ANALYSES OF
STEEL LINERS ON
CONCRETE STRUCTURES

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June, 1975

MASTER

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Prepared for the U.S. Energy Research and Development
Administration under Contract No. AT(45-1)-2170

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PRELIMINARY REPORT

This report contains information of a preliminary nature prepared in the course of work under Energy Research and Development Administration Contract AT(45-1)2170. This information is subject to correction or modification upon the collection and evaluation of additional data.

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POST-ACCIDENT HEAT REMOVAL STRUCTURAL EFFECTS ANALYSIS

1. Introduction

As a means of investigating the passive containment potential of the reactor containment building, loss of emergency core cooling following an HCDA was postulated. This hypothetical circumstance would lead to core meltdown and eventual penetration of the reactor vessel by molten fuel. Mechanistically following this approach, the molten fuel would also melt through the bottom of the guard vessel and the carbon-steel cover of the lower cavity, and come to rest at the bottom of the lower cavity. The liquid sodium would then flow out of the reactor vessel, fill the lower cavity, and rise to an approximate elevation of 500 ft in the reactor cavity as shown in Figure 1. It is further postulated that the fuel would continue to generate heat and transfer this decay heat to the molten sodium. The temperature of the molten sodium would rise to its boiling point of 1618°F and the pool of sodium would boil continuously, carrying away the decay heat generated by the fuel.

The condition of boiling sodium would persist for an indefinite period of time and the temperatures of the inner faces of the reactor cavity wall, the lower cavity wall, and the support ledge would all rise to approximately 1600°F as shown by the calculations for cases 404 and 405. The temperatures of the high-temperature liner and the low-temperature liner would also rise to 1600°F. These temperatures would be sustained for a long period of time as shown by the computer analyses. The pressure and temperature of the containment building would also rise to values greater than those of ambient conditions.

The purpose of this study is to determine the effects of the irradiation and the sustained high temperatures on the reactor cavity walls, the liners, the support ledge, and the containment building. The effects on the different components of their prolonged subjection to the high temperatures will depend primarily upon the properties of the affected materials at the elevated temperatures.

III. Materials

There are a number of materials whose properties must be known before an accurate analysis of the effects of the hypothesized sodium spill can be made. These include the concrete, the reinforcing steel, the steel of the high-temperature liner, the steel of the I-beam columns, and the steel of which the guard vessel is made. All of these will be subjected to temperatures ranging up to 1600°F. The temperature of the high-temperature liner may rise to 1680°F for a short period of time. In addition, the properties of the steel of which the containment building is made and of the plastics from which the penetration seals are made must be known for their operating temperatures (up to 350-400°F).

III-A. Summary of Material Properties

A summary of the properties of the materials used in the reactor cavity and the containment building is given below.

III-A.1 Steels The different types of steel used and the properties used in this analysis are tabulated below.

<u>Steel</u>	<u>Area of Use</u>	<u>Yield Strength, psi</u>		
		<u>Room Temp.</u>	<u>390°F</u>	<u>1600°F</u>
ASTM A36	Plate and structural shapes	36,000	31,000	790
ASTM A516 Grade 55	Low-temperature liner	30,000		660
ASTM A516 Grade 60	Containment vessel	32,000	28,000	
ASTM A516 Grade 70	High-temperature liner	36,000		790
ASTM A615 Grade 60	Reinforcing bar	60,000		1,320
Type 304 SST	Guard vessel	30,000		7,300

<u>Steel</u>	<u>UTS, psi</u>		<u>Stress for Rupture at 1600°F, psi*</u>	
	<u>RT</u>	<u>1600°F*</u>	<u>1000 hours</u>	<u>10,000 hours</u>
A36	58,000	4,180	190	100
A516-55	55,000	3,960	180	100
A516-70	70,000	5,040	230	120
A615-60	90,000	6,480	290	160
304 SST	70,000	13,700	2,100	1,250

III-A.2 Concrete Two types of concrete are used in the area of interest in this study. A magnetite-aggregate concrete with a specific weight of 220 lb/ft^3 is used in the reactor cavity walls above 479'-0" elevation and in the support ledge. Concrete using ordinary aggregate is used below 479'-0" elevation. See Figure 1. The properties used in this study are given below.

Magnetite-aggregate, portland-cement concrete used in reactor cavity wall

$$w = 220 \text{ lb/ft}^3$$

Compressive strength at room temperature after 28 days	7300 psi
" " " " "	90 days 8000 psi
" " " 1600°F	800 psi

Ordinary Concrete, $w = 149 \text{ lb/ft}^3$

Compressive strength at room temperature after 28 days	5970 psi
Compressive strength after being heated to 1000°F	410 psi

III. B Strength of Steel at Soaking Temperature

The high-temperature liner is made of ASTM A516 Grade 70 steel. This is a medium carbon steel which contains, typically, 0.30%C, 0.85-1.20% Mn and 0.15-0.30% Si. The steel will possess, at 100°F, a minimum yield strength of 38,000 psi and a minimum ultimate tensile strength of 70,000 psi. The steel is not intended for use at high temperatures, over 800°F, and data above that temperature are very sparse. However, from data available for other low- and medium-carbon steels it is possible to plot curves of the ratio of the yield strength at temperature to the yield strength at room temperature vs temperature. The ratio of ultimate tensile strength at temperature to ultimate tensile strength at room temperature can also be plotted vs temperature.

The plots of these ratios vs temperature are repeated here as Figures 2 and 3. A plot of typical total elongation plotted vs temperature is shown as Figure 4.

The values of strength and ductility for A516 Grade 70 steel at 1600°F were assumed to equal the following:

UTS at 1600°F = $0.072 \times \text{UTS at RT} = 0.072 \times 70,000 = 5,040 \text{ psi}$

Yield Strength at 1600°F = $0.022 \times \text{YS at RT} = 0.022 \times 38,000 = 840 \text{ psi}$

Total Elongation at 1600°F > 85%

The reinforcing steel used in the concrete is ASTM A615 Grade 60 steel. It has a nominal composition of 0.40% C and 1.25% Mn. It has a minimum ultimate tensile strength of 90,000 psi and a minimum yield strength of 60,000 psi at room temperature. This steel also fits into the category of a medium carbon steel. The ultimate tensile strength, yield strength, and ductility can be estimated in the same manner as was done for the A516 steel.

This gives the following values for a 1600°F temperature:

UTS = $0.072 \times 90,000 = 6,480 \text{ psi}$

Yield strength = $0.022 \times 60,000 = 1,320 \text{ psi}$

In addition, the creep strength of A615 steel was estimated at 1600°F

The estimated stress values, based on extrapolations from lower temperature data for medium-carbon steel are as follows:

Stress for rupture in 1,000 hours at 1600°F = 290 psi
Stress for rupture in 10,000 hours at 1600°F = 160 psi
Stress for 1% creep in 10,000 hours at 1600°F < 100 psi.

The low-temperature liner is made of ASTM A516 Grade 55 steel. This is similar to ASTM A516 Grade 70 steel, but has a slightly lower carbon and manganese content. The ultimate tensile strength will range between 55,000 and 65,000 psi and the minimum yield strength will equal 30,000 psi at room temperature. The yield strength at 1600°F will equal approximately 660 psi, see Figure 3. The total elongation is 35% or better at room temperature.

III. C Strength of Concrete at Soaking Temperatures

The concrete used in the reactor cavity wall is a high-density portland cement concrete made with magnetite aggregate, maximum size 3/4-inch. The proportion of cement used is 6 to 7 sacks per cubic yard. The W/C ratio was about 0.46. The specific weight was approximately 222 lb/ft³. Tests showed a 28-day compressive strength of 7290 psi and a 90-day compressive strength of 8130 psi. Other tests, Reference 3, showed a 90-day compressive strength as high as 9210 psi.

A search of the literature revealed no information on the properties of this type of concrete at 1600°F. Some information was obtained regarding the properties of other types of concrete at elevated temperatures, and a very small amount was found regarding the properties of magnetite concrete at temperatures up to 1112°F. Much of the data reported by different investigators is pertinent to the problem at hand.

Results of tests of portland cement concrete using 4.4 to 7.8 sacks of cement per cubic yard reported by Philleo⁽²⁾ showed a definite reduction in the modulus of elasticity of concrete with an increase in temperature. A plot of his data is shown in Figure 5. The data indicate a lower reduction in E with air-dried concrete than with moist-cured concrete. A slightly lower reduction in E for 28-day cured concrete as opposed to 90-day cured concrete was also evident. Extrapolation of his data, Figure 5, indicates that the modulus of elasticity at 1600°F for the concrete he tested would have ranged from 25% to 35% of the modulus at 75°F.

Ohgishi et al (9) concluded from their series of tests that there is no explicit difference between the modulus of elasticity of magnetite aggregate concrete and that of ordinary concrete.

Davis (11) reports a reduction in the modulus of elasticity for shielding concrete made with hydrous-iron aggregates to 69% at 200°C (392°F) and to 45% at 350°C (662°F). The value of E at the higher temperature is somewhat lower than was reported by Philleo, see Figure 4. Davis shows an even greater reduction in the modulus of elasticity with increasing temperature for ordinary concrete, to 28% at 350°C.

Harada et al (7) show a reduction in modulus of elasticity at 700°C (1292°F) and to 13-19% for river gravel-aggregate concrete, and to 45-47% for expanded shale-aggregate concrete.

The data obtained from different tests by different investigators vary widely and this variation has been noted by a number of people. The general consensus seems to be that the results will vary for different mixes, different cements, and different conditions. It is necessary to test the particular concrete to be used under the exact conditions of use to obtain accurate data. Another factor of importance is whether the concrete is tested at temperature or after it has cooled, and also whether the concrete is heated in the stressed condition. Consideration of these factors led to the conservative estimate of an 80% reduction in the modulus of elasticity for the concrete at 1600°F.

Ohgishi et al (9) found in their experiments that the modulus of elasticity for heated concrete dropped a greater amount when it was measured after the concrete had cooled than if it were measured while the concrete was still hot. Specifically, for concrete heated to 600°C (1112°F), the modulus of elasticity was found to equal 39% of the room temperature value when the measurement was made using the hot concrete. When the concrete was allowed to cool the measured modulus of elasticity equaled 28%.

There is quite a variation in the information reported in the literature on the effect of high temperature on the strength of concrete. The amount of the reduction in the strength at elevated

temperatures is a function of many variables and tests must be conducted with the particular concrete at the actual operating conditions to obtain an accurate value of the strength at temperature.

Davis reports that magnetite is quite stable at a high temperature and holds up at least as well as ordinary aggregate. He also points out that the coefficient of thermal expansion of the aggregate may be quite different from that of the cement paste and that the difference may cause cracking when the concrete cools.

Pihlajavaara (6) notes that the temperature at which the concrete is tested affects the value of the strength obtained. He found that the compressive strength retained ranged from 8% to 33% at 800°C (1472°F). He concluded, "The generalization of results obtained in a particular temperature test on concrete must be done very carefully or, very often, must be avoided. A statement which is true about the properties of a particular specimen, or a class of specimens, of cement paste or concrete is liable to be untrue if applied generally.."

Harada et al (7) reported a reduction in compressive strength to 60% of the original room temperature strength for concrete heated to 450°C (842°F). For concrete made with river gravel aggregate they showed a drop to 24-26% of original strength after heating to 700°C (1292°F). For concrete with expanded shale aggregate the compressive strength reduced to 64-73% of the original strength. In both cases the lower strength was for a concrete with a higher W/C ratio.

Weigler and Fischer (8) reported that the reduction of strength of concrete due to its exposure to a high temperature is reduced if the concrete is loaded in compression during the time it is heated. The optimal load they reported as being between 1/3 and 1/2 of the normal compressive strength. Their investigation showed the following results for portland cement concretes made with quartz and baryta aggregates heated to 600°C (1112°F) (age 197 days):

<u>Aggregate</u>	<u>Load During Heating</u>	<u>Retention of Normal Comp. Strength</u>
Quartz	None	42%
Quartz	1/3 normal comp. strength	53%
Baryta	None	58%
Baryta	1/3 normal comp. strength	117%

Ohgishi et al (9) report a greater retention of strength for heated concrete made with both serpentine and magnetite aggregates when the strength is measured at the high temperature--before the concrete has cooled. Specifically their test samples showed that dense concrete retained 82% of its compressive strength when it was heated to 500°C (932°F). After cooling from 500°C the retained strength equaled 62% of the original, room temperature strength. Dense concrete first heated to 600°C (1112°F) and then cooled, retained 41% of the original strength.

Peterson (5) stated that concrete subjected to a sustained temperature of 1200 to 1500°F can usually be taken apart by one's fingers.

Polivka (4) states that if concrete is made with heat-resisting aggregate it may retain 25 to 50% of its strength at temperatures as high as 1300 or 1400°F. He also states that if the concrete is subjected to wide and frequent fluctuations in temperature, the loss of strength of the concrete may be several times as great as that observed for concrete exposed to a constant high temperature.

Akroyd (10) states that when quartz is heated to 573°C (1063°F) the structure changes with a marked increase in volume which leads to spalling. He also states that when the temperature of concrete reaches 500-600°C (932-1112°F) the residual strength of the concrete is less than half of the initial strength.

Tests by Zoldner (3) showed that the strength retained by concrete at high temperatures is very markedly affected by the type of aggregate. Tests of different types of aggregate showed the following results:

<u>Test Temperature</u>	<u>Percent of Original Compressive Strength Retained</u>			
	<u>Gravel</u>	<u>Sandstone</u>	<u>Limestone</u>	<u>Expanded Slag</u>
700°C (1292°F)	26	30	42	49
800°C (1472°F)	15	21	24	30

Davis notes that thermal cycling has a substantial weakening effect on portland cement concrete, especially on the tensile strength and the modulus of elasticity. The greatest amount of deterioration occurs after one or two cycles and it increases somewhat for more cycles up to 20 or 30. The first thermal cycle can cause a large percentage of the damage.

A number of conclusions regarding concrete pertinent to this structural effects analysis can be drawn from the foregoing literature survey. These include the following:

- The proportion of the original strength of the concrete which is retained at an elevated temperature is increased if the concrete is loaded in compression during the heating.
- The proportion of the original strength of the concrete which is retained after the concrete is raised to a higher temperature is greater if the measurement is made at the elevated temperature. A considerable amount of strength may be lost during the cooling.
- The proportion of the original modulus of elasticity of the concrete which is retained after the concrete is raised to a higher temperature is greater if the measurement is made at the elevated temperature.
- A long-time soak at a high temperature may cause further deterioration in the strength of the concrete.

Because of the paucity of information available concerning magnetite aggregate concrete, and the divergences of the opinions expressed in the literature regarding the effects of exposure to high temperature, a series of tests was conducted to check the effects of high-temperature exposure. Four sample portland-cement concrete cylinders were heated. The cylinders were then cooled and tested for compressive strength. The concrete cylinders were six inches in diameter by 12 inches high. The four cylinders had the following characteristics:

<u>Cylinder No.</u>	<u>Type of Aggregate</u>	<u>Age (days)</u>	<u>Lab. Test Strength, psi</u>
1	Natural	28	5970
2	Magnetite	28	7200
3	Magnetite	91	8130
4	Magnetite	91	8130

The cylinders were heated as noted below and then cooled. Numbers 1, 2, and 3 were heated in air and cooled in air. Number 4 was heated and cooled in a nitrogen atmosphere. All were cooled slowly in the furnace. The power was turned off and furnace and contents were allowed to cool naturally. Cooling required approximately 30 hours.

