GENERAL ENGINEERING LABORATORY

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ELECTRONIC ANALOG COMPUTER STUDY OF A
NUCLEAR REACTOR STEAM PRESSURIZER

by

J. W. Bremer

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ELECTRONIC ANALOG COMPUTER STUDY

OF A

NUCLEAR REACTOR STEAM PRESSURIZER

(Report on G.E.L. Req'n. G591 "Steam Pressurizer Analysis"
For D.E. Davidson, SAR Mechanical Systems, KAPL)

BY

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January 20, 1956

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I. INTRODUCTION

If load is dropped from a nuclear reactor power plant suddenly, the pressure in the primary coolant loop will rise. This pressure rise is limited by a relief valve. Conversely if the load is picked up suddenly, the pressure in the loop will drop. This pressure drop is limited by boiling in the reactor. It is desired to minimize these pressure fluctuations due to power maneuvers by using a pressurizer. This pressurizer consists of a large steel vessel, partially filled with water, tied into the primary coolant loop, as shown in Figure 1. The rest of the vessel volume is filled with steam at the saturation temperature corresponding to steady state primary loop pressure. Also, on pressure rises, part of the incoming coolant will be sprayed into the steam in the pressurizer vessel. By using this scheme it is expected the pressure rises will be kept below the relief valve setting, and pressure drops will result in a pressure greater than the reactor boiling pressure.

FIGURE 1 - PRIMARY COOLANT LOOP

Tests of a small size pressurizer have been made by Combustion Engineering, Inc. of Chattanooga, Tennessee for the Westinghouse Electric Corporation on Contract 995-NP-Z. Determination of the performance of a

full sized pressurizer vessel as proposed by KAPL was done by an analog computer study. This study was made in two parts. The first part consisted of setting up on the computer the situation representing the test vessel. Several runs were made on the computer and the results compared with the Combustion Engineering data. Thus, the correctness of the analog computer simulation as compared with the experimental test vessel was determined to be within engineering accuracy. A second section of the study was the setting up of the full size pressurizer vessel on the analog computer. Runs were made using data supplied by KAPL for various power maneuvers. Performance of the pressurizer is to be measured by the pressure rise for a maximum insurge of primary coolant for the most violent power maneuvers.

Using the assumptions of homogeneous steam, no heat transfer to the vessel water, infinite film coefficient at the wall, and infinite insulation of the vessel, it was found that if 30% of the surge volume was used for spray, pressure rises were less than 200 psi. This report does not include the effects of power maneuvers resulting in outsurges.

II. COMPUTER REPRESENTATION

The pressurizer test vessel is constructed as shown in Figure 2. The two heads are 2 to 1 ellipses with 12 1/4 inch height. The height of this vessel is 4 feet 7 inches. The walls are made of carbon steel and are 2 1/2 inches thick. The internal volume of the vessel is approximately 15 cubic feet and the inside wall area is approximately 40 square feet. Referring to Figure 2, the spray water is sprayed into the top of the steam volume. Insurge or outsurge water enters the vessel from the bottom and is prevented from spraying to the top of the water surface by the baffle. The vessel is normally about 1/2 full of water. Theoretical

analyses performed at the General Engineering Laboratory indicate that pressurizer performance can adequately be found by considering the thermodynamic equations for the steam volume itself, assuming homogeneous steam. Thus, the equations for steam enthalpy, volume, and weight can be found and pressurizer vessel conditions determined on the analog computer as indicated in Figure 3, the pressurizer block diagram.

FIGURE 2 - PRESSURIZER VESSEL

The block diagram is a convenient way of illustrating the steam equations for ease in setting up the computer. The various symbols on Figure 3 represent multipliers, summers, dividers, and function generators as indicated by the symbols in the boxes. The signals in the system are then fed into these boxes which perform the necessary operations, in accordance with the steam equations. It is to be understood that these signals may be functions of time, as they represent the actual pressurizer variables. Thus, if the computer is properly set up, an analog representation of insurge and spray may be put into the computer as functions of time, and various pressurizer variables may then be measured as functions of time. If the computer is properly set up, these variables will correspond exactly with those of the actual pressurizer, for the same spray and surge conditions.

