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HRT MODIFIED PRESSURIZER DESIGN

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I. SUMMARY

This report outlines the work done on modification of the HRT pressurizer design. The introduction summarizes HRT mock up operations as they pertain to the pressurizer, and discusses establishment of the basic criteria for redesign. The design section presents a general description of the internal configuration of the modified pressurizer and its mode of operation. A flowsheet, conceptual drawing, and summary of design specifications are included. Calculations in the appendix justify values for the basic design parameters such as heater capacity, vaporization rate, water reserve, water level, and surge volumes. Appropriate references to unclassified literature are included.

Modification of the entire HRT pressurizing system has been a group effort by the Process Design Section. This report covers work done on the conceptual design of the pressurizer itself.

For this phase of the project acknowledgement is due R. G. McGrath for many of the calculations and design concepts, and to M. I. Lundin for his contributions and assistance as group leader. Special acknowledgement is also due R. L. Cauble for his excellent work in laying out and detailing the pressurizer.

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II. INTRODUCTION

Initial studies directed toward a redesign of the HRT steam pressurization system were started in October, 1955. At that time it became apparent from the results of HRT mockup operations with uranyl sulfate, that a pressurizer design requiring high inside pipe wall temperature ($> 300^{\circ}\text{C}$) in possible contact with fuel solution could not be used. The problem appeared to be that of excessive uranium precipitation on the inside walls of the heater legs, accompanied by considerable localized corrosion. Fuel solution coming in contact with the hot surfaces apparently formed a dense corrosive phase or precipitated. Operation under these conditions soon resulted in physical clogging of the legs and would eventually cause heater burnout.

What part the purge water stream of 10 gph (distributed equally to the three heater legs) played in the failure was not clear. The intended purpose of the purge stream from the recombiner condenser was to keep the heater legs filled with water, thereby preventing fuel solution from coming in contact with the hot surfaces. Overpressure was to be supplied by vaporizing this water, instead of fuel solution. It was originally thought that the purge would be essentially neutral (pH = 7) and free from contamination which might attack stainless steel. However, during operation of the HRT mockup with uranyl sulfate just prior to its failure, the purge stream was highly acid (pH 2.7-4.4), oxygen deficient, and contained considerable entrained uranium (1-900 ppm.).

Initial examination of the pressurizer after its failure revealed that only number two and three legs contained large amounts of precipitate. The first leg was nearly clean, although some localized pitting was in evidence. Later, during internal examination, it was found that the purge line instead of being in place along the axis of the pressurizer drum, was actually bent

over the baffle plate and into the dry pipe region in such a manner that only the first orifice hole appeared capable of delivering measurable flow. In all likelihood, during operation number two and three legs contained fairly high concentration fuel solution in contact with the hot pipe walls.

Because of this fluke in construction, it was difficult to evaluate true performance of the original design. Although this pressurizer might have functioned satisfactorily, if adequately purged with "clean water", it was now apparent that high wall temperatures had to be avoided, insofar as possible, in any region that the fuel solution might reach during all operating conditions. Isolation of the fuel solution from high wall temperatures was therefore adopted as a basic criteria for redesign of the HRT pressurizers.

Several design configurations were studied for steam pressurizing the HRT. Both an integral type unit incorporating a steam generator and fuel surge volume within the same vessel, and a two-unit system utilizing separate steam generator and surge volume vessels were considered. Both systems were based on the idea of taking a stream of "clean" water from the recombiner condenser and vaporizing it to obtain the desired steam overpressure. In either case a surge chamber was connected directly to the core loop to act as a steam chest and surge volume for the fuel solution.

It was theorized, and later verified by hydraulic test, that a flow of purge water into the hot liquid-vapor interface region of the connecting pipe from core to pressurizer would greatly reduce the concentration of fuel solution at this critical point, and thus decrease the probability of precipitation and/or phase separation. In effect, the concentration gradient would be shifted nearer the core inlet connection where the bulk temperature more nearly approached that of the fuel solution. Oxygen and acid deficiencies in the purge water were recognized as possible sources

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of trouble, but experimental data on precipitation and/or corrosion effects under simulated water purging of the fuel solution were not available. Design work went ahead on the assumption that if experience on the revised HRT mockup later pointed to oxygen and acid deficiency as a serious problem, appropriate design modification could then be made. The method of introducing this purge stream differed for the systems studied in that with the integral type unit it was done by direct overflow of excess purge water, while in the two unit system a separate purge line was required.

