the digital computer was sought.

4.2. Digital Computer

Given a mathematical model, it is sometimes possible to derive the information about the system by analytic means. Where this is not possible, it is necessary to use the numerical computing methods for solving the equations by making use of a digital computer. A rich variety of numerical computing methods have been developed for solving the equations of the mathematical models.

The digital computer contains simple discrete logical and arithmetic operations that can be executed in the proper sequence to solve the equations. Five basic components are employed in it.

1) The Input Unit - which is used to supply the necessary data to the computer.
2) The Memory Unit- in which data and the instructions given to it are stored.
3) The Arithmetic Unit - which performs the necessary arithmetic operations and provide the dicison making ability.
4) The Control Unit - which controls the computer operations.
5) The Output Unit - which provides and controls the output.

In the case of dynamic mathematical models, those that allow the changes of the system attributes to be derived as a function of time, a particular technique that has come to be identified as a system simulation is one in which all the equations of the model are solved simultaneously with steadily increasing value of the time.

In general, the computing languages can be grouped into three main catageories, namely,

1) Machine or assembly languages,

2) Procedural languages,

3) Problem oriented languages.

For writing the program in machine or assembly language, the problem must be broken into a sequence of simple tasks that the hardware of the computer can perform. This makes the programming unnecessorily tedious although it is only at this level the full power and flexibility of the computer can be exercised.

Procedural languages, Fortran, PL/1 etc. allow the user to define a computational procedure in a form which resembles the equivalent statement of the procedure of the problem. Good knowledge of the numerical methods is necessary for programming the continuous simulation problems in these procedural languages.

On the other hand, the problem oriented languages are designed in such a fashion that the computer handles the complications which are unrelated to the problem. They allow a continuous system simulation problem to be programmed on a digital computer in essentially the same way as is solved on an analog computer. They maintain the same general techniques developed to solve the problems with the analog computers, but in doing so they overcome the disadvantages of the analog computers. The user must preferably construct a functional block diagram in which each element is a specific operator. The language essentially consists of a collection of the predefined procedures which are called into the computer. For using these procedures, the information must be fed into the computer in a prescribed format. Some of the problem oriented languages are tabulated in the Table 4.1.

Problem oriented languages offer the user a greater freedom in describing a system. They make use of a Fortran like statement language, allowing the problem to be programmed directly from the equations of a

TABLE 4.1

EXAMPLES OF THE PROBLEM-ORIENTED LANGUAGES

	Name of the language	Area of application
1.	CSMP (Continuous System Modeling Program)	Modeling continuous dynamic systems
2.	GPSS (General Purpose System Simulator)	Modeling discrete dynamic systems
3.	ICES (Integrated Civil Engineering Systems)	Modeling of various types of Civil Engineering design situations
4.	ASKA (Automatic System For Kinematic Analysis)	Modeling structural dynamics
5.	APT (Automatically Programmed Tools)	Preparation of the data for numerically controlled machine tools
6.	ADAM (A Generalised Data Management System)	File management and information retrieval

mathematical model or the block diagram of the problem, rather than the equations to be broken into the functional elements. Alongwith the integration, addition etc. they include a variety of algeabraic and logical expressions to discribe the relations between the variables. They, therefore, extend the range of continuous system simulation by removing the orientation towards the differential equations which characterize the analog method. Thus an attempt has been made to hide the digital computer's true nature of being a discrete sequential device. One particular, that will be briefly described here is the S360/CSMP (Continuous System Modeling Program) also referred as IBM 360/CSMP.

4.2.1 S360/CSMP

The S360/CSMP is a problem oriented language designed to facilitate the digital simulation of the continuous systems on the large scale digital machines. The program provides an application oriented language that allows these problems to be prepared directly and simply from either a block diagram representation or a set of the ordinary differential equations. The feature of S360/CSMP permits the user to concentrate upon the phenomenon being simulated, rather than the mechanism for implementing the simulation.

The program includes a basic set of functional blocks with which the components of a continuous system may be represented, and accepts application oriented statements for defining the connection between these functional blocks. It also accepts most of the Fortran statements, allowing the user to readily handle non-linear and time-varient problems of considerable complexity. Input and output are simplified by means of user-oriented control statements. A fixed format is provided for printing (tabular format) and printplotting (graphic format) at selected increments of the independant variable.

It provides a basic set of 34 functional blocks, plus the means for the user to define functions specially suited to his particular simulation requirements. This complement is augmented by the Fortran library functions, and furthermore, special functions can be defined either through Fortran programming, or more simply, through a macro capability that permits individual existing functions to be combined into a larger functional block. The user is thereby given a high degree of freedom and flexibility for different problem areas.

4.2.2 Economic Evalution of S360/CSMP

Cost of operation for the different simulation techniques is considerably varied. Although none of those simulation techniques can be explicitly said to be the cheapest, an idea about the cost can be drawn from the studies made by Chubb (15) . He performed the economic study of various continuous simulation techniques for the two problems.

1) A five loop fifth order non-linear dynamic system for control purpose.

2) Seat ejection system with the fourth order dynamic equations and a coordinate transformation.

The comparison for the cost v/s the number of runs and the time required for the simulation v/s number of runs for the problem 1 is shown in Fig.4.1. The present problem consists of the solution of a set of differential equations as in the first problem mentioned above.

The cost analysis is based upon the data as mentioned below.

Cost of the labor	\$ 18 per hour.
Cost of IBM 360/65 Digital Comp.	\$ 200 per hour.
Cost of AD 256 Analog Computer.	\$ 18 per hour.
Cost of EAI 680 Hybrid Computer.	\$ 80 per hour.

Using this data for the cost estimation, the experiments for the

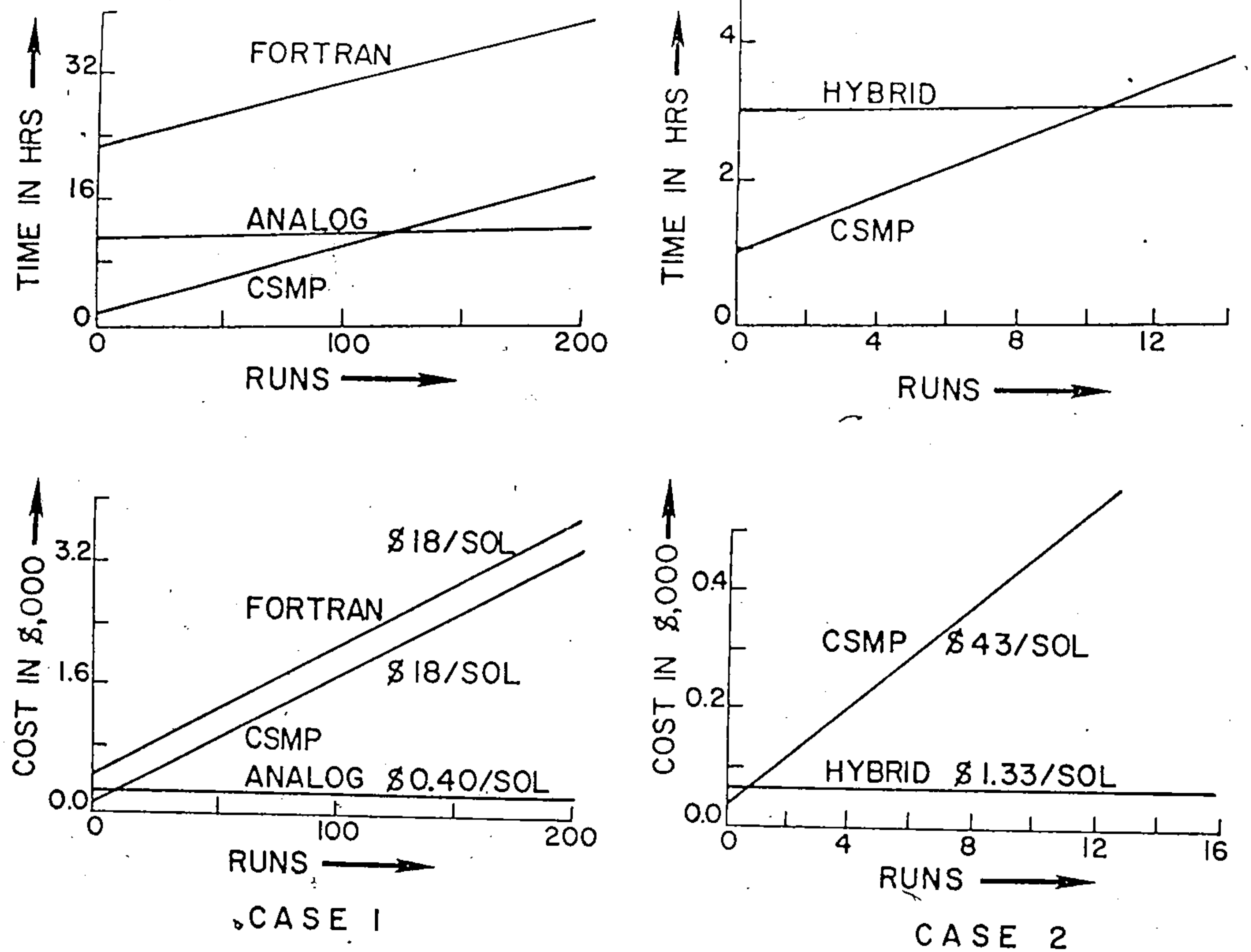

FIG. 4.1 TIME/COST INFORMATION FOR TYPICAL DYNAMIC SIMULATIONS

above two problems of simulation were carried out and the conclusions of the study can be summarized as follows.

1) For the first solution of the problem Fortran requires a long time, while S360/CSMP greatly reduces these, whereas hybrid computer falls between these two.

2) For the dynamic simulation, Fortran cannot be recommended because of the cost sonsideration.

3) Time required to generate the number of solutions for number of runs is remarkably less for the hybrid computer as compared to Fortran and S360/CSMP. It can be easily inferred from the Fig.4.1, that the time required for the hybrid computer solution, for any number of runs is almost constant.

From the foregoing paragraphs it is evident that S360/CSMP is the least expensive tool for the dynamic simulation, when only few runs are required, while hybrid computer is advantageous when a large number of runs are required.

The present simulation requires only very few runs, and hence the selection of S360/CSMP as the tool for simulation is also justified on the economic grounds.

4.3 Computing Procedures

All the cases described in the previous chapters were programmed and the pressure histories were predicted by executing the computer programs on an IBM 360/65 computer. Although the simulation is done for the NPD pressurizer, the theoretical models can be used to describe the surges of any other pressurizer of different geometrical dimensions and shape. The RKS method, the variable step-size fourth order Runge-Kutta method of numerical integration was employed in all the cases descriebed earlier. This feature of CSMP automatically adjusts the step size in order to ensure that the error is less than or equal to 0.0001, whereas the error estimation is performed by using the Simpson rule.

All the vapor and liquid properties were calculated based upon the heavy water. In this study the same simple polynomials approximated by Moeck (6) for the various saturation properties of the heavy water as a function of saturation pressure were used. Those polynomials, either linear or quadratic expressions, are valid over the pressure range from 35 bars to 75 bars and are summarized below, where in all cases,

P = Pressure in bars

v = Specific volume in CC/gram

h = Specific enthalpy in J/gram

$$h_{saw} = 747 + 9.85\ P - 0.034\ P^2$$

(maximum error 0.12%)

$$h_{sas} = 2790 + 0.898\ P - 0.0162\ P^2$$

(maximum error 0.02%)

$$h_{sas} - h_{saw} = \frac{10^6}{476 + 2.68\ P}$$

(maximum error 0.20%)

$$v_{saw} = 1.12 + 3.30\ 10^{-3}\ P$$

(maximum error 0.08%)

$$v_{sas} - v_{saw} = 2100/P - 3.97$$
(maximum error 0.43%)

$$v_{sas} = 2090/P - 2.46$$
(maximum error 0.40%)

$$\frac{dh_{saw}}{dP} = \left[\frac{0.241}{P} + 6.17\ 10^{-4}\right] h_{saw}$$
(maximum error 0.25%)

$$\frac{dh_{sas}}{dP} = \left[\frac{0.196}{P} - 2.35\ 10^{-4} - 1.26\ 10^{-5}\ P\right] (h_{sas} - h_{saw})$$
(maximum error 1.53%)

$$\frac{\partial v_{sus}}{\partial h_{sus}} = (2.32 - 0.0106\ P + 5.26\ 10^{-5}\ P^2)\ \frac{v_{sas} - v_{saw}}{h_{sas} - h_{saw}}$$
(maximum error 0.95%)

The last expression for the superheated steam is valid for the degree of superheat from 0°C to 25°C.

4.3.1 Pure Outsurge

The simulation of the pure outsurge case was made first to test the S360/CSMP as a tool for the simulation and subsequently the other cases were simulated. Two experimental runs (Runs A2 and A3) were used for this simulation.

The level function generation from the experimental data of the level and time of the NPD pressurizer was performed by using the special non-linear function generation capability (viz. NLFGEN) of CSMP. Since there is no abrupt deviation in the experimental data, non-linear function generation is more accurate than the linear function generation, and hence selected.

Derivative of the level history with respect to time was required for the simulation, which was computed by using the derivative package of CSMP (viz. DERIV (IC,L)). For using this function, the value of dL/dt at time equal to zero was required, which was calculated by forming a small difference table at the start of the level history.

The step size of the integration for the start of the computation was specified as 0.001. The printplot capability of the CSMP was used for obtaining the graphical representation of the pressure predictions. The computer program and the printplot results for Run A2 are given in Appendix 4.

4.3.2 Pure Insurge

In this case, the level history and the derivative of the level history with respect to time is obtained in the similar fashion as mentioned in pure outsurge case.

In the initial segament of the program, all the priliminary computations of pressurizer volume, cross sectional area, thermal diffusivity etc. are performed. It also contains the geometrical dimensions of the pressurizer, initial conditions, constants and other data input for the particular run.

The area exposed to the vapor is always changing and is a function of time and hence is calculated in the dynamic segament. Saturation temperature as a function of the saturation pressure is calculated as follows. First, a straight line was fitted by the least squares method for the temperature and pressure values for which the data is taken from Elliot (16).For the pressure range from 55 bar to 75 bars, the saturation temperature as a function of saturation pressure can be expressed as follows

$$T_{sas} = (P_{sas} - 55) \; 1.920852 \; + \; 976.416$$

where T is in radians.

Heat transfer rate is found out by using the method of solution of Fourier's differential equation of heat conduction. It has been found that the 12 nodal points were sufficient to yield stable and accurate solution, hence twelve nodal points were chosen thereby yielding twelve ordinary simultaneous differential equations, which were solved to get the temperature distribution history in the wall. The resulting temperature histories were integrated by Simpsons rule to obtain the heat content and finally, the rate of heat transfer was obtained by a numerical differentiation procedure. The details of this procedure are outlined in Appendix 3.

Since no data are available for the heavy water vapor in the superheated region in the range of interest, the superheated properties of the heavy water vapor are approximated by using the light water properties. The term $(\partial v_{sus}/\partial h_{sus})_P$ is calculated by using the expression given by Moeck(6). The value of K, the isentropic expansion coefficient is taken as 1.26 as suggested by Moeck.

The pressure predicted by using the adiabatic model, i.e. by neglecting the heat transfer to the pressurizer wall, is computed in the same program for the sake of comparison under the heading of " Pressure Prediction neglecting heat transfer ".

The computer program and printplot results for a typical run, i.e. Run 6, are given in Appendix 5.

4.3.3 Multiple Surges

These are the combinations of the previous basic surge processes. Their combination is attained in the Dynamic segment with the help of various logical signals.

4.3.3.1 Case 1, Outsurge Followed by Insurge

The level history for this particular run was assumed to be sinusoidal. The initial steady state level of 61 inches and the maximum level change of 20 inches in 60 seconds was considered and the level history as $L = L_i - (L_i - L_f) \sin(\pi t/120)$ was assumed.

Referring to Fig.3.2, as discussed earlier, the process a to b is simply a pure outsurge for which the Eq.(3.1.5) is used for the pressure prediction. Process b to c is the adiabatic compression of vapor, which is described by Eq.(3.2.7) with the term dQ/dt equal to zero. Implementation of changing outsurge to insurge in the program i.e. change from Eq.(3.1.5) to Eq.(3.2.7) at point b, when the rate of level change changes from negative to positive is triggered by logical function switches.

Process c to d is the pure inurge, during which the heat is being transferred from the system to the wall. Heat transfer from the wall starts from the point when the vapor temperature goes higher than the wall temperature. (Point c). For finding out the point c, it is necessary to compute the vapor temperature at every instant during the process, which is attained by using Eq.(3.3.1). It can be noted that the only one unknown in the expression is T, temperature, which can be computed by an iteration method. The Newton-Raphson method of iteration was employed for this purpose. Once the vapor temperature is computed, it is compared with the wall temperature at every instant, and heat transfer to the wall starts when the vapor temperature reaches the wall temperature. This condition is implemented in the program by means of the logical function switching capability of CSMP.

4.3.3.1.1 Newton-Raphson Method

$$\text{Let} \quad F(T) = \frac{P\,v}{R_o + \frac{1}{T}\sum_{n=1}^{5} C_n \left(\frac{P}{T}\right)^n} - T$$

Differentiating this expression with respect to T,

$$F'(T) = \frac{P\,v \sum_{n=1}^{5} C_n\,(n+1)\,P^n\,T^{-(n+2)}}{R_o^2} - 1$$

Newton-Raphson iteration function is defined as,

$$\Phi(T) = T - F(T)/F'(T)$$

In order to commence the iterations, an initial estimate of T is required. This is readily obtained by using the equation of state for an ideal gas as,

$$P V = M R T$$

where $R = 85.8/144$

and then the process of iteration is carried out until,

$$| \Phi(T) - T | \leq 2$$

which gave satisfactory results within the accuracy of $\pm 2^\circ$ R. Thus the value of ambient temperature was computed.