4

PRESSURIZER
BLOCK DIAGRAM

FIGURE 3

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Referring to the summer in the upper left hand corner of Figure 3, we see that the total enthalpy of the steam is found by summing the initial enthalpy of the steam and a term representing the work of compression -- the system pressure times the surge volume divided by a constant, and by subtracting the heats absorbed by the wall and by the spray water. The constant in the work of compression term is $144/778$ or $.185$. This is to convert the $P \times V_s$ units into btu's. Thus, we obtain:

$$H = H_o + .185 P V_s - H_c - H_{sp} \quad (1)$$

It should be noted that heat transfer to the water in the bottom of the pressurizer vessel is neglected. Thermodynamic analyses bear out this assumption.

In the next summer, the total volume of the steam in the pressurizer is found by subtracting two terms from the initial volume. The first of these is the surge volume (which includes the spray volume); the second term subtracted from the initial volume is the difference in volume between the steam condensed by the absorbed heats and the volume of the condensate. It is the heats absorbed by the wall and by the spray multiplied by the specific volume of the liquid and divided by the specific enthalpy of evaporation of the steam. The specific volume of the condensed liquid is fairly constant in the region of interest, and is taken as $.025$ cubic feet per pound. Thus:

$$V = V_o - V_s - (H_c + H_{sp}) \frac{.025}{h_{fg}} \quad (2)$$

A term representing the weight of the steam condensed by the vessel wall and the spray is subtracted from the initial steam weight to obtain the total steam weight. This term is equal to the heat absorbed by the wall and the heat absorbed by the spray water divided by the specific

enthalpy of evaporation. So, we obtain:

$$W = W_0 - (H_c / H_{sp}) \frac{1}{h_{fg}} \quad (3)$$

Thus, we see that the effect of the insurge is to decrease the steam volume, and to increase the steam enthalpy due to the work of compression. The effect of the spray is to decrease the steam enthalpy, and to condense some steam. This condensation decreases the steam volume and decreases the steam weight. The effect of the wall is similar to that of the spray. The initial steam weight for the test vessel is 40 lbs.; the corresponding figure for the full sized vessel is 276 lbs.

The total enthalpy and total volumes found previously, are now divided by total steam weights to obtain the specific enthalpy and specific volume, respectively.

$$h = \frac{H}{W} \quad (4)$$

$$V = \frac{V}{W} \quad (5)$$

These two signals are then fed into a function generator which represents the steam tables. This generator computes continuously the vessel pressure, temperature, and specific enthalpy of evaporation corresponding to the specific enthalpy and specific volume. This function generator is derived from a linearization of the steam table in the region of 2200 psi saturated; it works in either the superheat or two-phase state in this region. Thus, pressure is some function (1) of specific enthalpy and specific volume; and temperature is some function (2) of the same variables, or

$$P = F_1 (h, V) \quad (6)$$

$$T = F_2 (h, V) \quad (7)$$

Figures 4 and 5 are plots of pressure and temperature, respectively, versus specific enthalpy and specific volume. The lines in Figures 4 and 5 are taken from the computer representation, while the points shown are taken from the steam tables, and are used to check computer representation. Note that the breaks in the slope of the straight lines in Figures 4 and 5 occur when changing from the two-phase to the superheat conditions. The specific enthalpy of evaporation is of significance in only the two-phase region, therefore, it is a function of pressure only:

$$h_{fg} = 855 - 0.2 P \quad (8)$$

The heat absorbed in the spray water is determined under the assumption that the spray water reaches the temperature of the steam in the pressurizer vessel. This assumption is born out by tests performed by Combustion Engineering, Incorporated, that show that the spray water reaches the steam temperature in less than one foot of fall. Thus, this heat is equal to the specific heat of the spray water times its weight times its temperature rise; or, the specific heat of the spray water times the portion of the surge volume that is used for spray times the surge volume times the difference in temperature between the system temperature and the spray temperature all divided by the specific volume of the spray water. The specific heat divided by the spray water is very nearly a constant in the region of interest; this constant is equal to 61 btu's per degree F per cubic foot. Thus, we get:

