BEFORE THE UNITED STATES ATOMIC ENERGY COMMISSION

Application of PACIFIC GAS AND ELECTRIC COMPANY for a Class 104b. License to Construct and Operate a Nuclear Reactor as a Part of Unit No. 3 of Its Humboldt Bay Power Plant	Docket No. 50-133 Amendment No. 6

Now comes PACIFIC GAS AND ELECTRIC COMPANY (the Company) and amends its above-numbered application by submitting herewith Amendment No. 6, which consists of Addendum E to the Preliminary Hazards Summary Report (Exhibit B to said application). The information contained in Addendum E supplements and amends the Preliminary Hazards Summary Report and Amendments 2, 3, 4 and 5 to the above-numbered application. Specifically, Addendum E sets forth and answers the questions contained in the Commission's letter dated December 21, 1959.

In the event of a conflict the information in this amendment supersedes the information previously submitted.

Subscribed in San Francisco, California, this 3rd day of February, 1960.

Respectfully submitted,

PACIFIC GAS AND ELECTRIC COMPANY

By N. R. Sutherland
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By Philip A. Crane Jr.

Subscribed and sworn to before me this 3rd day of February, 1960

Rita J. Green (SEAL)
Rita J. Green, Notary Public in and for the City and County of San Francisco, State of California

My Commission expires July 16, 1963.

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APPENDICES

QUESTION 1.a.

What basic hydrodynamic principles were utilized in determining the behavior, size, and location of the steam nozzles in the suppression pool?

ANSWER

I. INTRODUCTION

The size and location of the steam nozzles were chosen to give an arrangement whose behavior could be confidently predicted on the basis of the tests performed on the large scale Condensing Test Facility at Moss Landing and on the Transient Test Facility at San Jose. The tests and facilities have been described in previous amendments. The test results are readily explained by established principles of hydrodynamics and heat transfer. As will be subsequently shown, the chosen design is consistent with a most conservative interpretation of the test results.

II. CONDENSATION MECHANISMS

A. Types

Observations at Moss Landing of single and multiple jets of different sizes show that the condensation mechanism of steam discharged into a pool depends on the flow rate. Different condensation patterns were observed for low, intermediate, and high steam flow rates.

At low steam flow rates, steam flow is intermittent. Steam condenses in the pipe until a layer of hot water forms next to the steam. Steam pressure then forces out the water in the pipe and the process of condensation can begin again.

At intermediate steam flow rates, the steam condenses outside the pipe at the surface of the jet which is approximately conical in shape and the jet condenses before it breaks up.

At high steam flow rates, steam issues from the pipe in a jet which breaks up into bubbles. Some condensation takes place at the surface of the jet, but most of the condensation occurs at the surface of the steam bubbles.

Observations of steam being injected into a vessel of water at Moss Landing, with a jet of steam from either 4", 6", or 8" pipes and where the flow rate was high enough so that condensation took place outside the pipe, showed that the jet breaks up into bubbles within one to two feet below the pipe outlet.

III. CONDENSATION THEORY FOR HIGH STEAM FLOW RATES

A. Length of Jet Before Break-up

The behavior of one fluid injected into another fluid has been investigated in classical physics studies. Lord Rayleigh[1] discussed this phenomena and

[1] Rayleigh, "Theory of Sound", Section 360

reported that instability will cause break-up of the jet. The length of the jet before it breaks up varies directly with the diameter of the orifice and directly with the jet velocity.

B. Bubble Size

A theory by G. I. Taylor of Great Britain, extended by others[1,2,3], states that when two fluids of different densities have a common interface which is accelerated in a direction perpendicular to the boundary, any small irregularity of the interface will tend to change in shape. The interface is unstable, i.e., the irregularities of the interface will grow with time, when the acceleration is directed from the lighter to the heavier medium.

The theory of Taylor instability establishes an upper limit for the bubble sizes formed from the steam jet, although it probably does not completely describe the process of bubble formation. Calculations in Appendix I, based on Taylor instability, give a maximum steam bubble diameter of approximately one half inch. Bubble sizes noted in the Moss Landing test were generally smaller than this. The size of the bubbles formed is a function of fluid properties and not of the pipe diameter.

C. Condensation of Steam Bubbles

With a known bubble size, heat transfer theory can be applied to determine how long it will take to condense the steam. In Appendix I the time of condensation is calculated to be less than .01 seconds.

IV. APPLICATION OF THEORY AND TEST RESULTS TO HUMBOLDT DESIGN

The theory described in the preceding paragraphs has been used with measured test values to determine performance of the Humboldt design. The calculations of jet length, bubble size, bubble condensation, and travel distance are presented in Appendix I. The results show that for the maximum steam flow from the Humboldt vents, the steam would be completely condensed in about five feet of travel. This distance is short compared with the available pool water circulation path, which is in the range of 15 to 30 feet.

V. EFFECT OF AIR ON STEAM CONDENSATION

At the start of the maximum credible operating accident, the air in the vent pipes and some of the air in the dry well would be forced into the pool ahead of any steam. This air would break up into small bubbles, just as the steam jet breaks up according to Taylor instability. Because the air would break up into small bubbles and leave the pool, it could not provide a channel through the water for the escape of steam, nor would it prevent the steam from

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[1] Bellman and Pennington, "Effects of Surface Tension and Viscosity on Taylor Instability", Quarterly of Applied Mathematics, 12, 1954, page 151-162.

[2] Allred and Blount, "Experimental Studies of Taylor Instability", U.S. Atomic Energy Commission Report LA-1600, November 1953.

[3] Birkhoff, "Taylor Instability and Laminar Mixing", Los Alamos Scientific Laboratory of the University of California Report LA-1862, [illegible]

condensing. Experimental evidence obtained at the Moss Landing tests[1] showed that air would rapidly escape from the pool. No steam was observed to escape with this air. The Transient Test Facility data on suppression chamber pressure revealed no evidence of uncondensed steam.

VI. JET SUBMERGENCE

During jet operation the momentum of the steam would depress the water level around the jet nozzle. The Humboldt vents have been designed to maintain submergence of the end of the vents during the maximum credible operating accident. The initial submergence is based upon Moss Landing tests. This subject has been described in pages 9-12, Section I of Addendum C.

VII. VENT PIPE DIAMETER

The selection of the Humboldt vent diameter is based on the Moss Landing tests and on the theory described in the preceding sections. The theory shows that the vent diameter directly affects the length of the jet before breaking up, but that fluid properties, not diameter, determine the bubble size formed by the jet break-up. Therefore larger vent diameters than the 14-inch selected for Humboldt could have been justified, but it was desired not to extrapolate too far from the Moss Landing tests.

VIII. MULTIPLE JET CONDENSATION

The foregoing paragraphs discuss the consistency of theoretical and actual performance for a single jet of steam in water. Since multiple jets are being used in the Humboldt Bay design, an examination of their behavior follows.

In the tests performed on the large scale Condensing Test Facility at Moss Landing, three four-inch jets were tested, spaced about one diameter apart. In the Transient Test Facility the 1" and 1-1/2" jets numbered up to 150 with about one diameter separation. In all cases, condensation was rapid and complete. This is consistent with the theories already explained. The jets break up into bubbles whether there are one or many jets. One diameter separation has appeared to be adequate to prevent the jets from merging; but even should they merge, it would only be a somewhat greater distance before break up occurred (at the tank bottom, if necessary). Once the jet breaks up, bubble size is independent of bubble source and condensation would continue as described.

IX. ADEQUACY OF WATER FOR MULTIPLE JET CONDENSATION

In order to insure condensation of steam, water must be continuously supplied to each of the jets, and the volume of pool water must be sufficient to condense all the steam.

Moss Landing tests proved the ability of a discharging vent to draw sufficient water into the jet to provide rapid and complete condensation. The Moss Landing

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[1] Addendum C, Section I, page 7.

compartment tests provide the data for selecting pool dimensions to obtain optimum mixing in the pool.

At Humboldt, the vent pipes discharge downward at the outer edge of the pool. The circulating path the pool water must take is down around the jets, to the bottom of the pool, across the pool, up the inner wall, and across to the vent pipes. There would be little cross flow of the pool water between different vent pipes because the vents are essentially equally spaced and discharge equal steam flow. Therefore, the behavior of each jet and its sector of the pool would tend to be independent of the others. When looked at in the above light, each Humboldt jet and its associated pool sector is similar to Moss Landing compartment tests.

As described in Addendum A, the maximum credible operating accident involves about 40 million Btu energy release which would raise the 1,200,000 pounds of water in the suppression pool from 80°F initially to about 115°F. Moss Landing compartment test results indicate that the pool temperature can rise to 180°F and still achieve complete condensation. From this it may be seen that the maximum credible accident uses only 1/3 of the condensing capacity of the pool water.

X. CONCLUSION

Test results obtained in the Pressure Suppression Development Program provide the principal design information for Humboldt Bay containment. These results are consistent with well accepted hydrodynamic and heat transfer principles. Specifically, consistency has been shown for the following conclusions:

1. Condensation of a steam jet in water under Humboldt Bay design conditions is rapid, complete, and takes place in a relatively short distance.

2. Jet nozzle size is unimportant as far as condensation is concerned.

3. Jet pipes will remain submerged.

4. Adequate water is available for multiple jet condensation.

5. The condensing capacity of the water pool is about three times that required for the maximum credible accident.

6. The arrangement of the Humboldt system is similar to arrangements tested successfully.

QUESTION 1.b.