This computation of vapor temperature is contained in the computer program under NOSORT section. Different kinds of logical branching and extensive use of Fortran statements are made in this section. The parameter KEYHET is used to start the computation of heat transfer to the wall at point c. The two parameters, KEYIN and KEYOUT changes their values when the rate of level changes the sign and control the correct use of the equation for describing the process.

Thus for the fixed initial and final level and the sinusoidal level change, the pressure history was found out. The CSMP program and the printplot results for this case of simulation of NPD pressurizer are given in Appendix 6.

4.3.3.2 Case 2, Insurge followed by Outsurge

Similar to the previous case, the sinusoidal level change was assumed for this run. Referring to Fig.3. 3, the process ab is the pure insurge which is described by Eq.(3.2.7). Process bc is the outsurge for which, as described in Chapter 3, the Eq.(3.2.7) is continued. Point c, is the point at which the vapor becomes saturated and the degree of superheat becomes zero. For finding out the point c, it is necessary to compute the vapor temperature, which is performed by the same method described in the previous case. The two parameters, KEYIN and KEYOUT change their values when the vapor becomes saturated and control the use of proper equation for describing the process. As soon as the vapor becomes saturated (point c),the pure outsurge expression is employed. Thus the process cde which is pure outsurge is described by the Eq.(3.1.5).

The CSMP program and the computer printplot results for this case are included in Appendix 7.

CHAPTER 5

RESULTS AND DISCUSSION

5.1 Pure Outsurge

Two experimental runs, i.e. Run A2 and A3 were used to check analytical model of the pure outsurge. The comparison between the predicted pressure and the experiments for both the runs of NPD pressurizer is made in Figs. 5.2 and 5.3 respectively. A good agreement between the predicted pressure behavior and the experiments is obtained for almost the entire outsurge. It can be seen that the predicted pressures are slightly lower than the experimental near the end of the outsurge. This can be explained by noting that no account for the heat inflow from the tank wall is made in the analysis. The effect of inflow would be much more significant if the outsurge had to occur over a longer period of the time. But as this outsurge is for a shorter period, this account of heat flow is not warranted.

There is a slight anamoly in the experimental data of Run A2. The pressure curve tends to flatten out more quickly at the end than would be expected by studying the level time curve. Without these anamolies the discrepencies would have been less.

It is noted, that the computational techniques described in the previous chapter are highly appropriate for the prediction of outsurge pressure behavior for the system and pressure range of the type encountered in the NPD surge tank.

5.2 Pure Insurge

The total six experimental runs, i.e. Runs 2-7(13) were used to check theoretical analysis of insurge. The comparison for a typical run i.e. Run 6 is presented in the Fig. 5.4. Alongwith the experimental and predicted pressures

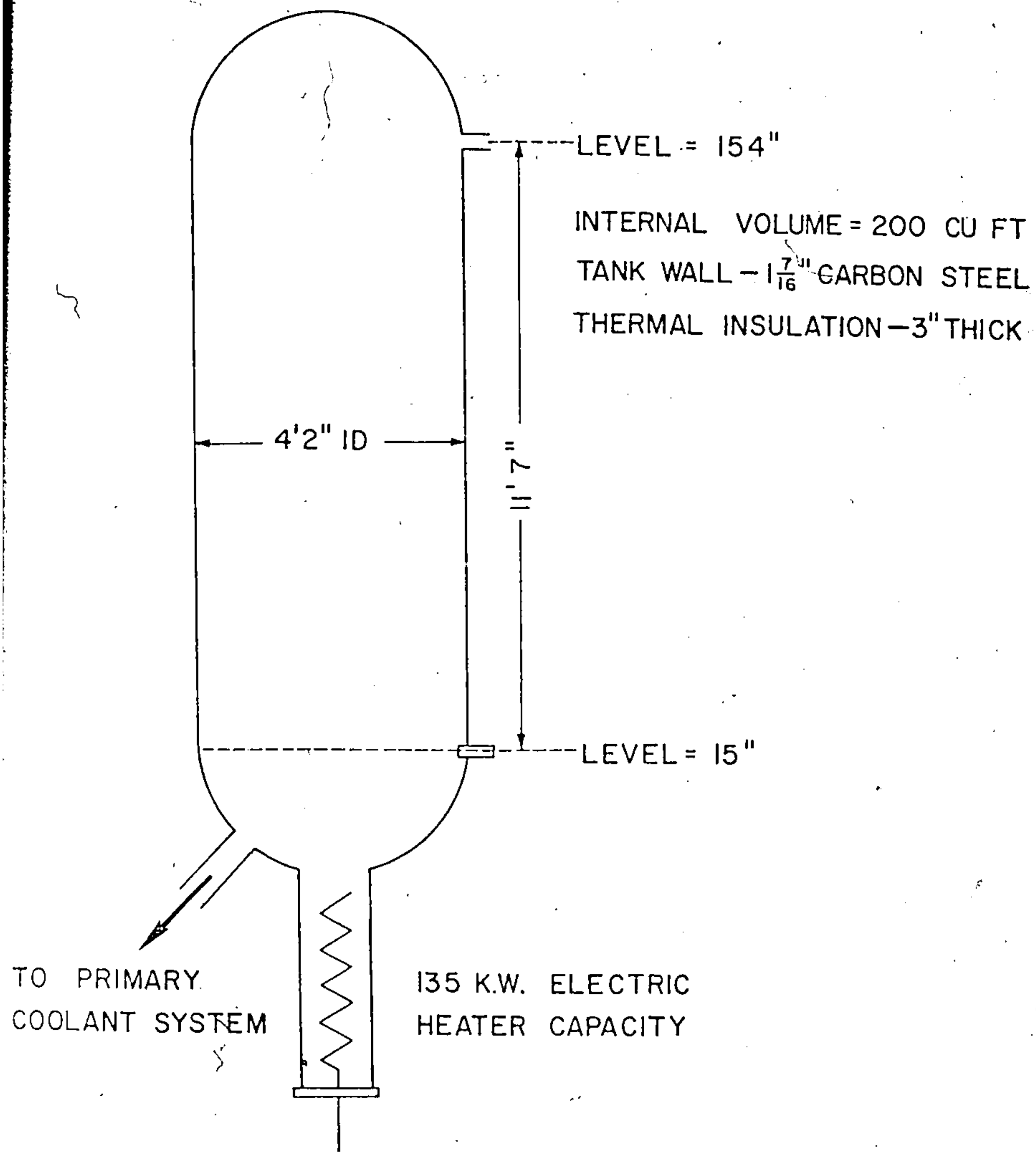

FIG. 5.1 SCHEMATIC DRAWING OF N.P.D. REACTOR PRIMARY COOLING SYSTEM STEAM SURGE TANK

the figure also contains the pressure behavior predicted by using the adiabatic model. It can be easily seen that the assumption of adiabatic compression would lead to extremely large errors for the insurges of magnitudes under investigation. The reason for poor predictions, based upon the adiabatic model as the compression proceeds, is because of the fact that there is an increasing temperature gradient in the walls building up in the early part of the compression. The predictions based upon the adiabatic model for these insurges are so poor that they are not presented for the further NPD pressurizer insurge tests reported here.

From Fig. 5.4 it is also obvious that a very good agreement between the predicted pressures and the experiments is obtained.

In order to gain some confidence in the S360/CSMP approach, a comparison was made between the present results and those of the iterative approach obtained by Gupta and Gorman (3). Both the approaches yield almost identical predictions.

Two of the tests discussed here (Runs 3 and 4) were conducted while the electric heating ellements in the liquid phase, each of 15 KW capacity, were turned on. It is easily inferred from the Figs. 5.6 and 5.7 that those heaters would have no appreciable effect on this compression processes.

Some more experimental results are compared with the predicted pressures in Figs. 5.5, 5.8 and 5.9 for the Runs 7, 2, and 5 respectively. It is observed that there is a good agreement for most of these insurges. The study of the experimental level and pressure curve reveals that there are small anamolies in the experimental data of Runs 3 and 4. Without these anamolies the discrepancies between the experimental and predicted values would have been less.

It is noted that near the end of the insurge, the predicted pressures are slightly higher than the experimental values. There are two factors in particular which tend to explain this behavior of the vapor. First the tank is considered in the calculations to be perfectly insulated whereas in the actual experimental tests, some heat is certain to be lost through the outer insulation. Secondly, in the calculations, the heat flow axially down the vessel wall is completely neglected, because the wall is considered to be a semi-infinite solid. It is inevitable that near the end of the compression, when the temperature of the vapor is relatively high, there is bound to exist a temperature gradient down the wall and some of the heat will be certainly lost in this manner. Both these conditions will contribute towards a small discrepancy near the end of the insurge, but as the discrepancy is small, the incorporation of these effects into the analysis is not warranted. The pressure prediction, which is on a higher side, leads to the safer design criteria.

5.3 Multiple Surges

The results for the outsurge followed by insurge and the insurge followed by the outsurge are given in Figs. 5.10 and 5.11 respectively. Since there is no experimental data available for the comparison, it is not possible to verify the validity of the analytical model presented in this thesis, quantitatively.However, some qualitative comments can be made. Firstly, the predicted pressure history obviously follows the level history trend, as is expected. Secondly, the multiple surges consist of pure insurge and pure outsurge, which have been dealt successfully before, with some modification, it is believed that the present multiple surge models should yield reasonably correct pressure prediction.

From the experiments it is evident that starting from the same initial steady state pressure, the drop of pressure during the outsurge, for a certain fixed level history, and level drop, is far lesser than the rise in pressure during the insurge for the same amount of level rise in the same time. This is probably because of the fact that, during insurge, large amount of work is done on the system for forcing the liquid in the pressurizer under a very high working pressure, whereas the outsurge process is entirely a waste process. This work is supplied to the system is partially responsible for this pressure rise during the insurge.

Exactly the similar trend is observed in both the cases of multiple surges. In Case 1, for the same amount of level change during both the outsurge and insurge, at the end of the cycle, the pressure of the system is higher than the initial pressure of the cycle. Similarly in Case 2, at the end of complete cycle, the system pressure is much higher than the initial pressure.

Combinations of Cases 1 and 2 alongwith pure insurge and pure outsurge cases can be used to describe other complicated multiple surges with proper modifications.