<u>Cylinder No.</u>	<u>Test No.*</u>	<u>Temp, °F</u>	<u>Time at Temp. hours</u>	<u>Atmosphere</u>	<u>Compressive Strength</u>
1	1	1000	24	Air	410
2	1	1000	24	Air	5230
3	2	1600	28	Air	1100
4	3	1600	168	N ₂	850

* cylinders nos. 1 and 2 were heated together

All sample cylinders were tested to failure to determine the compressive strength. Cylinder no. 1 (ordinary concrete) was badly cracked before testing and it crumbled during failure. The grout showed very little strength and there was no shearing of the aggregate. Spalling and cracking were also noted in the other three cylinders as a result of the heating. However, more strength was retained by the grout in these three magnetite-aggregate cylinders and there was a shearing of the aggregate in each of these three cylinders.

A handful of aggregate was heated in each test. The aggregate heated in air turned in color from gray-black to a reddish-brown. Pieces which were broken showed the same color change internally. The change in color indicates an oxidation from Fe_3O_4 to Fe_2O_3 . There was noticeably less color change in the aggregate heated in test no. 4. However, the aggregate in the concrete samples which were broken in testing all showed similar oxidation of the aggregate over the entire sheared-surface area.

From the test data, coupled with published information, it was assumed that the magnetite concrete would retain 10% of its compressive strength at 1600°F and that it would retain this amount of strength over a long period of time. It is believed that this is a conservative estimate since it gives no credit for the two following important factors which reports in the literature indicate serve to increase the strength of concrete which has been subjected to heating:

1. The concrete will be stressed in compression during heating, and
2. The stress condition will occur at the high temperature, i.e., the concrete will not be cooled after heating and before loading.

IV. Effect of Hot Sodium Soak on the Structural Integrity of the Reactor Cavity Walls and Liners and on the Vessel Support Ledge

It is hypothesized that the sodium would fill the reactor cavity to an elevation of 500 ft. The fuel would transfer heat to the sodium and cause it to boil. It is postulated that the transfer of heat from the fuel and the resulting boiling would continue for 1500 hours or more. The temperature within the reactor cavity would thus be maintained at 1618°F over an extended period of time. Thus it is necessary to analyze the effects of a long-term exposure of the reactor cavity materials to the high temperature and to a saturated sodium atmosphere at 1600°F.

IV-A Chemical Effect of Hot Liquid Sodium Soak on Steel Liners

The information which is available in the literature regarding the solubility of iron in sodium is not definitive. Different results reported from different tests by different testers indicate a solubility of iron in sodium at 1300°F ranging from a few tenths of a part per million to 10 ppm. A reasonable extrapolative estimate indicates a maximum solubility of perhaps 100 ppm of iron in sodium at 1600°F.

A corrosion rate of iron subjected to 1600°F sodium at a velocity of 20-30 ft/sec would probably equal 5 mils per year. Since the minimum thickness of the liners containing the liquid sodium is 1/4 inch, the rate of corrosion is quite safe. Moreover, the sodium pool will be essentially stagnant. The only velocity will be due to boiling and convective motion, so the velocity of the liquid in contact with the steel will be much lower than 20 ft/sec. The rate of corrosion will thus be much lower than 5 mils per year.

The rate of corrosion can also be checked against the solubility value of 100 ppm of iron in sodium. The liquid sodium will rise approximately 10 ft high in the reactor containment building. The ID of the lower portion of the containment building is 16.25 ft. The sodium will also fill the lower cavity. The dimensions of the lower cavity are, roughly, 20 ft diam. by 13 ft high. This represents a volume of sodium of 12,370 ft³ and an area of steel liner in contact with the sodium equal to 2670 ft². The amount of iron which could be held in solution at 1600°F would equal 12,370 x 46.2 x 100 ppm = 57.15 lb or 0.117 ft³. Assuming uniform erosion over the surface of

liner steel in contact with the sodium gives a reduction in the thickness of the liner equal to $(0.117/2670) \times 12 = 0.00053$ in.

There are two other possible sources of difficulty--dissolved oxygen and slag inclusions in the welds. Sodium carries oxygen which oxidizes the metals in contact with the sodium. The metallic oxides then come off the metal and are carried away by the sodium. Any excessive amount of dissolved oxygen would result in a concomitant increase in the amount of corrosion of the steel liner.

A much greater danger potential would result from slag inclusions in the welded joints. A slag inclusion would probably be either a sulphide or a silicate. Both of these are readily dissolved by high-temperature liquid sodium. Such dissolution would result in pits and areas of weakness in the welds. If the inclusion ran all the way through the weld, the result would be a hole through the weld and a leak of sodium through the protective liner.

IV-B Ability of Lower Reactor Cavity Wall to Withstand Cavity Pressure Plus Static Pressure Due to Head of Sodium

The pressure in the reactor cavity will reach a maximum of 5.46 psi as determined by the calculation of pressure vs. time. This maximum pressure will occur at time $T = 198$ hours. At 198 hours the temperature near the O.D. of the reactor cavity will equal 163°F for Case No. 404, or 343°F for Case No. 405. Of the two cases considered--Case No. 404 in which the reactor building cooling system remains operative and maintains a 120°F heat sink on the far side of the reactor cavity wall, and Case No. 405 in which the far side of the reactor cavity wall is considered to be insulated--Case No. 405 presents the most extreme conditions for resistance to internal cavity pressure.

If the surface of the liquid sodium rises to an elevation of 500 ft, there would be a static head pressure at elevation 489 ft of 11.0 ft of sodium, see Figure 1. At 1618°F this would equal $46.2/144 \times 11.0 = 3.53$ psi (sp. wt. of sodium = 46.2 lb/ft^3 at 1618°F). The total maximum pressure acting outward on the reactor cavity wall at elevation 489 ft would then equal $5.46 + 3.53 = 8.99$ psi. The cavity pressure and total pressure (cavity pressure plus head of liquid) will vary with time as shown in Figure 6. The variation of the temperature in the outer portion of the cavity wall with time is also shown in the same figure.

The internal pressure will act outward against the liner. The liner temperature would be in excess of 1600°F and would have little tensile strength. However, the liner is backed up by the concrete which would absorb the outward force and prevent the liner from expanding. Even if the inner layer of concrete should crumble and lose all of its strength due to the high temperature, it will still transmit the load to the outer section since it is completely contained.

The pressure load will produce a tensile force in the wall. The hoop force acting on a section of the wall would equal $p \times 195 = 195 \times p$ lb per inch of height or $2340 \times p$ lb per foot of height, see Figure 7.

The reactor cavity wall contains reinforcing steel bars which run circumferentially around the inside and outside of the wall, 3-1/4 in. in from the surface of the concrete. The rebars are welded to form continuous hoops. No. 11 size bars are used in both locations with a vertical spacing of nine inches on the outside of the wall and 12 inches at the inside. A no. 11 rebar has a cross-sectional area of 1.56 in^2 . This gives an area of steel to withstand the tensile hoop loading of 1.56 in^2 per foot of height at the inside of the reactor cavity wall and 2.08 in^2 per foot of height at the outside of the reactor cavity wall.

The rebar satisfies the requirements of ASTM A-615 Grade 60. The steel used had a nominal content of 0.40% C and 1.25% Mn. The minimum yield strength at room temperature is 60,000 psi. The minimum tensile strength at room temperature is 90,000 psi.

There are two situations which should be analyzed: 1. When the pressure is at a maximum, and 2. When the temperature of the reinforcing bars is at a maximum, i.e., when the steel is the weakest. Taking the maximum pressure case first gives--

$$\text{hoop load} = 2340 \times 8.99 = 21,036 \text{ lb/ft of height.}$$

Considering only the area of the reinforcing steel at the outer portion of the wall (figure 7) $S = 21,036/2.08 = 10,110$ psi tensile stress. The maximum temperature of the rebar at this time, as

determined by the thermal analysis (see Figure 6) would equal approximately 350°F. The steel would retain over 75% of its yield strength at this temperature, so the margin of safety would equal $(0.75 \times 60,000/10,110) - 1 = 3.5$ and the cavity wall would easily and safely withstand the pressure.

The second case, when the temperature of the reinforcing steel is at a maximum, would occur when the sodium had largely boiled away and the reactor cavity pressure had dropped to 1/2 psi or less. Assume new sodium was added and that the level of the sodium was raised to 494 ft giving a static head of 5.0 ft. The total pressure at the bottom of the reactor cavity would then equal $(5.0 \times 46.2/144) + 0.5 = 2.10$ psi.

The hoop force would equal $2340 \times 2.10 = 4910$ lb. The total area of rebar, inside plus outside, equals $2.08 + 1.56 = 3.64$ in.² per foot of height. The high-temperature liner would also act to contain the internal pressure. The liner is made of 3/8-in. plate, so it would add an area of $0.375 \times 12 = 4.50$ in.² per ft of height. The total area of steel would then equal $3.64 + 4.50 = 8.14$ in.² per ft of height. The stress in the steel would equal $4910/8.14 = 600$ psi. This compares with a yield strength of 1,320 psi and a stress for rupture in 1000 hrs. of 290 psi.

This is a considerable simplification of the actual case, see Figure 8. In the actual case, the pressure would act outward against the wall as shown in Figure 8. The force resulting from the pressure would be resisted by the wall as well as the reinforcing-steel hoops. The shear strengths of the wall and the vertical reinforcing steel will also resist the outward force acting on the wall. Both of these will add to the strength of the wall. The structure will successfully withstand the cavity pressure plus the static pressure due to the head of sodium under all hypothesized conditions with the proviso that if more sodium is added to the system, the amount added should be limited to that which would permit the liquid level to rise to an elevation of 494 feet.

IV-C Structural Effect of Hot Sodium Pool on High-Temperature and Low-Temperature Liners

After the fuel melt-through, the liquid sodium would flow out of the hole in the bottom of the vessel, fill the cavity beneath the vessel, and then rise up around the vessel in the reactor cavity. The volume of sodium is sufficient so that it would rise more than four feet above the top of the high-temperature liner. Thus, the high-temperature liner and approximately 4 feet of the low-temperature liner would be subjected to the high-temperature sodium, see Figure 1.

The temperature of the sodium when it first drained out of the vessel would equal 1100°F. Due to the influence of the heat given off by the fuel, the temperature of the sodium would later rise slowly to its boiling point of 1618°F.

It is desired to determine the effect of the hot sodium on the high-temperature liner and the low-temperature liner.

Conclusions:

The high-temperature liner and the low-temperature liner would be subjected to stresses well above the yield point when the temperature rose to 1618°F due to thermal expansion. But the steel is quite ductile at that temperature and would yield enough to relieve the thermally-induced stresses. No failure would occur from the thermally-induced stresses.

Another, much more serious, problem would arise from the heating of the concrete behind the low-temperature liner. It is probable that moisture in the amount of 5% or more of the total weight of the heated concrete would be driven out as the concrete was heated above 1000°F. The moisture would become steam and exert pressure on the low-temperature liner. Vent pipes would relieve the pressure behind the high-temperature liner, but no provision has been made to vent steam from behind the low-temperature liner. Potential pressure would be capable of causing rupture of the cylindrical portion of the low-temperature liner, the ring section above the cylindrical portion, and the structure above the ring. If the integrity of the

low-temperature liner is to be ensured, provision must be made to vent any steam that is trapped behind the different portions.

The welds will all be subjected to high thermally-induced stresses and forces. Care should be taken to ensure that all welds are full-size, well made, and structurally sound.

Discussion:

There would be essentially four steps in the sequence resulting from the flow of sodium out of the reactor vessel into the reactor cavity:

1. Hot sodium would flow out of the vessel and into the cavity. The sudden contact of hot sodium at approximately 1100°F would create a temperature differential across each liner with associated thermal stress.
2. The temperature of the two liners would rise to the temperature of the liquid sodium, say 1100°F.
3. The temperature of the sodium, and of the liners, would rise slowly to the boiling temperature of the sodium, 1613°F.
4. Steam would be generated from the moisture in the concrete backing the high-temperature and low-temperature liners as the temperature of the concrete rose.

The maximum temperature gradient across the high-temperature liner calculated by the computer analysis would occur at $t = 0.001$ hrs and would equal 61.9°F.* The average wall temperature at that instant would equal 195°F. At this temperature $\alpha = 6.4 \times 10^{-6}$ in./in.°F, $\nu = 0.27$, and $E = 27.7 \times 10^6$ psi. The thermal stress would equal

$$S = \frac{E\alpha\Delta T}{2(1-\nu)} = \frac{27.7 \times 10^6 \times 6.4 \times 10^{-6} \times 61.9}{2(1 - 0.27)} = 7500 \text{ psi.}$$

This is a safe stress, since the yield strength of the steel at this temperature would probably exceed 40,000 psi.

The other maximum danger point would occur when the temperature of the sodium and the liners rose to 1613°F. This would give a ΔT of approximately 1500°F

Approximate values of properties of A516 steel (medium carbon steel) are listed below. These are best estimates from available data and were obtained in part by extrapolation.

Coefficient of thermal expansion, α (mean value R.T. to 1100°F)		$8.35 \times 10^{-6}/^{\circ}\text{F}$
α (mean value R.T. to 1600°F)		$9.4 \times 10^{-6}/^{\circ}\text{F}$
E at 1100°F		15.5×10^6 psi
E at 1600°F		14×10^6 psi
Yield Strength: S_y at	1100°F	20,500 psi
	1300°F	8,000 psi
	1600°F ($0.22 \times S_y$ @ R.T.)	900 psi
UTS at 1100°F		43,000 psi
UTS at 1300°F		10,000 psi
UTS at 1600°F ($0.071 \times \text{UTS}$ @ R.T.)		5,000 psi
Shear ultimate at 1600°F ($0.75 \times \text{UTS}$)		3,750 psi
Elongation at Room Temp.		30%
at 1100°F		50%
at 1300°F		70%
at 1600°F		> 85%

Two grades of A516 steel, Grade 55 and Grade 70, are used for the containment system; see Figure 9. The Grade 70 steel will have a higher strength, especially at room temperature. However, the difference in the strengths of these two grades will diminish as the temperature increases. No differentiation between the properties was attempted in the tabulated data for the higher temperatures.

The high-temperature liner is surrounded by a 2-in. air space over most of its length, and there is also an air space at the bottom near the O.D.; see Figure 9.

The change in the radius of the high-temperature liner due to an increase in temperature to 1618°F would equal--

$$\Delta R = \alpha R \Delta T = 9.4 \times 10^{-6} \times 195.1 \times 1500 = 2.75 \text{ in.}$$

This is greater than the 2.0 in. air space, so the liner would be placed in compression, but only a small amount. Ignoring the yield strength and subtracting 2.0 from 2.75 gives the following:

Unit strain = $0.75/195.1 = 3.84 \times 10^{-3}$ or 0.38%. This is considerably less than the possible elongation at 1600°F, so no damage would result.

At its upper end the high-temperature liner is constrained and cannot expand radially or circumferentially. The total thermal strain must be absorbed in elastic and plastic strain. If the elastic strain is ignored--

$$\text{Strain} = \alpha \Delta T = 9.4 \times 10^{-6} \times 1500 = 0.0141 \text{ in./in.},$$

or a total strain of 1.4%. This is well below the elongation at 1600°F of 85%, so no rupture would occur. The upper end of the high-temperature liner would be swaged down 2.0 in. on the radius.