In the final analysis, the two-unit system offered advantages in space saving through greater flexibility in location, and better isolation of the steam generator from possible fuel contamination; but the integral type unit was basically simpler and offered decided advantage of maintaining very low dissolved solids concentration in the purge water being vaporized by removing 60% of the purge water feed through direct overflow. (Concentration factor 1.67). After demonstrating by conceptual designs that an integral type unit could be made from available materials and fitted into the reactor compartment, it was selected for detailing.

In determining the internal configuration of the unit, it was necessary to establish a second basic design criteria. It was apparent from the nature of the system selected, that continuous operation of the purge pump was essential to maintaining steam overpressure. Since this type of pump had been known to fail, it was decided that sufficient water should be stored in the steam generator to maintain full operating pressure for at least one hour after a purge pump failure. This appeared to be adequate time for either emergency repair of faults on the oil side of the purge pump system, or arrangement of an orderly shutdown. In order to minimize the water storage volume it was necessary to design for very low purge flows through the core connecting pipe after pump failure. What effect this short period of low flow might have on the precipitation and corrosion problem remains to be determined.

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III. DESIGN

A conceptual drawing of the final design of the HRT fuel pressurizer (Fig. 1), a complete flowsheet of the pressurizing system (Fig. 2), and a summary of the basic design specifications (Table 1) are included at the end of this section. A general outline of the basic design and system operation follows.

Water for steam pressurizing the core of the HRT comes from the recombiner condenser in the low pressure section of the fuel system. This water is assumed to be essentially neutral ($\text{pH} = 7$) and free from contamination which might contribute to localized corrosion in the steam generator. If, during initial testing of the HRT, it is found that the condensate does not meet these requirements, it may be necessary to introduce additional steps, such as distillation, to purify it.

The low-pressure condensate is pumped to loop operating pressure by a pulsafeeder type pump. Originally this pump supplied purge to both the pressurizer and circulating pump. It was recommended, however, that separate purge pumps be provided for the pressurizer and circulating pump, because the circulating pump purge represented a considerable drain on the storage water in the pressurizer in the event of a purge pump failure.

Water from the purge pump passes through the let-down heat exchanger where it is preheated to about 280°C , when thereactor is at power, and then enters the lower header of the steam generator. Clam-shell type heaters are attached to four pipes inclined upward from the lower header at 55° . Heat load calculations (Appendices A and B) indicate that four heaters (32 kw) are required under startup load conditions, if reasonably fast response is desired. At steady state the heat load becomes 12.4 kw: 4.4 kw for feedwater preheating,

6.6 kw for pressurizer heat losses, and 1.4 kw for direct heater losses to the cell. Part of the purge water entering the heater legs (40.5 lb/hrs, Appendix C) is vaporized to provide the desired steam overpressure. The remainder enters a storage pool in the main pressurizer drum. Natural recirculation of this water occurs through two downcomers. Entrained liquid in the generated steam is reduced by a steam baffle. The possibility of radiolytic gases becoming trapped in the apex of the steam baffle is minimized by bleed holes.

In selecting the size and internal configuration of the pressurizer it was necessary to consider the following requirements.

1. Minimum surge volume of 73 liters (2.58 ft^3).
2. Purge water storage sufficient for at least one hour of operation at pressure ($\sim 0.75 \text{ ft}^3$, Appendix E).
3. Constant purge water storage level by overflow of excess water to the core connecting pipe.
4. At least one inch difference in level between the purge water storage pool and the maximum level reached by the fuel solution before automatic dump is initiated due to a 20 liter (0.71 ft^3) adiabatic compression of the surge volume.
5. A minimum of 3.5 ft axial space requirement of four heater legs.
6. A minimum overall pressurizer size consistent with the total surge volume and purge water storage requirements.
7. Availability of materials and fabrication limitations.

A hypothetical pressurizer configuration that satisfies most of these general requirements consists of a closed right circular cylinder (12" - sch 160 pipe) about 6 ft. long containing a baffle plate (height equal to inside pipe radius) placed 2.5 ft. from one end. This gives a fuel surge volume of 0.71 ft^3 to the top of the plate and stays within the axial space requirement

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of 3.5 ft. for the heater legs and a total minimum pressurizer volume of 3.33 ft³. This arrangement, however, reduces the total surge volume below the acceptable minimum because the water storage volume to the top of the baffle is in excess of the minimum requirements. Also, it does not provide the required difference in levels between the purge water storage pool and the maximum fuel surge level. A convenient method of reducing the water volume and thus increasing the surge volume without changing the storage pool level, and at the same time establishing the required difference in levels between the storage pool and fuel surge level, is to insert a pipe from the surge chamber through the storage pool.