What methods were used to arrive at and to justify the design pressure of the principal containment components; the dry well, the suppression pool, the vent pipes and nozzles, and the refueling building?

ANSWER

I. DRY WELL PRESSURE DESIGN

The maximum dry well pressure can be computed by conventional thermodynamic considerations and steady state fluid flow calculations and is the basis for establishing the dry well vessel design pressure.

As long as the flow from the break into the dry well exceeds the flow out of the dry well through the vents the dry well pressure will increase. The pressure will reach a value calculated to be approximately 35 psig at the moment water is expelled from the submerged vent pipes (Addendum D, Section II, pg. 1). However, the pressure will continue to increase until the steam-water flow out of the vents equals the break flow into the dry well at which time the maximum dry well pressure is reached. The flow rate through the break is determined as a function of the break area and the pressure differential between the reactor and dry well. On the conservative basis of 100% carryover of water with the steam, the flow through the vent pipes is a flashing mixture of steam and water, and the critical-end-pressure at the vent pipe discharge is calculated for several flow rates. For each of these flow rates, and corresponding end of line pressures, the pressure drop through the vent piping is calculated. The sum of the critical-end-pressure and line pressure drop for each flow rate is the inlet pressure to the vent pipe, and thus the dry well pressure required to produce the corresponding flow rate. Both the "break" flow and the vent pipe flow are plotted as a function of dry well pressure, and the intersection of the two curves determines the maximum dry well pressure of 72 psig. A detailed explanation of this calculation is given in Appendix II. The conservatism of this calculation is described on page 5, Section II of Addendum C.

The design pressure for the guard pipes is computed in the same manner as for the dry well proper. The critical-end-pressure is calculated at the guard-pipe opening to the dry well, assuming a flashing water-vapor mixture flows inside the guard pipe after discharge from the primary system pipe break. To the critical-end-pressure is added the pressure drop in the annulus between the guard pipe and the primary system pipe to arrive at the maximum possible pressure in the guard pipe.

In setting the design pressure, no credit was taken for heat transfer to the dry well wall.

II. SUPPRESSION CHAMBER DESIGN PRESSURE

The maximum suppression chamber internal pressure would be resisted by the concrete structure surrounding the suppression chamber. A welded steel liner in the suppression chamber is provided to ensure against leaks. Due to other design considerations, the resultant concrete structure, based on ACI standards, is adequate for an internal pressure of 24.8 psig with normal working stresses. It is also adequate for a maximum of 33 psig based on allowable stresses for short time loading.

The suppression chamber maximum pressure is determined primarily by the ratio of the initial volume taken up by the air in the dry well and suppression chamber to the final air volume after all air has been transferred to and compressed in the suppression chamber. The maximum pressure was calculated to be 8 psig using the system design volumes that were contemplated when Addendum C was submitted. Detail structural design has resulted in some volume changes which results in a calculated pressure of 9.5 psig. Subject to review in the light of further detail study, the tentative design pressure is 10 psig. This pressure is well within the strength of the concrete structure, assuming normal working stresses.

In setting the design pressure for the suppression chamber, no credit was taken for heat transfer to the dry well and suppression chamber walls. An outline of the method of calculations used is given in Appendix III.

III. DESIGN OF VENT PIPES

The nominal design pressure of the vent pipes is the same as the dry well design pressure (72 psig). However, the actual vent pipe wall thickness is selected for reasons of mechanical strength and rigidity and would permit much higher pressures without exceeding design stresses. The pipes are supported and braced to withstand all static and dynamic forces due to weight, vent pipe flow and reaction forces.

IV. REFUELING BUILDING DESIGN

The structural elements of the refueling building have been designed to resist loadings due to earthquake, wind and live load. Due to these design considerations, the structure will withstand, at stresses as permitted by the building code, an external pressure of 7.2 inches of water.

The refueling building is designed to retain radioactive materials which may be released due to fuel handling or fuel loading accidents, or any materials released into it from dry well and suppression chamber design leakage following the maximum credible operating accident. To accomplish this, the normal ventilation equipment will be running and the refueling building gas treatment equipment will be turned on prior to refueling operations, as described in Addendum C, Section IV, pg. 1. In the event of abnormal emissions of radioactive material, suitable radiation monitoring devices will sound audible alarms and trip closed the leaktight valves in the normal ventilation air inlet and discharge lines. The negative pressure thus developed by the gas treatment equipment will assure that the building air contents will pass through the gas scrubbing system and out the stack. The scrubber is designed to remove 95% of the halogens and particulate matter.

The negative pressure to be maintained by the gas treatment equipment has been selected as 1/4 inch of water and with the design leakage rate is sufficient to maintain a negative pressure under exposure to wind velocities up to 20 mph and barometric and temperature changes. Wind velocities above 20 mph could cause exfiltration due to existenc. of areas of reduced pressure on the lee side of the building. However, the existence of high winds would make such leakage not critical as inversion conditions would no longer exist.

QUESTION 1.c.

Will steam which has become superheated by expanding from 1,000 psi to 72 psi condense just as well as the 100 psi saturated steam used in the Moss Landing tests? What are the experimental and theoretical considerations involved in reaching this conclusion?

ANSWER

Experimental and theoretical considerations described in the following paragraphs indicate that steam expanding from 1000 psi will condense as well as the steam used for the Moss Landing tests. It will be shown that superheat would have a negligible effect on the heat transfer rate, and on the amount of heat to be transferred. Further, the thermodynamic state of the steam in the condensing jets tested at Moss Landing is close to the state of steam that would be expected at Humboldt.

The state of the steam in the condensing jets at Moss Landing was determined by measuring the temperature and pressure before the steam entered the tank. Knowing these two state functions, the enthalpy was determined to be about 1188 Btu/lb. Assuming that the flow through the test piping and vents was at constant enthalpy, the steam temperature can be calculated for any pressure. If 18 psia is chosen as an average static pressure in the jet (corresponding to atmospheric pressure plus a submergence of about 7.6 feet), the average steam temperature in the jet tested at Moss Landing is found to be 298.4°F.

The state of the steam in the Humboldt condensing jets depends on the assumptions made as to the origin of the steam. The following table shows five conceivable steam jet conditions at Humboldt and compares them with the Moss Landing conditions. The column "initial state" refers to the fluid state before expansion to the jet. Constant enthalpy expansion is assumed, since it results in the highest possible jet temperature for a given initial state.

Initial State	Jet Enthalpy Btu/lb	Jet Temperature °F	Jet Superheat °F
1. Saturated steam at 1265 psia	1180	275	[illegible]
2. Saturated steam at 1000 psia	[illegible]	299	[illegible]
3. Saturated steam at 87 psia	1185	284	62
4. Saturated steam at 450 psia (max. enthalpy for saturated steam	1205	326	[illegible]
5. Steam-water mixture at 87 psia	Less than 1154	222	0
6. Moss Landing Tests	1188	[illegible]	70

The table indicates that the steam jet temperatures which could occur at Humboldt would not differ greatly from those at Moss Landing. Consequently, the heat transfer rate from the steam will be essentially the same.

The effect of superheat on the energy to be transferred is negligible since at 18 psia and 300°F the energy in superheat is 37.9 Btu/lb, as compared with a latent heat of 963.6 Btu/lb.

In addition to the above thermodynamic considerations, the evidence of test results obtained with film type condensation of superheated steam supports the claim that superheat would have no significant effect on the condensation rate. Experiments on film condensation on a solid surface[1] show the heat transfer rate for superheated steam to be the same as, or slightly higher than, for saturated steam. A superheat of 180°F produced a heat transfer rate 3% greater than was measured with saturated steam under similar conditions.

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[1] W.H. McAdams, Heat Transmission, Third Edition, p. 351

QUESTION 2.a.

What assurance is there that the equilibrium pressure in the suppression pool is also the maximum pressure? Is it not possible that at some time during the release a portion of the steam might not be condensed, and that the partial pressure due to it would make the total larger than the equilibrium value obtained later after the steam had condensed?

ANSWER

On the basis of complete condensation of steam obtained in the suppression pool, as demonstrated in the answer to Question 1.a., the maximum pressure in the suppression chamber has been calculated as that resulting from transfer of water, steam, and heat energy from the dry well and reactor vessel to the suppression chamber and without the benefit of any cooling by the chamber walls. This is explained more fully in the answer to Question 1.b.

The pressure rise in the suppression chamber due to an accident would not be uniform. The San Jose transient tests indicated a peak pressure occurring almost immediately after the start of the accident which was a result of the adiabatic compression and heating of the suppression chamber air due to the sudden injection of the air from the vent pipes and the dry well. For Humboldt, such a peak is calculated to be about 8 psig, which is lower than the 9.5 psig peak obtained by the method described in the answer to Question 1.b. and Appendix III.

The determination of maximum pressure in the suppression chamber, as stated above, is based on complete condensation of the steam. Both theory and tests indicate that the steam will condense completely in the water pool. Consequently it does not appear possible for significant amounts of uncondensed steam injected into the pool to escape to the vapor space and appreciably raise the pressure of the suppression chamber.

Condensation of steam was complete in all of the transient tests at San Jose and all of the compartment tests at Moss Landing. These tests have been described in previous amendments. However, further remarks on the limits of detection of steam by visual observation are given below.