The high-temperature liner is also constrained from expanding upward at its upper end. The length of the liner is 7.0 ft. The elongation from top to bottom would equal--

$$9.4 \times 10^{-6} \times 84 \times 1500 = 1.184 \text{ in.}$$

This expansion can easily be accommodated by the 2-in. air space at the bottom of the high-temperature liner. However, the movement of the outer circumference downward relative to the inner portion would produce stress in the liner. This case can be approximated as shown in Figure 10, a circular flat plate with outer edge fixed and supported, inner edge fixed, uniform load along inner edge. Case 20, page 222, Roark, fourth edition.

$$y = \frac{3F(m^2 - 1)}{4\pi m^2 E t^3} \left[a^2 - b^2 - \frac{4a^2 b^2}{a^2 b^2} \left(\ln \frac{a}{b} \right)^2 \right]$$

$$E = 14 \times 10^6 \text{ psi}$$

$$a = 195.1$$

$$b = 183.1$$

$$t = 0.25$$

$$m = 1/\nu = 1/0.25 = 4.0$$

$$y = 1.025 \times 10^{-6} \times F (4538.4 - 4532.3) = 6.253 \times 10^{-6} \times F$$

for $y = 1.184 \text{ in.}$ $F = 189,400 \text{ lb.}$

The maximum apparent stress would occur at the inner edge--

$$S = \frac{3F}{2t^2} \left[1 - \frac{2a^2}{a^2 - b^2} \left(\ln \frac{a}{b} \right) \right] = 7.6394 \times F (1 - 1.0648)$$

$$S = 1.447 \times 10^6 (1 - 1.0648) = -93,800 \text{ psi.}$$

This is considerably above the yield strength of the thermal liner at 1600°F (900 psi), so the plate would yield. If the calculated excess stress is converted into strain--

$$\delta = S/E = 92,900/14,000,000 = 0.00664 \text{ in./in. or } 0.66\% \text{ strain.}$$

This is well below the elongation at this temperature, so no rupture would occur.

The low-temperature liner is constrained on the O.D. so it cannot move radially. The thermal strain resulting from a rise in temperature to 1618°F would equal--

$$9.4 \times 10^{-6} \times 1500 = 0.014 \text{ in./in. or } 1.4\%.$$

The low-temperature liner would compress 1.4%. The bulk of this would be plastic strain, which amount can easily be accommodated, so no danger will result.

The low-temperature liner would also expand vertically, but it is not constrained in this direction. Point "A", Figure 9, would rise--

$$9.4 \times 10^{-6} \times 30 \times 1500 = 0.423 \text{ in.}$$

This is similar to the case previously treated. This case can also be approximated as a circular flat plate with outer edge fixed and supported, inner edge fixed, uniform load along inner edge. See Figure 11.

The same equation for vertical deflection would apply.

$$E = 14 \times 10^6 \text{ psi}$$

$$a = 195.1 + 14.9 = 210.0 \text{ in.}$$

$$b = 195.1 \text{ in.}$$

$$t = 0.25$$

$$m = 1/0.25 = 4.0$$

$$y = 1.025 \times 10^{-6} \times F (6036.0 - 6025.1) = 11.173 \times 10^{-6} \times F$$

$$\text{for } y = 0.423 \text{ in.} \quad F = 37,900 \text{ lb.}$$

The maximum apparent stress would equal--

$$S = 7.6394 \times 37,900 (1 = 1.0754) = 21,800 \text{ psi.}$$

The yield strength = 900 psi, so the plastic strain would equal--

$$\delta = 20,900/14,000,000 = 0.00149 \text{ in./in. or } 0.15\% \text{ which is safe.}$$

The flat deck portion of the low-temperature liner, Figure 12, is fixed at the O.D. The ring section would tend to expand. For analysis the ring can be considered as a short length of a heavy-walled cylinder. The increase in the outer radius due to a 1500°F rise in temperature would equal--

$$\Delta R = 210.0 \times 9.4 \times 10^{-6} \times 1500 = 2.96 \text{ in.}$$

The external pressure required to produce an equivalent ΔR --

$$\Delta b = p \frac{b}{E} \left(\frac{a^2 + b^2}{b^2 - a^2} - \nu \right) \quad \nu = 0.25 \text{ at } 1600^\circ\text{F}$$

$$a = 195.1 \text{ in.} \quad b = 210.0 \text{ in.}$$

$$\Delta b = 210 p/E (13.61 - 0.25) = 2805.6 \times p/E$$

$$E = 14 \times 10^6.$$

For $\Delta b = 2.96 \text{ in.}$ --the equivalent pressure would equal--

$$2.96 \times 14 \times 10^6 / 2805.6 = 14,770 \text{ psi.}$$

The radial shear force acting on the 1/4-inch long cylinder would equal-- $0.25 \times p = 3690 \text{ lb/in.}$

The weld at the O.D. of the ring is specified as a 3/8-in. weld.

Assuming a 3/8-in. wide bead gives an apparent shear stress of--

$$S = 3690/0.375 = 9840 \text{ psi.}$$

However, the yield strength of the steel is only 900 psi at 1600°F.

The steel would be in compression radially and circumferentially. The shear stress in the weld would thus equal $900 \times 1.414 = 1273 \text{ psi}$, which is above the shear yield but well below the ultimate shear strength of 3750 psi. There would be yielding in the weld, but not failure.

The residual moisture in the concrete would become steam when the temperature of the contacting steel plates (high-temperature and low-temperature liners) rose to 1600°F and raised the temperature of the concrete to a high temperature. Tests have shown that as much as 80%

of the water used in mixing concrete can be driven off by heating the concrete to 1000°F (HEDL-TME-73-48, FFTF Secondary Sodium Fire Protection System Test F 1, R. K. Hilliard and J. M. Yatabe, April 1973).

The steam that was generated behind the high-temperature liner would push against the liner and force it away from its backing enough to permit passage of the steam to a vent pipe. It would then pass harmlessly out the vent pipe. Only slight damage would occur to the high-temperature liner from this cause, providing that the vent pipes have sufficient capacity to vent the steam without allowing the back pressure to rise to a dangerously high level.

The low-temperature liner, however, presents a different problem. It is backed by concrete over its entire area and there is no provision to vent any trapped steam. The steam would act against the liner and the deck as shown in Figure 13.

The tensile stress in the inner cylinder and on the outer weld due to the steam pressure can be determined by first calculating the portions of the pressure load held at the inside diameter and the outside diameter.

Taking moments about the inner edge (see Figure 14) gives--

$$F_2 \times 14.9 = \int s dF \quad dF = 2\pi(R_1 + s)ds \times p$$

$$F_2 = \frac{2\pi p}{14.9} \left[\int R_1 s ds + \int s^2 ds \right]_0^{14.9} = 0.4217 p (R_1 s^2/2 + s^3/3)$$

$$F_2 = 0.4217 p (195.1 \times 111.00 + 1102.65) = 9537.4 \times p$$

Taking moments about the outer edge gives--

$$F_1 \times 14.9 = \int s dF \quad dF = 2\pi(R_2 - s)ds \times p$$

$$F_1 = \frac{2\pi p}{14.9} \left[\int R_2 s ds - \int s^2 ds \right]_0^{14.9} = 0.4217 p (R_2 s^2/2 - s^3/3)$$

$$F_1 = 0.4217 p (210.0 \times 111.00 - 1102.65) = 3364.8 \times p$$

The total pressure load equals $\pi(R_2^2 - R_1^2) \times p = 18962.6 \times p$, which checks. ($F_1 + F_2 = 9364.8 + 9597.4 = 18,962.2$)

The axial tensile stress in the low-temperature liner would equal--

$$S = F_1 / (2\pi R \times t) = 9364.8 \times p / (2\pi \times 195.1 \times 0.25) = 5,000 \text{ psi UTS.}$$

So a pressure of 164 psi would stress the low-temperature liner to its ultimate strength of 5000 psi. A pressure of 30 psi would stress the liner to its yield strength of 900 psi. The pressure would also act upward on the ring section. The initial stress would be well above the yield stress, so the ring section would deform upward, held at the inner edge by the low-temperature liner and at the outer edge by the weld. At 1600°F the steel will elongate 85%. Minimum stress would occur when the flat section yielded to form a semicircular shape. This would occur with an elongation of 57%. The stress can be estimated by ignoring the radius of 192.4 inch.

$$\text{Hoop stress} = PD/2t = p \times 14.9/2 \times 0.25 = 5,000 \text{ psi UTS.}$$

A pressure of 168 psi would stress this section to its ultimate strength of 5000 psi. A pressure of 30 psi would stress this section to its yield strength of 900 psi. However, the actual pressures would be noticeably lower than these, since the steel would thin down as it elongated. Steam pressure would also act on the outside of the low-temperature liner. The internal, cavity, pressure would be negligible. The effect of the external steam pressure is discussed in section IV-C.2.

IV-C.1 Effect of High Temperature on Low-Temperature Liner Panels

The high-temperature liner is installed in such a manner that it is free to expand outward and downward, see Figure 9. The low-temperature liner, however, is welded to the embedments in sections 30 in. high by 41 in. wide, as shown in Figure 9-a. Since the temperature of the liner will rise much more rapidly than the temperature of the concrete, the liner will expand more rapidly than the concrete or the embedments. Thus the low-temperature will be constrained from expanding either vertically or horizontally.

The low-temperature liner forms a circular cylinder supported continuously on the outside, so the circumferential expansion will be absorbed uniformly around the periphery. The liner steel

is relatively ductile, total elongation of 37% at room temperature, and will be strained in compression. The total thermal elongation which will occur will be on the order of one percent, so the circumferential expansion will be absorbed uniformly around the periphery without any harmful effect. The vertical expansion will be resisted by the welds to the horizontal embedments. These welds will be stressed in shear as shown in Figure 9-b.

The low-temperature liner is made of ASTM A516 Grade 55 steel. At room temperature this steel has a minimum yield strength of 30,000 psi and a minimum ultimate strength of 55,000 psi. Both yield and ultimate strengths for a given lot of steel may be higher, but an increase in one would be matched by a corresponding increase in the other. This steel is also relatively ductile and will exhibit a total elongation of 37% or better. The ultimate shear strength of a carbon steel such as this equals approximately 70% of the ultimate tensile strength. The shear yield strength equals approximately 55% of the tensile yield.

The weld joint between plates and to the embedments is shown in Figure 9-b. In each case the plate is backed up by another plate on the other side of the weld, so the lateral force on the weld is balanced. In addition, the welds are longer than the plate is thick, so that the shear stress in the weld is less than the compressive stress in the attached plate. However, regardless of the size of the weld, the shear area cannot exceed the thickness of the plate divided by the cosine of 45° or $t \times 1.414$. The maximum shear area for the liner plates would thus equal $0.25 \times 1.414 = 0.354 \text{ in.}^2/\text{inch}$.

If the yield strength of the plate in compression is less than the shear strength of the weld on the plate, then the plate would yield before the weld would shear. For the liner steel the ultimate tensile strength is approximately twice the yield strength at room temperature. Thus the ultimate shear strength would equal 140% of the compressive yield strength. The shear strength of the welded joint divided by the compressive yield strength of the plate would equal (taking a one-inch wide section)--

$$(60,000 \times 0.70 \times 1.414)/30,000 = 1.98.$$

The same approximate safety factor of two for the strength of the welded joint compared with the compressive strength of the weld persists throughout the temperature range from room temperature to 1600°F as shown below, data from graphs in Figures 2 and 3.

Temperature °F	UTS ksi	Yield Strength ksi	<u>Strength of Welded Joint</u> <u>Compressive Strength of Plate</u>
Room temp.	60	30	1.98
200	60	29.1	2.04
400	60	26.4	2.25
600	58.2	22.4	2.57
800	48.0	17.4	2.73
1000	28.2	12.3	2.27
1200	13.8	7.2	1.90
1400	6.9	3.0	2.28
1600	3.6	0.6	5.94

It should be emphasized that this analysis assumes a weld which is solid and has a shear path which is at least 1.5 times as long as the plate is thick as is shown in Figure 9-b. If only a fillet weld is used as is shown in Figure 9-c, which shows liner plate details for some of the cells, the shear area will equal $0.707 \times t$. For this case the shear strength of the weld compared with the compressive strength of the plate would equal--

$$(60,000 \times 0.70 \times 0.707) / 30,000 = 0.990,$$

or the shear strength of the weld would be approximately equal to the compressive strength of the plate. If the thickness of the fillet weld should be less than the thickness of the plate at any point, the danger of weld failure would be increased. A thin plate or a plate which was dimpled at the midpoint would probably buckle and relieve the force acting on the weld. However, there would be little, if any, relief at the corners of the panel and failure could still be expected at those locations.

Some of the low temperature panels in the cells are attached with fillet welds. Failure of some of these welds can be expected as a result of an HCDA and subsequent sodium spill, especially

near the corners of a panel and particularly if the thickness of the base or the height of a fillet is less than the thickness of the plate.

In addition to the gross strength of the welded joint compared with the compressive strength of the plate there is also the problem of the effect of the temperature on the welded joint itself. A variation of temperature across the weld would produce a thermal stress in the amount of $E \alpha \Delta T$. The modulus of elasticity for the plate material will vary from 28×10^6 psi at room temperature to 14×10^6 psi at 1600°F , giving an average value of 21×10^6 over the temperature range.

The temperature difference between the plate and the concrete will be quite high at times. However, the gradient through a section of steel will be much smaller due to the high thermal conductivity of the steel. If a temperature gradient as high as 500°F should exist across a weld, the apparent thermal stress in the weld would equal--

$$S = 21 \times 10^6 \times 6.5 \times 10^{-6} \times 500 = 68,250 \text{ psi.}$$

The strain would equal $\alpha \Delta T = 6.5 \times 10^{-6} \times 500 = 0.00325$ or 0.325% .

The weld would yield, but the material is sufficiently ductile so that there would be no danger of failure for even this extreme assumed temperature differential.

IV-C.2 Effect of Generation of Steam from Water in Cavity Wall Concrete on Low Temperature Liner Panels

The inside of the reactor cavity wall is lined with steel plate to prevent the sodium from contacting the concrete in case of a spill. The liner above elevation 496'-0 is termed the low-temperature liner. It is made of 1/4-inch thick steel plate of ASTM A-516 grade 55. The steel plate is welded to horizontal and vertical angle irons and I-beams which are embedded in the concrete. The low-temperature liner is thus composed of a number of small panels.

If hot, liquid sodium should be spilled into the cavity, heat would pass through the liner and heat the concrete in the wall behind the liner. The water contained in the concrete would be driven out in the form of steam. If the steam were not vented it would produce pressure behind the panels. In some cases, depending upon the temperature, the pressure could rise high enough to cause rupture of the panels.

When concrete sets up it shrinks and thus would draw away slightly from the embedments. This would give a small passage for migration of the steam. However, the flow of steam through a thin passage is too slow to solve the problem of venting the steam, see Appendix 4 page 17. Moreover, the concrete would not draw away from all of the embedments uniformly and so a continuous thin passage would not be available for passage of the steam out of the trapped spaces behind the individual panels.

The driving force (pressure) behind the steam to cause it to flow through the connecting lines is limited by the allowable stress on the panels. The panels are curved to fit the inside of the cavity wall. However, the radius of curvature of 16'-3 or 18'-0 is large and so, to simplify the analysis, the panels are considered as flat plates.