$$H_{sp} = 61 \times V_s (T - T_{sp}) \quad (9)$$

Referring again to Figure 3, the pressurizer block diagram, the only part not yet determined is the configuration labeled "vessel wall." As seen, the heat absorbed by the vessel wall is a function only of the temperature of the steam, or:

$$H_c = F_3(T) \quad (10)$$

Function (3) is, of course, also a function of time. During insurges, there will always be steam condensing at the surface of the wall. This condensation gives a very high film coefficient of heat transfer at the surface between the steam and the wall. For this representation, an infinite film coefficient will be assumed. The present study does not include outsurges, therefore, the film coefficient during outsurge is not under consideration. The pressurizer design includes sufficient insulation at the outside of the vessel; we therefore assume perfect insulation. A conventional, ten section, nodal lumped-parameter model of the wall was used for the computer representation.

The carbon steel used for the test vessel has a conductivity of 20 btu's per degree F per hour per foot, a specific heat of .12 btu's per pound per degree F, and a density of 518 lbs. per cubic ft. These constants give a time constant of 4.86 seconds per section for the test vessel. Figure 6 shows a Hottle chart from McAdams "Heat Transfer," 3rd edition, page 38, Figure 3-5, for a slab with infinite insulation and infinite film coefficient with an applied step of temperature. The dotted line on Figure 6 shows absorbed heat versus time, plotted on the same scale, for the computer model, also for a step of temperature. Inspection of Figure 6 shows that the model is of sufficient accuracy at the low times for which runs will be made. This computer model will now give the absorbed heat for any time function of temperature at the steam wall surface.

Different constants are used for the full size pressurizer vessels design. This vessel is approximately 120 inches high and 37 inches in inside diameter. Since approximately 60 per cent of this vessel will be filled with steam in the steady state, this steam sees a wall area of

9,600 square inches and a volume of 44.8 cubic feet. The type 304 stainless steel walls are 4 inches thick. The steel conductivity is 16.6 btu's per hour per foot per degree F, the specific heat is .112 btu's per lb. per degree F and the density is .3 lbs. per cubic inch. For a ten section model, the time constant per section is 17.6 seconds. Figure 7 gives the Hottle chart comparison for the full size vessel.

We now have all the necessary equations to set the problem up on the analog computer. The Goodyear Aircraft Corporation L-3 analog computer with various Philbrick nonlinear components performs the operations indicated in the block diagram, Figure 3. However, dividers are not available, so a multiplier with an amplifier feedback around it must be used. Otherwise, the computer diagram is merely a more detailed block diagram. When the computer is set up, time-varying voltages corresponding to surge volumes and spray temperatures may be put into the computer. The signals, pressure and temperature, may then be monitored with a Sanborn four channel recorder and these will correspond with the pressure and temperature in the pressurizer vessel for a corresponding surge condition.

III. TEST DATA RUNS

Ten runs from Combustion Engineering, Incorporated "Report on Contract 955-NP-Z," representing a wide range of surge and spray conditions, were put into the computer to determine the validity of the representation. These runs are tabulated on Figure 8, where V_s/V_i is the surge ratio, or surge volume divided by the initial steam volume; the initial steam state is 2000 psi saturated. The peak pressure and peak temperature for the experimental data and the computer runs are also tabulated. Note that the surge volume does not include the spray

Z - SECS.