In the modified HRT pressurizer design the baffle is extended to close the drum and the pipe required to adjust the storage pool and surge volumes and levels passes through the diaphragm and storage pool to another baffle at the opposite end of the pool. It is still necessary to restrict the storage pool to an area immediately over the heaters in order to reduce the pool storage volume, so both end caps become part of the surge volume. For this modified design, calculations show (Appendix D) that if no letdown is assumed automatic dump is initiated when the fuel solution rises 3.8 inches into the pressurizer void space. The storage pool level at the center line of the pressurizer drum is 1.25 inches above this point.

The storage pool level is maintained constant by allowing excess purge water to be removed through a series of 1/8" orifice holes located in the end plate of the steam generator at about the centerline of the pressurizer drum. It is possible that the spray action created by this arrangement will help to dissolve some of the radiolytic gases present. A 1/2" hole centered 5/8" above the orifice holes is provided as an overflow in case the orifice holes become clogged.

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A drain line is placed along the bottom of the pressurizer from the end cap region of the steam generator to the surge chamber. This line returns any liquids which might expand through the surge pipe into the end cap, and also provides a path for the purge water coming from the orifice holes. Some difficulty might be experienced in draining a small pool of liquid behind the drain line. However, the concentration of fuel solution which might collect at this point is greatly reduced by the purge water flow.

A liquid level indicator is provided on the steam generator in order to keep the reactor operator informed as to the volume of water in storage. This is particularly important during purge pump failure, since a history of the liquid change is necessary in order to estimate the time remaining before mandatory shutdown. This instrument also provides a means of controlling the liquid level at a point below the orifice holes in the event that purge flow to the core connecting pipe is not wanted. This control could be coupled to a bypass on the purge pump.

A dry pipe is placed along the top of the steam generator to provide additional entrainment separation for the vent line connection. The vent line acts as the connecting link to the pressure balancing system between the core and blanket pressurizer. It also serves as a letdown line for radiolytic gases through its connection to a vent valve and the dump tanks. The dry pipe has holes along the bottom to return any entrained liquid, and also a slot in its end plate to prevent the formation of a gas pocket behind it.

The diaphragm separates the steam generator from the main surge chamber of the pressurizer. A single dry pipe and vent line is used for the entire unit. The amount of fluid carryover through this pipe is minimized by a baffle plate reaching the center line of the dry pipe, and by separate drain holes on the fuel side. A splash plate is provided to reduce the amount of entrained liquid which might reach the dry pipe in case of a rapid fuel volume surge.

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A single pipe connects the pressurizer surge chamber to the main core loop. A liquid level controller maintains the liquid level in this pipe at a point about ten inches below the inside diameter of the pressurizer drum. A thermocouple well located just above this point is required for initial adjustment of the controller. Another thermocouple well placed in the main surge chamber serves as a backup for the liquid level controller and also as an indicator of non-condensable gas accumulation. If a slow level rise should occur due to a malfunction of the controller, it will be detected as a drop in the thermocouple temperature as soon as the lower temperature core fluid reaches it. The slow accumulation of non-condensable gases is detected by a slow drop in the thermocouple temperature due to a drop in the required partial pressure of the water vapor.

The design of this pressurizer was reviewed for conformance with the ASME Code for Unified Pressure Vessels. Some necessary reinforcement was added to the connecting pipe from the pressurizer to the main core loop.

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HRT Pressurizer
Fig. 4

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HRT Pressurizer Nomenclature

1. Thermocouple Weld
2. Splash Plate
3. Dry Pipe
4. Diaphragm
5. Steam Baffle
6. Surge Pipe
7. Drain Pipe
8. End Plate
9. Downcomer
10. Lower Header
11. Heater
12. Heater Leg

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Figure 2

HRT Pressurizer Flowsheet

DESIGN SPECIFICATIONS

Installed Heater Capacity	32 kw
Purge Water Flow Rate	109 lb/hr
Steady State Vaporization Rate	40.5 lb/hr
Volumes	
Pressurizer drum	
Metal (inside)	4.8 liters
Water storage	20.9 liters
Void space	74.1 liters
Heaters, downcomers and lower header	12.4 liters
Pressurizer drum void to center line	27.0 liters
Temperatures and Heat Fluxes	
(a) Steady State (12.4 kw)	
Heat load	
Feedwater preheating	4.4 kw
Direct heater losses	1.4 kw
Pressurizer heat losses	6.6 kw
Inward heat flux at pipe surface	6,500 B/hr ft ²
Inward heat flux at inside wall	9,200 B/hr ft ²
Temperature distribution	
Bulk fluid	636 °F
Inside wall	649 °F
Outside wall	664 °F
Overall heat transfer coefficient	33 B/hr ft ² °F
(Aluminum at 835 °F)	

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TABLE I (cont'd.)