The minimum amount of steam leaving the pool which could be detected by visual observation is estimated to be 100 lbs/hr or less. This compares with a steam flow in the compartment tests of 85,000 lbs/hr. If a fraction 100/85,000 of the total steam to be condensed in the Humboldt pool escaped to the vapor space, the resulting increment in vapor space pressure would be .6 psi. This is small compared with the allowable pressure of the space of 33 psig as described in the answer to Question 1.b.

QUESTION 2.b.

What are the sizes, locations, and number of penetrations of the dry well? What will be their effect on the integrity and function of the dry well? What are the design criteria for the dry well wall to assure the integrity of the concrete from the initial blast of steam?

ANSWER

I. DRY WELL VESSEL PENETRATIONS

There are a total of 56 penetrations in the dry well vessel, plus a 14 ft. diameter flanged and gasketed top access head for refueling and a 6 ft. diameter flanged and gasketed head for control rod drive removal. The majority of the penetrations are weld end type from 4" to 40" pipe sizes. Smaller connections are socket welds or pipe nipples. There are five 24" flanged connections which contain the electrical leads mounted through the flange plates. If leakage develops at any time in these connections, the entire flange can be replaced with a new flange plate and connectors. There are also four 24" flanged connections which contain the 64 hydraulic control piping entries socket welded to the flange plates, and one 6" flanged connection for the in-core flux monitor calibration wires. The location of the various dry well penetrations are shown on Drawing No. 55455.

Each primary system pipe enters the dry well vessel through a guard pipe which, in effect, is a continuation of the dry well vessel. The guard pipe closes onto the primary pipe by welding to the reactor side of the isolation valve body. The bellows are employed in the guard pipe to permit freedom of movement of the primary system pipe so no undue stresses can be developed in it.

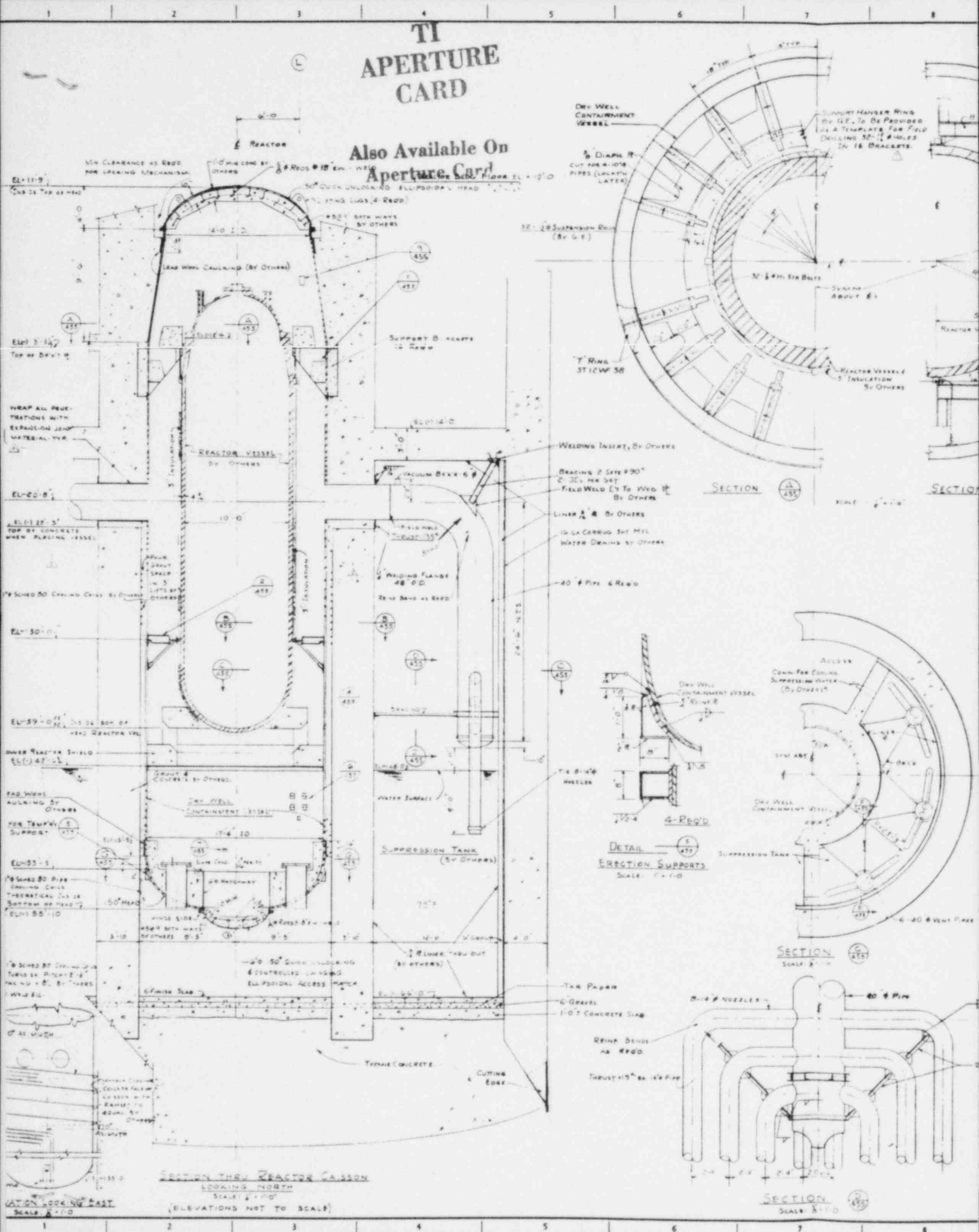
The dry well vessel purge line to the stack employs two valves in series, one locked closed and the other normally closed. The purge can also be diverted into the refueling building gas treatment system. The drain from the dry well also employs two normally closed valves in series. Details of these lines and other lines servicing the dry well vessel are shown schematically in Addendum C, Section II, Figure 1-B-10.

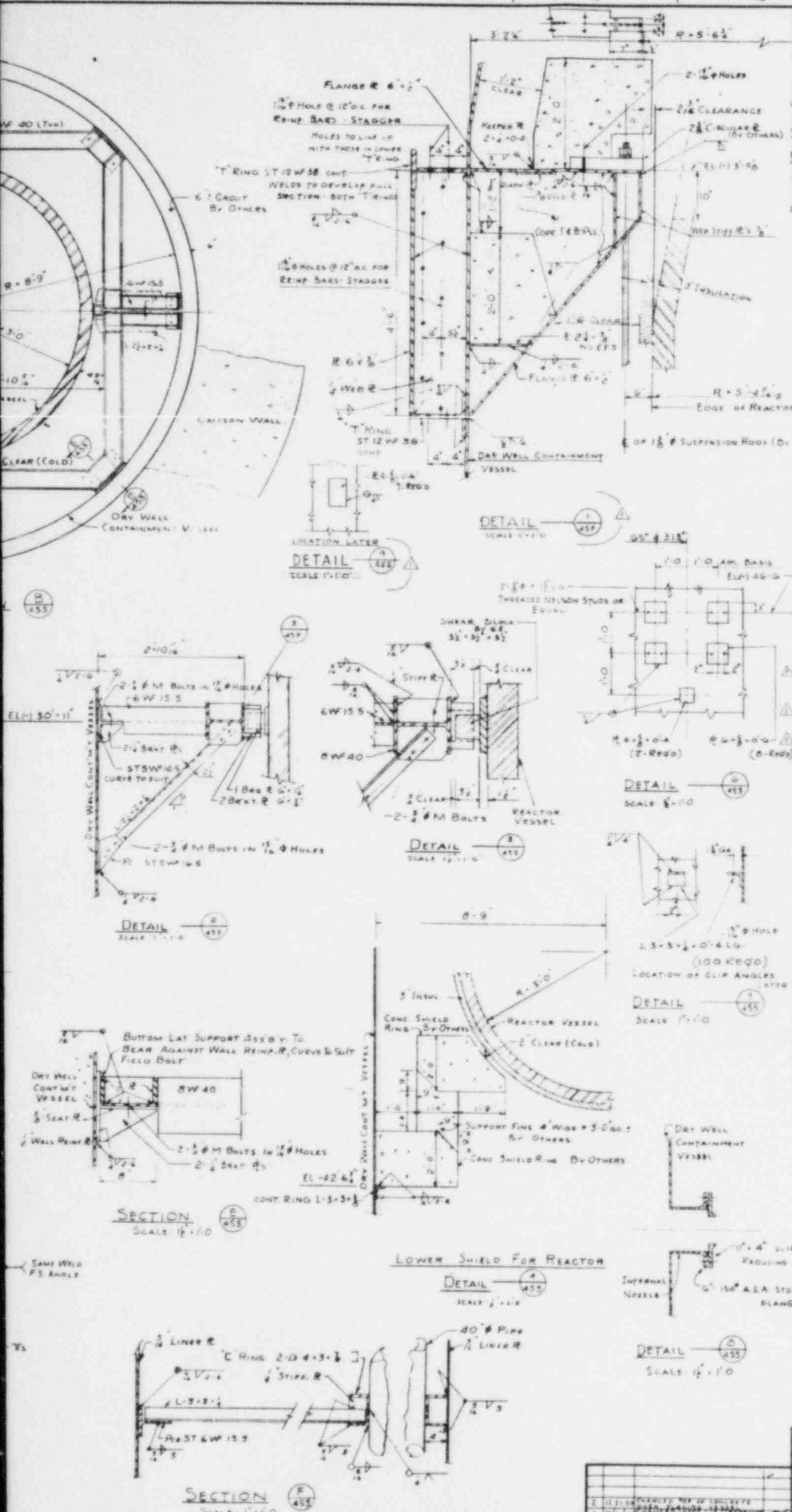
II. EFFECT OF PENETRATIONS ON DRY WELL INTEGRITY

The integrity of the dry well vessel is maintained by utilizing all-welded connections except for the top and bottom flanged access heads, and the electrical and hydraulic control piping flanges described above. The vessel, with its penetrations will be designed and constructed in accordance with the ASME code, including code-required reinforcements at openings and penetrations. The flanged access openings can be leak-tested by a low pressure leak test each time the heads are replaced as is done on vapor containment enclosures. Conventional instrumentation as developed and used for vapor containment, such as valve position signals and interlocks, will be installed to insure that the operators are warned in the event of a failure or breach of containment integrity.