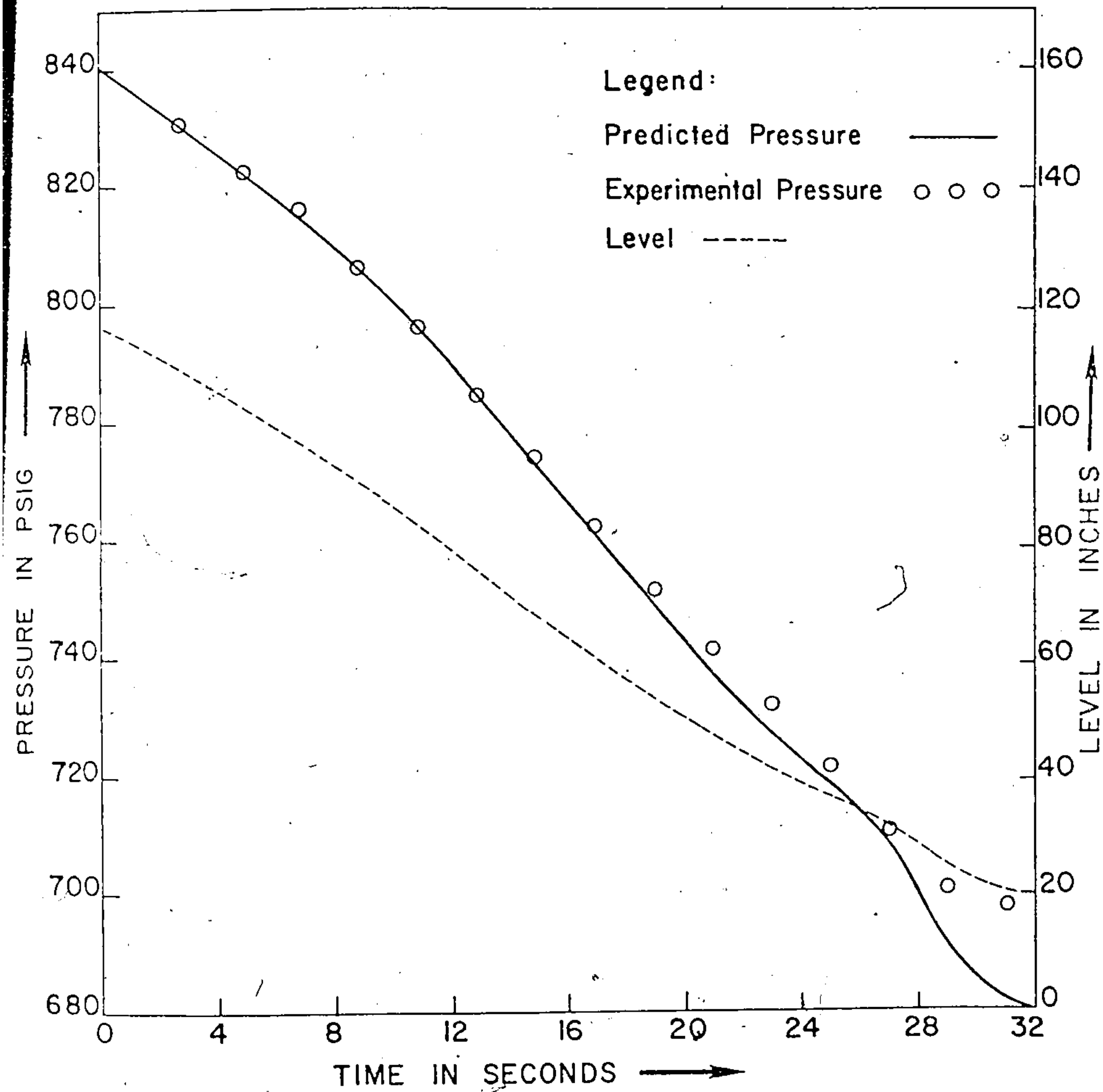

FIG. 5.2 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN A2

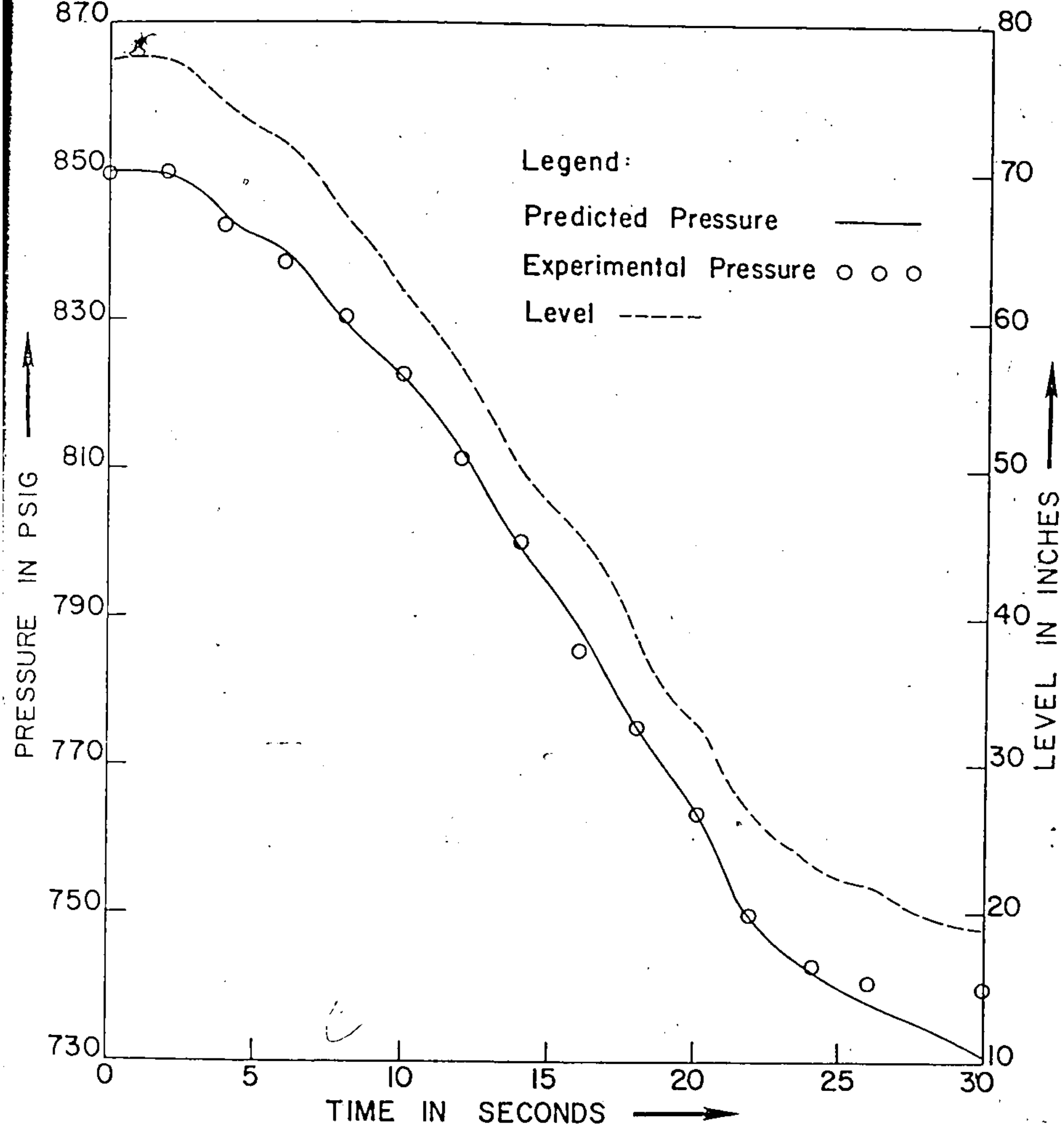

FIG. 5.3 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN A3

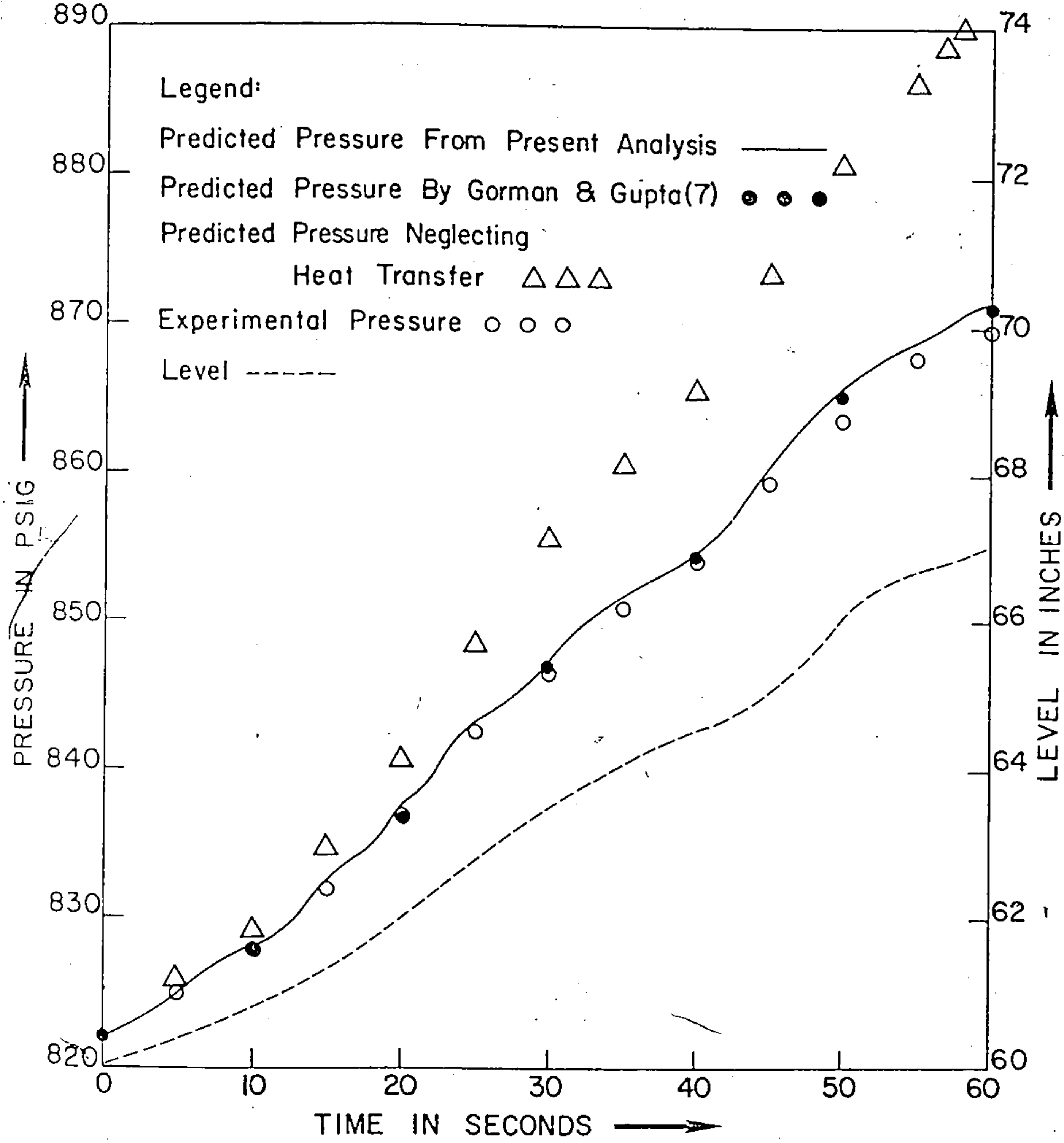

FIG. 5.4 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 6

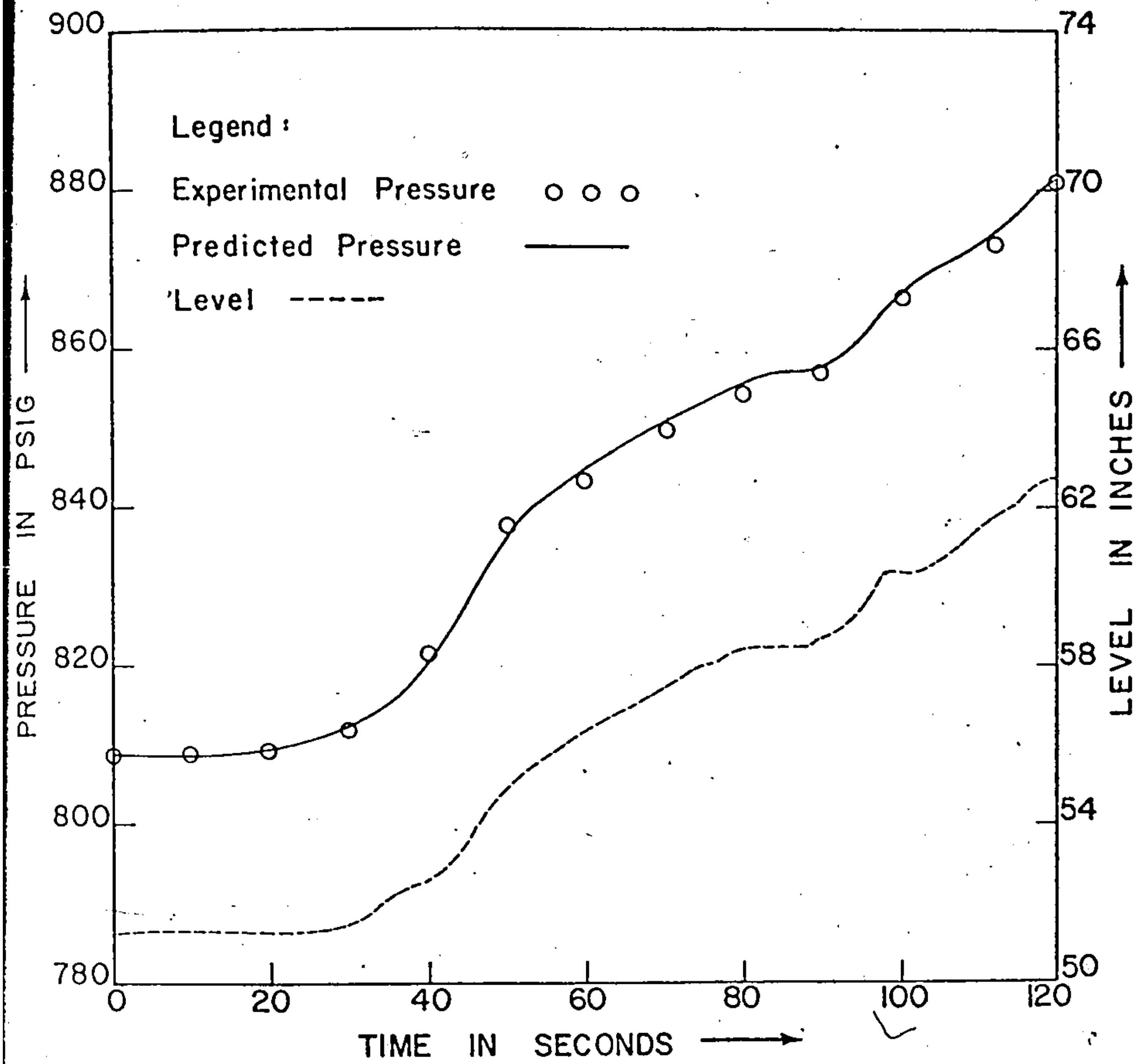

FIG. 5.5 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 7

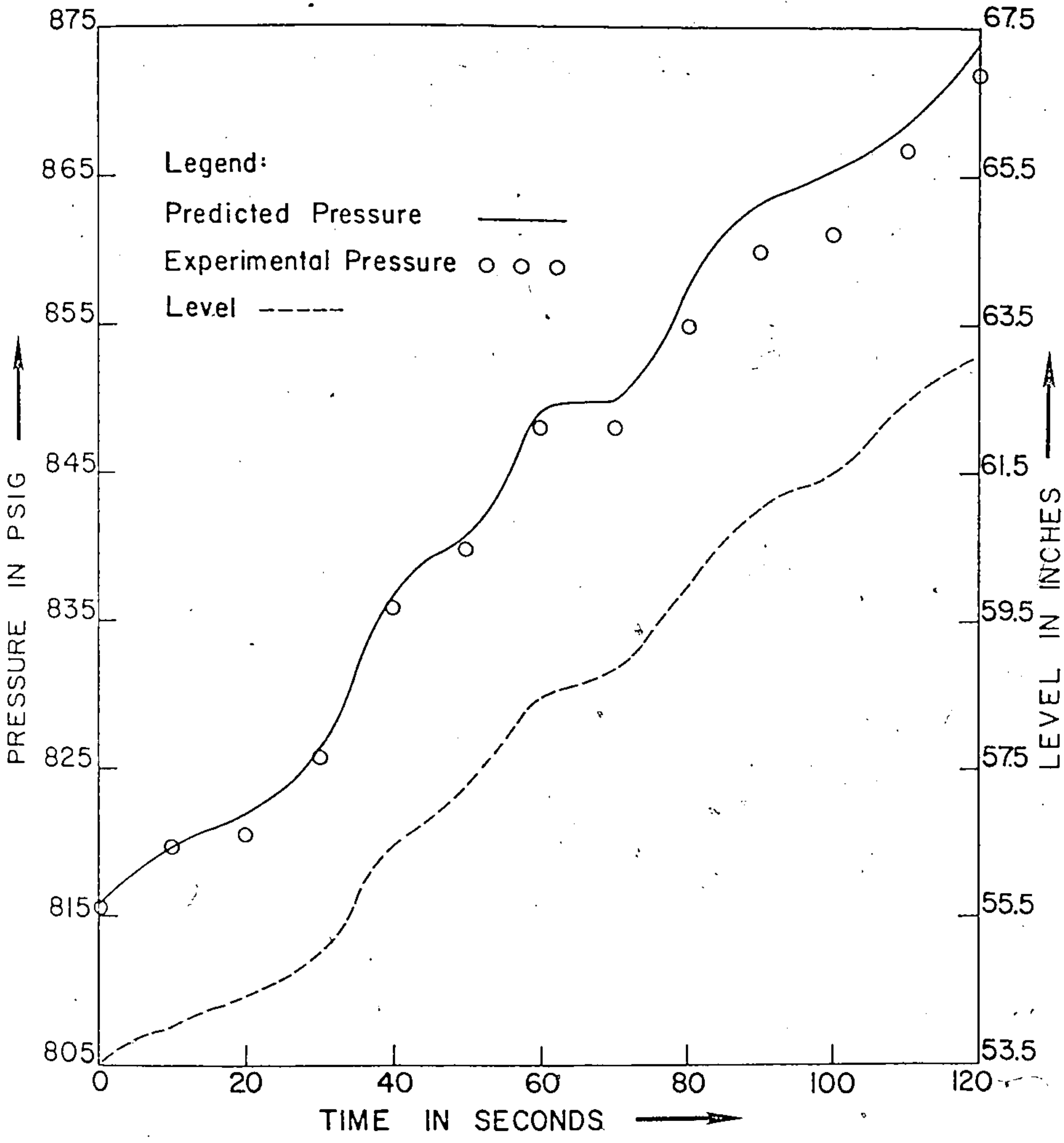

FIG. 5.6 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 3

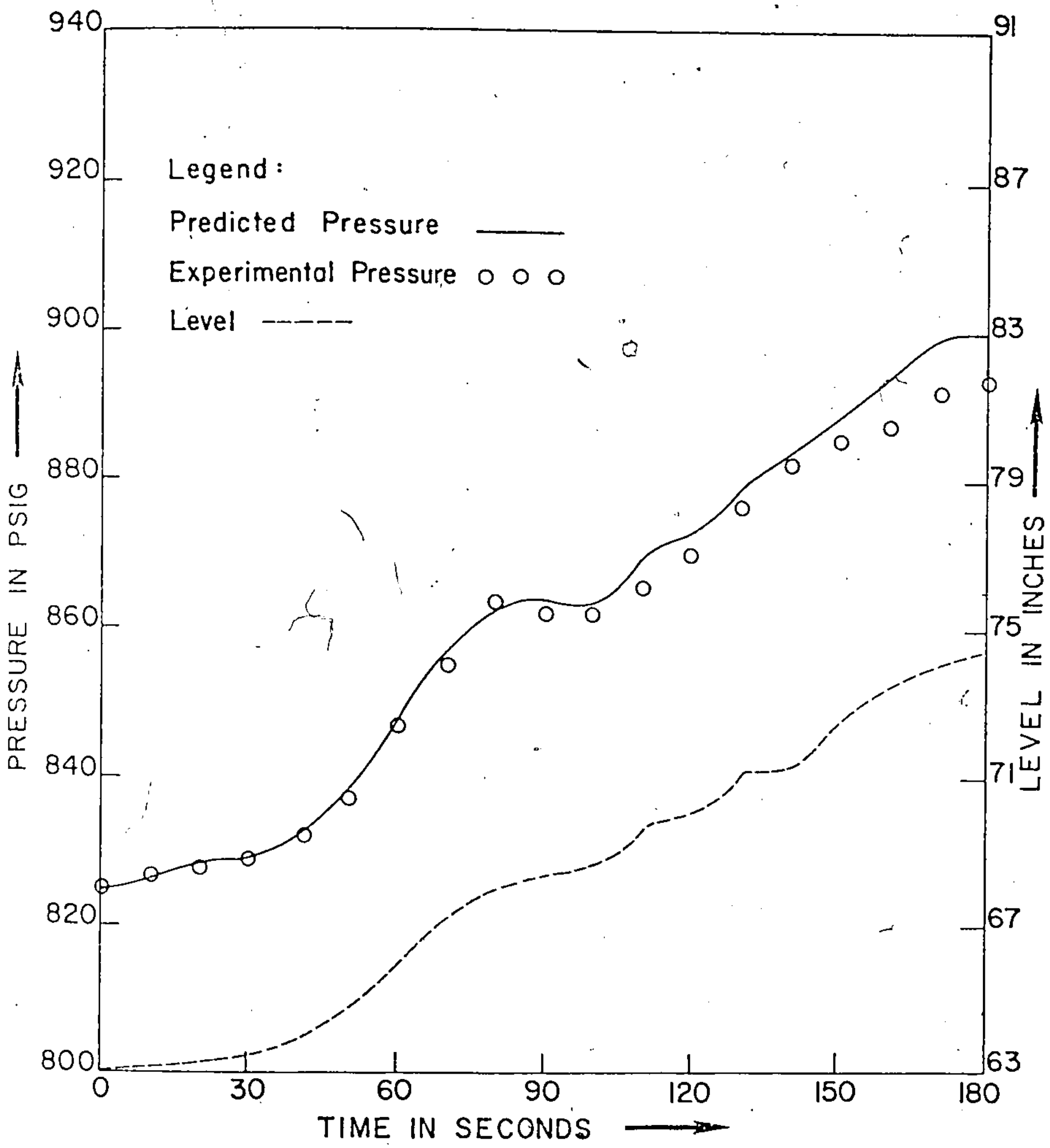

FIG. 5.7 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 4

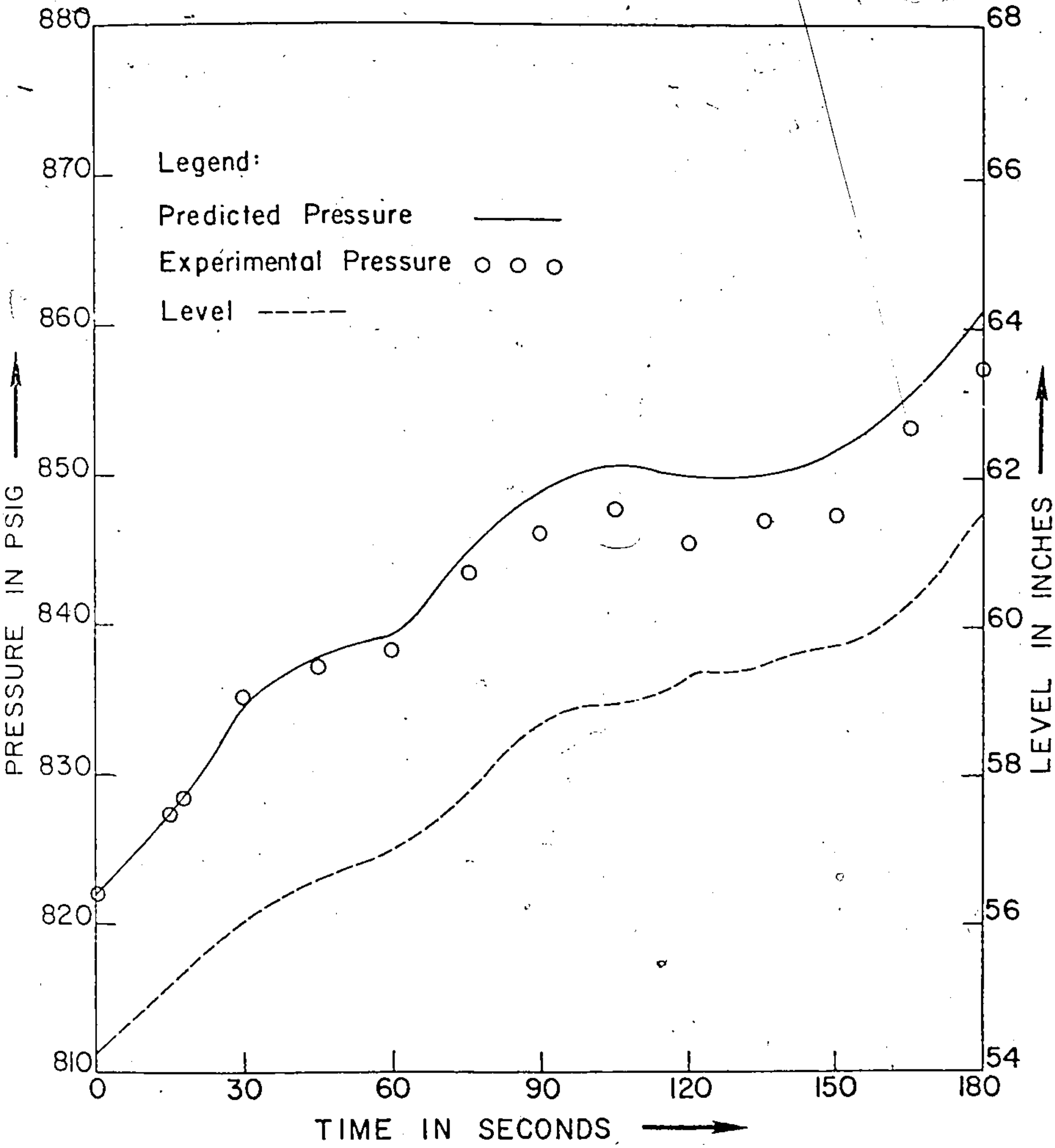

FIG. 5.8 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 2

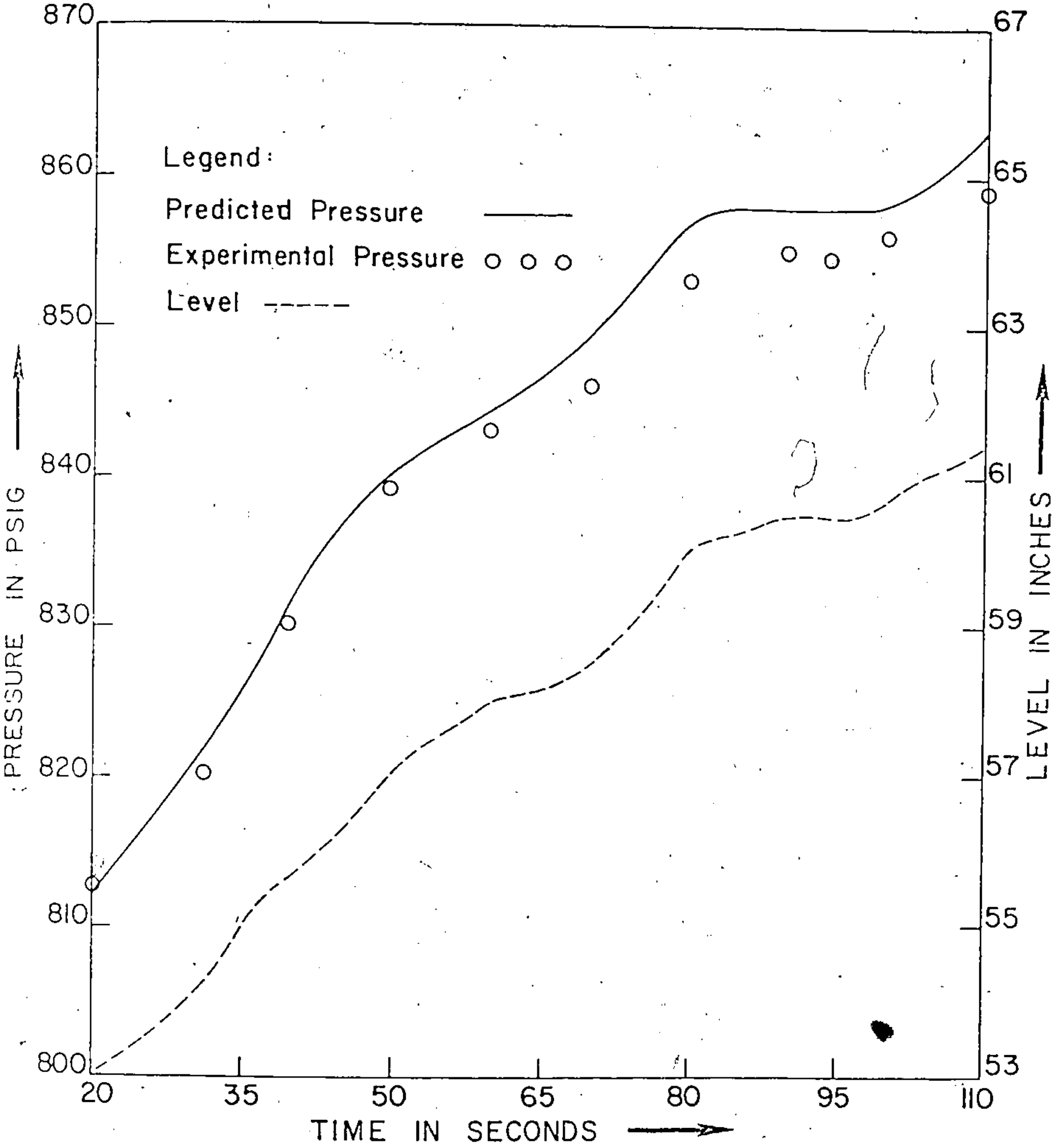

FIG. 5.9 COMPARISON BETWEEN PREDICTED PRESSURE HISTORY WITH EXPERIMENTS FOR RUN 5

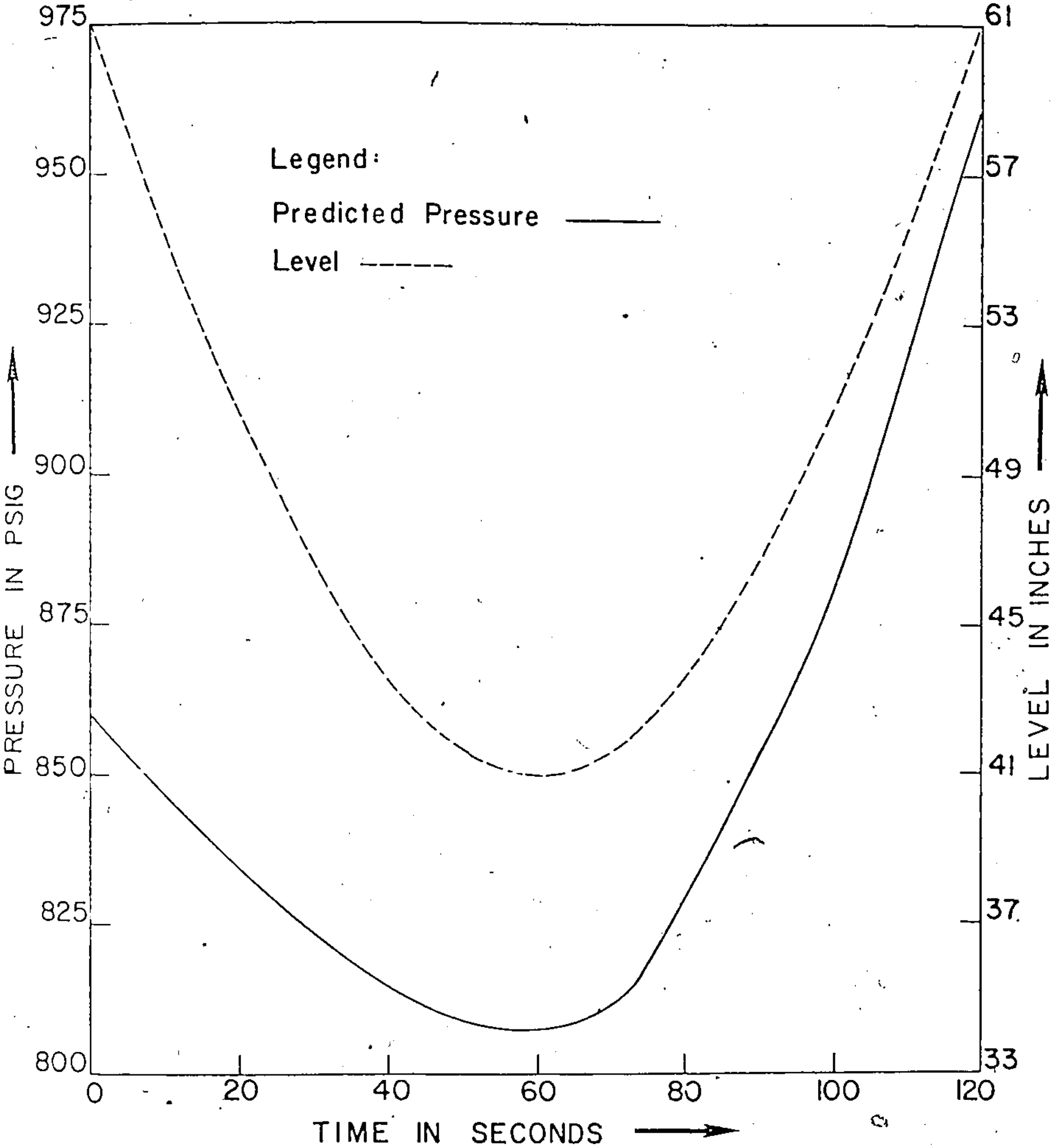

FIG. 5.10 PRESSURE PREDICTION & LEVEL HISTORY FOR CASE 1, OUTSURGE FOLLOWED BY INSURGE

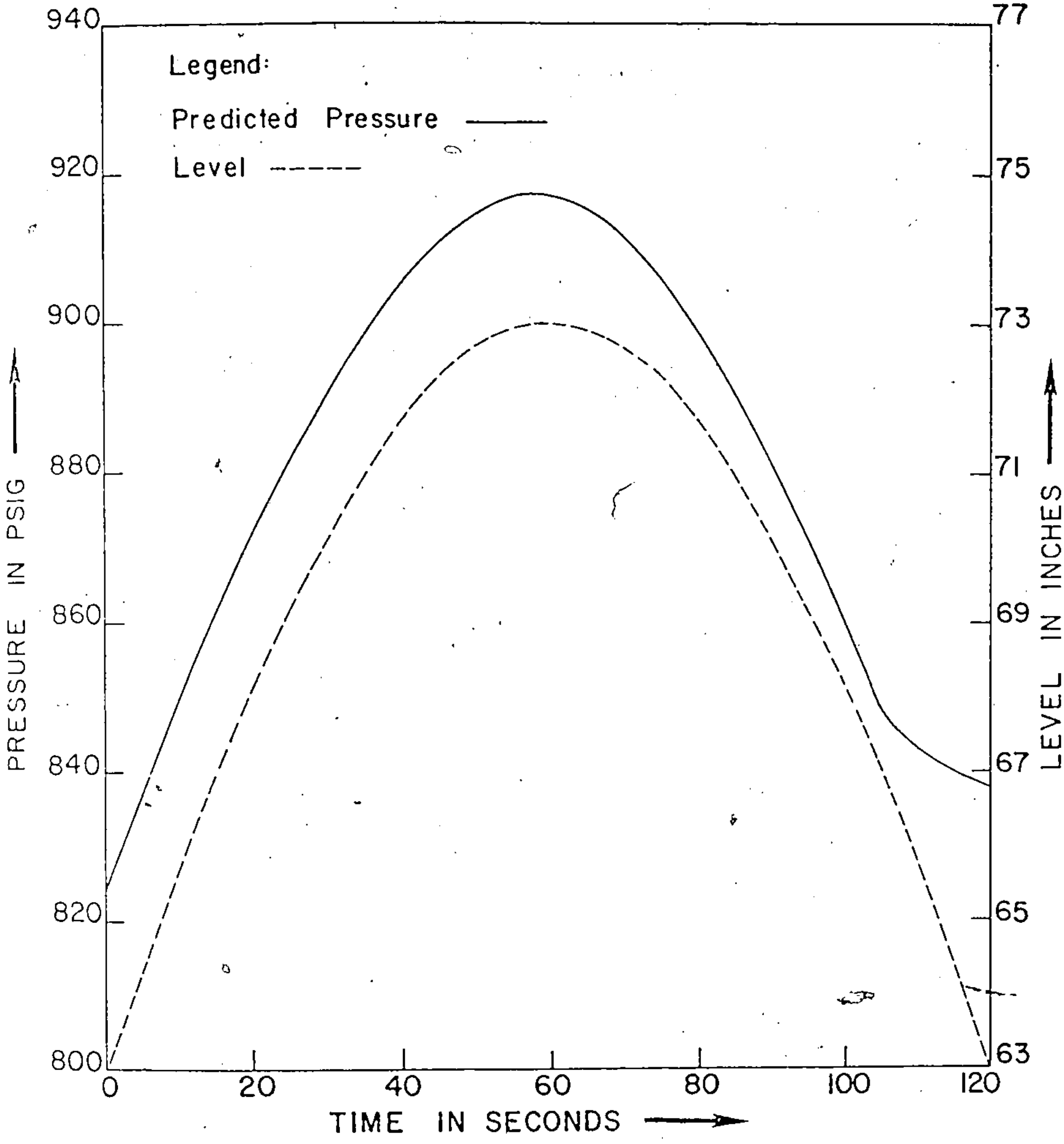

FIG. 5.11 PRESSURE PREDICTION & LEVEL HISTORY FOR CASE 2, INSURGE FOLLOWED BY OUTSURGE

5.4 Discussion and Conclusions

It is evident that the theoretical models and the computational procedures presented in this thesis provide a very powerful means for predicting the performance of the surge tanks of a nuclear power plant, during the surges of the liquid. Although the experimental data have been verified for the limited range, the included computer programmes can be used for the analysis over a wide range of pressures.

Though the computer programme has been written for the vertical cylindrical tank with the hemispherical domes, there would not be any difficulty in introducing the modifications to handle any particular geometry of the pressurizer.