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(b) Maximum Theoretical (30.4 kw)

Inward heat flux at pipe surface	17,900 B/hr ft ²
Inward heat flux at inside wall	25,400 B/hr ft ²
Temperature distribution	
Bulk fluid	636 °F
Inside wall	662 °F
Outside wall	710 °F
Overall heat transfer coefficient	90 B/hr ft ² °F
(Aluminum at 835 °F)	

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IV. APPENDICES

A. Heat Load Calculations

These design calculations are based, in part, on modifications of work done by F. C. Zapp.⁽¹⁾

A heat transfer coefficient for losses by radiation can be expressed as:⁽²⁾

(Assuming the hot enclosed body is small in area as compared to the cooler enclosure and is convex throughout, and the enclosure is concave-throughout)

$$h_r = \frac{0.173 \epsilon \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]}{T_1 - T_2}$$

h_r = radiation heat transfer coefficient, B/hr ft² °F

ϵ = emissivity of the hot surface

T_1 = temperature of the hot surface, °R

T_2 = temperature of the enclosure (ambient), °R

A heat transfer coefficient for losses by natural convection from a hot horizontal pipe to cooler air at ordinary temperature and atmospheric pressure can be expressed as:⁽³⁾

$$h_c = 0.5 \left[\frac{T_1 - T_2}{d_o} \right]^{1/4}$$

h_c = convection heat transfer coefficient, B/hr ft² °F

d_o = outside pipe diameter, inches

T_1, T_2 = defined previously

A correct value for the constant in this equation is difficult to evaluate due to density, viscosity, conductivity, and air velocity corrections which must be made to reflect expected reactor cell conditions of 7.5 psia, 150°F, and

estimated forced convection air currents of about 75 fpm due to the space cooler fans. Although the true cell conditions are not accurately known, it is apparent that the increased heat removal due to forced convection is tending to balance the decrease due to subatmospheric pressure. In view of the desired accuracy and unknowns, and the relative importance of convective cooling, it was concluded that the basic equation, without alteration, is sufficiently accurate for this design calculation. A constant value of 0.4 was used by Zapp in evaluating similar losses on the basis of a reference to earlier work by McAdams.

The overall heat transfer coefficient, $h(\text{B/hr ft}^2 \text{ } ^\circ\text{F})$, for losses from a hot horizontal pipe in the reactor cell due to combined convection and radiation can now be expressed as:

$$h = h_c + h_r = 0.5 \left[\frac{T_1 - T_2}{d_o} \right]^{1/4} + \frac{0.173 \epsilon \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right]}{T_1 - T_2}$$

McAdams ⁽⁴⁾ gives a value of 0.52 and 0.65 for the emissivity of 347 S.S. at 450^oF and 1650^oF respectively. To be conservative, an emissivity of 0.7 is assumed for all temperatures in the range 400-900^oF.

The total heat loss, $Q(\text{B/hr})$, from the system is:

$$Q = h A(T_1 - T_2)$$

Where A = area of hot surface, ft^2

The ambient temperature of the reactor cell is 150^oF (610^oR). The pressurizer equipment is assumed to be at 636^oF (1096^oR). An insulation efficiency of 90% is assumed. Spiwak ⁽⁵⁾ indicates that the HRT type aluminum foil insulation has an efficiency of approximately 94%. Heat loss by condensing at the liquid-vapor interface of the fuel solution is assumed to be zero since

the purge water flow should keep this interface at saturation temperature without condensing.

The convective heat transfer coefficient for a long vertical pipe is the same as for a horizontal pipe,⁽⁶⁾ so the same basic equation can be used for most of the heat loss calculations. Intricate shapes, such as flanges, are reduced to cylinders having the same exposed surface area, but with an equivalent diameter, d_{eq} (inches), calculated on the basis of maintaining the same overall length.