TI APERTURE CARD

Also Available On Aperture Card





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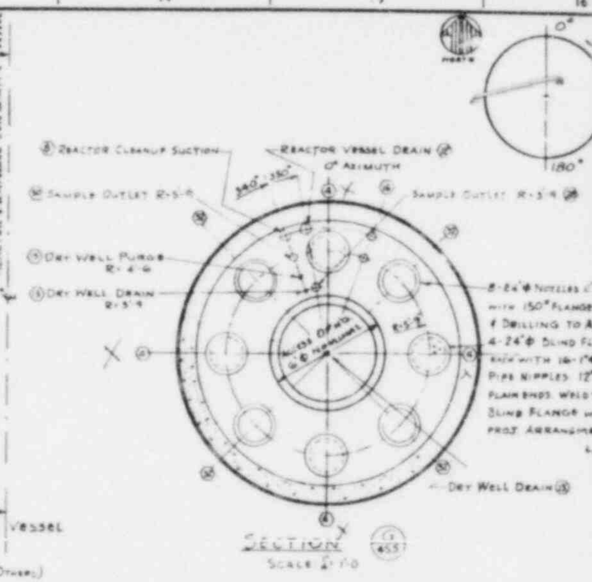


NOZZLE LIST

Desig-nation	No Req'd	Nom Diam	Type	Design Press PSIG	Length	Elev	Azimuth Degrees	Remarks
1	1	24"	Weld End	250	5'-6"	4'-7 3/8"	180°	Steam Out
2	1	20"	"	"	4'-9"	(-)19'-6"	180°	Feed Water
3	1	18"	"	"	4'-9"	(-)1'-6	180°	Shut Down Ret
4	1	18"	"	"	4'-9"	(-)12'-3	270°	Shut Down Suct
5	1	12"	"	"	4'-9"	(-)1'-6	90°	Emer Cond Ret
6	1	8"	"	"	5'-9"	2'-3	0°	Liquid Poison
7	1	8	"	"	5'-9"	4'-3	0°	Cool Spray
8	1	8"	"	"	0'-8	See Sec (G)	340°	Reactor Cleanup
9	4	10"	"	125	5'-9	(-)25'-2	[illegible]	Relief Valve Dischar
10	6	40"	"	72	5'-9"	(-)20'-8	60°, 105°, 150°, 210°, 255°, 300°	Dry Well Vent Pi
11	4	24"	Flg	300 ASA	2'-9	See Sec (G)	0°, 90°, 180°, 270°	Scram Control En
12	1	6"	Weld End	250	0'-8"	See Sec (G)	350°	Reactor Vessel Dr
13	2	2"	Nipple	72	0'-6"	See Sec (G)	340° [illegible]	Dry Well Drain
14	1	8"	Weld End	250	4'-9"	6'-0	90°	Spare
15	2	[illegible]	[illegible]	72	[illegible]	[illegible]	340°	Dry Well Purge
16	2	6"	Weld End	"	0'-8"	See Sec (G)	20°	Cooling Water
17	6	6"	"	"	See Main Sect (Suppr. Tank)		60°, 105°, 150°, 210°, 255°, 300°	Vacuum Br'k on Ven
18	2	2"	Soc Weld	—	Coupling	[illegible]	345°	Dry Well Level Con
19	1	1/4"	"	—	"	(-)25'-4	340°	Dry Well Press
20	1	1"	Pipe Tap	—	"	(-)25'-4	330°	Dry Well Temp
21	1	4"	Weld End	250	4'-9"	(-)8'-0	0°	Spare
22								
23	1	6"	"	250	7'-6"	(-)18'-6	350°	Level Control
24	1	6"	"	"	7'-6	(-)14'-6"	350°	" "
25	1	6"	"	"	7'-6"	(-)18'-6"	340°	" "
26	1	6"	"	"	7'-6"	(-)14'-6"	340°	" "
27	1	6"	"	"	7'-6"	(-)16'-6	350°	Sample
28	1	6"	"	"	0'-8"	See Sec (G)	350°	"
29	1	6"	"	"	7'-6"	(-)16'-6	340°	"
30	1	6"	"	"	0'-8"	See Sec (G)	340°	"
31	1	6"	Flg	300 ASA	See Dtl	(-)31'-6	345°	Flux Monitor
32	4	24"	"	300 ASA	2'-9"	See Sec (G)	[illegible]	Electrical Outle
33	1	24"	"	300 ASA	4'-0	(-)11'-0"	191°	Elec Out (Bimcha [illegible]
34								
35								
36								
37								

DESIGN DATA

1. Weight of dry well containment vessel = (to be determined by fabr[icator])
2. Weight of concrete inside containment vessel = 235^k
3. Weight of water inside containment vessel when flooded to top of reactor core = 310^k (after [illegible] only)
4. Max weight of loaded reactor vessel = 578^k

Lateral Forces:

Seismic Force acting on reactor vessel = 0.25g
Center of gravity of reactor vessel (estim.) = 18.0' from [illegible] bottom [illegible]
or
Jet Force of 18.4^k acting at any point inside the containm[ent] vessel resulting from a 1" Ø break in the reactor vessel. Whe[re] containment vessel is backed up by inner caisson wall, it is a[s]sumed that caisson will resist jet force. Spread jet force over 4 ft diameter at heads, in conjunction with 72 psig int. press[ure].

Design Pressures & Temperature

Internal Pressure = 72 psig (Peak)
External Pressure = 0.5 psig
Temperature Rise = 140° F (70°F at installation to 210°F when reach 72 psi[g])

No.	Date	Description	Appd
2	12-21-59	Changed top of concrete [illegible]	
1	11/3/59	[illegible] issued for quotations	
0	10-6-59	Issue for quotation	

TABLE OF CHANGES

BECHTEL CORPORATION
SAN FRANCISCO, CALIFORNIA

Job 29[illegible]

Approved By | GM 141436 | Drawing List
By [illegible] | Dr [illegible] | Ch [illegible] | O.K. | Date 9-17-59 | Scales As Shown

DRY WELL CONTAINMENT VESSEL

HUMBOLDT BAY POWER PLANT—UNIT NO. 3
PACIFIC GAS AND ELECTRIC COMPANY
SAN FRANCISCO, CALIFORNIA

Bill of Material
Supersedes
Superseded By
Sheet No.
Drawing No. 55455

III. DRY WELL CONCRETE WALL DESIGN

The concrete behind the dry well wall is designed to resist the force resulting from the initial blast of steam. This force is transmitted through the dry well vessel wall directly into the reinforced concrete as the vessel is grouted into place over its entire length. This force has been calculated to be 165,000 lbs based on the MCOA break pressure of 1265 psia. This is spread over 101 sq. inches resulting in a loading of 1640 psig on the concrete with no credit being given to the spread of the jet stream in the space between the reactor vessel and the dry well wall.

The stresses due to this pressure are within the allowable working stresses for the concrete in accordance with the American Concrete Institute Standards for short time loading. The concrete will have a minimum compressive strength of 3000 psi. Temperature effect on the concrete is not significant. In their paper entitled "Effect of Long Exposure of Concrete to High Temperature" the Portland Cement Association reports that concrete has been successfully used for structural purposes at continuous exposures of up to 600°F. For the maximum credible operating accident the maximum steam temperature would be 575°F and the duration of exposure is from 10 to 12 seconds.

The top and bottom heads of the vessel which are not backed by concrete utilize 150 psig design for steel thickness and, in addition, have at least 12 inches of concrete poured inside the heads. This would serve to distribute the jet loading over a larger area of the steel plate so that the plate would not be overstressed.

APPENDIX I
CALCULATION OF DISTANCE STEAM TRAVELS BEFORE CONDENSATION IN POOL

I. SUMMARY

The distance traveled by steam leaving a vent pipe is the sum of (1) the length of the jet before it breaks up into small bubbles, and (2) the distance the bubbles travel while the remaining steam condenses. These distances for the Humboldt maximum credible operating accident are calculated as 5.3 feet and 2 inches respectively. Details of the calculations are shown in this appendix.