The work could be extended by carrying out the series of experiments on either an actual surge tank or a model surge tank, and producing the accurate experimental data for the multiple surges for the purpose of verification of the validity of the theoretical analysis. It is reasonable to expect that a good agreement would be obtained between the predicted pressure history and the experiments, provided the accurate experimental data are available for this purpose. A pilot project of constructing a small surge tank has been initiated at the University of Ottawa and the information and the experience gained from it will be used for constructing a larger and more sophisticated surge tank of the size encountered in the practice, so that the experimental studies can be continued on the various aspects of the problems. The experimental study is very important both for checking the analytical model and for better understanding of certain phenomena occuring in the surge tank which are almost impossible to investigate by the theoretical models. Specifically inspray and condensation phenomena during insurge and subsequent evaporation during case 1 of multiple surge can only be fully understood by experiments.

The work could also be extended by considering the effect of inspray of subcooled liquid droplets through a nozzle in the vapor region. This type of a fine spray is often used to decrease the pressure rise during the insurge. However, there would not be any difficulty in making the appropriate modifications in the analysis.

Another area of interest which does not appear to have been explored by the investigators, involves the liquid insurge and outsurge to the system where the vapor contains the non-condensible gases. This problem may become of more interest to nuclear designers if non-condensible gases are used extensively in the system.