880

1760

2640

0

FIGURE 7

CHECK OF
PRESSURIZER WALL
SIMULATION
FOR FULL-SIZE VESSEL

12300

 H_c
Btu

24600

volume; the spray in the Combustion Engineering report was given in gallons per minute, and not as a percentage of the surge volume. The computer diagram was corrected to include this condition. In the absence of other data, the insurge was assumed to be a linear function of time. Figures 9 and 10 are bar graphs comparing the data for peak pressure and peak temperature respectively. The solid lines represent the experimental data, and dashed lines the results of the computer runs. The correlation between the experimental data and the results from the computer runs is quite good, considering the fact that the methods used to get the experimental data were quite inaccurate. For example, the Combustion Engineering report indicates that measurements of surge volume are accurate to no better than 30%. Also, no method for determining the initial quality of the pressurizer steam was included in the experimental setup. Perfect correlation between the computer representation and the experimental data could be obtained for any given run either by varying the insurge volume by 30%, or by including initial moisture in the steam. In most cases, errors between the computed runs and the experimental data are conservative, that is, peak pressures and temperatures are higher than experimental data indicates. It is thus felt that the computer representation is valid for the pressurizer vessel, within engineering accuracy.

IV. COMPUTER RUNS ON THE FULL-SIZE VESSEL

The computer setup was changed to fit the new conditions for the full sized vessel, as shown in Figure 11, the computer diagram. Runs were received from KAPL representing various power maneuvers to which the pressurizer would be subjected. These runs consisted of two curves of temperature versus time. The first of these was spray temperature. This was fed directly into the computer. The second was the bulk average

PEAK TEMP. OF-Comp.	PEAK TEMP. OF--Exp.	PEAK PRESSURE psi-Comp.	PEAK PRESSURE psi-Exp.	SPRAY TEMP. OF	SPRAY VOL. gpm	SURGE RATE cfm	TIME OF SURGE-SECS.	V/V ₁	FROM PAGE NO. IN REPORT	RUN NO.
702	666	2680	2450	--	0	6.78	20	.30	D-14	1
676	660	2420	2380	--	0	3.30	27.5	.20		2
671	640	2370	2125	--	0	7.86	8.0	.14		3
6415	652	2125	2225	560	5.0	2.76	37.5	.23	D-13	4
649	652	2170	2175	569	5.0	5.10	11.5	.13		5
659	650	2310	2275	505	3.0	8.59	10.5	.20	D-12	6
657	648	2260	2300	500	3.0	9.00	8.0	.16		7
632	640	1940	2100	457	2.0	1.08	45	.11	D-11	8
686	660	2530	2350	460	2.0	9.00	15	.30		9
660	655	2310	2245	545	0.5	6.30	10	.14	D-8	10

FIGURE 8
RUNS FOR TEST VESSEL

16

FIGURE 9RESULTS FOR TEST VESSEL RUNS - PRESSURE

— EXPERIMENTAL DATA

--- COMPUTED DATA

12/30/55

254 017

17

FIGURE 10RESULTS FOR TEST VESSEL RUNS - TEMPERATURE

$$\begin{aligned}
 h' &> 720 \text{ V'} && \text{FOR SUPERHEAT, } < \text{FOR TWO-PHASE} \\
 P' &= 7.8 h' - 15040 \text{ V'} && \text{SUPERHEAT} \\
 T' &= .97 h' - 1250 \text{ V'} && \\
 P' &= 2.6 h' - 11440 \text{ V'} && \text{TWO-PHASE} \\
 T' &= .06 P' && \\
 h_{fg} &= 459 - .198 P' && \\
 V_L &= .025 &&
 \end{aligned}$$

$$h' = \frac{1}{W} [H_o + .185 P h_s - 61 X V_s (T' - T_{sp}') - H_c]$$

$$V' = \frac{1}{W} [V_o - V_s - W_c V_L]$$

$$W = W_o - W_c = W_o - \frac{H_o + H_{sp}}{h_{fg}}$$

$$R = \frac{L}{KA} = \frac{.4}{16.6 \times 9660} = .11 \frac{\text{OF SEC.}}{\text{BTU}} \quad C = C_p W_s \Delta T = .142 \times .3 \times 4 \times 9660 = 164 \frac{\text{BTU}}{\text{OF}}$$