II. LENGTH OF JET BEFORE BREAK-UP[1]

$$\frac{L_H}{L_M} = \frac{V_H}{V_M} \; \frac{D_H}{D_M} \qquad (1)$$

where L is the length of jet before breaking up
V is the initial jet velocity
D is the orifice diameter
H is a subscript referring to Humboldt
M is a subscript referring to Moss Landing tests

Typical set of Moss Landing test data:[2]

V_M = 1,000 ft/sec (calculated from test measurements)

D_M = 0.335 feet

L_M = 0.5 to 1.0 feet

For steam flow at acoustic velocity, with the Humboldt nozzle diameter:

V_H = 1,600 ft/sec

D_H = 1.104 ft

Using equation (1) and L_M = 1.0 ft

$$L_H = \frac{1600}{1000} \times \frac{1.104}{0.335} \times 1.0 \text{ ft} = 5.3 \text{ feet}$$

---

1 Lord Rayleigh "Theory of Sound" Section 360.

2 Photograph shown in Addendum A, Appendix I, Figure 5.

III. LENGTH OF TRAVEL OF STEAM BUBBLES DURING CONDENSATION

The length of travel of a steam bubble during condensation depends on (1) the initial size of the bubble (2) the heat transfer rate and (3) the velocity of the bubble. These three factors will be considered separately.

A. Bubble Size

Taylor instability[1] may be used to predict conservatively the size of steam bubbles formed when the steam jet breaks up. If an interface exists between a gas and a liquid and the interface is accelerating in the direction of the liquid, irregularities in the surface will grow and bubbles will tend to form with a diameter in the range of:

$$2\pi\sqrt{\frac{3\sigma g}{a(\rho_L-\rho_g)}} > D > 2\pi\sqrt{\frac{\sigma g}{a(\rho_L-\rho_g)}} \qquad (2)$$

where D = diameter bubble, ft

σ = liquid surface tension (0.004 lb/ft at 222°F)

$\rho_L - \rho_g$ = difference in the liquid and gas densities (59.5 lb/ft^3 at 18 psia)

g = gravity constant

a = acceleration of interface (=g)

Substitution in equation (2) gives a calculated maximum bubble diameter of half an inch.

B. Condensation of Steam Bubble

R = radius of steam bubble, ft.

Q = energy, Btu

θ = time required to condense the steam bubble

Δt = temperature difference for heat transfer, °F

U = heat transfer coefficient, Btu/hr °F ft^2

ρ_g = steam density lbs/ft^3

H_{f_g} = latent heat of vaporization

Consider a small change in radius R caused by condensation

$$\Delta Q = 4\pi R^2 \Delta R \, \rho_g H_{fg} \qquad (3)$$

[1] Bellman and Pennington, "Effects of Surface Tension and Viscosity on Taylor Instability", Quarterly of Applied Mathematics 12, 1954, page 151 - 162.

By the heat transfer equation

$$\frac{\Delta Q}{\Delta \theta} = 4\pi R^2 U \Delta t$$

$$\Delta Q = 4\pi R^2 U \Delta t \Delta \theta \qquad (4)$$

Equation (3) and (4) are combined to give:

$$U \Delta t \Delta \theta = \Delta R \, \rho_g H_{fg} \qquad (5)$$

Consider ΔR and $\Delta \theta$ as differentials, integrating with $R = R_o$ at $t = 0$ and assuming Δt, U, ρ_g and H_{f_g} to be constant

$$\int_0^\theta d\theta = \frac{\rho_g H_{fg}}{U \Delta t} \int_{R_o}^0 -dR \qquad (6)$$

$$\theta = \frac{\rho_g H_{fg} R_o}{U \Delta t} \qquad (7)$$

In evaluating equation (7), the following parameter values have been used.

Δt = 100°F

ρ_g = .0450 lbs/ft^3

H_{f_g} = 963.6 Btu/lb

Take R_o = $\frac{.54}{2 \times 12}$ inches = 0.0223 ft. (the largest value from A)

U = 5000 Btu/hr °F ft^2

The value of U = to 5000 is justified by high speed movies taken of nucleate boiling[1], where bubble size and times for collapse were measured. Using these values with equation (7) the time for condensation of a pure steam bubble is:

$$\theta = \frac{.045 \text{ lbs/ft}^3 \times 963.6 \text{ Btu/lb} \times .0223 \text{ ft} \times 3600 \text{ sec/hr}}{5000 \text{ Btu/hr°F ft}^2 \times 100°\text{F}}$$

$$= .007 \text{ seconds}$$

---

[1] Gunther, F. C., "Transactions" ASME, 1951, p. 115 or see work of Scherrer, McLean, and Hoffman of the Naval Research Laboratory in 1955.

C. Distance Bubble Travels to Condense

The distance S that a steam bubble travels in condensing depends upon its velocity V and the time to condense θ.

$$S = V\theta$$

The bubble velocity is estimated to be about 20 ft/sec from observations of the Moss Landing tests.

The distance a .54 inch bubble would travel before being condensed would be about:

$$S = 20 \text{ ft/sec} \times .007 = 0.14 \text{ ft.}$$

APPENDIX II

CALCULATION OF DRY WELL DESIGN PRESSURE

I. FLOW THROUGH PRIMARY SYSTEM BREAK

The first step of the dry well pressure computation is to determine the flow through break and vent pipes. The break is considered an orifice for which the general flow equation is:

$$Q = CA\sqrt{2gh} \qquad (1)$$

where Q = ft^3/sec
A = ft^2
h = ft
g = ft/sec^2

and C is the discharge coefficient.

The maximum credible size of this break has previously been established as equivalent to the transvers area of a 12" Sch. 80 pipe, or 0.703 ft^2. It is assumed that the flow through this break is 100% water. This is a conservative assumption inasmuch as any steam present, due to its much greater volume, would reduce the mass flow rate through the break and therefore result in a lower pressure buildup in the dry well.

The discharge coefficient C has a value of approximately 0.61 for a sharp-edged orifice or a pipe stub at high heads. Although a well rounded orifice having a coefficient higher than 0.61 may possibly be more correct for a break in one of the smaller nozzles located on the reactor vessel, for a location where the maximum credible size break could occur, a higher coefficient than 0.61 is not conceivable. In addition, where flow of saturated water through orifices of significant length is considered, flashing would occur in the orifice thus reducing the discharge through the orifice. For the considered break, the choking effect of the flashing would be quite pronounced. An arbitrarily selected test run from WAPD test report #CTA-EL-3581, dated February 28, 1957, by R. T. Graulty, and dealing with saturated water orifice tests, indicates an orifice coefficient of 0.51 for saturated water @ 1005 psia flowing through an orifice 0.28" long with a 1/4" bore. However, no credit is taken in the calculations for the flow reduction due to the choking effect, a conservative approach. The orifice formula above is transformed into:

$$W = 12CA\sqrt{2g\frac{(P_1 - P_2)}{\bar{v}_{f1}}} \qquad (2)$$

where W = lbs/sec break flow
P_1 = Upstream or Reactor press., psia
P_2 = Downstream or Dry well press., psia
$\bar{v}_{f1}$ = Spec. vol. of water @ P_1, ft^3/lb
$2g$ = 64.4 ft/sec^2

Immediately after the break the reactor pressure would begin to decrease. However, for the time interval between the break and the occurrence of dry well peak pressure, the conservative approach is taken that the reactor pressure remains constant.

For P_1 = const. = 1265 psia
$\bar{v}_{f1}$ = 0.0225 ft^3/lb
A = 0.703 ft^2, break area
C = 0.61

the above equation becomes:

$$W = 275.3\sqrt{1265 - P_2} \qquad (3)$$

With constant reactor pressure this equation yields the total break flow as a function of dry well pressure, P_2.

II. PRESSURE TO EXPEL WATER FROM VENTS

As the pressure builds up in the dry well and thereby in the vent system, the water contained in the submerged vent pipes will be displaced. The initial pressure buildup required in the dry well to displace this water has been calculated, as shown on page 1 of Addendum D, Section II, to be approximately 35 psig. This pressure is considerably less than the dry well pressure required to maintain a flow through the vent system equal to the break flow. Therefore, the dry well peak pressure will occur after the water in the vent pipes has been expelled and flow through the vents has been established.

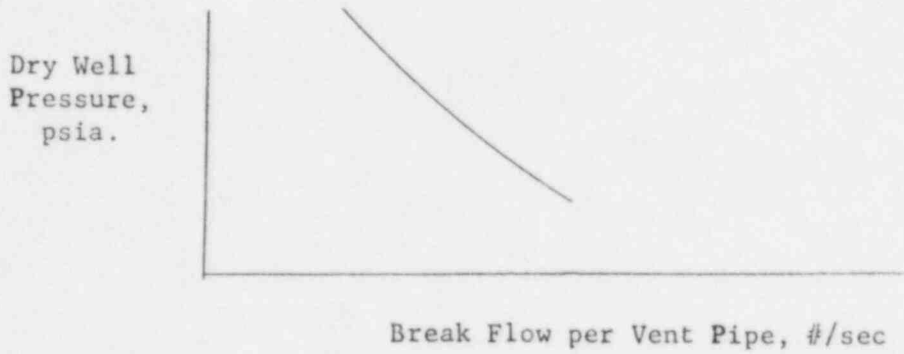
III. PEAK PRESSURE

Since the flow carrying capacity of the vent pipes increases with dry well pressure and the discharge through the break into the dry well decreases with increased dry well pressure, the peak dry well pressure will occur when the mass flow out of the break equals the mass flow through the vents.