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APPENDIX 1

DERIVATION OF THE PURE INSURGE

During the insurge, the liquid and the vapor phases will no longer be in thermodynamic equillibrium and vapor will be superheated and the liquid subcooled. The boundary constraints for our system (refer Fig. 3.1) are as mentioned below.

$$V_d = V_s + V_w$$

$$V_s = v_{sus} M_s$$

$$V_w = v_{suw} M_w$$

$$\frac{dV_d}{dt} = \frac{d}{dt} (v_{suw} M_w + v_{sus} M_s) = 0 \qquad (A.1)$$

Equation of continuity and the equation of energy for the two seperate phases are

$$\dot{M}_w = W_{fd} - W_7 \qquad (A.2)$$

$$\dot{M}_s = W_{gd} - W_2 \qquad (A.3)$$

$$\frac{d}{dt}(M_w h_{suw}) = \frac{M_w v_{suw}}{J} \frac{dP}{dt} + W_{fd} h_{saw} - W_7 h_{suw} \qquad (A.4)$$

$$\frac{d}{dt}(M_s h_{sus}) = \frac{M_s v_{sus}}{J} \frac{dP}{dt} + W_{gd} h_{sas} - W_2 h_{sus} + \frac{dQ}{dt} \qquad (A.5)$$

where the kinetic and the potential energies were assumed to be negligibly small. The liquid and the vapor properties can be expressed in several ways; the most convenient of which are

$$v_{suw} = v_{suw}(h_{suw}) \quad \text{and} \quad \dot{v}_{suw} = \frac{dv_{suw}}{dh_{suw}} \dot{h}_{suw} \tag{A.6}$$

$$v_{sus} = v_{sus}(P, h_{sus})$$

and

$$\dot{v}_{sus} = \left(\frac{\partial v_{sus}}{\partial P}\right)_h \frac{dP}{dt} + \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{dh_{sus}}{dt} \tag{A.7}$$

From Eq. (A.2) and Eq. (A.4), we get,

$$\frac{dh_{suw}}{dt} = \frac{v_{suw}}{J} \frac{dP}{dt} + \frac{W_{fd}}{M_w}(h_{saw} - h_{suw}) \tag{A.8}$$

From Eq. (A.3) and Eq. (A.5); we get,

$$\frac{dh_{sus}}{dt} = \frac{v_{sus}}{J} \frac{dP}{dt} - \frac{W_{gd}}{M_s}(h_{sus} - h_{sas}) + \frac{1}{M_s} \frac{dQ}{dt} \tag{A.9}$$

The Eqs. (A.1), (A.3) and (A.6) can be combined into the form

$$M_w \frac{dv_{suw}}{dh_{suw}} \frac{dh_{suw}}{dt} + v_{suw}(W_{fd} - W_7) + v_{sus}(W_{gd} - W_2)$$

$$+ M_s \left[\left(\frac{\partial v_{sus}}{\partial P}\right)_h \frac{dP}{dt} + \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{dh_{sus}}{dt}\right] = 0 \tag{A.10}$$

Rearranging Eq. (A.8), (A.9) and (A.10) as below,

$$\frac{dh_{sus}}{dt} - \frac{v_{sus}}{J} \frac{dP}{dt} = -\frac{W_{gd}}{M_s}(h_{sus} - h_{sas}) + \frac{1}{M_s} \frac{dQ}{dt}$$

$$\frac{dh_{suw}}{dt} - \frac{v_{suw}}{J} \frac{dP}{dt} = \frac{W_{fd}}{M_w}(h_{saw} - h_{suw})$$

$$M_s \frac{\partial v_{sus}}{\partial h_{sus}} \frac{dh_{sus}}{dt} + M_w \frac{dv_{suw}}{dh_{suw}} \frac{dh_{suw}}{dt} + M_s \frac{\partial v_{sus}}{\partial P} \frac{dP}{dt} = -v_{suw}(W_{fd} - W_7)$$
$$-v_{sus}(W_{gd} - W_2)$$

From the above expressions an array was formed as shown below and the equations were solved for dP/dt

$$\frac{dP}{dt} = \frac{\begin{vmatrix} 0 & 1 & +\frac{W_{fd}}{M_w}\left(h_{saw}-h_{suw}\right) \\ 1 & 0 & -\frac{W_{gd}}{M_s}\left(h_{sus}-h_{sas}\right) + \frac{1}{M_s}\frac{dQ}{dt} \\ M_s\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P & M_w\frac{dv_{suw}}{dh_{suw}} & -v_{suw}\left(W_{fd}-W_7\right) \quad -v_{sus}\left(W_{gd}-W_2\right) \end{vmatrix}}{\begin{vmatrix} 0 & 1 & -\frac{v_{suw}}{J} \\ 1 & 0 & -\frac{v_{sus}}{J} \\ M_s\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P & M_w\frac{dv_{suw}}{dh_{suw}} & M_S\left(\frac{\partial v_{sus}}{\partial P}\right)_h \end{vmatrix}}$$

Thus
$$\frac{dP}{dt} = \frac{1}{D}\Big[v_{suw}(W_7 - W_{fd}) + v_{sus}(W_2 - W_{gd}) + W_{gd}(h_{sus} - h_{sas})\frac{\partial v_{sus}}{\partial h_{sus}}$$
$$- W_{fd}(h_{saw} - h_{suw})\frac{dv_{suw}}{dh_{suw}} - \frac{\partial v_{sus}}{\partial h_{sus}}\frac{dQ}{dt}\Big] \qquad (A.11)$$

where
$$D = M_s\frac{\partial v_{sus}}{\partial h_{sus}}\frac{v_{sus}}{J} + M_s\frac{\partial v_{sus}}{\partial P} + \frac{v_{suw}}{J}M_w\frac{dv_{suw}}{dh_{suw}}$$

The equations presented above constitute a dynamic model of the steam drum as shown in the figure 3.1.It is desirable to check them againest the experimental data. Hence the equations can be adapted for the NPD pressurizer by setting

$$W_{fd} = W_{gd} = W_1 = W_2 = W_3 = W_6 = 0$$

Thus the Eq. (A.11) reduces to,
$$\frac{dP}{dt} = \frac{1}{D}\Big[v_{suw}W_7 - \frac{dQ}{dt}\frac{\partial v_{sus}}{\partial h_{sus}}\Big] \qquad (A.12)$$

and D remains the same.

However, Eq.(A.2) gives,
$$W_7 = -\frac{dM_w}{dt}$$

and substituting this into Eq.(A.12) yields,

$$\frac{dP}{dt} = \frac{1}{D}\Big[-v_{suw}\frac{dM_w}{dt} - \frac{\partial v_{sus}}{\partial h_{sus}}\frac{dQ}{dt}\Big] \qquad (A.13)$$

Eq.(A.8) gives,

$$\frac{dh_{suw}}{dt} = \frac{v_{suw}}{J}\frac{dP}{dt} \qquad (A.14)$$

Substituting this value into Eq.(A.6) we have,

$$\dot{v}_{suw} = \frac{dv_{suw}}{dh_{suw}}\frac{v_{suw}}{J}\frac{dP}{dt} \qquad (A.15)$$

Now

$$\frac{dV_w}{dt} = v_{suw}\frac{dM_w}{dt} + M_w\frac{dv_{suw}}{dt}$$

$$\frac{dh_{suw}}{dt}\frac{dv_{suw}}{dh_{suw}} = \frac{1}{M_w}\left[\frac{dV_w}{dt} - v_{suw}\frac{dM_w}{dt}\right]$$

$$\frac{dM_w}{dt} = \frac{1}{v_{suw}}\frac{dV_w}{dt} - \frac{M_w}{v_{suw}}\frac{dv_{suw}}{dh_{suw}}\frac{dh_{suw}}{dt} \qquad (A.16)$$

Substituting Eq.(A.14) into Eq.(A.16), we get,

$$\frac{dM_w}{dt} = \frac{1}{v_{suw}}\frac{dV_w}{dt} - M_w\frac{dv_{suw}}{dh_{suw}}\frac{1}{J}\frac{dP}{dt} \qquad (A.17)$$

Substituting Eq.(A.17) into Eq.(A.13), yields,

$$D\frac{dP}{dt} = -v_{suw}\left[\frac{1}{v_{suw}}\frac{dV_w}{dt} - M_w\frac{dv_{suw}}{dh_{suw}}\frac{1}{J}\frac{dP}{dt}\right] - \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{dQ}{dt}$$

$$= -\frac{dV_w}{dt} + v_{suw}\frac{M_w}{J}\frac{dv_{suw}}{dh_{suw}}\frac{dP}{dt} - \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{dQ}{dt}$$

Hence

$$\left(M_s\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{v_{sus}}{J} + M_s\left(\frac{\partial v_{sus}}{\partial P}\right)_h + \frac{v_{suw}}{J}M_w\frac{dv_{suw}}{dh_{suw}}\right)\frac{dP}{dt}$$

$$= -\frac{dV_w}{dt} + M_w\frac{v_{suw}}{J}\frac{dv_{suw}}{dh_{suw}}\frac{dP}{dt} - \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{dQ}{dt}$$

i.e.

$$M_s\left[\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{v_{sus}}{J} + \left(\frac{\partial v_{sus}}{\partial P}\right)_h\right]\frac{dP}{dt} + \frac{dV_w}{dt} + \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P\frac{dQ}{dt} = 0$$

(A.18)

Using the various identities Moeck (6) has shown that,

$$\left[\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{v_{sus}}{J} + \left(\frac{\partial v_{sus}}{\partial P}\right)_h \right] \frac{1}{v_{sus}} = -\frac{1}{PK} \qquad \text{(A.19)}$$

Also $\quad V_w = a + b\,L$

$$V_d - V_s = a + b\,L$$

$$V_s = V_d - a - b\,L$$

$$M_s = \frac{V_d - a - b\,L}{v_{sus}}$$

$$\frac{dV_w}{dt} = b\,\frac{dL}{dt}$$

Substituting the above equations into Eq.(A.18) we get,

$$-\frac{1}{P\,K}\,(V_d - a - b\,L)\,\frac{dP}{dt} + b\,\frac{dL}{dt} + \left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{dQ}{dt} = 0$$

or in a more workable form,

$$\frac{dP}{dt} = \frac{P\,K\,b\,\frac{dL}{dt} + P\,K\left(\frac{\partial v_{sus}}{\partial h_{sus}}\right)_P \frac{dQ}{dt}}{(V_d - a - b\,L)} \qquad \text{(A.20)}$$

APPENDIX 2

RATE OF HEAT TRANSFER TO THE VESSEL WALL DURING INSURGE

For computing the rate of heat transfer to the pressurizer wall, the temperature distribution inside the wall should be calculated first. The following assumpsions are made in this respect.

1) The inner surface of the wall follows the saturation temperature history.

2) Heat content in the wall at time equal to zero is zero. As the initial temperature distribution in the wall is uniform, it is convinient to assume this as the datum for finding out the total amount of heat transferred.

3) At the start of the insurge, all the contents of the tank are at the uniform saturation temperature corresponding to the initial pressure of the system.

4) The pressurizer has the perfect thermal insulation on the outer surface, so that the heat transfer to the atmosphere from the outer surface of the wall is completely abscent.

5) For the pressurizer under investigation, thickness to diameter ratio is less than 0.1 and hence is assumed as the flat plate.

6) Tank is considered as the infinite plate in the two directions having thickness ℓ, so that the heat transfer to the wall is uni-directional.

7) The temperature of the incoming liquid does not affect the process, that is no significant mixing of the surging liquid and the main liquid volume occurs.

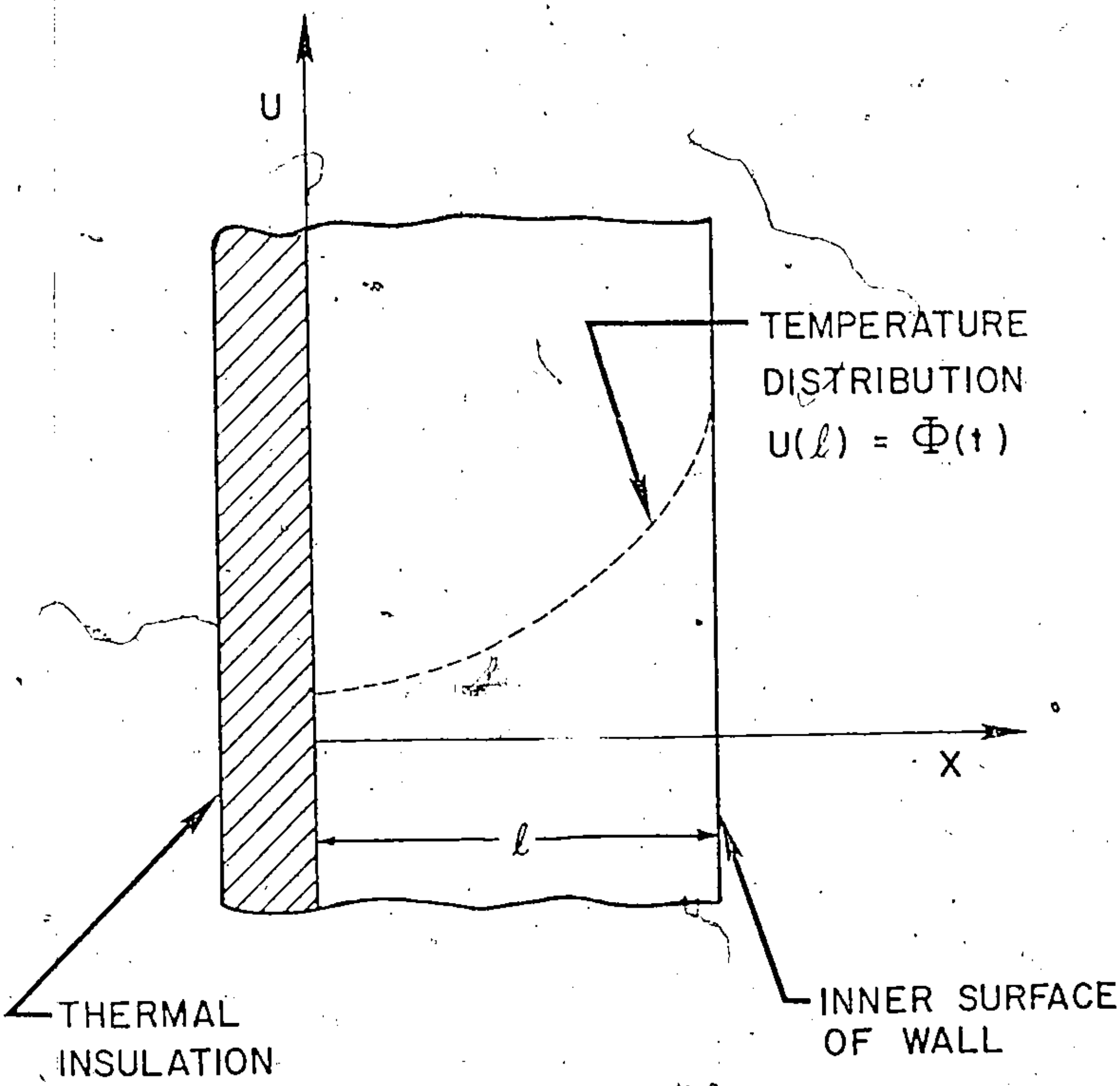


FIG. A.2.1 PRESSURIZER VESSEL WALL

The temperature distribution inside the pressurizer wall can therefore be obtained from Carslaw and Jaeger (14) as follows.

$$u_x = \frac{2}{\ell} \sum_{n=0}^{\infty} e^{-\alpha(2n+1)^2 \pi^2 t/4\ell^2} \cos \frac{(2n+1)\pi x}{2\ell}$$

$$\frac{(2n+1)\pi\alpha(-1)^n}{2\ell} \int_0^t e^{\alpha(2n+1)^2 \pi^2 \zeta/4\ell^2} \phi(\zeta)\, d\zeta \qquad (A.2.1)$$

Substituting, $\frac{(2n+1)^2\pi^2 \alpha}{4\ell^2} = a_n$ into Eq.(A.2.1), yields

$$u_x = \frac{2}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \cos \frac{\sqrt{a_n} x}{\sqrt{\alpha}} \sqrt{\alpha a_n} (-1)^n \int_0^t e^{a_n \zeta} \phi(\zeta)\, d\zeta \qquad (A.2.2)$$

Heat content in the wall at any time t is given by the expression as follows

$$Q = \int_0^{\ell} (A\, dx)\, C_p\, u_x\, \rho$$

Since the diameter to thickness ratio of the vessel wall is very high, we can assume that the area exposed to the vapor remains constant with respect to x, the length coordinate, but is the function of t only.

In order to permit the calculations of the heat content of the wall at any time, we have to calculate the following integral.

$$\int_0^{\ell} u_x\, dx = \int_0^{\ell} \frac{2}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \cos \frac{\sqrt{a_n} x}{\sqrt{\alpha}} \sqrt{\alpha a_n} (-1)^n$$

$$\times \int_0^t e^{a_n \zeta} \phi(\zeta)\, d\zeta$$

$$= \frac{2}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \sqrt{\alpha a_n} \, (-1)^n \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta \int_0^{\ell} \cos \frac{\sqrt{a_n} \, x}{\sqrt{\alpha}} \, dx$$

$$= \frac{2}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \alpha \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta \qquad \text{(A.2.4)}$$

Eq.(A.2.4) is obtained by using the identity:-

$$\sin \frac{(2n+1)\pi}{2} = (-1)^n$$

Substituting Eq.(A.2.4) into Eq.(A.2.3), we get,

$$Q = \frac{2 A C_p \rho \alpha}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta$$

or

$$Q = \frac{2 A k}{\ell} \sum_{n=0}^{\infty} e^{-a_n t} \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta \qquad \text{(A.2.5)}$$

using the definition of thermal conductivity as $k = C_p \alpha \rho$

Notice that heat transfer from time 0 to t, to the wall is equivalent to heat content of the wall at time t (as heat content at time = 0 is 0). Hence upon one one differentiation of Eq.(A.2.5) with respect to time, the rate of heat transfer is obtained.

$$\frac{dQ}{dt} = \frac{2k}{\ell} \left[\sum_{n=0}^{\infty} \frac{dA}{dt} e^{-a_n t} \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta + A e^{-a_n t} (-a_n) \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta + A e^{-a_n t} e^{a_n t} \phi(t) \right]$$

$$= \frac{2k}{\ell} \left[\sum_{n=0}^{\infty} e^{-a_n t} \left[\frac{dA}{dt} - A a_n \right] \int_0^t e^{a_n \zeta} \phi(\zeta) \, d\zeta + A \phi(t) \right] \qquad \text{(A.2.6)}$$

Eq(A.2.6) gives the rate of heat transfer from the vapor to the pressurizer wall due to the condensation of the vapor on the pressurizer wall.

APPENDIX 3

SOLUTION OF THE PARTIAL DIFFERENTIAL EQUATION.

Rate of the heat transfer to the wall, dQ/dt can be found by taking the derivative of Q with respect to the time, so it is necessary to compute Q first. From Appendix 2,

$$Q = A\ C_p\ \rho \int_0^{\ell} u_x\ dx$$

u_x is the temperature distribution in the wall and is described by the Fourier heat conduction equation as

$$\frac{\partial^2 u}{\partial^2 x} = \frac{1}{\alpha}\frac{\partial u}{\partial t}$$

where x is the coordinate along the thickness of the wall. In order to obtain the temperature distribution in the wall at any instant of time, the above partial differential equation must be solved. The initial and boundary conditions for it are:-

1) $u(0,t) = \phi$

   i.e. the inner surface of the pressurizer wall follows the saturation temperature history.

2) $u(x,0) = 0$

   i.e. at the start of the insurge, the pressurizer wall is at the uniform temperature and as discussed earlier, taking that as the datum.

3) $\left.\frac{du}{dx}\right|_{x=\ell} = 0$

   i.e. at the other end of the pressurizer wall, there is a perfect thermal insulation, so that the heat transfer from that end is zero.