In addition to the heat losses by convection and radiation to the reactor cell, the heaters on the pressurizer must supply energy for preheating the incoming purge water to the saturation temperature. The magnitude of this load is dependent upon the temperature of the purge water as it leaves the letdown heat exchanger. This temperature is estimated to be 280°C (536°F) when the reactor is at full power. The preheating load is calculated on the basis of compressed light water since the required enthalpy values are more readily available. A 12-gph pump is assumed.

$$Q_p = W(h_2 - h_1)$$

$$Q_p = \text{water preheating load, B/hr}$$

$$W = \text{mass flow rate, lb/hr}$$

$$h_2 = \text{enthalpy of saturated water at } 636^{\circ}\text{F}$$

$$h_1 = \text{enthalpy of compressed water at } 536^{\circ}\text{F}$$

$$Q_p = 109(672 - 533)$$

$$= 15,180 \text{ B/hr} \quad \text{or } 4.44 \text{ kw}$$

Thus, each pressurizer heater must supply 1.1 kw of heat for feedwater preheating.

The total heat load for each pressurizer and heater can now be estimated.

CALCULATIONS

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	T °F	do in.	deq in.	A ft ²	hc B/hr ft ² °F	hr B/hr ft ² °F	Q Uninsulated B/hr	Q Insulated kw
Pressurizer drum (1)	636	12.75		24.	1.25	3.25	52,500	1.54
Heater legs (4) (Part not covered by heaters)	636	2.375		1.70	1.90	3.25	4,250	0.13
Lower header (1)	636	4.00		4.20	1.65	3.25	10,000	0.30
Downcomers (2)	636	1.315		2.75	2.20	3.25	7,290	0.21
Heater (direct loss) (1)	835	4.25		2.60	1.78	4.62	11,400	0.34
2" flange (set)	636		8.8	2.88	1.35	3.25	6,450	0.19
Controller (1)								
Insulated part	636		9.1	5.50	1.34	3.25	12,300	0.36
Uninsulated part	636	3.5		0.23	1.75	3.25	556	(0.16)
Uninsulated part	400		8.0	2.38	1.22	1.98	1,905	(0.56)
Interconnecting pipe (1/2)	636	2.375		3.11	1.9	3.25	7,800	0.23
Allowance for pressure balancing valve and rupture disc flange (1/2)	636							0.50

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		<u>per pressurizer</u>	<u>per heater</u>
Direct heater losses to cell		1.4 kw	.35 kw
Feedwater preheating		4.4	1.10
Pressurizer heat losses			
Drum	1.6 kw		
Legs, downcomers, headers	0.7		
2" flange (set)	0.2		
Liquid level controller (2)	2.2		
Liquid level indicator (1)	1.1		
Interconnecting pipe (1/2)	0.3		
Pressure balancing equip. (1/2)	0.5	6.6	1.65
Total:		12.4 kw	3.1 kw

B. Heat Flux and Temperature Distribution in Heater Legs.

Each clamshell type heater on the pressurizer must supply, at steady state, a net inward heat flux equivalent to 1.1 Kw for feedwater preheating and 1.65 Kw for other system heat losses. The length of pipe covered by the heater is 28 inches (1.45 ft²). The resulting net inward heat flux at the pipe surface is 6500 B/hr ft². The heat flux from the inside wall (1.02 ft²) to the boiling water is 9,200 B/hr ft².

Work reported by McAdams⁽⁷⁾ indicates that a boiling film coefficient of about 1,000 B/hr ft² °F. can be expected. On this basis the film drop is 10 °F. Using a stainless steel conductivity of 11.5 B/hr ft °F, and a calculated log mean pipe area of 1.24 ft², the temperature drop across the stainless steel wall is about 18 °F. Therefore, at steady state;

$$\begin{aligned}T_{\text{bulk fluid}} &= 636^{\circ}\text{F} \\T_{\text{inside wall}} &= 646^{\circ}\text{F} \\T_{\text{outside wall}} &= 664^{\circ}\text{F}\end{aligned}$$

Between the outside wall of the heater leg and the clamshell heater a finite temperature drop occurs due to the lack of a perfect bond. The magnitude of this drop depends upon such factors as trueness of heater bore and pipe diameter, differences of thermal expansion, and axial trueness of the pipe. The overall heat transfer coefficient (based on pipe surface area) required for steady state operation, assuming a maximum allowable heater temperature of 835 °F, is 33 B/hr ft² °F. This allows a uniform temperature drop of 171 °F across the heater to pipe interface. Coefficients of this order are now being obtained on the HRT mock up loop.