The panels are made of a very ductile steel, see Figure 4. The total elongation varies from 35% at room temperature to a minimum of 27% at 400°F and then increases to 86% at 1600°F. Thus the panels can yield a small amount without any danger of rupture. Moreover, a slight amount of yielding, on the order of 3-5%, would not cause any serious dimensional problems in the cavity.

The flat-plate stresses for the different panels rise rapidly as the result of only small pressure differentials. In the case of one row of low-temperature liner panels the flat-plate stress equals $6,785 \times \text{pressure}$, see Appendix 3 page 1. However, a small amount of yielding of the panel changes the stress pattern from one of bending stress at the welded edge of the plate to hoop stress at the mid-point of the plate. If a strain of 5% is assumed, the stress lowers to $117 \times \text{pressure}$, a reduction of 98%. Since the panel steel is quite ductile at all temperatures, the permissible pressure behind the plate is based on permitting 5% strain across the smaller transverse dimension. This approach permits adequate pressure behind the panels to allow safe venting of the steam.

It should be noted that a bi-axial strain condition can markedly reduce the maximum elongation before failure, Reference 13. However, the most extreme reduction occurs when the transverse strains are equal, i.e. ratio equals 1.0. When the ratio of longitudinal to transverse strain exceeds 2.0 there appears to be little or no reduction in the elongation in the longitudinal direction before rupture. But the mechanism is not well understood and the condition becomes exaggerated at higher temperatures. However, the assumed elongation of 5% is only $1/9$ of the indicated maximum strain before failure of 45%. For plain carbon steel the maximum expected reduction in strain before failure for a strain ratio of one at an elevated temperature is 80%, or a reduction by a factor of five. Since the condition considered is not extreme and the assumed elongation of 5% is much less than even the minimum permissible elongation

for the extreme condition of 9.2% strain for a bi-axial strain condition with a ratio of one, there should be no danger of rupture.

The rate at which the water is driven out of the concrete at different times can be determined from the data from computer case no. III. These data show the amount of water released in a small increment of time at a specified elapsed time plus the total water released at each time. From these data the rate of steam generation can be calculated for the increment of time; this is a semi-instantaneous rate of steam generation. An average rate of steam generation during the intervals between the elapsed times can also be calculated from the total.

Data are given for heat structures nos. 2, 3, and 6.

Heat Structure No. 2	Upper cavity wall	3,296 ft ²
Heat Structure No. 3	Lower portion of upper cavity wall down to high-temperature liner	358 ft ²
Heat Structure No. 6	Behind the high-temperature liner in the lower portion of the reactor cavity	1,147 ft ²

The semi-instantaneous rates of steam generation as well as the average rates of steam generation based on the gross weights released for the different heat structures are calculated in Appendix 4, pages 13 through 16. The individual rates are then totaled and plotted. The curves are given in Appendix 4, pages 19 and 20.

The rate of steam generation is a maximum near time zero and then falls rapidly as shown in the graph.

It is assumed that all of the steam is piped to the area behind the high-temperature liner and then vented through vent pipes. The maximum average rate of steam evolution shown by the computer analysis equals 7,953 lb/hr at an elapsed time of 0.054 hours.

For this analysis it is assumed that the maximum rate of steam generation shortly after time zero would equal 8500 lb/hr. The maximum rates of steam generation in lb/hr-ft² for the three different heat structures are listed below, see also Appendix 4.

<u>Heat Structure No.</u>	<u>Rate of Steam Generation</u> <u>lb/hr-ft²</u>
2	1.05
3	3.33
6	4.10

The liner in the uppermost 17.82 ft of cavity, from elevation 513'-4 to the underside of the support ledge, is manufactured by EFCO. The EFCO liner consists of a group of 18 individual panels which are connected together. The dimensions of one of these panels are shown in Figure A4-1. Two panels are connected vertically.

Below the EFCO liner is the low-temperature liner by Becntel and below the low-temperature liner is the high-temperature liner. These are shown in Figures 9, 9a, and 9b.

It is assumed that the temperature of the liner in heat structure no. 2 is equal to the temperature of the cavity atmosphere above the pool of sodium. For the purpose of this analysis it is assumed that the temperature, 312°F, is constant over the entire surface of heat structure no. 2. Similarly, the temperature of the liner in heat structure no. 3 is equal to 1083°F over its entire area. The temperature of the high-temperature liner, which covers the surface of the cavity in the area of heat structure no. 6 is taken to equal 1113°F. The temperature of the steam behind the liner panels of the different heat structures is assumed to equal the temperature of the panel.

The individual panels are all welded to embedments in the concrete of the cavity wall. The EFCO liner is skip-welded to the angle iron embedments, eight inches of weld bead on each side of the angle leg for each 12 inches of angle length. This gives a two-inch overlap at each end of each eight-inch length of weld, assuming the sections of weld are evenly spaced. The weld is

thus essentially continuous and there is no open space available for steam flow.

In the case of the high-temperature liner and of the low-temperature liner below the EFCO liner, the liners are made of individual panels welded on all four edges to I-beam embedments as shown in Figures 9a and 9b. There is no passage for the escape of steam generated behind the panels.

When the concrete sets up, it will shrink a small amount and pull away from the embedments. This will leave a thin passage around the embedments through which steam could flow. The shrinkage will not be uniform and it will not be large. The concrete could pull away from an embedment over its entire surface or only over part of the surface. For the purpose of estimating the rate at which steam might escape past an embedment assume a slot 0.010 inch thick. The length around an I-beam embedment would equal 1.063 ft, see Figure 9-a. For a condition of 500°F superheated steam with an upstream pressure of 15 psig and a downstream pressure of one atmosphere (15 psi drop across the I-beam) the flow rate of steam through such a slot one foot wide would only equal 0.00737 lb/sec. The calculations are given in Appendix 4. This low rate of flow through a slot coupled with the indeterminacy of the existence of a continuous slot, rule out the possibility of venting the steam past the embedments through clearance resulting from shrinkage of the concrete.

The vertical embedments in the EFCO liner section are spaced as much as four feet apart, see Figure A4-1. The vertical embedments in the remainder of the low-temperature liner section are spaced 48.5 in. to 49.1 in. apart. It can be assumed that steam will have to be vented from a series of four-foot wide sections running vertically from beneath the reactor support ledge down to the cavity floor. The rates of steam generation behind the panels in a four-foot wide section over the height of the cavity are shown in Figure A4-2.

The sizes of the holes in the liner panels and of the connecting tubes will depend upon the rate at which the steam must be vented, the properties of the steam, and the pressure which the panels can withstand on the back side.

The larger sized EFCO liner panels are 1.27 ft x 4.00 ft and are made of 1/4-in. thick plate. The temperature of these panels equals 312°F. They are made of A516 grade 55 steel. The yield strength at 312°F equals $0.93 \times 30,000 = 27,900$ psi (Ref. Figure 3). If the panel is allowed to yield 5% across the slot dimension the hoop stress would equal $59.44 \times p$, see Appendix 3. This means that the pressure behind the panel could rise somewhat above a pressure equal to $27,900/59.44 = 469$ psi without causing the panel to yield more than 5%. The steel from which the panels are made has a total elongation of 27.5% at 312°F so the panel could safely withstand a pressure of 469 psi or greater based on this criterion.

The panels are welded to the embedments with 3/16-in. fillet welds on both sides of the angle leg. The welds are discontinuous in lengths of 8 in. every 12 in. This gives an average overlap of two inches at each end of each weld. The fillet welds do provide full strength. Each panel will have a fillet weld along 66.7% of each edge. The panels are 15.24 in. x 48.00 in. A 3/16-in fillet weld provides $0.188 \times 0.707 = 0.133 \text{ in}^2$ per linear inch shear area. So the shear area per panel would equal

$$0.667 \times 2 (15.24 + 48.00) \times 0.133 = 11.22 \text{ in}^2.$$

Shear yield stress at 312°F equals $27,900 \times 0.55 = 15,345$ psi

Pressure on panel to stress fillet welds to shear yield--

$$p \times (15.24 \times 48.00) = 11.22 \times 15,345 \quad p = 235 \text{ psi}$$

The saturation pressure of steam at 312°F is 80 psia which would give a differential pressure across the EFCO panels of 65 psi. The panels could withstand that pressure with only a minor amount of plastic strain.

Steam is produced at the rate of 5.35 lb/hr behind each of the full-sized EFCO panels. If each panel is connected to the header with a 6-in. length of 5/16 x 0.035 in. tubing the pressure drop for this rate of steam flow will be less than one psi, see Appendix 4, page 5.

All the steam from a four-foot wide section is collected into one 3/4 x 0.035 wall tube which is 10 ft long and discharges into the space behind the high-temperature liner. The pressure drop required for steam in the amount of 74.84 lb/hr to flow through 10 ft of tubing is less than one psi, see Appendix 4, page 6.

The steam generated behind the sections of low-temperature liner below the EFCO liner can be vented into the space behind the high-temperature liner in the same fashion. The largest panel section occurs between I-beams 6 and 7, see Figures 9 and 9-a. The panel is made of A516 grade 55 steel. If the panel is permitted to strain 5% it will safely withstand a differential pressure as high as 87 psi based on yield strength at 1083°F. The analysis is shown in Appendix 3, page 1, which gives $S = .117 \times p$ after the panel has strained 5%.

A516 grade 55 steel has a yield strength equal to $0.34 \times 30,000 = 10,200$ psi. So $p = 10,200 / .117 = 87$ psi.

The panel will withstand a total elongation of 46% at 1083°F before failure.

If connecting tubes of 5/16 x 0.035 in. tubing are used to connect the low-temperature panels to a header, the maximum pressure drop will occur for the largest flow. The largest

flow occurs from the panel between I-beams nos. 6 and 7, see Appendix 4, Figure A4-3. The flow rate equals 33.3 lb/hr. The pressure drop requirement for this rate of flow through a 5/16 x 0.035 tube six inches long would equal approximately 6.5 psi, see Appendix 4 page 8.

The pressure drop required to drive the total steam flow of 132.62 lb/hr generated behind a four-foot wide strip of low-temperature panel below the EFCO liner through a 3/4 x 0.035 inch tubing to the area behind the high-temperature liner would equal less than 2 psi, see Appendix 4 page 9. —

There are at present two vent pipes passing through the reactor cavity wall. These are made of 1-1/2-in. schedule 40 pipe and are 11 feet long with two 90° elbows.

The pressure of the steam behind the high-temperature liner should not exceed 5 psig to prevent collapse of the liner at 1113°F, see Appendix 3 page 3. This gives a maximum pressure drop available to drive the steam through the vent pipes of 5 psi. This pressure head would produce a flow of approximately 0.206 lb/sec through each of the two vent pipes. The total maximum rate of steam generation is approximately 8500 lb/hr or 2.36 lb/sec. See Appendix 4 page 15 and Figure A4-2.

It might be noted that the expected maximum rate of steam generation in heat structure no. 6 equals $4.1 \text{ lb/hr ft}^2 \times 1147 \text{ ft}^2 = 4703 \text{ lb/hr}$ or 1.306 lb/sec. The two vent pipes would handle less than 32% of this if the pressure behind the high-temperature liner were held to no more than 5 psig.

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Section Showing Reactor, Guard Vessel, and Reactor Cavity

Plot of Ratio

USS Steel - Med. Carbon Steel (0.35% C)
+ AISC 1015 515 115 1.23
Δ AISC 1015 515 115 1.23

Ultimate Tensile Strength at Temperature
Ultimate Tensile Strength at Room Temperature
vs. Temperature for Medium-Carbon Steel

FIGURE 2

Δ Carbon Steel (.4% C) STP 192-15-17
○ Carbon Steel (.4% C) STP 192-15-17
+ U.S.S. 4140 - Mel. (Carbon Steel) (.35% C)

Plot of Ratio $\frac{\text{Yield Strength at Temperature}}{\text{Yield Strength at Room Temperature}}$ vs. Temperature
for Medium-Carbon Steel

FIGURE 3

Plot of Typical Total Elongation
vs. Temperature for Medium-Carbon Steel

FIGURE 4

Variation of Modulus of Elasticity, E, of Concrete with Temperature from data reported by R. Philleo, "Some Properties of Concrete at High Temperatures," Journal of the Concrete Institute, April 1958

Figure 5

1200.

Temperature at Outside of Reactor Cavity Wall and Pressure
in Reactor Cavity Plotted vs. Time for Location Near Bottom
of High-Temperature Liner, Case No. 405

1000

800

600

400

200

10

8

6

4

2

Pressure - psig

Total Pressure At Elevation 489 ft. (Includes Static Head)

Pressure In Reactor Cavity

Time - hours

FIGURE 6

Force Acting on Reactor Cavity Wall Due to Pressure of Sodium

FIGURE 7

Diagram Showing Distribution of Pressure
Acting Against Lower Reactor Cavity Wall

FIGURE 8

High Temperature and Low Temperature Liners

FIGURE 9

Detail Showing Method of Attaching Low-Temperature Liner to Cavity Wall

FIGURE 9-a

Section Showing Shearing Force Acting on Welds at Upper and Lower Ends of Section of Low-Temperature Liner

Fillet Weld

FIGURE 9-c

Bottom End of High-Temperature Liner

FIGURE 10

Schematic Representation of Cylinder-to-Ring Portion of Low-Temperature Liner

FIGURE 11

Section of Horizontal Ring of Low-Temperature Liner

FIGURE 12

Upper End of Lower Cylindrical Portion of
Low-Temperature Liner

FIGURE 13

Pressure Loading on Ring Section of Low-Temperature Liner

FIGURE 14

For a flat plate with all edges fixed and a uniform load (pressure) over its entire surface

Take lowest panel section in low temperature liner.

thickness = 0.25 in.

Ref. Roark, 4th Ed. p 227

Case No 41

$$S = \beta \frac{p b^2}{t^2}$$

$$a/b = \frac{49}{30} = 1.633$$

β (from Table in Roark) 0.4712

$$S = 0.4712 \frac{p \times 30^2}{0.25^2} = 6,785 \times p$$

Flat plate stress

For 5% elongation across short dimension

Chord length = 15

$$R \theta = 1.05 \times R \sin \theta$$

$$\frac{\theta}{\sin \theta} = 1.05$$

$$\theta = 30^\circ - 51' \text{ or } 0.5384 \text{ radians}$$

$$\sin \theta = 0.51279$$

$$\cos \theta = 0.85851$$

$$R = \frac{15}{\sin \theta} = 29.2517$$

$$b = R - R \cos \theta = 0.14149 \times R = 4.13883 \text{ in.}$$

$$\text{Stress} = \frac{F R}{t} = \frac{p \times 29.2517}{0.25} = 117.0 \times p$$

Determine strain in 49-inch direction

$$R^2 = (R - b)^2 + 24.5^2$$

$$R = \frac{b^2 + 24.5^2}{2b} = 74.5839$$

$$\alpha = \sin^{-1} \frac{24.5}{R} = \sin^{-1} 0.32849 = 19^\circ 13.63'$$

$$\alpha = 0.33470 \text{ radians}$$

$$2\alpha = 0.66940 \text{ radians}$$

$$\text{Arc length} = R \times 2\alpha = 49.9265 \text{ in.}$$

$$\text{Strain} = \frac{0.9265}{49} = 0.01891 \text{ or } 1.891\%$$

$$\text{Ratio of strains} = \frac{0.0500}{0.01891} = 2.644$$

EFCO Liner Panel 1.27 ft x 4.00 ft or 15.24 in. x 48 in.