The concept used here for solving the partial differential equation is to discretize the independant variable x, and keep the other independant variable, t, as the continuous variable of S360/CSMP. For discretizing x, the wall of the pressurizer was assumed to be having 12 nodal points, making $\Delta x = \ell/12$.

There are three basic finite difference methods for solving a partial differential equation, i.e.,

1) Forward difference method
2) Backward difference method
3) Central difference method

Here for discretizing x, the central difference method was selected. Hence $\partial u/\partial x$ is represented as,

$$\frac{\partial u}{\partial x} = \frac{u_{i-1} - u_{i-1}}{2\,\Delta x}$$

Similarly;

$$\frac{\partial^2 u}{\partial x^2} = \frac{u_{i-1} - 2\,u_i + u_{i+1}}{(\Delta x)^2}$$

Substituting these expressions of the derivatives in the partial differential equation of heat conduction, we get,

$$\frac{\partial u}{\partial t} = \alpha\,\frac{u_{i-1} - 2\,u_i + u_{i+1}}{(\Delta x)^2}$$

In the above equation, the partial differential equation reduces to a total differential equation as,

$$\frac{du}{dt} = \alpha\,\frac{u_{i-1} - 2\,u_i + u_{i+1}}{(\Delta x)^2}$$

or $$\dot{u}_i = \alpha \frac{u_{i-1} - 2\,u_i + u_{i+1}}{(\Delta x)^2}$$

where $i = 1, 2, 3, 4 \ldots\ldots\ldots\ldots 12$

Denoting $\alpha / (\Delta x)^2 = S$, these twelve simultaneous ordinary differential equations can be written out as:-

$$\dot{u}_1 = S\,(u_0 - 2\,u_1 + u_2)$$

$$\dot{u}_2 = S\,(u_1 - 2\,u_2 + u_3)$$

$$\dot{u}_3 = S\,(u_2 - 2\,u_3 + u_4)$$

$$\dot{u}_4 = S\,(u_3 - 2\,u_4 + u_5)$$

$$\dot{u}_5 = S\,(u_4 - 2\,u_5 + u_6)$$

$$\dot{u}_6 = S\,(u_5 - 2\,u_6 + u_7)$$

$$\dot{u}_7 = S\,(u_6 - 2\,u_7 + u_8)$$

$$\dot{u}_8 = S\,(u_7 - 2\,u_8 + u_9)$$

$$\dot{u}_9 = S\,(u_8 - 2\,u_9 + u_{10})$$

$$\dot{u}_{10} = S\,(u_9 - 2\,u_{10} + u_{11})$$

$$\dot{u}_{11} = S\,(u_{10} - 2\,u_{11} + u_{12})$$

$$\dot{u}_{12} = S\,(u_{11} - 2\,u_{12} + u_{13})$$

These are the simple ordinary differential equations and can be solved simultaneously by S360/CSMP.

Implementation of Initial and Boundary Conditions

Initial and boundary conditions are:-

$$u(0,t) = \phi$$

$$u(x,0) = 0$$

$$\left.\frac{du}{dx}\right|_{x=\ell} = 0$$

At any time t, the temperature of the first nodal point is equal to ϕ, hence we have, $u_0 = \phi$. Since u at any nodal point at time equal to zero is zero, the initial conditions of the above differential equations is zero, giving, $u_i(0) = 0$

From the third initial condition, as the derivative of temperature with respect to time at the other end is zero, it can be written by the central difference formula as,

$$\frac{du}{dx} = \frac{u_{i-1} - u_{i+1}}{2\,\Delta x}$$

Hence at $x = \ell$, as $i = 12$, the above expression leads,

$$\left.\frac{du}{dx}\right|_{x=\ell} = \frac{u_{11} - u_{13}}{2\,\Delta x} = 0$$

i.e. $u_{13} = u_{11}$

Eliminating the imaginary temperature of the wall, u_{13}, from the above mentioned twelve differential equations, we get twelve equations for the twelve unknowns.All those twelve differential equations are in the solvable form by the S360/CSMP, and thus the temperature distribution in the wall at any instant can be obtained.

The number of stations in the wall are chosen arbitrarily. But by changing the number of nodal points in the wall to 1o,16 and 20, the computation was carried out and it was observed that the difference in the pressure prediction is in the fifth place. This implies that the choice of twelve nodal points is quite sufficient to render the accurate solution.

Once the partial differential equation of heat conduction is solved and the temperature distribution in the wall is obtained, for computing the rate of heat transfer, we need to evaluate $\int_0^{\ell} u\, dx$. As this integration is with respect to x, and not with t, the CSMP INTGRL statement cannot be used for integration. It is performed by the Simpson Rule of integration.

$$\int_0^x y\, dx = (y_0 + 4(y_{n-1}) + 2(y_{n-2}) + y_n)\ \Delta x/3$$

The truncation error of this formula is very small and the correction to it can be achieved by adding the following term. (25).

$$- \frac{\Delta x}{90} \{ \delta^4 y_1 + \delta^4 y_3 + \delta^4 y_5 \cdots\cdots\cdots \delta^4 y_{n-1} \}$$

$$+ \frac{\Delta x}{756} \{ \delta^4 y_2 + \delta^4 y_4 + \delta^4 y_6 \cdots\cdots\cdots \delta^4 y_{n-2} \}$$

Where the above quantities in the correction term can be found out by forming a difference table.

Since in our case, u is relatively small and the equation is particularly stable, and it is a small part of the main computation, this correction term can be omitted. Hence,

$$\int_0^{\ell} u\, dx = (u_0 + 4 (u_1 + u_3 + u_5 + u_7 + u_9 + u_{11}) +$$

$$+ 2 (u_2 + u_4 + u_6 + u_8 + u_{10}) - u_{12}))\ell/36$$

This value of the integral will change with the time as the value of u_i is changing with time.

Calculation of the Rate of Heat Transfer

By substituting the value of the integral in the Eq.(A.2.3), the total heat content in the wall can be obtained. Once the heat transferred to the wall is known, the rate of the heat transfer can be obtained by differentiating it with respect to time. This is accomplished by using the derivative capability of S360/CSMP.

$$\frac{dQ}{dt} = \text{DERIV (DQDTO , Q)}$$

Where the quantity DQDTO is the rate of heat transfer at time equal to zero. As the temperature distribution in the wall is initially uniform, the rate of heat transfer is zero.

Hence $$\frac{dQ}{dt} = \text{DERIV (0. , Q)}$$

The above expression gives the rate of heat transfer from the vapor to the pressurizer wall, due to the condensation of vapor.

APPENDIX 4

COMPUTER PROGRAM AND OUTPUT FOR PURE OUTSURGE

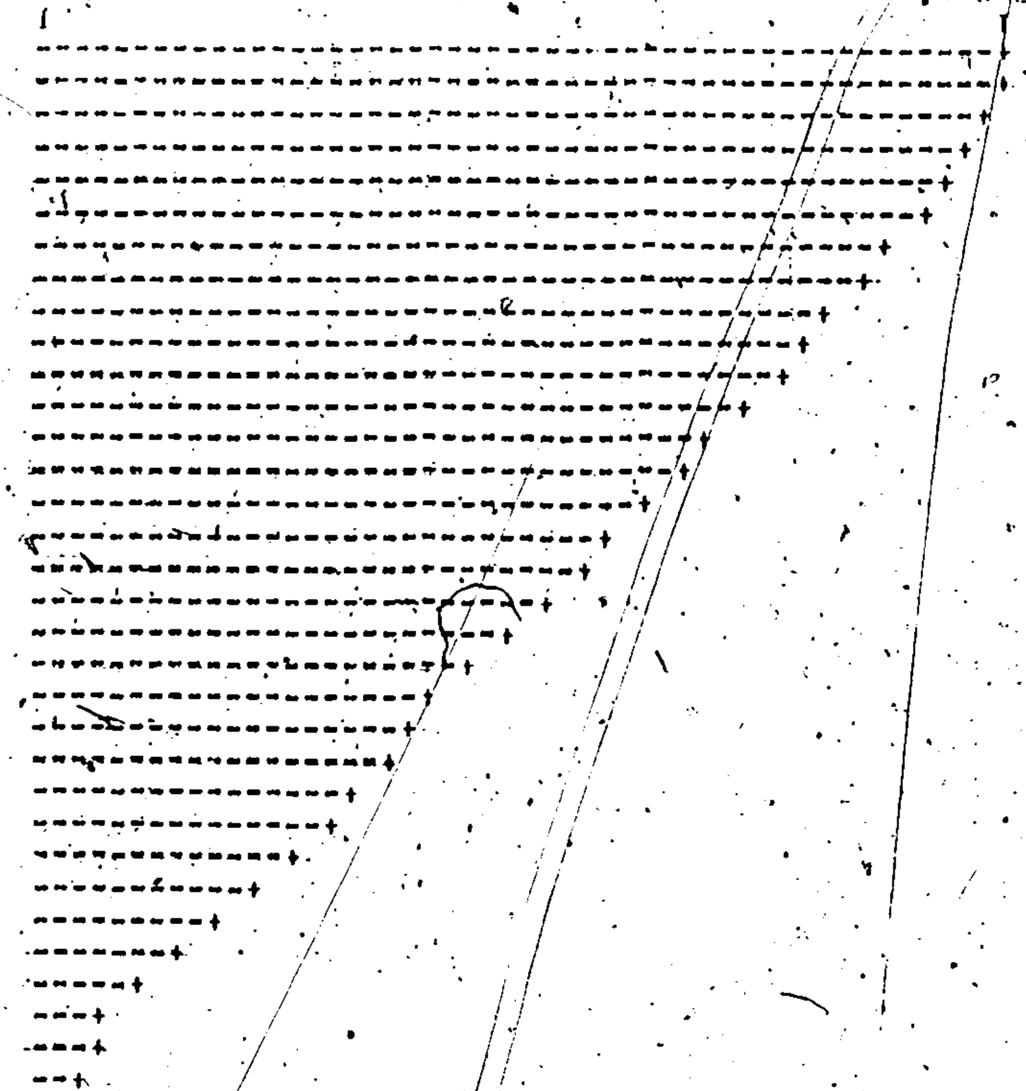
```
****CONTINUOUS SYSTEM MODELING PROGRAM****

***PROBLEM INPUT STATEMENTS***

*************************************************************************
*            THIS PROGRAM SIMULATES THE SURGE TANK IN THE...
*            PRIMARY CIRCUIT OF THE NUCLEAR POWER PLANT.
*            PRESSURE PREDICTION DURING OUTSURGE FOR RUN A2
*            PROGRAMMED BY KIRAN KULKARNI
*************************************************************************

RENAME  TIME = T

*-----------------------------------------------------------------------
*            START OF INITIAL SEGAMENT
*-----------------------------------------------------------------------
INITIAL
PARAM     A = 0.111, J= 10.
          PZERO = 840.* .068948
          PI = 4.0 * ATAN (1.00)
          VD = 200.*(2.54**3.)*(10.**(-6.))*12.*12.*12.
          B = PI* DIAM* DIAM/400.
          DIAM = 50. * .0254
FUNCTION LEVEL = 0.,116.,5.,103.,10.,86.,15.,68.,20.,50.,25.,37., ...
30.,22.,35.,19.

DYNAMIC

*-----------------------------------------------------------------------
*            LEVEL FUNCTION GENERATION
*-----------------------------------------------------------------------

          AL = NLFGEN (LEVEL, T)
          ALDOT = DERIV (-2.6,AL)
          LDOT = ALDOT * 2.54
          L = AL * 2.54
```

```
*-----------------------------------------------------------------------
*          THERMODYNAMIC PROPERTIES
*-----------------------------------------------------------------------

       DVG = -2000.*(PA*PA)
       HF = 747.+ 9.85*PA-.0543*PA*PA
       HFG= 10.**6./(476.+ 2.68*PA)
       HG = 2790.+.898*PA-.0162*PA*PA
       VF = 1.12+ .0033*PA
       VG = 2090./PA -2.46
       DHF = (.241/PA-.000617)*HF
       DHG = (.0196/PA -.000235- .0000126* PA)*HFG
       VFG = 2100. /PA - 3.97

*-----------------------------------------------------------------------
*          EXPRESSION FOR THE PURE OUTSURGE
*-----------------------------------------------------------------------

       DLDP = (A* F1 + VD* F2)/R + F1*L
       DPDL = 1. / DLDP
       PDOUT = DPDL * LDOT
       SOUT = ( VG/VF* DHF - DHG)/HFG
       F1 = SOUT + DVG/ VG
       MOUT = ( DHG- VG/J) / HFG
       F2 = MOUT- DVG/VG
       PA = P+1.
       P = INTGRL (PZERO, PDOUT)
       PRESS = P/.068948

TIMER.  FINTIM = 32., DELT = .001,OUTDEL = 1.
PRTPLT  PRESS
        END
        STOP
```

PRESS VERSUS T

MINIMUM 6.7349E 02 — MAXIMUM 8.4000E 02

T	PRESS
0.0	8.4000E 02
1.0000E 00	8.3677E 02
2.0000E 00	8.3351E 02
3.0000E 00	8.3021E 02
4.0000E 00	8.2687E 02
5.0000E 00	8.2350E 02
6.0000E 00	8.1878E 02
7.0000E 00	8.1419E 02
8.0000E 00	8.0957E 02
9.0000E 00	8.0487E 02
1.0000E 01	8.0009E 02
1.1000E 01	7.9486E 02
1.2000E 01	7.8959E 02
1.3000E 01	7.8422E 02
1.4000E 01	7.7873E 02
1.5000E 01	7.7312E 02
1.6000E 01	7.6738E 02
1.7000E 01	7.6150E 02
1.8000E 01	7.5548E 02
1.9000E 01	7.4929E 02
2.0000E 01	7.4293E 02
2.1000E 01	7.3835E 02
2.2000E 01	7.3354E 02
2.3000E 01	7.2862E 02
2.4000E 01	7.2359E 02
2.5000E 01	7.1844E 02
2.6000E 01	7.1214E 02
2.7000E 01	7.0581E 02
2.8000E 01	6.9931E 02
2.9000E 01	6.9259E 02
3.0000E 01	6.8561E 02
3.1000E 01	6.8418E 02
3.2000E 01	6.8274E 02



APPENDIX 5

COMPUTER PROGRAM AND OUTPUT FOR PURE INSURGE

```
****CONTINUOUS SYSTEM MODELING PROGRAM****

***PROBLEM INPUT STATEMENTS***

*****************************************************************************
*               THIS PROGRAM SIMULATES THE SURGE TANK IN THE...
*               PRIMARY CIRCUIT OF THE NUCLEAR POWER PLANT
*               PRESSURE PREDICTION DURING INSURGE FOR RUN SIX
*               PROGRAMMED BY KIRAN KULKARNI
*****************************************************************************

RENAME  TIME = T
*----------------------------------------------------------------------------
*               START OF INITIAL SEGAMENT
*----------------------------------------------------------------------------
INITIAL
PARAM     K = 1.26
          PI = 4.0 * ATAN (1.00).
          PZERO = 822. * .068948

*----------------------------------------------------------------------------
*               GEOMETRY OF THE PRESSURIZER
*----------------------------------------------------------------------------
PARAM     A= 0.111
          DIAM = 50. * .0254
          HEIGHT = 154. * 2.54
          AHEMS = PI * DIAM* DIAM/2.
          THW = 23.* .0254 / 16.
          VD = 200.*(2.54**3.)*(10.**(-6.))*12.*12.*12.
          B = PI* DIAM* DIAM/400.

*----------------------------------------------------------------------------
*               THERMAL PROPERTIES OF THE WALL
*----------------------------------------------------------------------------
PARAM     SPHEAT=0.111, AK= 26., DGDT0 = 0.
          DX = THW/12.
          S = ALPHA/(DX**2.)
          SPHE = SPHEAT* 1054.8
          RHO = 490. / ((12.*.0254)**3.)
          COND = AK * 1054.8 / (3600.*12.*.0254)
          ALPHA =(AK/(RHO*SPHEAT))*.0254*.0254*144./3600.
```

```
DYNAMIC

*------------------------------------------------------------------------
*            CALCULATION OF THE VAPOR SURFACE AREA
*------------------------------------------------------------------------

      AWALL = PI* DIAM* (HEIGHT-L)/100.
      AREA= AHEMS + AWALL

      P = INTGRL (PZERO,PDOT)
      PDOT = (PA K* B* LDOT - M)/(VD-A-B*L)
      M = K*DRDH*DODT
      PRESS = P/.068948
*------------------------------------------------------------------------
*            SATURATION TEMP HISTORY OF WALL
*------------------------------------------------------------------------
      TSAT = 976.416 +(P-55.) * 1.920852
      PHI0 = 976.416 + (PZERO- 55.) * 1.920852
      PHIM = TSAT - PHI0

*------------------------------------------------------------------------
*            LEVEL FUNCTION GENERATION
*------------------------------------------------------------------------
FUNCTION LE= 0.,60.,10.,60.75,20.,62.,30.,63.5,40.,64.5,50.,66.,60.,67.
      ALI= AFGEN(LE,T)
      L = ALI*2.54
      ALDOT = DERIV(0.75,ALI)
      LDOT = ALDOT*2.54

*------------------------------------------------------------------------
*            CALCULATION OF THE DENSITY TERM
*------------------------------------------------------------------------
      PA = P+ 1.0
      HFGH= (10.**(6.))/(476.+2.68*PA)
      DRDH = (2.32-.0106*PA+5.26*(10.**(-5.))*PA*PA)*VFGH/HFGH
      VFGH= (2100. -3.97*PA)*(10.**(-6.))
```