With the six equally sized vent pipes sharing the flow equally, the flow per vent pipe can be established:

Dry well pressure:	40	65	100	120, psia
Total break flow:	9636	9539	9396	9319, lbs/sec
Flow per vent pipe:	1606	1590	1566	1553, lbs/sec

This relationship is plotted:



IV. EFFECT OF WATER CARRYOVER ON PEAK PRESSURE

When the high pressure saturated water is discharged through the break opening into the dry well at the much lower pressure, flashing occurs. With a dry well pressure of 87 psia, for example, approximately 30% of the water will flash off as steam. How much of the water remaining in the dry well would be carried over to the suppression pool by the sweeping action of the steam is not predictable. For this reason the conservative assumption was made that all the water in the dry well is carried over into the vent pipes. The effect of the carryover on peak dry well pressure has been investigated, and it is estimated that with 0% and 50% carryover the dry well peak pressure would be approximately 70% and 85% respectively of that calculated for 100% carryover.

V. CRITICAL END PRESSURE

On the basis of 100% water carryover, the flow through the vent pipes is two-phased, i.e., a mixed flow of water and water vapor with continued flashing as its pressure decreases from the dry well pressure existing at the vent pipe inlet to that pressure which prevails at the vent pipe outlet. At this latter point a "critical-end-of-pipe-pressure" is established. The existence of this critical pressure was extensively investigated by M. W. Benjamin and J. G. Miller and discussed in Transactions of the ASME, Vol. 64, 1942. A brief outline of the theory is given below, followed by its application to the dry well design and a discussion of the design margin indicated by some recent test results.

A. Theory of Critical End Pressure

When saturated water travels from an initial saturation pressure to a reduced pressure, the rate of flow is governed by the increase in volume due to the flashing of the liquid and the acceleration of mass resulting from the reduction of pressure. The energy made available (vdP) by the reduction in pressure, varies approximately as a linear function with the specific volume, whereas the resultant kinetic energy increase (VdV/g), (where V is velocity) and overall frictional resistance ($KV^2/2g$) vary as the square of the increase in specific volume. Therefore, at a certain reduced pressure the energy made available by an increment of pressure reduction will just equal the resultant increases in kinetic energy and friction. Since this phenomenon is found only at the end of a line which is followed immediately by an increase of pipe size or flow-area, it is called the critical-end-pressure. The critical-end-pressure will represent the actual pressure at the end of the line only when the back pressure is less than this critical-end-pressure.

B. Application to Dry Well Design

For flow of fluids at zero heat transfer the equation for conservation of energy can be expressed as follows:

$$\frac{dP}{\rho} + \frac{VdV}{g} + \frac{f}{D}\cdot\frac{V^2}{2g}\cdot dL \pm dH = 0 \qquad (4)$$

which for pipe length dL = 0 and elevation difference dH = 0 becomes:

$$\frac{dP}{\rho} + \frac{VdV}{g} = 0 \qquad (5)$$

Since $V = \frac{W}{\rho A}$ or $dV = -\frac{W}{A}\frac{d\rho}{\rho^2}$ and $\frac{dP}{\rho} = 144\frac{dp}{\rho}$

equation (5) can be transformed to:

$$\frac{1}{2}\rho^2\left(\frac{dp}{d\rho}\right) = \frac{(W/A)^2}{288\,g} \qquad (6)$$

Symbols for equations (4), (5), and (6) are defined as follows:

ρ = Density, lbs/ft^3
P = Pressure, lbs/ft^2
V = Velocity, ft/sec
D = Pipe Diameter, ft
f = Friction factor
p = Pressure, lbs/in^2
A = Pipe Area, ft^2
W = Pipe flow rate, lbs/sec

This equation applies at the end of the line where the critical pressure occurs. It can be used to determine this critical pressure by the trial and error method. However, it is necessary first to determine the value of both $dp/d\rho$ and ρ of the mixture as a function of reduced pressure.

The density of the mixture is computed stepwise at reduced pressures in the total range between the estimated discharge and dry well pressure on the basis of an initial saturation pressure of 1265 psia. For this computation, it is assumed that the process is an expansion at constant enthalpy. Actually, the expansion through the vent piping is neither isenthalpic (constant enthalpy) nor isentropic but a polytropic process.

However, calculations show that a somewhat greater dry well pressure results from calculations based on isenthalpic expansion; therefore, this more conservative method is chosen. The mixture density is calculated by use of the following equations:

$$\text{Flashing fraction} = q = (h_{f_1} - h_{f_2})/h_{fg_2} \qquad (7)$$

$$\bar{v}_2 = \bar{v}_{f_2} + q \cdot \bar{v}_{fg_2} \qquad (8)$$

$$\rho_2 = 1/\bar{v}_2 \qquad (9)$$

where the symbols follow the terminology used in Keenan & Keyes steam tables from which all values were taken.

From the tabular computation of the density ρ both Δp and $\Delta\rho$ are easily determined as the "step" in pressure and change in ρ respectively. Therefore, for a given value of W/A, a value of p is determined for which the values of ρ and $\Delta p/\Delta\rho$ satisfy equation (6) for critical pressure.

C. Margin Indicated by Test Results

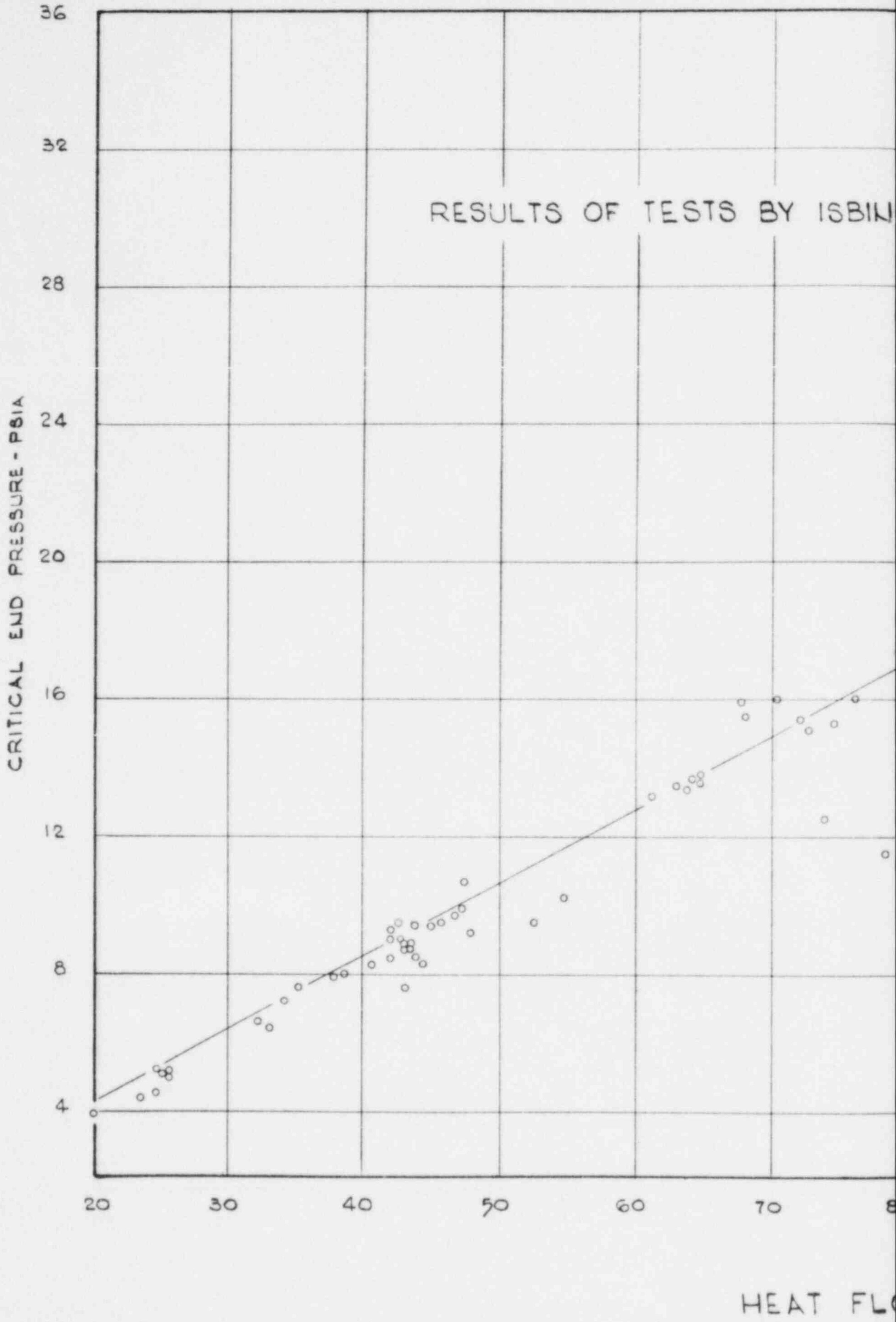
Recently, extensive tests of critical flows for a two-phase steam-water mixture were carried out by Messrs. H. S. Isbin, J. E. Moy, and A. J. R. Da Cruz at the Department of Chemical Engineering of the University of Minnesota[1] Briefly, the test arrangement was as follows: Steam and water flows were mixed, passed through a horizontal test section and then discharged into an enlarged pipe which was connected directly to a condenser. The total mass flow to the test section was readily set and controlled - independent of the pressure regulation in the condenser - by throttling separately the supply of steam and water to the mixer. Various methods of mixing employed a variety of spray nozzles, various length of test sections and cold and preheated water with proportionately different steam flows to provide the same desired exit quality. It was found that the values determined for the critical pressure were independent of the method of mixing.