CONDITIONS - BASE

$$\begin{aligned}
 P_B &= 2200 \text{ psi} \\
 h_B &= 1119 \text{ BTU/lb} \\
 V_B &= .01625 \text{ ft}^3/\text{lb} \\
 T_B &= 650 \text{ }^\circ\text{F}
 \end{aligned}$$

STAINLESS STEEL VESSEL 74.7 ft³, 9660 in², 4" th. USED
44.8 ft³ 276" STEAM (INITIALLY)

FIGURE 11

254 019

COMPUTER DIAGRAM
FULL SIZE PRESSURIZER

SAR-SPA

jack bremer

1/13/56

SCALE FACTORS

$$\begin{aligned}
 1 \text{ V} &= 1 \text{ BTU/lb} & 1 \text{ V} &= 1000 \text{ BTU} \\
 &= .001 \text{ ft}^3/\text{lb} & &= .2 \text{ ft}^3 \\
 &= 1^\circ\text{F} & &= 1^\circ \\
 &= 10 \text{ psi} & &
 \end{aligned}$$

temperature of the primary coolant loop (See Figure 1); this must be multiplied by .44 cubic ft. per degree F to obtain the insurge volume. This constant is the average temperature coefficient for the primary coolant loop water under these conditions. This last function, $V_s(t)$, was obtained on the computer by feeding a ramp voltage function of time into a function generator which was set to give the desired function of time. Forty-two runs were made, representing various surge, spray, and spray temperature conditions. Computer records showing surge volume, spray temperature, pressure, and temperature as a function of time are given in Appendix 1. It should be noted that on a few of the runs, the computer saturated and is indicated as such on the runs. Pressure and temperature values after the machine saturates should not be used. This saturation is due to excessive voltage in the multipliers under certain conditions, and does not affect the rest of the solutions. The variables and scale factor for all the runs are labeled on run #1; all runs use 2200 psi saturated as the starting condition. Pages 1 and 2 of Appendix 1 give the surge, spray, and spray temperature conditions for the runs. Peak pressure obtained is also tabulated there, for convenience. Because the computer was not set up to include outsurges, all data that occurs after the peak insurge volume is reached cannot be correct. This is not important because only peak pressure and temperature is of interest.

Inspection of the peak pressures as tabulated on pages 1 and 2 of Appendix 1 indicates that spraying 30% of the surge volume will result in a pressure rise of less than 200 psi. Analyses performed too late to be included in the computer setup as described, indicate minor changes in the computer representation that would modify the terms for second order effects. However, it is felt that these changes would affect the

results shown in Appendix 1 only slightly; and that the conclusions based on these results would not be affected; namely, that a 30% spray keeps the pressure rises within the specified 200 psi range. It should be understood that the above conclusion has been proved valid only for the surge conditions corresponding to the maximum power maneuvers supplied by KAPL. A large surge in a short time, with a high spray temperature, may produce pressure rises greater than 200 psi even with 30% spray. However, run B-9 is considered at this time to yield the maximum insurge during a power maneuver. Thus, the full size pressurizer vessel design as proposed by KAPL appears to be adequate with regard to insurges.

NOMENCLATURE

H	-	Total Enthalpy of Steam	-	btu
H _o	-	Initial Enthalpy of Steam	-	btu
P	-	System Pressure	-	psi
V _s	-	Total Surge Volume Including Spray	-	ft ³
H _{sp}	-	Heat Absorbed in the Vessel Wall	-	btu
H _c	-	Heat Absorbed in the Spray Water	-	btu
V _o	-	Initial Steam Volume	-	ft ³
V	-	Total Steam Volume	-	ft ³
h _{fg}	-	Specific Enthalpy of Evaporation	-	btu/#
W	-	Total Steam Weight	-	#
W _o	-	Initial Steam Weight	-	#
h	-	Specific Enthalpy	-	btu/#
v	-	Specific Volume	-	ft ³ /#
T	-	System Temperature	-	°F
X	-	Portion of Surge Volume Sprayed	-	_____
T _{sp}	-	Incoming Spray Temperature	-	°F