Flat plate stress

$$S = p \frac{p b^2}{t^2} = 0.50 \frac{p \times 15.24^2}{0.25^2} = 1858 \times p$$

Check stress fr. strain

$$R \beta = 1.05 \times R \sin \beta$$

$$\frac{\beta}{\sin \beta} = 1.05 \quad \beta = 30^\circ 51' \text{ or } 0.5384 \text{ radians}$$

$$\sin \beta = 0.51279$$

$$\cos \beta = 0.85851$$

$$R = \frac{7.62}{\sin \beta} = 14.860 \text{ in.}$$

$$b = R - R \cos \beta = 0.14149 \times R = 2.1025 \text{ in.}$$

$$\text{Hoop stress} = \frac{p R}{t} = \frac{p \times 14.860}{0.25} = 59.44 \times p$$

p = pressure in psi

High-Temperature Liner - External Pressure

The high-temperature liner can be considered as a short tube with the ends held circular. If the pressure behind the high-temperature liner gets too high the liner will fail due to elastic instability.

The equation for elastic instability is, from Roark, page 354, case 51

$$P = 0.807 \frac{E t^2}{L r} \left[\left(\frac{1}{1-\nu^2} \right)^3 \left(\frac{t}{r} \right)^2 \right]^{\frac{1}{4}}$$

The high-temperature liner is made of A516 grade 70 steel.

At spill conditions -

$$Y_S = 38,000 \times 0.32 = 12,160 \text{ psi}$$

$$E = 20 \times 10^6 \text{ psi}$$

$$\nu = 0.3$$

The dimensions include -

$$r = 16' - 3" = 195 \text{ in}$$

$$L = 84 \text{ in.}$$

$$t = 0.375 \text{ in.}$$

$$P = 0.807 \frac{20 \times 10^6 \times 0.1406}{84 \times 195} \left[\left(\frac{1}{0.91} \right)^3 \left(\frac{0.375}{195} \right)^2 \right]^{\frac{1}{4}}$$

$$P = 138.54 (1.327 \times 3.698 \times 10^{-6}) = 138.54 \times 0.00471 = 6.53 \text{ psi}$$

check hoop stress at 6.53 psi

$$S = \frac{PR}{t} = \frac{6.53 \times 195}{0.375} = 4630 \text{ psi}$$

Assume plate forms an arc of a circle as it deforms. If it elongates 20%:

$$R \cdot \theta = 1.20 \times R \sin \theta$$

$$\frac{\theta}{\sin \theta} = 1.20$$

$$\theta = 1.028 \text{ radians or } 57.93^\circ$$

$$\sin \theta = \theta / 1.20 = 0.8567$$

For an arc length of 33.0 in

$$R \sin \theta = 15$$

$$\text{or } R = \frac{15}{\sin \theta} = \frac{15}{0.8567} = 17.51 \text{ in.}$$

$$\text{Hoop stress} = \frac{PD}{2t} = \frac{P \times 2 \times 17.51}{2 \times 0.25} = 70.04 P$$

Similarly —

Elongation %	θ radians	θ degrees	$\sin \theta$	R in.	Hoop Stress psi
15	0.9029	51.73	0.7851	19.11	76.42 P
20	1.028	58.90	0.8567	17.51	70.04 P
25	1.1311	64.81	0.9349	16.58	66.31 P
30	1.2202	69.91	0.9386	15.98	63.92 P
35	1.3015	74.57	0.9641	15.56	62.23 P
40	1.3732	78.68	0.9809	15.29	61.17 P
50	1.5700	90.00	1.0000	15.00	60.00 P

The values of $\frac{\sin \theta}{\theta}$ can be checked the old of the series

$$\frac{\sin \theta}{\theta} = 1 - \frac{\theta^2}{6} + \frac{\theta^4}{120} - \frac{\theta^6}{5040} + \frac{\theta^8}{362880} - \dots$$

EFCO Panel
Figure A4-1

$$3.083' \times 4.000' \times 1.05 \text{ lb/ft}^2_{hr} + 0.500' \times 4.000' \times 3.33 \text{ lb/ft}^2_{hr} = 19.609 \text{ lb/ft}^2_{hr}$$

$$19.609 + 43.05 = 62.659 \text{ lb/hr. or } 0.01740 \text{ lb/sec}$$

Rate of Steam Generation In A 4-foot Width	I Beam	Δ Rate lb/sec	Total Past I-Beam lb/sec
		0.01740	
	4	0.00370	0.01740
	5	0.00648	0.02111
	6	0.00925	0.02759
	7		0.03684

Figure A4-3

Compressible Fluid Flow

The rate of compressible fluid flow through an orifice or short tube can be calculated with the aid of the following relationship:

$$W = 8.02 A_t C_d P_1 \frac{Y}{\sqrt{RT_1}}$$

W = Rate of flow in lb/sec

A_t = Cross-sectional or flow area in in.²

C_d = Orifice coefficient

P_1 = Upstream pressure in psia

R = Gas constant = 85.59 ft/lb-mole

T_1 = Upstream temperature in °R

Y = Compressibility factor

$$Y = \left(\frac{k}{k-1} \right)^{\frac{1}{2}} \left(r^{\frac{2}{k}} - r^{\frac{k+1}{k}} \right)^{\frac{1}{2}}$$

$$r = \frac{\text{Downstream Pressure}}{\text{Upstream Pressure}}$$

k for steam = 1.23

$$\text{For steam } Y = 2.1381 (r^{1.5625} - r^{1.78125})^{\frac{1}{2}}$$

Steam Flow Pressure Drop Analysis, EFco Liner -1

$$\text{Flow rate } 5.35 \text{ lb/hr.} = 0.001486 \text{ lb/sec.}$$

$$\text{Assume } 5/16 \times 0.035 \text{ wall tubing } ID = 0.243 \text{ in.}$$

$$h_L = 1.5 \frac{v^2}{2g} + f \frac{L}{D} \frac{v^2}{2g} \quad \text{or } 0.0203 \text{ ft}$$

Take limiting condition of sat. steam at 312°F

$$\text{Steam properties } p = 80 \text{ psia}$$

$$V = 5.472 \text{ ft}^3/\text{lb}$$

$$\mu = 0.0155 \text{ centipoise}$$

$$0.0155 / 4.72 \times 10^{-3} = 3.24 \times 10^{-7} \text{ lb sec/ft}^2$$

$$\text{Assume 6 in. length of connector } L/D = 6.00 / 0.243 = 24.7$$

$$Q = 0.001486 \times 5.472 = 0.00812 \text{ ft}^3/\text{sec}$$

$$A_{ID} \text{ of tubing} = 0.0463 \text{ in.}^2 \quad \text{or } 3.219 \times 10^{-4} \text{ ft}^2$$

$$v = (81.3 \times 10^{-4} \text{ ft}^3/\text{sec}) / (3.219 \times 10^{-4} \text{ ft}^2) = 25.26 \text{ ft/sec} \quad \frac{v^2}{2g} = 9.902$$

$$Re = \frac{\rho D v}{\mu} = \frac{0.1827 / 0.0203 \times 25.76}{32.2 \times 3.24 \times 10^{-7}} = 9,158$$

$$\epsilon/D = 0.000,005 / 0.0203 = 0.000,246 \text{ for drawn tubing}$$

$$f = 0.031$$

$$h_L = 1.5 \times 9.908 + 0.031 \times 24.7 \times 9.908 = 2.266 \times \frac{v^2}{2g} = 22.45 \text{ ft}$$

$$\text{specific weight of steam } w = 0.1827 \text{ lb/ft}^3$$

$$\Delta P = w \times h_L = 0.1827 \times 22.45 = 4.10 \text{ lb/ft}^2 \quad \text{or } 0.028 \text{ psi}$$

Check assuming compressible flow through a short tube

$$\text{for two velocity heads loss } h = 2 \frac{v^2}{2g} \quad \text{or } v = 0.71 \sqrt{2gh}$$

and 1.0 psi drop

$$W = 8.02 A_t C_d P_1 \sqrt{\frac{Y}{RT_1}} = 8.02 \times 0.0463 \times 0.71 \times 20 \times \frac{0.1119}{\sqrt{85.591772}} = 0.0091 \text{ lb/sec}$$

This is considerably greater than the required flow rate of

Steam Flow Pressure Drop Analysis, EFCO Liner-2

The total amount of steam must flow through the header to the high-temp. liner. When the connecting tube passes through the liquid sodium the temperature of the steam will rise. Assume it rises to 1080°F

Pressure drop through 13 ft length

$$Q = 0.02077 \text{ lb/sec} \times \frac{80}{79} \times 5.472 = 0.1152 \text{ ft}^3/\text{sec}$$

$$v = 0.1152 / 0.00252 = 45.71 \text{ ft/sec} \quad \frac{v^2}{2g} = 32.44 \text{ ft}$$

$$Re = \frac{\rho D v}{\mu} = \frac{0.180 \times 0.0567 \times 45.71}{32.2 \times 3.24 \times 10^{-7}} = 24,700$$

$$\epsilon/D = 0.000,005 / 0.0567 = 0.000,088 \quad f = 0.022$$

$$h_L = f \frac{L}{D} \frac{v^2}{2g} = 0.022 \frac{13}{0.0567} \times 32.44 = 163.6 \text{ ft}$$

$$\Delta p = (0.180 \text{ lb/ft}^3) \times (163.6 \text{ ft.}) = 29.45 \text{ lb/ft}^2 \text{ or } 0.205 \text{ psi}$$

Assume steam is at mean pressure of 78 psia and 1080°F through last 10.0 ft. of line

$$\text{Specific volume } V = 11.72 \text{ ft}^3/\text{lb} \quad \mu = 0.0295 \text{ centipoise}$$

Steam Flow Pressure Drop Analysis, EFCO Liner - 3

$$h_L = \frac{v^2}{2g} + f \frac{L}{D} \frac{v^2}{2g} = \left(1 + 0.022 \frac{10}{0.0567} \right) 145.15 = (1 + 3.88) \frac{v^2}{2g}$$

$$h_L = 708 \text{ ft} \quad \Delta P = \frac{708 \text{ ft}}{11.72 \text{ ft}^2/\text{lb}} = 60.4 \text{ lb/ft}^2 \text{ or } 0.42 \text{ psi}$$

$$0.42 + 0.21 = 0.63 \text{ psi}$$

So 3/4-in. tubing is adequate to handle flow of steam from a 4-foot-wide strip of EFCO liner.

The required ΔP is less than one psi. In the actual case there is a ΔP of almost 60 psi available. Thus the steam would expand further and choking flow would take place at the discharge end.

Steam Flow, Low-Temperature Liner -1

Assume the steam generated behind the panels of the low-temperature liner flows through connecting tubes into a header. The individual flows can be taken from Figure A4-2

①H	8.61 lb/hr	312°F
	17.22 lb/hr.	312°F
②H	17.22 lb/hr.	312°F
③H	19.61 lb/hr.	574°F
④H	13.32 lb/hr.	1083°F
⑤H	23.31 lb/hr.	1083°F
⑥H	33.33 lb/hr.	1083°F
⑦H		

$$\text{Total} = 132.62 \text{ lb/hr.} = 0.0366 \text{ lb/sec}$$

The most critical requirement is the largest flow rate at the higher temperature - 33.33 lb/hr. at 1083°F, 0.00926 lb/sec.

Assume 20 psig pressure behind panel

$$\text{Steam at 35 psia and } 1083^\circ\text{F} \quad V = 26.22 \text{ ft}^3/\text{lb} \quad w = 0.0381 \text{ lb/ft}^3$$

$$Q = 26.22 \times 0.00926 = 0.243 \text{ ft}^3/\text{sec}$$

$$\mu = 0.029 \text{ cgs.}$$

$$\text{For a } 5/16 \text{ x } 0.035 \text{ tubing} \quad \text{I.D.} = 0.243 \text{ in.} = 0.0203 \text{ ft.}$$

$$A_{10} = 0.0463 \text{ in.}^2 = 3.219 \times 10^{-4} \text{ ft.}^2$$

Calculate pressure drop through a 6-in. length of tubing

$$h_L = 1.5 \frac{v^2}{2g} + f \frac{L}{D} \frac{v^2}{2g}$$

$$v = (0.243 \text{ ft}^3/\text{sec}) / (3.219 \times 10^{-4} \text{ ft.}^2) = 755 \text{ ft/sec}$$

$$\text{Check sonic velocity} \quad c = \sqrt{k g R T} = \sqrt{1.28 \times 32.2 \times 85.59 \times 1543}$$

Steam Flow, Low-Temperature Liner - 2

The error introduced by treating the flow as incompressible is on the order of $M^2/4$ for low Mach Nos.

$\frac{M^2}{4} = 0.0262$ So the error should not be appreciable and the calculation will be satisfactory for an approximation

$$Re = \frac{\rho v D}{\mu} = \frac{0.0321 \times 0.0203 \times 755}{32.2 \times 0.027/47,111} = 29,900$$

$$e/D = 0.003035/0.0203 = 2.46 \times 10^{-4} \quad f = 0.024$$

$$h_L = 1.5 \frac{v^2}{2g} + 0.02 + \frac{L}{D} \times \frac{v^2}{2g} = (1.5 + 0.593) \times 2851 = 18,500$$

$$\Delta p = 18,500/26.22 = 706 \text{ lb/ft}^2 = 4.90 \text{ psi}$$

check as short tube with $C_d = 0.69$ and $r = \frac{30}{35} = 0.857$

$$W = 8.02 C_d A_t P_1 \frac{Y}{\sqrt{RT_1}} = 8.02 \times 0.69 \times 0.0463 \times 35 \times \frac{0.345}{\sqrt{85.57 \times 1543}}$$

$W = 0.00851 \text{ lb/sec}$, which is 8.1% less than the required flow rate of 0.00926 lb/sec.