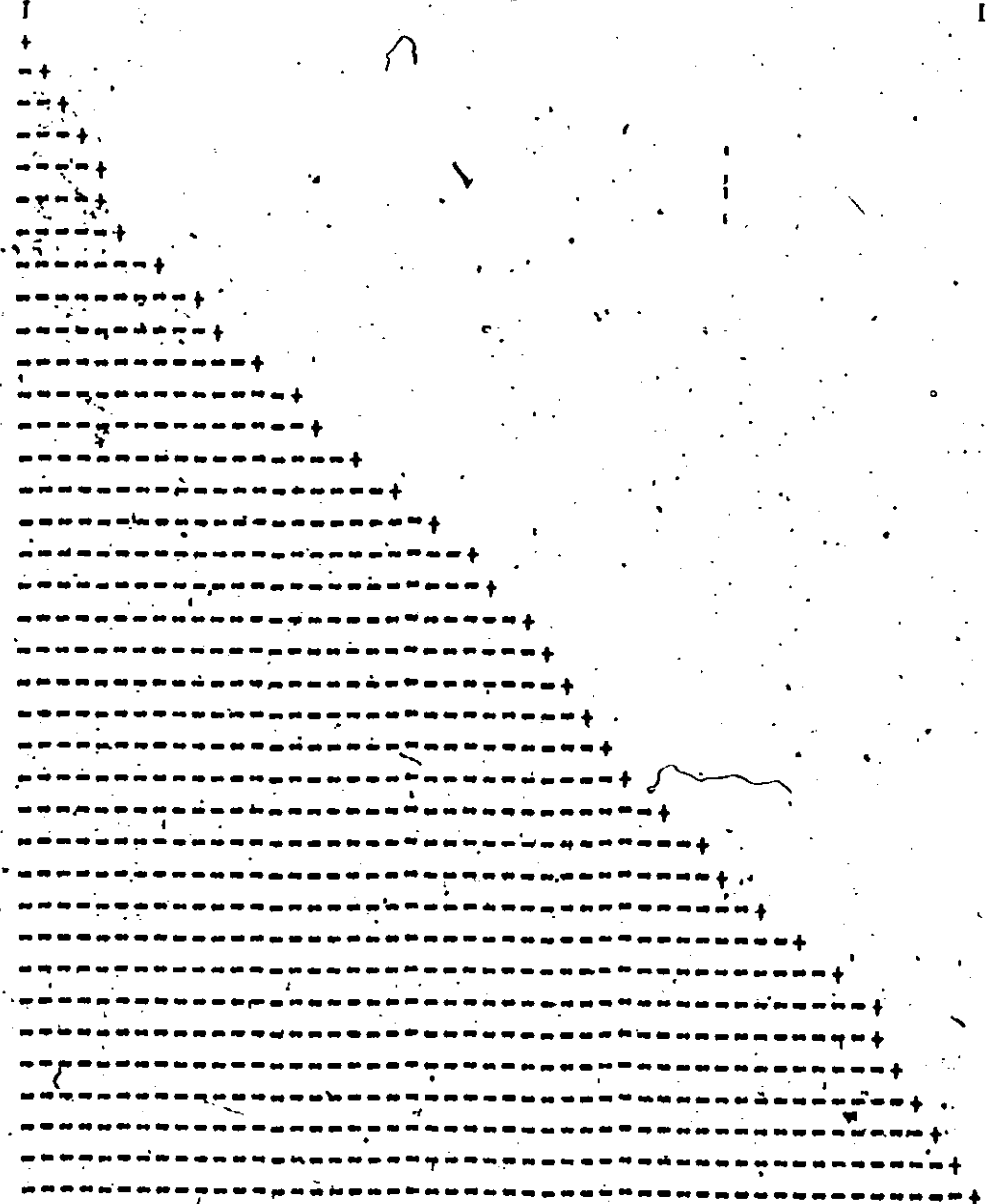
```
*---------------------------------------------------------------------
*              CALCULATION OF THE HEAT TRANSFER TERM
*              SOLUTION OF THE PARTIAL DIFF EQUATION
*---------------------------------------------------------------------

      U0 = PHIN
      UD1 = S*(U0-2.*U1+U2)
      UD2 = S*(U1-2. * U2 +U3)
      UD3 = S* (U2 -2.*U3+U4)
      UD4 = S* (U3- 2. *U4 +U5)
      UD5 = S* (U4 - 2.*U5 + U6)
      UD6 = S* ( U5 -2.*U6+U7)
      UD7 = S* ( U6 - 2.*U7 + U8)
      UD8 = S* (U7 - 2.* U8 + U9)
      UD9 = S* ( U8 - 2.*U9 +U10)
      UD10 =  S* (U9- 2.0*U10+ U11)
      UD11 = S* (U10-2.0*U11+ U12)
      UD12 = S* (U11-U12)*2.
      U1 = INTGRL (0.0,UD1)
      U2 = INTGRL (0.0, UD2)
      U3 = INTGRL (0.0,UD3)
      U4 = INTGRL (0.,UD4)
      U5 = INTGRL (0., UD5)
      U6 = INTGRL (0., UD6)
      U7 = INTGRL (0.,UD7)
      U8 = INTGRL (0.0, UD8)
      U9 = INTGRL (0.0, UD9)
      U10= INTGRL (0.0, UD10)
      U11 = INTGRL (0.0, UD11)
      U12  = INTGRL (0.0, UD12)

*---------------------------------------------------------------------
*              INTEGRATION BY THE SIMPSON RULE
*---------------------------------------------------------------------

      SIGMA=DX*(U0+4.*(U1+U3+U5+U7+U9+U11)+2.*(U2+U4+U6+U8+U10)+U12 )/3 .

      Q = AREA* RHO *SPHE* SIGMA
      DQDT= DERIV(DQDT0,Q)
```

```
*----------------------------------------------------------------------------
*              PRESSURE PREDICTION NEGLECTING HEAT TRANSFER
*----------------------------------------------------------------------------
         PN = INTGRL(PZERO, PNDOT)
         PNDOT= PN* K *B* LDOT/(VD- A- B*L)
         PNOHT = PN/.068948

TIMER    FINTIM = 60.,  DELT = .0001, OUTDEL = 2., PRDEL = 5.
PRINT    PRESS, PNOHT, Q
PRTPLT   PRESS
LABEL    PRESSURE PREDICTION FOR NPD PRESSURIZER. RUN SIX
         END
         STOP
```

PRESS VERSUS T

MINIMUM 8.2200E 02 — MAXIMUM 8.7179E 02

T	PRESS
0.0	8.2200E 02
1.6667E 00	8.2303E 02
3.3333E 00	8.2403E 02
5.0000E 00	8.2502E 02
6.6667E 00	8.2598E 02
8.3333E 00	8.2694E 02
1.0000E 01	8.2788E 02
1.1667E 01	8.2967E 02
1.3333E 01	8.3129E 02
1.5000E 01	8.3287E 02
1.6667E 01	8.3443E 02
1.8333E 01	8.3597E 02
2.0000E 01	8.3751E 02
2.1667E 01	8.3947E 02
2.3333E 01	8.4134E 02
2.5000E 01	8.4318E 02
2.6667E 01	8.4501E 02
2.8333E 01	8.4683E 02
3.0000E 01	8.4865E 02
3.1667E 01	8.4954E 02
3.3333E 01	8.5059E 02
3.5000E 01	8.5167E 02
3.6667E 01	8.5274E 02
3.8333E 01	8.5382E 02
4.0000E 01	8.5490E 02
4.1667E 01	8.5689E 02
4.3333E 01	8.5873E 02
4.5000E 01	8.6053E 02
4.6667E 01	8.6232E 02
4.8333E 01	8.6411E 02
5.0000E 01	8.6589E 02
5.1667E 01	8.6671E 02
5.3333E 01	8.6770E 02
5.5000E 01	8.6872E 02
5.6667E 01	8.6974E 02
5.8333E 01	8.7076E 02
6.0000E 01	8.7179E 02



APPENDIX 6

COMPUTER PROGRAM AND OUTPUT FOR CASE 1

OUTSURGE FOLLOWED BY INSURGE

```
****CONTINUOUS SYSTEM MODELING PROGRAM****

***PROBLEM INPUT STATEMENTS***

*****************************************************************************
*             CASE ONE
*             THIS PROGRAM SIMULATES THE SURGE TANK IN THE...
*             PRIMARY CIRCUIT OF THE NUCLEAR POWER PLANT.
*             PROGRAMMED BY KIRAN KULKARNI
*****************************************************************************

RENAME   TIME = T
/          DIMENSION    C(5)
/          EQUIVALENCE   (L1,C(1)),(L2,C(2)),(L3,C(3)),(L4,C(4)),(L5,C(5))
           PHIMIN = 1.
*----------------------------------------------------------------------------
*             START OF INITIAL SEGAMENT
*----------------------------------------------------------------------------
INITIAL
PARAM      K= 1.26, DO = 860., J = 10.,RO = 0.8720, L1 = -1102.666
PARAM      L2= 465.7617, L3 =2045.262, L4= -31[illegible]5.968, L5 = 1274.154
           KEYSAT = 0.
           RAPX = 85.8/144.
           PZERO = DO * .068948
           KEYHET = 0.

*----------------------------------------------------------------------------
*             DATA FOR THE RUN
*----------------------------------------------------------------------------
PARAM      LINITI = 61., LFINAL = 41.
           W = PI/120.
           LCMS = LINITI * 2.54


*----------------------------------------------------------------------------
*             GEOMETRY OF THE PRESSURIZER
*----------------------------------------------------------------------------
PARAM      A= 0.111
           B = PI* DIAM* DIAM/400.
```

```
      VD = 200.*(2.54**3.)*(10.**(-6.))*12.*12.*12.
      RAD = DIAM /2.
      HEIGHT = 154. * 2.54
      AHEMS = PI * DIAM* DIAM/2.
      THW = 23.* .0254 / 16.
      PI = 4.0 * ATAN (1.00)
      PI = 4.0 * ATAN(1.00)
      DIAM = 50. * .0254

*------------------------------------------------------------------------
*           THERMAL PROPERTIES OF THE WALL
*------------------------------------------------------------------------
PARAM SPHEAT=0.111, AK= 26., DODT0 = 0.
      DX = THW/12.
      S = ALPHA/(DX**2.)
      SPHE = SPHEAT* 1054.8
      RHO = 490. / ((12.*.0254)**3.)
      COND = AK  * 1054.8 / (3600.*12.*.0254)
      ALPHA =(AK/(RHO*SPHEAT))*.0254*.0254*144./3600.

DYNAMIC

*------------------------------------------------------------------------
*           LEVEL FUNCTION GENERATION
*------------------------------------------------------------------------

      ALI = LINITI -C2
      C2 = - ( LFINAL - LINITI) * SIN (WAT)
      L = ALI*2.54
      LDOT = ALDOT*2.54
      ALDOT= ( LFINAL - LINITI) *WACOS (WAT)

*------------------------------------------------------------------------
*           CALCULATION OF THE VAPOR SURFACE AREA
*------------------------------------------------------------------------

      AWALL = PI* DIAM* (HEIGHT-L)/100.
      AREA= AHEMS + AWALL
```

```
*------------------------------------------------------------------------
*               SATURATION TEMP HISTORY OF WALL
*------------------------------------------------------------------------
        TSAT = 976.416 +(P-55.) * 1.920852
        PHI0 = 976.416 + (PZERO- 55.) * 1.920852
        PHIM = TSAT - PHI0

*------------------------------------------------------------------------
*               THERMODYNAMIC PROPERTIES
*------------------------------------------------------------------------

        PA = P+ 1.0
        DVG = -2090./(PA*PA)
        HF = 747.+ 9.85*PA-.0343*PA*PA
        HFG= 10.**6./(476.+ 2.68*PA)
        HG = 2790.+.098*PA-.0162*PA*PA
        VF = 1.12+ .0033*PA
        VG = 2090./PA -2.46
        DHF = (.241/PA-.000617)*HF
        DHG = (.0196/PA -.000235- .0000126* PA)*HFG
        VFG = 2100. /PA - 3.97

*------------------------------------------------------------------------
*               CALCULATION OF THE DENSITY TERM
*------------------------------------------------------------------------
        HFGH= (10.**(6.))/(476.+2.68*PA)
        VFGH= (2100. -3.97*PA)*(10.**(-6.))
        DRDH = (2.32-.0106*PA+5.26*(10.**(-5.))*PA*PA)*VFGH/HFGH

*------------------------------------------------------------------------
*               CALCULATION OF THE HEAT TRANSFER TERM
*               SOLUTION OF THE PARTIAL DIFF EQUATION
*------------------------------------------------------------------------


        U0 = FCNSW(KEYHET, 0.,0.,PHIM)
        UD1 = S*(U0-2.*U1+U2)
        UD2 = S*(U1-2. * U2 +U3)
        UD3 = S* (U2 -2.*U3+U4)
        UD4 = S* (U3- 2. *U4 +U5)
```

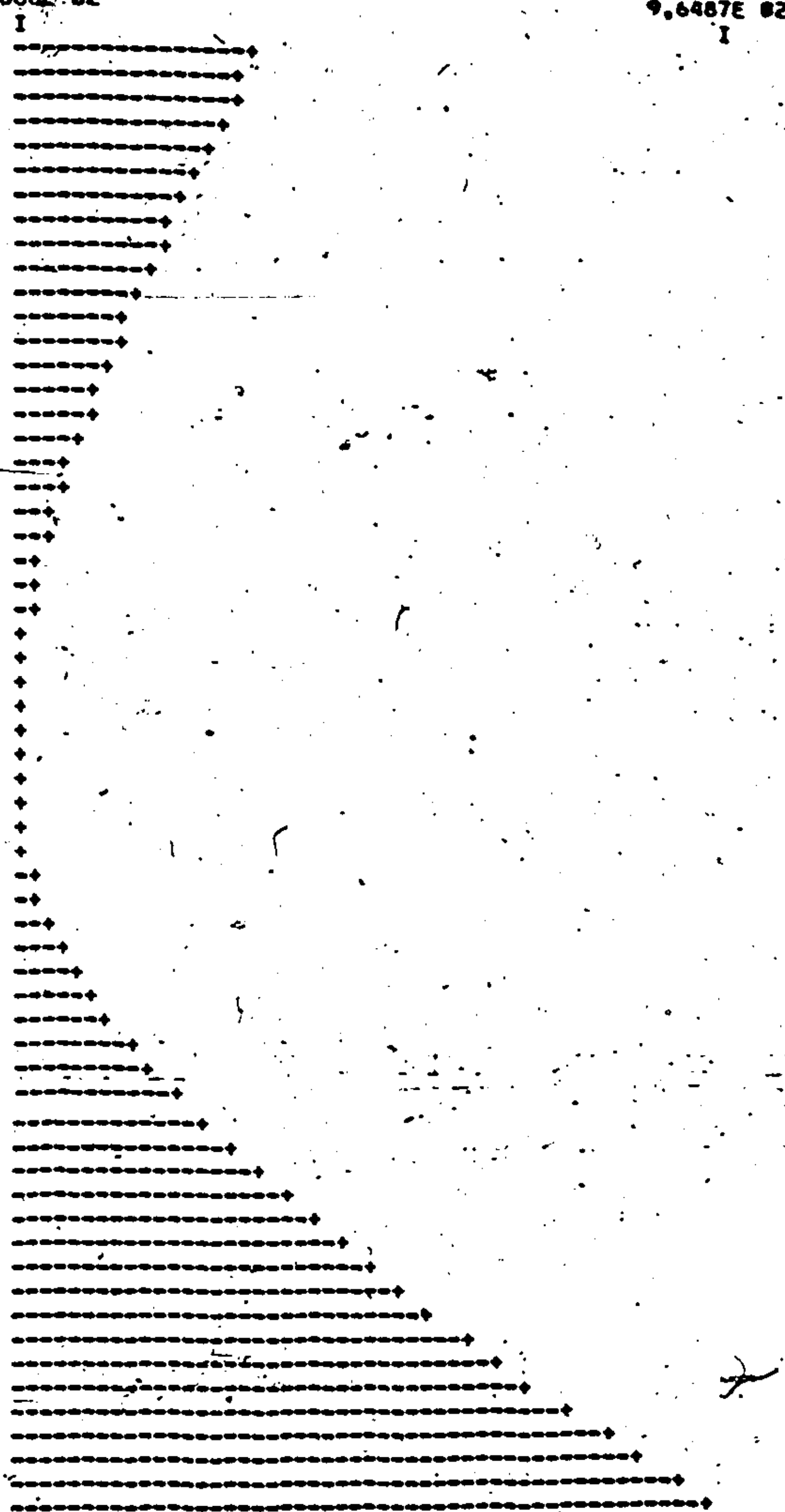
```
UD5 = S* (U4 - 2.*U5 + U6)
UD6 = S* ( U5 -2.*U6+U7)
UD7 = S* ( U6 - 2.*U7 + U8)
UD8 = S* (U7 - 2.* U8 + U9)
UD9 = S* ( U8 - 2.*U9 +U10)
UD10 =  S* (U9- 2.0*U10+ U11)
UD11 = S* (U10-2.0*U11+ U12)
UD12 = S* (U11-U12)*2.
U1 = INTGRL (0.0,UD1)
U2 = INTGRL (0.0, UD2)
U3 = INTGRL (0.0,UD3)
U4 = INTGRL (0.,UD4)
U5 = INTGRL (0., UD5)
U6 = INTGRL (0., UD6)
U7 = INTGRL (0.,UD7)
U8 = INTGRL (0.0, UD8)
U9 = INTGRL (0.0, UD9)
U10= INTGRL (0.0, UD10)
U11 = INTGRL (0.0, UD11)
U12  = INTGRL (0.0, UD12)
*------------------------------------------------------------------------
*               INTEGRATION BY THE SIMPSON RULE
*------------------------------------------------------------------------

SIGMA =DX*(U0+4.*(U1+U3+U5+U7+U9+U11)+2.*(U2+U4+U6+U8+U10)+U12)/3

Q = AREA* RHO *SPHE* SIGMA
DQDT= DERIV(DQDT0,Q)
M = K*DRDH*DQDT
VOLWAT = VD- A - B* L
PDIN = ( P* K* B* LDOT - M )/VOLWAT
KEYIN = FCNSW(LDOT, 0.,1.,1.)
KEYOUT = NOT(KEYIN)
*------------------------------------------------------------------------
*               EXPRESSION FOR THE PURE OUTSURGE
*------------------------------------------------------------------------

DLDP = (A* F1 + VD* F2)/B + F1*L
DPDL = 1. / DLDP
PDOUT = DPDL * LDOT
```

```
      SOUT = ( VG/VF* DHF - DHG)/HFG
      F1 = SOUT + DVG/VG
      HOUT = ( DHG- VG/J) / HFG
      F2 = HOUT- DVG/VG

      PDOT = KEYIN*PDIN + PBOUT*KEYOUT
      P = INTGRL (PZERO,PDOT)
      D = P/.068948
      G = VOLWAT / (.3048**3.)
      MS = (VO-A-B*LCMS)/(.3048**3.*.52954)
      SPVO = G/ MS

*------------------------------------------------------------------------------
*               CALCULATION OF STEAM TEMPERATURE
*------------------------------------------------------------------------------
NOSORT
      IF(KEYIN.EQ.0) GO TO 22
      IF (PHIMIN.GT.1.) GO TO 71
      PHIMIN = PHIMIN*TSAT
71    CONTINUE
      TRAPX = D* SPVO/RAPX
      TRAPX = TRAPX + 150.
      FYT2 = TRAPX
80    T2 = FYT2
      Z = T2
      Y3 = 0.
      N = 0
45    N = N+1
      X3 = C(N)*(D**N)*(N+1) *(1./Z**(N+2))
      Y3 = X3 + Y3
      IF (N.LT.5.) GO TO 45
      R= RO+(1./Z)*(C(1)*D/Z + C(2)*(D/Z)**2.+ C(3)*(D/Z)**3. ...
      + C(4) *(D/Z)**4. + C(5)*(D/Z)**5. )
      FT2 = D* SPVO/R -T2
      FDT2 = (D*SPVO*Y3/(R*R)) -1.
      FYT2 = T2 - FT2/FDT2
      DIFF = FYT2 -T2
      IF (ABS(DIFF).GT.2.) GO TO 80
      KEYSAT = KEYSAT+1.
      IF(KEYSAT.LE.1.) GO TO 41
```

```
          GO TO 42
 41       T20 = FYT2
 42       CONTINUE
          T2 = FYT2
          TAMB = T2 + PHIMIN - T20
          KEYHET = COMPAR(TAMB,PHI0)
          GO TO 81

 22       DEGREE = 0.
          TAMB = TSAT
 81       CONTINUE
SORT
          PRESS = D
TIMER     FINTIM = 120., DELT = .0001, PRDEL = 2., OUTDEL = 2.
PRTPLT    PRESS
          END
          STOP
```