The problem of heater design for this pressurizer, however, is that of the transient rather than steady state. In order to bring the reactor up to pressure rapidly (or recover a pressure loss) a large heater capacity is required in the pressurizer. With this objective in mind, the pressurizer heaters were designed with an installed capacity of 8 Kw each. Of this, a maximum of 7.6 Kw is theoretically available as inward heat. If the same assumptions are made as outlined in the previous temperature calculations, a new temperature distribution is established for the maximum theoretical inward heat flux of $17,900 \text{ B/hr ft}^2$ at the pipe wall:

$$T_{\text{bulk fluid}} = 636^{\circ}\text{F}$$

$$T_{\text{inside wall}} = 662^{\circ}\text{F}$$

$$T_{\text{outside wall}} = 710^{\circ}\text{F}$$

The allowable uniform temperature drop across the heater to pipe interface now becomes 125°F , and the required overall heat transfer coefficient, based on the pipe surface area and a maximum heater temperature of 835°F , becomes $90 \text{ B/hr ft}^2 ^{\circ}\text{F}$. This is somewhat higher than has been obtainable on the HRT mock up. Earlier work of Hafford and Spiewak⁽⁸⁾, however, indicates that this coefficient should be attainable. Some thought might be given to using higher heater temperatures if the interface temperature drop cannot be reduced.

It should be pointed out that during reactor startup heat load on the pressurizer is considerably larger due to an additional water preheating requirements. The letdown heat exchanger remains essentially unheated until the reactor fuel is brought up to temperature and circulation begins. The resulting added load is nearly 13 Kw. It might be reduced by intermittent purge pump operation, if the resulting low purge flow to the core connecting

pipe can be tolerated. This mode of operation should be investigated further.

C. Steady State Vaporization Rate

A flow rate of 12 gph is assumed for the purge pump. This is equivalent to approximately 109 lb/hr of heavy water at the ambient cell temperature of 150°F.

The steam generator must produce steam at a rate equivalent to the heat losses from the system exclusive of the direct heater losses, purge water preheating, and losses from areas filled with water. It is assumed that the liquid level controllers and indicator heat losses are supplied only by condensing steam.

Steam heat losses:

Pressurizer drum (~ 80%)	1.2 kw
2" flange (set)	0.2
Liquid level controllers (2)	2.2
Liquid level indicator (1)	1.1
Interconnecting pipe (1/2)	0.3
Allowance for pressure balancing valve and rupture disc flanges (1/2)	0.5
	<hr/>
	5.5 kw

This heat loss requires a steam rate of about 40.5 lb/hr (assuming light water heat of condensation).

Since all steam condensate is assumed to be recoverable by the system (exclusive of venting or leakage), the purge rate to the core connecting pipe during steady state operation will be 109 lb/hr.

D. Surge Volumes

The final pressurizer design contains the following volumes:

Pressurizer drum

Metal (inside)	4.8 liters
Water storage	20.9
Void	<u>74.1</u>
	99.8
Heater legs, downcomers and lower header	12.4
Pressurizer drum void to center line	27
Pressurizer drum void to (center line - 1.25 in.)	20

If no letdown is assumed (a worst condition), automatic dump is initiated when the core pressure reaches 2800 psi. This pressure results from an adiabatic steam volume reduction of 20 liters. For this condition, the fuel solution rises about 38 inches into the pressurizer (taking no credit for possible surge volume in core connecting pipe). In order to protect the steam generator from flooding it is essential that the water level be above the fuel level. A practical design clearance of 1.25 inches was selected.

E. Water Reserve Requirements

In order to operate for a period of at least one hour after purge pump failure, the following water reserve volume is required:

Heat losses with unrecoverable condensate:

Liquid level controllers (2)	2.2 kw
Pressurizer drum not over storage pool area (1/2)	<u>.8</u>
	3.0 kw

This represents a water loss of 22 lb/hr (assuming light water heat of condensation).

Using a value of 43.4 lb/ft³ for the density of heavy water at 636°F, the required minimum storage volume becomes approximately 0.5 ft³. In view of the uncertainty as to the amount of insulation allowable on the liquid level controller, the inherent unknowns in the original heat transfer equations, and the possibility of vent valve steam leakage, a 50% safety factor was applied to this figure in order to assure continued operation for at least one hour after pump failure. A design storage volume of 0.75 ft³ was therefore selected. This larger storage volume will prove advantageous under startup conditions if intermittent operation of the purge pump is utilized.

V. REFERENCES

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