If assume 6.5 psi drop $r = 0.814$ $Y = 0.382$

$W = 0.00942 \text{ lb/sec}$ so $5/16 \times 0.035$ tubing is large enough

Assume total steam flow is piped to high-temperature liner through a $3/4 \times 0.035$ tube

$$Q = 0.0368 \times 26.22 = 0.965 \text{ ft}^3/\text{sec}$$

$$v = (0.965 \text{ ft}^3/\text{sec}) / (0.00252 \text{ ft}^2) = 383 \text{ ft/sec}$$

$$\frac{v^2}{2g} = 2278 \text{ ft}$$

$$M = \frac{383}{2333} = 0.164$$

So incompressible flow relationship will give reasonable accuracy

$$h_L = f \frac{L}{D} \frac{v^2}{2g}$$

assume 5 ft. length

$$L = 5$$

$$D = 0.035$$

Steam Flow, Low-Temperature Liner - 3

$$Re = \frac{0.0381 \times 0.0567 \times 383}{32.2 \times 0.029/47,800} = 42,350$$

$$\epsilon/D = 0.000,088 \quad f = 0.025$$

$$h_L = 0.025 \times 88.2 \times 2278 = 5023 \text{ ft.}$$

$$\Delta p = 5023 \text{ ft} \times 0.0381 \text{ lb/ft}^3 = 191 \text{ lb/ft}^2 = 1.33 \text{ psi}$$

3/4 x 0.035 in. tubing is more than adequate

Steam Flow Through Vent Pipes

Assume 5 psi drop through vent pipe

Steam at 1113°F

20 psia

$$V = 46.81 \text{ ft}^3/\text{lb}$$

$$\mu = 0.326 \text{ centipoise}$$

$$= 5.86 \times 10^{-7} \text{ lb sec}/\text{ft}^2$$

For approximation use incompressible flow relationship.

$$h_L = 1.5 \frac{v^2}{2g} + f \frac{L}{D} \frac{v^2}{2g}$$

90° elbow equivalent $\frac{L}{D} = 30$

$$\frac{L}{D} = \frac{11}{0.134} + 2 \times 30 = 82 + 30 = 112$$

$$h_L = 5 \text{ psi} = 720 \text{ lb}/\text{ft}^2 \quad 720 \text{ lb}/\text{ft}^2 \times 46.81 \text{ ft}^3/\text{lb} = 33,702 \text{ ft}$$

Assume $f = 0.022$

$$33,700 = 1.5 \frac{v^2}{2g} + 0.022 \times 112 \times \frac{v^2}{2g} = 4.624 \frac{v^2}{2g}$$

$$v = 685 \text{ ft}/\text{sec}$$

Check value of f

$$Re = \frac{PDv}{\mu} = \frac{0.134 \times 685}{46.81 \times 32.2 \times 5.86 \times 10^{-7}} = 104,000$$

$$\epsilon/D = 0.00015/0.134 = 0.00111$$

$$f = 0.022$$

$$\text{Mach No.} = \frac{685}{\sqrt{1.28 \times 32.2 \times 85.56 \times 1573}} = \frac{685}{2355} = 0.291$$

So error should not be large

$$Q = 685 \times 0.0141 = 9.66 \text{ ft}^3/\text{sec}$$

$$W = 9.66 \text{ ft}^3/\text{sec} / 46.81 \text{ ft}^3/\text{lb} = 0.206 \text{ lb}/\text{sec per vent pipe}$$

$$\text{Maximum rate of steam generation} = 8500 \text{ lb}/\text{hr} = 2.36 \text{ lb}/\text{sec}$$

The maximum rate of steam generation in heat structure no. 6 is $4.10 \text{ lb/ft}^2 \text{ sec}$. The area of heat structure no. 6 equals 1177 ft^2 . $4.10 \times 1177 = 4793 \text{ lb/hr} = 1.306 \text{ lb/sec}$

Each vent pipe will vent less than 0.206 lb/sec

$$1.306 / 0.206 = 6.34$$

Estimate of Rate of Steam Flow Through a Thin Passage

The concrete will draw away from the embedments a slight amount when it sets, assume 0.010 inch. This will leave a small passage for steam flow past the embedment. The rate of flow for a given pressure drop can be estimated with the aid of the relationship given by Grinnell, Ref. 12. For the assured slot the flow will be laminar. The length of the flow path around the I-beam equals 12.75 in. This gives a height-to-length ratio equal to 7.8×10^{-4} which is below the limiting value of 1/1000. The flow equation is--

$$W = \frac{bh^3}{24\mu RTL} (P_1^2 - P_2^2)$$

For case considered

W = flow rate, lb/sec

b = width of slot, ft

1.0 ft

h = height of slot or thickness, ft

8.333×10^{-4} ft

μ = viscosity, lb sec/ft²

5.23×10^{-7} lb sec/ft² (0.025 cpse.)

R = gas constant, ft/°R

85.8 ft/°R

T = temperature, °R

960° R assumed

L = length of slot, ft

1.063 ft

P_1 = upstream pressure, lb/ft² abs.

P_2 = downstream pressure, lb/ft² abs. 2115.36 lb/ft² abs (1.0 atmos.)

$$W = \frac{1.0 \times 5.787 \times 10^{-10}}{24 \times 5.23 \times 10^{-7} \times 85.8 \times 960 \times 1.063} (P_1^2 - P_2^2)$$

$$W = 5.268 \times 10^{-10} (P_1^2 - P_2^2) \text{ for pressure in lb/ft}^2$$

$$W = 10.92 \times 10^{-6} (P_1^2 - P_2^2) \text{ for pressure in psi}$$

The rate of steam flow past a one-foot length of 3 x 7.5 I-beam embedment for the assumed conditions--0.010 in. gap, 500°F steam--can be calculated using the given equation. Assuming a discharge pressure of 15 psia gives the following flow rates for different upstream pressures:

P_{up} psia	P_{down} psia	Δp psi	Steam Flow Rate lb/sec	Vol. Rate ft ³ /sec
20	15	5	0.00191	0.0535
25	15	10	0.00437	0.096
30	15	15	0.00737	0.137
45	15	30	0.01966	0.242
60	15	45	0.03686	0.337
75	15	60	0.05897	0.429
90	15	75	0.08600	0.519
105	15	90	0.11794	0.605

The volume flow rate can be estimated by calculating the specific volume of steam for the given upstream pressure at 500°F and multiplying by the weight rate of flow. Saturated steam at 20 psi has a temperature of 228°F (688°R) and a specific volume of 20.09 ft³/lb.

$$V = \frac{960}{688} \times 20.09 = 28.03 \text{ ft}^3/\text{lb @ 500}^\circ\text{F} \quad 28.03 \times 0.00191 = 0.0535 \text{ ft}^3/\text{sec}$$

Rate of Average Generation of Steam Plotted Against Time

Data from Case III

Figure A4-4

Steam Rate lb/ft² hr

Time - hours

Heat Structure No. 6

Heat Structure No. 5

Heat Structure No. 2

Total Rate of Steam Generation, klb/hr

Rate of Steam Generation from 1700 to 1730
in Hall and Floor 13 Reactor Core
Plotted vs. Time from Accident

○ Average Rate of Steam Generation
□ Momentary Rate of Steam Generation

Time - hours

From: Reactor Safety
Date: January 6, 1975
Subject: MIGRATION OF MOISTURE FROM CONCRETE
IN REACTOR CAVITY WALL THROUGH VENT
LINES TO A POINT OUTSIDE THE REACTOR
CAVITY

Hanford Engineering Development Laboratory

To: D. D. Stepnewski

cc: RD Peak
DE Simpson LMP File/LB

During reactor operation the temperature in the reactor cavity will be held to 120°F. It is assumed that the temperature of the firebrick behind the high-temperature liner will also be at 120°F. It is furthermore assumed that the atmosphere in the containment building outside the reactor cavity wall will be at 76°F and 20% R.H.

The space behind the high-temperature liner is connected to the space outside the reactor cavity wall through two vent pipes. The difference in the conditions at the two ends of the vent pipes will cause water vapor to diffuse through the vent pipes and discharge to the atmosphere outside the reactor cavity wall. This migration of moisture will result in a drying-out of the concrete of the reactor cavity wall. However, calculations show that the rate of migration of moisture out of the concrete through the two vent lines, for the conditions considered, would equal less than one pound per year.

A copy of the analysis is attached.

L. M. Polentz
L. M. Polentz

LMP:1o
Attachment

RATE OF DIFFUSION OF WATER VAPOR THROUGH VENT LINES IN THE REACTOR CAVITY

L. M. Polentz

INTRODUCTION:

During the operation of the reactor, the temperature in the reactor cavity will be held at 120°F. It is assumed that the temperature in the space behind the high-temperature liner will also be held at 120°F. The moisture in the concrete will evaporate until the pressure of the water vapor in the space behind the reactor-cavity liner reaches the saturation pressure of water at 120°F. The partial pressure of the water vapor in the atmosphere outside the reactor cavity will be much less than the partial pressure of the water vapor in the space outside the reactor-cavity liner. The space is connected to the outside by means of two 1-1/2-inch schedule 40 pipe lines. The water vapor will diffuse from the space where its vapor pressure is relatively high to the outside where the vapor pressure is lower. The purpose of this analysis is to estimate the rate of migration of the water from the concrete through the vent lines to the space outside the reactor cavity walls due to the difference in temperature between the inside and outside of the reactor cavity wall.

CONCLUSION:

Calculations show that the rate of migration of moisture out of the concrete through the two vent lines for the conditions considered would be on the order of 0.127 lb/year.

DISCUSSION AND CALCULATION:

The reactor cavity is a thick-walled cylinder of concrete which is lined on the inside with steel plates. The liners form a sealed barrier so that no flow of gas or liquid can take place between the reactor cavity and the space behind the liners. Two vent pipes connect the space behind the high-temperature liner to the containment building as shown in Figure 1.

The temperature in the reactor cavity will be maintained at 120°F. It is assumed that the temperature of the inside face of the firebrick lining will also be maintained at 120°F. If moisture migrates from the concrete through the pores in the firebrick, then that moisture will also be held at 120°F. The moisture will evaporate until the partial pressure of the moisture reaches the saturation pressure of water at 120°F. The rise in the pressure of the water vapor will tend to cause an increase in the pressure in the space behind the high-temperature liner. This will cause a slight flow of moisture-laden air out of the vent lines until the internal pressure stabilizes at one atmosphere. Since there would be a relatively large supply of water available, the air in the space behind the high-temperature liner would be saturated with water vapor. The partial pressure of the water vapor would be maintained at the saturation pressure of water at 120°F, or 1.692 psia.

The temperature in the containment building at the discharge end of the vent lines is assumed to be held constant at 76°F. It is also assumed that the relative humidity will remain constant at 20%. The saturation pressure of water at 76°F equals 0.444 psia. Thus the partial pressure of the water vapor at the discharge end of the vent lines would equal 0.0888 psia.

The difference in the partial pressures of the water vapor at the two ends of the vent lines would cause a diffusion of the water vapor from the higher pressure area to the lower pressure area. For the purpose of this approximation it will be assumed that this is a case of steady-state diffusion of one gas (water vapor) through a second stagnant gas (air).

The equation for steady-state diffusion is given in Ref. 1 as:

$$N_A = \frac{D_v P}{RTZ} \times \frac{P_{B2} - P_{B1}}{P_{BM}} = \frac{D_v P}{RTZ} \times \frac{P_{A1} - P_{A2}}{P_{BM}}$$

where:

N_A = Rate of transfer of gas A (water vapor). lb-mol/(hr ft²)

D_v = Diffusivity, ft²/hr

P = Total pressure = one atmosphere

R = Gas constant = 0.728 atm. ft³/(lb-mole °R)

T = Temperature, °R

Z = Thickness of layer of gas B through which gas A diffuses, ft.

P_{B2} = Partial pressure of gas B (air) at outside end of vent line, atmos.

P_{B1} = Partial pressure of gas B at inside end of vent line, atmos.

$P_{BM} = (P_{B2} - P_{B1}) / \ln(P_{B2}/P_{B1})$

P_{A1} = Partial pressure of gas A at inside end of vent line, atmos.

P_{A2} = Partial pressure of gas A at outside end of vent line, atmos.

$D_v = aT^2$ where D_v for water vapor in air at 32°F is given as 0.853 ft²/hr in the International Critical tables. The value of D_v for the mean temperature of 98°F (temperature at midpoint of vent line) would equal $0.853 \times (558/492)^2 = 1.097$ ft²/hr. (a is an empirical constant.)

For this approximation assume the temperature is constant and equals the mean temperature of $(76 + 120)/2 = 98^\circ\text{F}$ or 558°R .

The total length of a vent line is 11.0 ft. The water molecules would have to travel this distance to escape to the area outside the reactor cavity wall. The distance Z , the thickness of the layer of air through which the water molecules would have to travel, was taken as 11.0 ft. The effect of the two elbows in the vent lines was assumed to be negligible and was ignored.

$$N_A = \frac{D_v P}{RTZ} \times (P_{B2} - P_{B1}) \times \frac{\ln(P_{B2}/P_{B1})}{P_{B2} - P_{B1}}$$

$$N_A = \frac{1.097 \times 1.0}{0.728 \times 558 \times 11.0} \ln(14.607/13.004) = 2.854 \times 10^{-5} \text{ lb-mol/hr ft}^2$$

One pound mol of water equals 18 lb

The cross-sectional area of one vent line equals

$$0.785 \times 1.610^2 = 2.035 \text{ in.}^2 = 0.0141 \text{ ft}^2$$

$$N_A = 2.854 \times 10^{-5} \times 18 \times 0.0141 = 7.243 \times 10^{-6} \text{ lb/hr per vent line.}$$

There are two vent lines and 8760 hours per year

$$7.243 \times 10^{-6} \times 8760 \times 2 = 0.127 \text{ lb/year}$$

The total rate of diffusion of water through the two vent lines
would equal 0.127 lb/year.

The units can be checked by converting standard engineering units.

$$P = 2116 \text{ lb/ft}^2 \text{ abs}$$

$$R = 1544/18 = 85.78 \text{ ft/}^\circ\text{R}$$

$$T = 558^\circ\text{R}$$

$$Z = 11.0 \text{ ft}$$

$$N_A = \frac{1.097 \times 2116}{85.78 \times 558 \times 11.0} \times 0.1162 \frac{(\text{ft}^2/\text{hr}) \times \text{lb/ft}^2}{(\text{ft/}^\circ\text{R}) (^\circ\text{R}) (\text{ft})} = 5.123 \times 10^{-4} \text{ lb/hr ft}^2$$

$$5.123 \times 10^{-4} \times 0.0141 = 7.22 \times 10^{-6} \text{ lb/hr per vent line}$$

The method of Treyball, Ref. 2, gives a similar result.

$$N_A = \frac{D_{AB} P}{RTZ P_{BM}} (P_{A1} - P_{A2})$$

$$D_{AB} = 0.258 \text{ cm}^2/\text{sec at } 25.9^\circ\text{C or } 0.305 \text{ cm}^2/\text{sec at } 59^\circ\text{C}$$

$$\text{interpolation } (D_{AB} = aT^2) \text{ gives } 0.272 \text{ cm}^2/\text{sec at } 37^\circ\text{C}$$

$$P = 1.0 \text{ atmosphere}$$

$$R = 82.06 \text{ cm}^3 \text{ atm/(gm-mole } ^\circ\text{k)}$$

$$T = 310^\circ\text{k}$$

$$Z = 335.3 \text{ cm}$$

$$P_{A1} = 1.692/14.696 = 0.1151 \text{ atm.}$$

$$P_{A2} = 0.0888/14.696 = 0.006042 \text{ atm.}$$

$$P_{B1} = 1 - P_{A1} = 0.8849 \text{ atm}$$

$$P_{B2} = 1 - P_{A2} = 0.99396 \text{ atm}$$

$$P_{BM} = \frac{P_{B1} - P_{B2}}{\ln(P_{B1}/P_{B2})} = 0.9389$$

$$N_A = \frac{0.272 \times 1.0 \times 0.1091}{82.06 \times 310 \times 335.3 \times 0.9389} = 3.705 \times 10^{-9} \text{ gm-mole/(sec-cm}^2\text{)}$$

$$\text{gm-mole} = \text{lb-mole}/454$$

$$\text{sec} = \text{hr}/3600$$

$$\text{cm}^2 = \text{ft}^2/929$$

$$3.705 \times 10^{-9} \text{ gm-mole/(sec-cm}^2\text{)} \times 3600 \times 929/454 = 2.73 \times 10^{-5} \text{ lb-mole (hr ft}^2\text{)}$$

$$2.73 \times 10^{-5} \times 18 = 4.91 \times 10^{-4} \text{ lb/hr ft}^2$$

REFERENCES:

1. Sherwood, Thomas K. and Robert L. Pigford, "Absorption and Extraction", McGraw-Hill Book Co., Inc., 1952
2. Treybal, Robert E., "Mass-Transfer Operations", McGraw-Hill Book Co., Inc., 1955

From: Reactor Safety
Date: July 11, 1974
Subject: Effect of Temperature on the Strength
of Magnetite-Aggregate, Portland-
Cement Concrete.