The total energy of the discharge stream was determined by measuring separately the flow rates and enthalpies of the steam and water feeds to the mixer. Four full-bore test sections were used consisting of stainless steel pipes 1/4, 1/2, 3/4, and 1-inch diameter, and two annuli were formed by using a 1/4-inch brass pipe supported inside the 3/4- and 1-inch pipes by 3 brass pins. The discharge of each test section had a sharp, flat cross-section.

Of the test results only the ones were used for which the total energy (Btu/lb) as measured is below the maximum value possible for saturated water, i.e., water at the critical pressure of 3206.2 psia. In this way the mixture tested could as well have been produced by flashing of initially saturated water as by mixing of steam and water.

Figure E-1 shows the values of measured critical pressure as a function of the product of flow rate per unit area and heat content. This plot shows in a definite pattern that the critical pressure is proportional to the heat flow rate ($Btu/ft^2 \cdot sec$) independent of the test pipe size.

[1]AIChE Journal, September 1957, pages 361-365



RESULTS OF TESTS BY ISBIN
CRITICAL END PRESSURE - PSIA
36
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HEAT FL

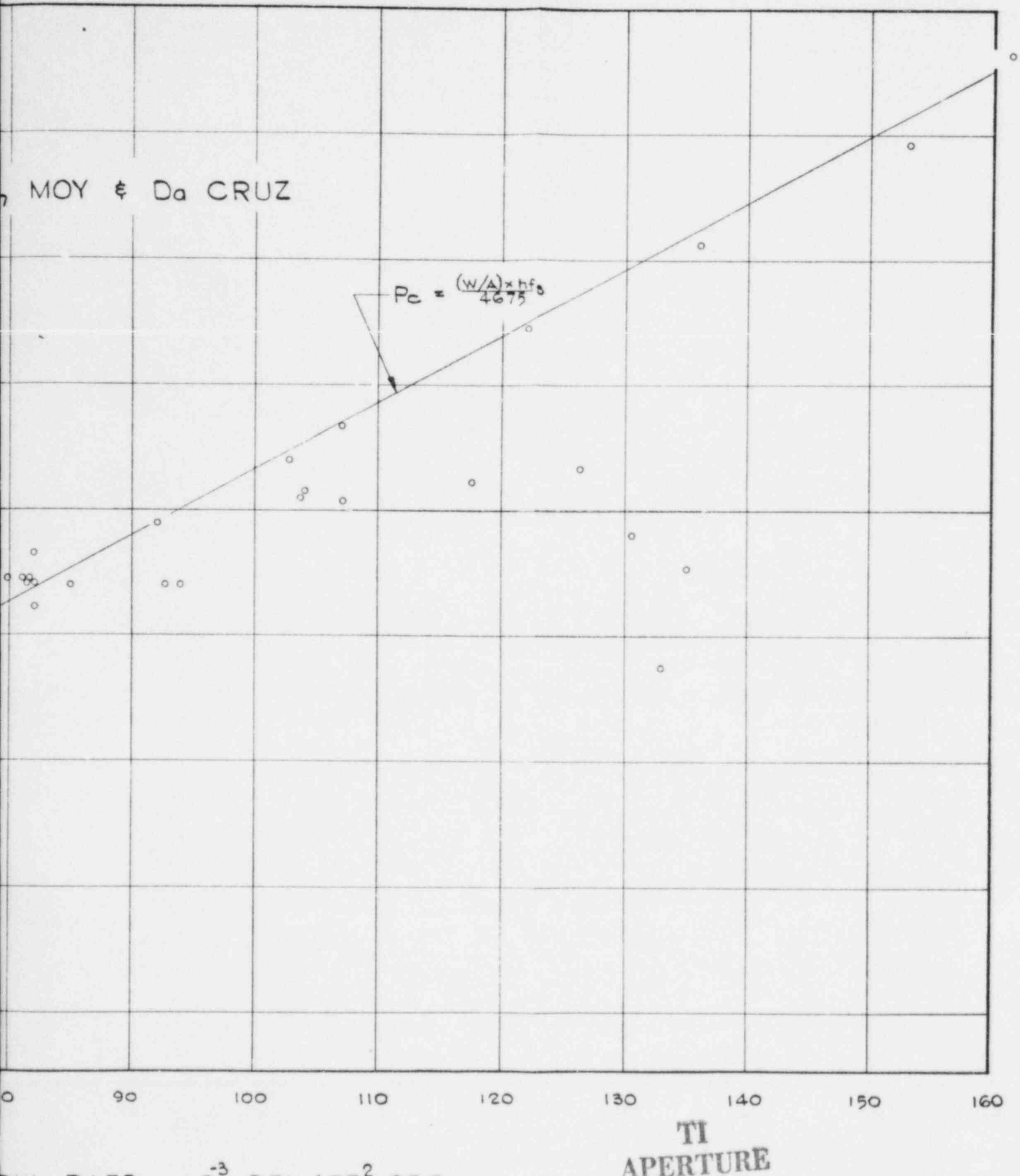

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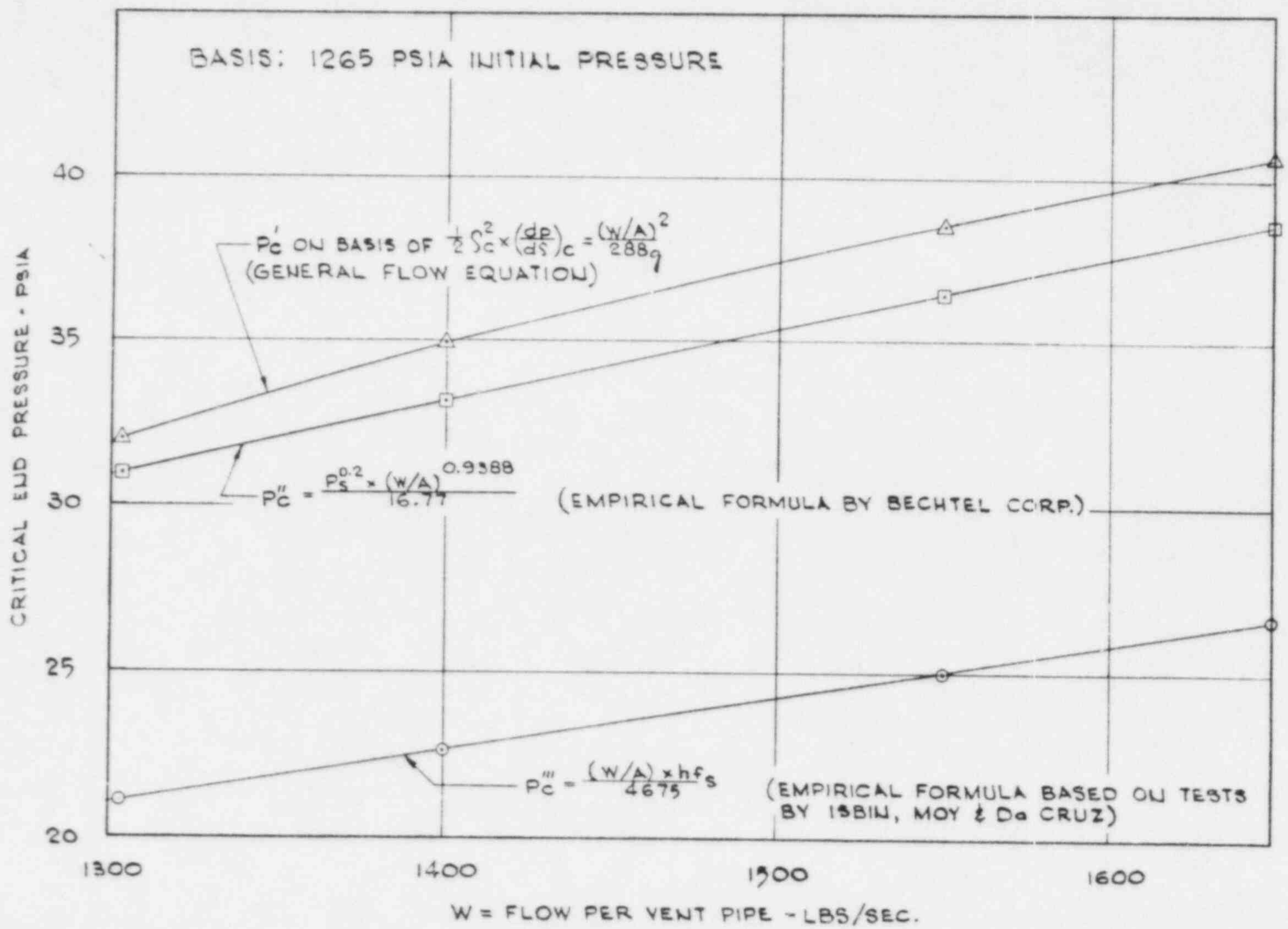
FIG. E-1



CRITICAL END OF LINE PRESSURE VS. FLOW

FIG. E-2

Applying flow rates and heat contents similar to those of the Humboldt design to this empirical relationship, the value of critical pressure is found to be only approximately 65% of that calculated on the basis of the general flow equation (6). The reason for this difference is probably that the latter is based on a completely homogeneous mixture which is hardly obtainable in reality. This comparison is shown on Figure E-2. This test substantiates the critical pressure phenomenon and indicates conservatism in the formulas used for the Humboldt design, as no credit has been taken for this margin.