APPENDIX I
COMPUTER RECORDS

RUN DATA

<u>RUN</u>	<u>KAPL RUN FOR V_s</u>	<u>PER CENT SPRAY</u>	<u>SPRAY TEMP.</u>	<u>PEAK PRESSURE</u>
1	B1-D	0	552	2500
2	B1-D	10	552	2420
3	B1-D	30	542	2320
4	B1-D	30	552	2330
5	B1-D	30	562	2340
6	B1-D	50	552	2250
7	B3-D	0	562	2540
8	B3-D	10	552	2460
9	B3-D	30	542	2340
10	B3-D	30	552	2350
11	B3-D	30	562	2360
12	B3-D	50	552	2240
13	B4-D	0	552	2530
14	B4-D	10	552	2450
15	B4-D	30	542	2330
16	B4-D	30	552	2340
17	B4-D	30	562	2350
18	B4-D	50	552	2240
19	B-6	0	552	2570
20	B-6	10	552	2460
21	B-6	30	542	2340

28

RUN DATA

<u>RUN</u>	<u>KAPL RUN FOR V_S</u>	<u>PER CENT SPRAY</u>	<u>SPRAY TEMP.</u>	<u>PEAK PRESSURE</u>
22	B-6	30	552	2360
23	B-6	30	562	2370
24	B-6	50	552	2270
25	B-9	0	552	2620
26	B-9	10	552	2530
27	B-9	30	542	2380
28	B-9	30	552	2400
29	B-9	30	562	2420
30	B-9	50	552	2290
31	B14-D	0	552	2560
32	B14-D	10	552	2470
33	B14-D	30	542	2350
34	B14-D	30	552	2360
35	B14-D	30	562	2370
36	B14-D	50	552	2260
37	B17-D	0	552	2540
38	B17-D	10	552	2460
39	B17-D	30	542	2320
40	B17-D	30	552	2340
41	B17-D	30	562	2360
42	B17-D	50	552	2240

254-025

$V_s - FT^3$

8
6
4
2
0

650
600
550
500

$T_{SP} - ^\circ F$

BI-D
No SPRAY

1

$P - PSI$

2600
2400
2200

$T - ^\circ F$

690
670
650

TIME - SECS.

0 20 40 60 80 100 120 140 160 180

30

B1-D
10%
550°F

2

BAD
30%
542°F

3

R1-10
20/10
5577

4

37

BI-D
309°
562°F

5

34

B1-D
50%
552 of

6

35

BB-D 55211
NO SPRAY

7

B3-D 552°F
107%

8

37

BB-D +10°K
3070

9

B-D 5529F
3090

10

39

SANBORN Recording Permapaper

B3-D 30%

L-D -10°F

70

11

254 036

B3-D 552°F
50%

(12)

B4-D
NO SPRAY

55295

13

42

B4-D
1090 5520

14

B4-D
30%
542°F

15

44

B4-D
30% 552°F

16

45

B4-D
302 562°F

17

46

SANBORN Recording Permapaper

B4-D
50% 552°F

18

47

SANBORN recording Permapaper

B6
NO SPRAY

19

48

B6
107° 552° F

(20)

49

B6
30% 542°F

(21)

136
30.7% 562°F

(23)

52

1

1

B6
50% 552°F

(24)

54

SANBORN Recording

B-9 +100%
30%

(27)

56

B-9 56095
3090

28

B-9 -10°K
30%

29

B-9 560°K
50%

30

59

B14D
NO SPRAY

31

60

B14D
10% 552°F

32

61

B140
30% 542 °F

33

62

B14D
30% 552°F

34

63

B14D
30% 562°F

35

64

B14D
50% 552°F

36

65

B 17-D
NO SPRAY

(37)

66

BIID
10% 552°F

38

67

SANBORN

empty paper

1

1

1

B17D
307, 542°F

39

254 064

68

B17D
30% 5520

40

69

B17D
307.562°F.

41

70

B17D
50% 552°F

42

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1-70

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