PRESS VERSUS T

MINIMUM 8.0688E 02 — MAXIMUM 9.6487E 02



T	PRESS
0.0	8.6000E 02
2.0000E 00	8.5732E 02
4.0000E 00	8.5464E 02
6.0000E 00	8.5196E 02
8.0000E 00	8.4930E 02
1.0000E 01	8.4666E 02
1.2000E 01	8.4404E 02
1.4000E 01	8.4146E 02
1.6000E 01	8.3892E 02
1.8000E 01	8.3643E 02
2.0000E 01	8.3399E 02
2.2000E 01	8.3162E 02
2.4000E 01	8.2932E 02
2.6000E 01	8.2709E 02
2.8000E 01	8.2496E 02
3.0000E 01	8.2291E 02
3.2000E 01	8.2096E 02
3.4000E 01	8.1912E 02
3.6000E 01	8.1738E 02
3.8000E 01	8.1576E 02
4.0000E 01	8.1427E 02
4.2000E 01	8.1290E 02
4.4000E 01	8.1166E 02
4.6000E 01	8.1056E 02
4.8000E 01	8.0959E 02
5.0000E 01	8.0877E 02
5.2000E 01	8.0809E 02
5.4000E 01	8.0756E 02
5.6000E 01	8.0718E 02
5.8000E 01	8.0696E 02
6.0000E 01	8.0688E 02
6.2000E 01	8.0709E 02
6.4000E 01	8.0773E 02
6.6000E 01	8.0879E 02
6.8000E 01	8.1027E 02
7.0000E 01	8.1218E 02
7.2000E 01	8.1451E 02
7.4000E 01	8.1726E 02
7.6000E 01	8.2044E 02
7.8000E 01	8.2404E 02
8.0000E 01	8.2807E 02
8.2000E 01	8.3252E 02
8.4000E 01	8.3739E 02
8.6000E 01	8.4269E 02
8.8000E 01	8.4840E 02
9.0000E 01	8.5454E 02
9.2000E 01	8.6107E 02
9.4000E 01	8.6712E 02
9.6000E 01	8.7346E 02
9.8000E 01	8.8002E 02
1.0000E 02	8.8680E 02
1.0200E 02	8.9377E 02
1.0400E 02	9.0096E 02
1.0600E 02	9.0835E 02
1.0800E 02	9.1593E 02
1.1000E 02	9.2369E 02
1.1200E 02	9.3163E 02
1.1400E 02	9.3973E 02
1.1600E 02	9.4798E 02
1.1800E 02	9.5637E 02
1.2000E 02	9.6487E 02

APPENDIX 7

COMPUTER PROGRAM AND OUTPUT FOR CASE 2

INSURGE FOLLOWED BY OUTSURGE

```
****CONTINUOUS SYSTEM MODELING PROGRAM****

***PROBLEM INPUT STATEMENTS***

*************************************************************************
*               CASE TWO
*               THIS PROGRAM SIMULATES THE SURGE TANK IN THE...
*               PRIMARY CIRCUIT OF THE NUCLEAR POWER PLANT.
*               PROGRAMMED BY KIRAN KULKARNI
*************************************************************************
RENAME  TIME = T
/         DIMENSION     C(5)
/         EQUIVALENCE    (L1,C(1)),(L2,C(2)),(L3,C(3)),(L4,C(4)),(L5,C(5))

*------------------------------------------------------------------------
*               START OF INITIAL SEGMENT
*------------------------------------------------------------------------
INITIAL
        KEYIN = 1.
PARAM   K= 1.26, DO = 825., J = 10.,RO = 0.8720, L1 = -1102.666
PARAM   L2= 465.7617, L3 =2045.262, L4= -3103.968, L5 = 1274.154
        RAPX = 85.8/144.
        PI = 4.0 * ATAN (1.00)
        PZERO = DO * .068948

*------------------------------------------------------------------------
*               DATA FOR THE RUN
*------------------------------------------------------------------------
PARAM   LINITI = 63., LFINAL = 73.
        W = PI/120.
        LCMS = LINITI * 2.54

*------------------------------------------------------------------------
*               GEOMETRY OF THE PRESSURIZER
*------------------------------------------------------------------------
PARAM   A= 0.111
        VO = 200.*(2.54**3.)*(10.**(-6.))*12.*12.*12.
```

```
         DIAM = 50. * .0254
         RAD = DIAM /2.
         HEIGHT = 154. * 2.54
         AHEMS = PI * DIAM* DIAM/2.
         THW = 23.* .0254 / 16.
         B = PI* DIAM* DIAM/400.

*------------------------------------------------------------------------
*              THERMAL PROPERTIES OF THE WALL
*------------------------------------------------------------------------
PARAM    SPHEAT=0.111, AK= 26., DQDT0 = 0.
         DX = THW/12.
         S = ALPHA/(DX**2.)
         SPHE = SPHEAT* 1054.8
         RHO = 490. / ((12.*.0254)**3.)
         COND = AK  * 1054.8 / (3600.*12.*.0254)
         ALPHA =(AK/(RHO*SPHEAT))*.0254*.0254*144./3600.

DYNAMIC

*------------------------------------------------------------------------
*              LEVEL FUNCTION GENERATION
*------------------------------------------------------------------------

         ALI = LINITI -C2
         C2 = - ( LFINAL - LINITI) * SIN (W*T)
         L = ALI*2.54
         LDOT = ALDOT*2.54
         ALDOT= ( LFINAL - LINITI) *W*COS (W*T)

*------------------------------------------------------------------------
*              CALCULATION OF THE VAPOR SURFACE AREA
*------------------------------------------------------------------------

         AWALL = PI* DIAM* (HEIGHT-L)/100.
         AREA= AHEMS + AWALL
```

```
*------------------------------------------------------------------
*                SATURATION TEMP HISTORY OF WALL
*------------------------------------------------------------------
         TSAT = 976.416 +(P-55.) * 1.920852
         PHI0 = 976.416 + (PZERO- 55.) * 1.920852
         PHIM = TSAT - PHI0

*------------------------------------------------------------------
*                THERMODYNAMIC PROPERTIES
*------------------------------------------------------------------

         VFG = 2100. /PA - 3.97
         DVG = -2090./(PA*PA)
         HF  = 747.+ 9.85*PA-.0343*PA*PA
         HFG= 10.**6./(476.+ 2.68*PA)
         HG  = 2790.+.898*PA-.0162*PA*PA
         VF  = 1.12+ .0033*PA
         VG  = 2090./PA -2.46
         DHF  = (.241/PA-.000617)*HF
         DHG = (.0196/PA -.000235- .0000126* PA)*HFG

*------------------------------------------------------------------
*                CALCULATION OF THE DENSITY TERM
*------------------------------------------------------------------
         PA = P+ 1.0
         HFGH= (10.**(6.))/(476.+2.68*PA)
         DRDH = (2.32-.0106*PA+5.26*(10.**(-5.))*PA*PA)*VFGH/HFGH
         VFGH= (2100. -3.97*PA)*(10.**(-6.))

*------------------------------------------------------------------
*                CALCULATION OF THE HEAT TRANSFER TERM
*                SOLUTION OF THE PARTIAL DIFF EQUATION
*------------------------------------------------------------------

         U0 = PHIM
         UD1 = S*(U0-2.*U1+U2)
         UD2 = S*(U1-2. * U2 +U3)
         UD3 = S* (U2 -2.*U3+U4)
         UD4 = S* (U3- 2. *U4 +U5)
```

```
UD5 = S* (U4 - 2.*U5 + U6)
UD6 = S* ( U5 -2.*U6+U7)
UD7 = S* ( U6 - 2.*U7 + U8)
UD8 = S* (U7 - 2.* U8 + U9)
UD9 = S* ( U8 - 2.*U9 +U10)
UD10 =  S* (U9- 2.0*U10+ U11)
UD11 = S* (U10-2.0*U11+ U12)
UD12 = S* (U11-U12)*2..
U1 = INTGRL (0.0,UD1)
U2 = INTGRL (0.0, UD2)
U3 = INTGRL (0.0,UD3)
U4 = INTGRL (0.,UD4)
U5 = INTGRL (0., UD5)
U6 = INTGRL (0., UD6)
U7 = INTGRL (0.,UD7)
U8 = INTGRL (0.0, UD8)
U9 = INTGRL (0.0, UD9)
U10= INTGRL (0.0, UD10)
U11 = INTGRL (0.0, UD11)
U12 = INTGRL (0.0, UD12)
*-----------------------------------------------------------------------
*             INTEGRATION BY THE SIMPSON RULE
*-----------------------------------------------------------------------
SIGMA =DX*(U0+4.*(U1+U3+U5+U7+U9+U11)+2.*(U2+U4+U6+U8+U10) ...
+ U12) /3.

Q = AREA* RHO *SPHE* SIGMA
DQD = DERIV(DQDT0,Q)
DQDT = FCNSW(DQD,0.,0.,DQD)
M = K*ORDH*DQDT
VOLWAT = VD- A - B* L
PDIN = ( P* K* B* LDOT - M )/VOLWAT
PDOT = KEYIN*PDIN + PDOUT*KEYOUT
P = INTGRL (PZERO,PDOT)
D = P/.068948
KEYOUT = NOT(KEYIN)
```

```
*------------------------------------------------------------------------
*              EXPRESSION FOR THE PURE OUTSURGE
*------------------------------------------------------------------------

         DLDP = (A* F1 + VD* F2)/B + F1*L
         DPDL = 1. / DLDP
         PDOUT = DPDL * LDOT
         SOUT = ( VG/VF* DHF - DHG)/HFG
         F1 = SOUT + DVG/ VG
         MOUT = ( DHG- VG/J) / HFG
         F2 = MOUT- DVG/VG

*------------------------------------------------------------------------
*              RATE OF STEAM CONDENSATION
*------------------------------------------------------------------------

         DMCDT = DQDT/ (HFGH* 1000.)
         MC = INTGRL(0.,DMCDT)


         G = VOLWAT / (.3048**3.)
         MS = (VD-A-B*LCMS)/(.3048**3.*.55049) -MC/.453592
         SPVO = G/ MS

*------------------------------------------------------------------------
*              CALCULATION OF STEAM TEMPERATURE
*------------------------------------------------------------------------
NOSORT
         IF(KEYIN.EQ.0) GO TO 22
         TRAPX = D* SPVO/RAPX
         TRAPX = TRAPX + 150.
         FYT2 = TRAPX
 80      T2 = FYT2
         Z = T2
         Y3 = 0.
         N = 0
 45      N = N+ 1
         X3 = C(N)*(D**N)*(N+1) *(1./Z**(N+2))
         Y3 = X3 + Y3
         IF (N.LT.5.) GO TO 45
```

```
          R= RO+(1./Z)*(C(1)* D/Z + C(2)*(D/Z)**2.+ C(3)*(D/Z)**3. ...
          + C(4) *(D/Z)**4. + C(5)*(D/Z)**5. )
          FT2 = D* SPVO/R -T2
          FDT2 = (D*SPVO*Y3/(R*R)) -1.
          FYT2 = T2 - FT2/FDT2
          DIFF = FYT2 -T2
          IF (ABS(DIFF).GT.2.) GO TO 80
          IF (T.EQ.0.) GO TO 41
          GO TO 42
 41       T20 = FYT2
 42       CONTINUE
          T2 = FYT2
          TAMB = T2 + PHI0 -T20
          DEGREE = TAMB + TSAT

*---------------------------------------------------------------------
*                 CHECK FOR THE VAPOR SATURATION
*---------------------------------------------------------------------

          IF((T.GE.60.).AND.(DEGREE.LE.0.))GO TO 20
          GO TO 21
 20       KEYIN = 0.
 22       DEGREE = 0.
          TAMB = TSAT
 21       CONTINUE
SORT
          PRESS = D
TIMER     FINTIM = 120., DELT = .0001, PRDEL = 2., OUTDEL = 2.
PRTPLT    PRESS
LABEL     PRESSURE PREDICTION FOR NPD PRESSURIZER CASE TWO
          END
          STOP
```

PRESSURE PREDICTION FOR NPD PRESSURIZER CASE TWO

PAGE 1

PRESS VERSUS T

MINIMUM 8.2500E 02 I

MAXIMUM 9.1698E 02 I

T	PRESS
0.0	8.2500E 02
2.0000E 00	8.2979E 02
4.0000E 00	8.3457E 02
6.0000E 00	8.3933E 02
8.0000E 00	8.4407E 02
1.0000E 01	8.4878E 02
1.2000E 01	8.5343E 02
1.4000E 01	8.5802E 02
1.6000E 01	8.6254E 02
1.8000E 01	8.6697E 02
2.0000E 01	8.7130E 02
2.2000E 01	8.7552E 02
2.4000E 01	8.7960E 02
2.6000E 01	8.8355E 02
2.8000E 01	8.8733E 02
3.0000E 01	8.9095E 02
3.2000E 01	8.9437E 02
3.4000E 01	8.9760E 02
3.6000E 01	9.0062E 02
3.8000E 01	9.0342E 02
4.0000E 01	9.0598E 02
4.2000E 01	9.0829E 02
4.4000E 01	9.1035E 02
4.6000E 01	9.1214E 02
4.8000E 01	9.1366E 02
5.0000E 01	9.1491E 02
5.2000E 01	9.1586E 02
5.4000E 01	9.1653E 02
5.6000E 01	9.1690E 02
5.8000E 01	9.1698E 02
6.0000E 01	9.1676E 02
6.2000E 01	9.1625E 02
6.4000E 01	9.1545E 02
6.6000E 01	9.1436E 02
6.8000E 01	9.1298E 02
7.0000E 01	9.1133E 02
7.2000E 01	9.0940E 02
7.4000E 01	9.0722E 02
7.6000E 01	9.0478E 02
7.8000E 01	9.0210E 02
8.0000E 01	8.9920E 02
8.2000E 01	8.9607E 02
8.4000E 01	8.9274E 02
8.6000E 01	8.8921E 02
8.8000E 01	8.8548E 02
9.0000E 01	8.8156E 02
9.2000E 01	8.7746E 02
9.4000E 01	8.7320E 02
9.6000E 01	8.6880E 02
9.8000E 01	8.6427E 02
1.0000E 02	8.5962E 02
1.0200E 02	8.5488E 02
1.0400E 02	8.5005E 02
1.0600E 02	8.4517E 02
1.0800E 02	8.4392E 02
1.1000E 02	8.4272E 02
1.1200E 02	8.4151E 02
1.1400E 02	8.4028E 02
1.1600E 02	8.3904E 02
1.1800E 02	8.3779E 02
1.2000E 02	8.3654E 02