VI. TRIAL-AND-ERROR CALCULATION OF PRESSURE DROP

From the plot of dry well pressure versus flow per vent pipe (see page 3), two vent pipe flow rates are selected, one corresponding to a dry well pressure lower and one corresponding to a dry well pressure higher than expected. For each flow rate the critical-end-pressure is computed as outlined above, and, for the same flow rates, the line loss or pressure drop in the vent piping is determined.

This pressure drop calculation for flashing flow is also a trial and error calculation using a form of the general flow equation,

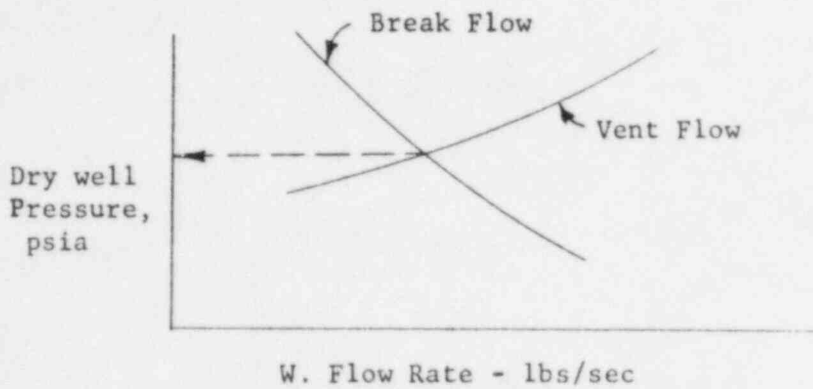
$$\frac{f}{D}\cdot dL = \frac{\int_{p_2}^{p_1}\rho\,dp + \frac{1}{144}\int_0^H \rho^2\,dH}{(W/A)^2/288g} - 2\ln\left(\frac{\rho_1}{\rho_2}\right) \qquad (10)$$

This equation for downward vertical flow is derived by combining the General Energy Equation (4) as stated previously (see page 4) with the Continuity Equation $V = W/\rho A$.

Equation (10) is closely approximated by:

$$\Sigma K + \frac{fL}{D} = \frac{\sum_{p_2}^{p_1}\rho\Delta p + \rho_1\frac{\frac{\rho_2}{2}\cdot H}{144}}{(W/A)^2/288g} - 2\ln\left(\frac{\rho_1}{\rho_2}\right) \qquad (11)$$

where p_1 and ρ_1 correspond to the piping inlet conditions (or dry well pressure), p_2 and ρ_2 correspond to the piping outlet conditions (or critical-end-pressure), H is the elevation difference, A the piping cross-sectional area and ΣK the sum of all resistance coefficients for, and L the length of, the piping system. By trial and error for each flow rate, a value of p_1 can be determined which satisfies the equation. The summation of $\rho\Delta p$ is carried out in the previously mentioned tabular calculation of ρ as a function of the reduced pressure. By plotting the calculated values of p_1 (dry well pressure) versus vent pipe flow rate on the same graph as mentioned previously (see page 3), the intersection of the two lines, where both simultaneously satisfy the flow versus dry well pressure relationship, establishes the equilibrium pressure, i.e., the actual dry well peak pressure.



The formula stated above for determination of the critical-end-pressure for flashing mixture results in slightly higher pressures than a simplified formula developed and used by Bechtel Corporation engineers. This comparison is shown on Figure E-2. In connection with the formulas for calculation of pressure drop in lines carrying flashing mixtures, this simplified formula has been used successfully for a number of years by Bechtel Corporation in power plant design.

APPENDIX III
CALCULATION OF SUPPRESSION CHAMBER MAXIMUM PRESSURE

The maximum pressure which can be attained in the suppression chamber has been determined by means of energy and mass balances for incident and peak pressure conditions using the system design volumes and the maximum operating conditions and assumptions stated below.

Initial Conditions at Time of Incident

1. The reactor is operating at 200 MWT power, 1265 psia, and with water level at normal maximum. Total net reactor vessel volume is 2650 cubic feet, including 860 cubic feet of steam and 1790 cubic feet of saturated water.
2. The reactor vessel (including core structure) is at same average temperature as the coolant (574°F). Its weight is 400,000 lbs.
3. The dry well, including vent pipes, has a net volume of 12,100 cubic feet and is filled with dry air at 150°F.
4. The dry well vessel weight is 177,000 lbs. and is also at 150°F.
5. Suppression chamber water pool is at 80°F and has a volume of 20,025 cubic feet.
6. Air in suppression chamber (above the pool) is at 100°F and dry, with a volume of 32,375 cubic feet.

Peak Pressure Conditions

1. Reactor scram is initiated at occurrence of MCOA. Decay heat totals 4.0 million Btu during the first 120 seconds after scram. Decay heat released after 120 seconds does not contribute to the peak pressure.
2. Feedwater continues to flow into the reactor at its maximum rate of 227 lbs/sec and with an average enthalpy of 240 Btu/lb for 60 seconds and 60 Btu/lb thereafter.
3. No energy other than 1) and 2), above, is added to the system, and no energy is lost from the system for the first 120 seconds.
4. All air is displaced to the suppression chamber air space. Air temperature is assumed same as the pool temperature.
5. Dry well and reactor vessels are filled with steam saturated at the same pressure existing in the suppression chamber.
6. Reactor vessel and core structure is at the same temperature as contained steam.

7. Dry well vessel is at 150°F.

8. All steam expelled from the dry well into the vent pipes is condensed in the pool.

9. Masses of water (all forms), air, and metal remain unchanged except for addition of feedwater.

The suppression chamber peak pressure in this system is determined largely by the ratio of the sum of the initial air volumes in the system to the final air volume. Increased temperature and specific volume of the pool water, inflow of condensed steam and feedwater, and other minor changes, however, require a mass and energy balance to determine a more precise peak pressure for the assumed conditions.

Utilizing the system design volumes and the above stated conditions and assumptions, total weights and corresponding energies can be determined for each fluid or material based on published standard values of specific volumes, specific heats, and heat contents. The total initial energy of the system is thus readily established.

The final system energy at the peak pressure condition is obtained by adding the reactor decay heat and the energy of the influent feedwater to the "initial" energy calculated previously. The total weight of water and vapor in the system is obtained by adding the weight of the influent feedwater to the sum of the water and vapor determined for the initial condition. The peak suppression chamber pressure is then determined by balancing the partial energies and masses in the system with the total final energies and masses, using a trial and error method.

An estimated temperature, volume, and energy are first selected for the pool water. These can be closely approximated by assuming, for the first trial, that all water is resident in the pool and estimating the heat transferred to the pool. The remaining volume in the suppression chamber is then available for the total weight of air previously calculated. Since the estimated pool temperature is also the air temperature, the air partial pressure is determined by use of the standard gas laws. The partial pressure of the water vapor in the chamber air space is then determined from the steam tables at the chamber air temperature. The total pressure in the air space is then the sum of the air and vapor partial pressures.

Utilizing this total chamber pressure and the assumptions previously made, the volumes, weights, and energies are determined for air, metal, and water vapor in the remainder of the system. The total of the weights and energies thus calculated are compared with the correct totals, and the pool volume, temperature, and energy are adjusted accordingly and new calculations are made. This procedure is repeated until the actual values are determined. This is the peak pressure in the chamber air space. For the Humboldt design this peak pressure has been calculated as 9.5 psig. The maximum pressure at any point in the suppression chamber is then determined by adding the hydrostatic pressure of the pool water to the pressure in the air space determined above. Therefore, the required design pressures for the suppression chamber have been selected as 10 psig in the air chamber and 19 psig at the bottom of the suppression pool.

Following is a mass (weight), volume, and energy balance before and after the MCOA:

	Pressure PSIA	Temp. °F	Volume cu.ft.	Weight lbs	Btu/lb	Energy 1000 Btu
1. Before MCOA						
Reactor vessel	--	574	--	400,000	Base	--
Reactor water	1265	574	1,790	79,555	580.6	46,190
Reactor steam	1265	574	860	2,527	1180.3	2,983
Dry well air	14.7	150	12,100	789	12	10
Chamber air	14.7	100	32,375	2,299	Base	--
Pool water	14.7	80	20,025	1,246,000	48	59,808
2. Energy Added						
Decay heat	--	--	--	--	--	4,000
Feedwater, 2 min.	--	--	--	27,000	--	4,100
Totals			67,150	1,758,170		117,091
3. After MCOA						
Reactor vessel	--	240.5	--	400,000	-36.69	-14,676
Steam, R & DW	(24.2)	240.5	14,750	880	1160	1,020
Chamber air	22.1	128.5	30,410	3,088	6.6	20
Moisture in air	2.1	128.5	(30,410)	186	1117.2	210
Pool water	--	128.5	21,990	1,354,016	96.4	130,517
Totals	24.2 psia = 9.5 psig		67,150	1,758,170		117,091

classification review not required. [illegible] 2/11/60

PACIFIC GAS AND ELECTRIC COMPANY

AMENDMENT NO. 6
ADDENDUM E
TO THE
PRELIMINARY HAZARDS SUMMARY REPORT
HUMBOLDT BAY POWER PLANT
UNIT NO. 3

JANUARY 29, 1960