Hanford Engineering Development Laboratory

To: D. D. Stepnewski

cc: HS Davis PR Shire
 RK Hilliard RJ Slember
 RH Sanders CC Steele
 F. Schack LM Polentz File/LB
 DE Simpson

A series of tests was conducted to obtain information regarding the strength of magnetite-aggregate, portland-cement concrete at 1600°F. The information was required for the structural effects analysis of the reactor cavity wall following a PAHR containment transient.

Four concrete cylinders were heated, cooled, and tested as follows:

<u>Type Aggregate</u>	<u>Heat Cycle</u>		<u>Test Strength After Heating psi</u>
	<u>Temp. °F</u>	<u>Time days</u>	
Natural	1600	1	410
Magnetite	1000	1	5,230
Magnetite	1600	1	1,100
Magnetite	1600	7*	850

* in nitrogen atmosphere, others in air.

Based on the results of the tests, plus reports of investigations on the properties of portland-cement concrete at high temperatures, a value of 850 psi was used for the compressive strength of magnetite-aggregate concrete at 1600°F. This value, based on information presently available, is believed to be conservative for the following reasons:

- Concrete retains a greater proportion of its strength when heated if it is loaded in compression while being heated.
- Heated concrete retains a greater proportion of its strength if it is tested when hot than if it is cooled and then tested.

Since the cylinders which were tested were not loaded while being heated and were cooled before being tested for compressive strength, the strength values obtained should be conservative. In the case being analyzed the cavity wall would be loaded during heating and it would have to withstand compressive stress at the high temperature.

There are two factors which the results of the tests indicate may have an important bearing on the strength of magnetite-concrete at high temperature and which should be investigated further. These are:

July 11, 1974

1. The effect of extended time at temperature on the strength of the concrete, and 2. The relative effect on the strength of magnetite aggregate of holding it for an extended period of time at high temperature in air vs. holding it in a neutral (nitrogen) atmosphere.

A copy of the report of the test is attached.

L. M. Polentz

L. M. Polentz

LMP:lo
Attachment

TEST OF STRENGTH OF MAGNETITE-AGGREGATE CONCRETE AT HIGH TEMPERATURE

Lloyd M. Polentz

Introduction: Tests were conducted to determine the effect of temperature on the strength of magnetite-aggregate concrete. Three test cylinders of magnetite-aggregate concrete plus one cylinder of natural-aggregate concrete were heated to temperatures of 1000 and 1600°F, cooled, and tested for compressive strength.

Conclusions: The compressive strengths of the different test cylinders after heating were as follows:

Type Aggregate	Days	Lab. Cured Strength psi	Heat Cycle		Test Strength After Heating psi
			Temp. °F	Time days	
Natural	28	5970	1600	1	410
Magnetite	28	7200	1000	1	5230
Magnetite	91	8130	1600	1	1100
Magnetite	91	8130	1600	7*	850

* in nitrogen atmosphere, others in air

The effects on strength of heating magnetite-aggregate concrete in air vs. heating it in a neutral (nitrogen) atmosphere, and of holding the concrete at a high temperature for an extended period of time should both be investigated further.

Discussion: Information regarding the strength of portland-cement, magnetite-aggregate concrete after the concrete had been exposed to temperatures of 1000 and 1600°F was necessary to determine the effect of temperature on the strength of the reactor cavity wall and on the vessel support structure.

A survey of the literature unearthed very little information regarding the effect of high temperatures on the strength of magnetite-aggregate concrete. However, there was some useful information regarding the effect of temperature on portland-cement concrete. Different investigators have found that the reduction of strength of this type of concrete due to its exposure to a high temperature is reduced if the concrete is loaded in compression

during the time it is heated. Investigators have also found that portland-cement concrete retains a greater proportion of its strength after it has been heated if the concrete is tested at the high temperature. It has also been found that concrete continues to lose strength if it is subjected to thermal cycling, but that the bulk of the strength loss occurs during the first few cycles, primarily the first cycle. It is hypothesized that the loss of strength occurs largely as a result of cooling, probably due to the effect of differential coefficients of thermal expansion. Based on these findings it was concluded that if the concrete samples were heated in the unloaded condition and cooled before testing, the strengths obtained would be conservative. That is, the strength of the concrete in the reactor cavity wall would be markedly greater after being raised to a high temperature than would be indicated by the test results, since the concrete in the reactor cavity would be loaded throughout the time it was heated and would also be subjected to the required loading at the high temperature.

Test cylinders 6-in. in diameter by 12-in. long made of the same concrete as was used in the reactor cavity wall, plus one test cylinder of concrete made with natural aggregate, were obtained from the Bechtel Corp. The magnetite-aggregate concrete is a high-density concrete with a specific weight of approximately 222 lb/ft³. The proportion of cement used is 6 to 7 sacks per cubic yard. The W/C ratio is about 0.46. The aggregate is magnetite from Lovelock, Nevada and has a maximum size of 3/4 inch.

The test cylinders were heated in a heat-treating furnace in the Technical Shops at HEDL. The procedure was as follows:

One magnetite-aggregate cylinder and a handful of magnetite aggregate were placed in the furnace and the temperature was brought up to 1000°F. It took about 1-1/2 hours to raise the temperature. After the temperature in the furnace had been brought up to 1000°F it was held at that temperature for 24 hours. The test cylinder was then removed from the furnace, packed in ashes, and allowed to cool slowly.

A second magnetite-aggregate concrete cylinder, the natural-aggregate concrete cylinder, and a handful of magnetite aggregate were then placed in the hot furnace and the temperature was raised to 1600°F. It took about 2 hours for the temperature in the furnace to rise to 1600°F. The temperature was maintained for about 28 hours. The heat was then turned off and the furnace and contents allowed to cool overnight.

For the third test a nitrogen bottle was connected to the furnace. A stream of nitrogen was bled through the furnace throughout the test including heatup and cooldown to maintain a neutral test atmosphere. A magnetite-aggregate cylinder and a handful of magnetite aggregate were placed in the furnace and the temperature was raised to 1600°F. The temperature was maintained for seven days. Then the heat was turned off and the furnace and contents were allowed to cool slowly over the weekend.

After the concrete cylinders had been heated and cooled they were taken to the FFTF site where they were tested in compression. The results of the tests of the four different cylinders were as follows:

- Natural-aggregate, portland-cement concrete cylinder -- This cylinder was heated to 1600°F in air, held at temperature for 24 hours, and then cooled slowly in the furnace. This cylinder was heated at the same time as one of the magnetite-aggregate concrete cylinders. The cylinder was severely cracked after the heat cycle, and broke into three pieces while it was being capped for the compression test. The cylinder was taped together and tested in compression. The compressive strength equaled 410 psi. The cylinder crumbled during failure. The grout showed very little strength and there was no shearing of the aggregate.
- Magnetite-aggregate, portland-cement concrete cylinder -- This cylinder was heated to 1000°F in air, held at temperature for 24 hours, and then cooled slowly. Spalling was noted at four places and shallow cracks were apparent on the surface of the cylinder. The compressive strength of the cylinder equaled 5,230 psi at failure. The cylinder failed in a normal manner with a cone-shaped shear plane. The sheared faces of the pieces of aggregate were reddish in color, whereas the normal color is slate gray.
- Magnetite-aggregate, portland-cement concrete cylinder -- This cylinder was heated to 1600°F in air, held at temperature for 28 hours, and then cooled slowly in the furnace. This cylinder was heated at the same time as the natural-aggregate concrete cylinder. The cylinder showed many mosaic-like hairline cracks in the surface. The compressive stress equaled 1,100 psi at failure. The break was almost horizontal rather than conical. About half of the aggregate

in the plane sheared. The color of the grout had changed to a yellowish brown and the aggregate was a reddish color both on the outer surfaces of the unsheared particles and on the shear faces of the sheared particles.

- Magnetite-aggregate, portland-cement concrete cylinder -- This cylinder was heated to 1600°F in a nitrogen atmosphere, held at temperature for seven days (168 hours) and then cooled slowly in the furnace. The nitrogen atmosphere was maintained by bleeding nitrogen into the furnace from a standard cylinder. Nitrogen was bled into the furnace before the temperature was increased to purge the furnace, and throughout the heating and cooling cycle. There were no facilities for measuring the flow rate and the flow adjustment was made with a small needle valve. The system was unattended the first two days (over a weekend) and it is possible that the nitrogen bottle may have emptied and the flow of nitrogen could have stopped for a short time.

This cylinder also showed the mosaic-like pattern of surface cracking which was noted on the previous cylinder, but the cracking on this cylinder was more severe. The compressive stress equaled 850 psi at failure. Some crumbling was noticed and about half of the aggregate sheared. The color of the grout was reddish brown, as in the case of the previous cylinder and the aggregate was a reddish color, as before.

The aggregate which was heated in air to 1000°F for 24 hours changed color on the outside from the original slate gray to a reddish-brown rust color. Some pieces of aggregate were broken in two after the test and the faces of the breaks were tinged with reddish-brown. However, the bulk of the area of the break faces was a dark gray. The magnetic attraction of the heated aggregate was about the same as that of unheated aggregate.

Aggregate samples which were heated in air to 1600°F for 28 hours changed color to a reddish brown on the outside. Some of the pieces were broken in two after being cooled and the faces of the breaks showed a change of color over their complete areas. The color change was much greater than that of the pieces heated to 1000°F. Very little of the area of the break faces remained the original gray color. The color ranged from a slight

reddish tinge to yellowish and even somewhat chalky on the broken face of the sample. The magnetic attraction of the pieces of aggregate ranged from the same as for unheated aggregate down to, in one case, very little attraction. One of the pieces of aggregate of this sample group which was broken in two seemed to have lost a large portion of its strength.

Aggregate samples which were heated in a nitrogen atmosphere to 1600°F for seven days changed color on the outside to a reddish brown, similar to the color changes experienced with the other samples. Some of the pieces were broken in two after they had cooled and the broken faces showed splotches of reddish-brown color over approximately half of the broken surface area. The color change on the inside of the pieces of aggregate was much less than the change noted on samples which had been heated to 1600°F in air. The force of magnetic attraction of these samples ranged from that of unheated magnetite to very weak. One piece with essentially no magnetic attraction was broken in two and the broken faces were reddish in color over the full surface area of the break.

In addition to the samples discussed above, a piece of natural-aggregate, portland-cement concrete was run through all three heat cycles. This piece of concrete lost the bulk of its strength and could be crumbled by hand rather easily after passing through the three heating cycles.

The results of this short series of tests indicates that magnetite-aggregate portland-cement concrete loses a considerable portion of its strength after being heated to 1600°F.

Based on the one test in which natural-aggregate and magnetite-aggregate concrete cylinders were heated to 1600°F and then tested for compressive strength, it appears that magnetite-aggregate concrete retains a greater portion of its strength than does natural-aggregate concrete. The reason is not clear since the cement seemed to lose almost all of its binding strength in the case of the natural aggregate concrete while it appeared to retain a fair portion of its binding strength in the case of the magnetite-aggregate concrete. The same cement was used for both cylinders.

From the appearance of the batches of aggregate which were heated to 1600°F, it appears that the magnetite aggregate may undergo a chemical change when heated, especially in the presence of air. It is possible that this change could be accompanied by a reduction in strength. The change could be expected

to be greater if the time at temperature was increased.

A comparison of the result of heating to 1600°F and holding the concrete at that temperature for one day, or seven days indicates that holding the concrete a longer time at the high temperature may result in a greater reduction in strength.

The effects of heating magnetite-aggregate concrete in air vs. heating it in a neutral (nitrogen) atmosphere should be investigated further as should the effects of holding the concrete at the high temperature for a long period of time.

Acknowledgment: The author would like to acknowledge the assistance provided by Mr. Frans Schack, who supplied the concrete cylinders and the samples of aggregate used for testing and who handled the compression tests of the cylinders.

From: Reactor Safety
Date: February 21, 1975
Subject: "TEST OF AIR LEAKAGE THROUGH REACTOR
CAVITY WALL" LETTER, L. M. POLENTZ
TO D. D. STEPNEWSKI, FEBRUARY 4, 1975
AND ACCOMPANYING REPORT 'PERMEABILITY
OF THE REACTOR-CAVITY WALL', by LMP

Hanford Engineering Development Laboratory

To: D. D. Stepnewski

cc: JC Chandler	DI Hulbert	DE Simpson	
RR Derousseau	HD Lenkersdorfer	RJ Slember	
RK Greenwell	RD Peak	CC Steele	
RK Hilliard	RH Sanders	SA Weber	LMP File/LB

Reference:

1. "Flow of a Compressible Fluid for a Thin Passage",
S. K. Grinnell, Transactions ASME, May 1956.

The subject letter, and accompanying report reviewed the results of an analysis of the permeability of the reactor-cavity wall, and estimated the rate at which steam might flow through the wall. It did not consider the rate at which steam might flow past the embedments in the wall to which the panels are welded. This latter case had previously been briefly studied. The study and the results were reported. At that time the rate of flow around the embedments was found to be insufficient to vent the steam at the rate at which it was expected to be produced. Pertinent pages from that report are attached.

Since the time of a previous study a more refined analysis has been made of the heat transfer and rate of steam generation in the cavity.

The conditions of the hypothesized sodium spill have also changed since the original analysis was made, in that it is now assumed that the sub-cavity is filled with concrete and that the hot, liquid sodium will rise to an elevation of 505'-6. In addition, the newly calculated rate of steam generation is less than the rate calculated earlier. Using the new data the original analysis of the flow of steam past the embedments can be restudied and amplified. Such a re-analysis indicates that the expected flow of steam past the lowermost embedment would equal less than 2.0% of the required flow rate.

The maximum average rate of steam generation behind the low-temperature liner and that portion of the Bechtel liner which is below the level of the sodium pool surface as determined by the latest analysis equals 2.26 lb/hr-ft². For the purpose of this analysis it has been assumed that steam would be generated at this rate behind the panels from the location of the embedment just above the level of the sodium pool (elevation 507'-2) to the embedment just above the top of the high-temperature liner (elevation 496'-0).

The rate of steam generation per foot of circumference, i.e., a vertical strip one foot wide, would equal the linear length of wall between elevations 507'-2 and 496'-0, including the 1.75 feet of ledge, times the rate of steam generation in lb/hr-ft². The product equals $(11.17 + 1.75) \times 2.26 = 29.19$ lb/hr. The variation in the amount of steam which would be generated per circumferential foot due to the change in diameter of the reactor cavity above and below the ledge is small and would not seriously affect the results of this analysis. Actually, the difference in diameters means that a 1.11 ft. width of panel above the ledge would feed into a 1.00 ft. width of panel below the ledge. This means that the amount of steam generated behind the panels above the ledge should be increased by the factor of 1.11. This would mean an increase of $8.17 \text{ ft} \times 0.11 \times 2.26 = 2.03$ lb/hr in the required flow rate of steam past the bottommost embedment.

The steam generated behind the uppermost panel would have to flow across the embedment at elevation 502'-1, see Figure 1. This steam would then combine with the steam generated behind the next lower panel and the combined amount would have to flow past the embedment at elevation 499'-6, and so on. The flow past the lowest embedment would have to equal the total rate at which the steam was generated or 29.19 lb/hr., (31.22 lb/hr if the effect of the larger diameter above the ledge is included).

There would be a pressure drop across each embedment. If the conditions across each embedment are considered to be the same (equivalent flow paths and same steam properties) and laminar flow is assumed, then the pressure drop across each embedment will be proportional to the rate of flow. The maximum safe pressure behind the uppermost panel considered equals 20 psi, see subject report. It is assumed that at least 1 psi would be required to force the steam through the vent lines passing through the wall of the reactor cavity. This gives a total of 16 psi to force the steam past the five embedments in series. The pressure drops, distribution of pressure, and individual flow rates are shown in Figure 1. (These pressure and flow distributions ignore the effect of the larger cavity diameter above the ledge.)

As previously noted, all of the steam would have to flow past the lowermost embedment. If the temperature of the steam is taken as 950°F (midway between the surface temperature of the concrete at 800°F and the temperature of the outer surface of the liner at 1100°F) and the pressure as 8.73 psig (23.4 psia), the specific volume of the steam would equal 35.8 ft³/lb. The required volumetric rate of flow past one foot of the lower embedment would then equal $29.19 \times 35.8 = 1045$ ft³/hr or 0.290 ft³/sec. The required weight rate of flow would equal $29.19/3600 = 0.00811$ lb/sec or, adjusted for the larger reactor-cavity diameter above the ledge, $31.22/3600 = 0.00867$ lb/sec per foot of length of embedment at 496'-0 elevation.

Concrete hardens through hydration of the cement. As hydration proceeds the volume of the space initially occupied by water (volume of voids) is reduced. The reduction in volume results from the fact that the hydrated "gel" has a volume which is greater than that of the unhydrated cement.

For a water/cement ratio of about 0.35 by weight, the volume of the gel resulting from complete hydration will be sufficient to fill the original void volume due to water. With a higher W/C ratio, a portion of the original void volume will remain after hydration. The W/C ratio of the concrete used in the wall of the reactor cavity was approximately 0.47, so there would be some small volume of voids remaining after hydration of the cement. It is possible that some of this void volume could be adjacent to the embedments. The possibility of the void volume being so distributed that it would form a thin passage around each embedment is extremely remote.

The concrete for the reactor-cavity wall was vibrated into place after being poured. Small "pencil" vibrators were used in constricted spaces. A fairly uniform, homogeneous wall filling all of the space behind the forms was obtained by this method. The embedments were checked by tapping them after the concrete had hardened to determine if any of them were loose or if there were voids around them. A few of the embedments emitted a ringing sound indicating that they were not tightly held overall, i.e., that there was some small space between the embedment and the concrete. The thickness of this space, or thin passage, could only be estimated. The engineer on the job said he thought an estimate of 0.010-inch passage thickness was too much. A more realistic figure was thought to be 0.005 inch or less.

If a passage thickness of 0.005 in. is assumed and a length of 1.06 feet (Reference 1) the rate of steam flow for the previously described conditions can be estimated by the method of Grinnell, Reference.

$$W = \frac{bh^3}{24\mu RTL} (P_1^2 - P_2^2)$$

For steam at the conditions considered:

$$T = 950^\circ\text{F} = 1410^\circ\text{R}$$

$$R = 85.67 \text{ ft}^\circ\text{R (from steam table)}$$

$$\mu = 0.028 \text{ centipoise} = 5.86 \times 10^{-7} \text{ lb sec/ft}^2$$

$$P_1 = 8.73 \text{ psig} = 23.4 \text{ psia} = 3370 \text{ lb/ft}^2 \text{ abs}$$

$$P_2 = 4.00 \text{ psig} = 18.7 \text{ psia} = 2693 \text{ lb/ft}^2 \text{ abs}$$

$$b = 1.00 \text{ ft}$$

$$h = 0.005 \text{ in.} = 4.167 \times 10^{-4} \text{ ft}$$

$$L = 1.06 \text{ ft}$$

$$W = \frac{1.00 \times (4.167 \times 10^{-4})^3}{24 \times 5.86 \times 10^{-7} \times 85.67 \times 1410 \times 1.06} (3370^2 - 2693^2)$$

$$W = 165 \times 10^{-6} = 0.000165 \text{ lb/sec}$$

This equals $0.000165/0.00867 = 0.019$ or 2.0% of the required steam flow rate.

February 21, 1975

This is probably the maximum flow rate that could be expected around a one-foot length of embedment. There would be many places where the thickness of the passage between the concrete and the embedment would be less than 0.005 in. and the flow would thus be considerably less, since the rate of flow is proportional to the thickness of the passage to the third power. On the other hand, there would be some spaces at the ends of the horizontal segments of the embedments which could be open. These could occur where the horizontal segments are welded to the vertical members. However, it is probable that these spaces would be filled with concrete and there would not be any open flow passages at most of these locations.

L. M. Polentz

LMP:lo

Attachment

Steam Generation Rate And Vertical
Pressure Distribution Behind Low-Temperature Panels
In Reactor Cavity Following An HcDA and Fuel
Melt-through

Figure 1

LMP. 2-21-75

thus essentially continuous and there is no open space available for steam flow.

In the case of the high-temperature liner and of the low-temperature liner below the EFCO liner, the liners are made of individual panels welded on all four edges to I-beam embedments as shown in Figures 9a and 9b. There is no passage for the escape of steam generated behind the panels.

When the concrete sets up, it will shrink a small amount and pull away from the embedments. This will leave a thin passage around the embedments through which steam could flow. The shrinkage will not be uniform and it will not be large. The concrete could pull away from an embedment over its entire surface or only over part of the surface. For the purpose of estimating the rate at which steam might escape past an embedment assume a slot 0.010 inch thick. The length around an I-beam embedment would equal 1.063 ft, see Figure 9-a. For a condition of 500°F superheated steam with an upstream pressure of 15 psig and a downstream pressure of one atmosphere (15 psi drop across the I-beam) the flow rate of steam through such a slot one foot wide would only equal 0.00737 lb/sec. The calculations are given in Appendix 4. This low rate of flow through a slot coupled with the indeterminacy of the existence of a continuous slot, rule out the possibility of venting the steam past the embedments through clearance resulting from shrinkage of the concrete. Special provisions will have to be employed to let the steam escape without letting the pressure behind the panels rise to a dangerous point.

The vertical embedments in the EFCO liner section are spaced as much as four feet apart, see Figure A4-1. The vertical embedments in the remainder of the low-temperature liner section are spaced 48.5 in. to 49.1 in. apart. It can be assumed that steam will have to be vented from a series of four-foot wide sections running vertically from beneath the reactor support ledge down to the cavity floor. The rates of steam generation behind the panels over the height of the cavity are shown in Figure A4-4.

Detail Showing Method of Attaching Low-Temperature Liner to Cavity Wall

Figure 9-a

Section Showing Shearing Force Acting On Welds At Upper and Lower Ends of Section of Low-Temperature Liner

Figure 9-b

Estimate of Rate of Steam Flow Through a Thin Passage

The concrete will draw away from the embedments a slight amount when it sets, assume 0.010 inch. This will leave a small passage for steam flow past the embedment. The rate of flow for a given pressure drop can be estimated with the aid of the relationship given by Grinnell. For the assumed slot the flow will be laminar. The length of the flow path around the I-beam equals 12.75 in. This gives a height-to-length ratio equal to 7.8×10^{-4} which is below the limiting value of 1/1000. The flow equation is--

$$W = \frac{bh^3}{24\mu RTL} (P_1^2 - P_2^2)$$

For case considered

W = flow rate, lb/sec

b = width of slot, ft

1.0 ft

h = height of slot or thickness, ft

8.333×10^{-4} ft

μ = viscosity, lb sec/ft²

5.23×10^{-7} lb sec/ft² (0.025 cpse.)

R = gas constant, ft/°R

85.8 ft/°R

T = temperature, °R

960° R assumed

L = length of slot, ft

1.063 ft

P_1 = upstream pressure, lb/ft² abs.

P_2 = downstream pressure, lb/ft² abs. 2115.36 lb/ft² abs (1.0 atmos.)

$$W = \frac{1.0 \times 5.787 \times 10^{-10}}{24 \times 5.23 \times 10^{-7} \times 85.8 \times 960 \times 1.063} (P_1^2 - P_2^2)$$

$$W = 5.268 \times 10^{-10} (P_1^2 - P_2^2) \text{ for pressure in lb/ft}^2$$

$$W = 10.92 \times 10^{-6} (P_1^2 - P_2^2) \text{ for pressure in psi}$$

The rate of steam flow past a one-foot length of 3 x 7.5 I-beam embedment for the assumed conditions--0.010 in. gap, 500°F steam--can be calculated using the given equation. Assuming a discharge pressure of 15 psia gives the following flow rates for different upstream pressures:

P_{up} psia	P_{down} psia	Δp psi	Steam Flow Rate lb/sec	Vol. Rate ft ³ /sec
20	15	5	0.00191	0.0535
25	15	10	0.00437	0.096
30	15	15	0.00737	0.137
45	15	30	0.01966	0.242
60	15	45	0.03686	0.337
75	15	60	0.05897	0.429
90	15	75	0.08600	0.519
105	15	90	0.11794	0.605

The volume flow rate can be estimated by calculating the specific volume of steam for the given upstream pressure at 500°F and multiplying by the weight rate of flow. Saturated steam at 20 psi has a temperature of 228°F (688°R) and a specific volume of 20.09 ft³/lb.

$$V = \frac{960}{688} \times 20.09 = 28.03 \text{ ft}^3/\text{lb @ } 500^\circ\text{F} \quad 28.03 \times 0.00191 = 0.0535 \text{ ft}^3/\text{sec}$$

Rate of Average Generation of Steam Plotted
Against Time

Data from Case 1

Figure A-4-4

Steam Rate lb/ft² hr

12 1/2" 10 X 10 TO THE INCH 40 0700
7 X 10 INCHES
MADE IN U.S.A.
KEUFFEL & ESSER CO.

K-E 10 X 10 TO THE INCH 46 0700
 7 X 10 INCHES
 KRUPP & ESSER CO.

Total Rate of Steam Generation, klb/hr

From P.H.P. Chart

Rate of Steam Generation is for 2000 lbs.
 10 min. One Hour at 2000 lbs.
 Started vs. Time from Accident

○ Average rate of Steam Generation
 □ Momentary rate of Steam Generation

9

8

7

6

5

4

3

2

1

From: Reactor Safety
Date: July 8, 1974
Subject: Test of the Effect of Plastic Strain
on the Ability of a Flat Panel to
Withstand Pressure Applied Over One
Face

Hanford Engineering Development Laboratory

To: D. D. Stepnewski

cc: MJ Anderson RK Peak RJ Slember RK Hilliard
RK Brown JA Ryan AB Webb
GL Fox RH Sanders SA Weber
RG Hill LK Severud EO Weiner
GW McHair MD Simmons R. Winkle
RA Moen DL Simpson LMP File/LB.

A test was conducted by applying pressure to one face of a flat steel sheet 1/8-in. thick. The steel sheet was welded to a heavy backing plate on all four edges. The pressure was applied to the space between the steel sheet and the backing plate and was increased in steps until the weld failed. Failure, in the form of a small crack at one place in the weld, occurred at a pressure of 620 psi after the sheet had strained plastically approximately 4% in the center portion.

The data from the test indicated that a flat panel made of ductile material will deform plastically before failing when it is uniformly loaded on one face. It was also concluded from the test results that the membrane stress in the panel can be calculated with reasonable accuracy with the simple hoop stress formula after the strain exceeds 2%.

A copy of the report of the test is attached.

L. M. Polentz

LMP:lo
Attachment

EFFECT OF PLASTIC STRAIN ON ABILITY OF FLAT PLATE TO WITHSTAND APPLIED PRESSURE

Lloyd M. Polentz

Introduction:

A test was performed to determine how well a membrane stress analysis would apply to the stress in a flat plate which was pressurized on one side after it had strained elastically.

A rectangular flat sheet of 1/8-in. thick, low-carbon steel was welded to a one-inch thick backing plate on all four edges. Hydraulic pressure was applied to the back side of the sheet, between the sheet and the plate, and increased until failure occurred in the weld. The plastic strain was measured and a comparison was made between the yield stress and the applied pressure.

Conclusions:

Test results showed a good correlation between the measured pressure for a given strain and the pressure as calculated with the aid of the simple hoop stress relationship for strains in the steel sheet in excess of 2.0%. Specifically, a comparison of the measured and calculated values is as follows:

<u>Plastic Strain</u> <u>%</u>	<u>Measured</u> <u>Pressure</u> <u>Psi</u>	<u>Calculated</u> <u>Pressure</u> <u>Psi</u>	<u>Error</u> <u>%</u>
2.15	445	403	-9.4
2.50	490	450	-8.1
3.15	550	541	-1.6
4.0	605	641	6.1

Discussion:

The low-temperature liner in the reactor cavity is installed in the form of a series of small rectangular panels. The longer dimension of the rectangle is in the circumferential direction, the shorter dimension is in the vertical or axial direction. The stress due to a uniform pressure over the surface of a flat rectangular plate loaded in this manner is greater across the short dimension than across the long dimension, so the bulk of the strain will take place across the shorter dimension.

The larger panels in the lower portion of the low-temperature liner, that portion which would be subjected to the highest temperature in the case of a sodium spill, are 30 in. high by 49 in. long. The 49-in. length is in the circumferential direction on a radius of 16'3-1/8", or 195.1 inch. The length of the arc is 0.265% longer than the chord, assuming that the chord is 49 inches

2.
long, see Appendix for calculations. So the panel would be subjected to a small plastic strain if it changed from concave to convex. However, as previously noted, the bulk of the strain would take place in the vertical dimension, not the circumferential dimension.

To simplify the analysis and reduce the time and cost of the experiment, a flat plate was tested rather than a curved one.

A test sample was made by welding a piece of 1/8-in. thick hot-rolled sheet of ASTM A 569 steel (0.15% C, 0.25 to 0.60% Mn) onto the face of a piece of 1-in. thick steel plate, see Figure 1. The L/W ratio of 18/11 = 1.64 was made the same as the L/W ratio of the low-temperature panel, 49/30 = 1.63. Extra heavy welds were used around the edges to ensure that the shear strength of the weld was greater than the tensile strength of the sheet. The heavy backing plate was used to ensure that the backing plate would not bend and that all plastic deformation would take place in the 1/8-in. thick sheet.

The space behind the steel sheet was pressurized with hydraulic fluid as shown in Figure 2. The pressure was raised to a pre-determined value and then released. The resulting deflection of the center of the sheet was then measured with calipers and a scale. The pressure was increased in steps until the weld failed by cracking at one point when the pressure reached 620 psi and the system leaked. The recorded data are listed in Table 1.

The steel sheet was painted with a brittle lacquer so that the development of the pattern of strain could be seen as the pressure loading was increased. Sketches of the strain patterns after different pressure loadings are shown in Figures 3 through 7.

The A 569 steel is a low-carbon steel, 0.15% C, with 0.25 to 0.60% Mn. No curves could be found for the effect of strain hardening on this steel. However, stress-strain curves showing the increase in apparent yield strength for A 515 grade 70 steel were found in the literature, "The Effect of Slow-Constant Strain Rates on the Tensile Properties of Selected Alloys at Room and Elevated Temperatures", C. L. Dotson, ASME paper 73-Mat-BB.

The A 515 grade 70 steel tested had an analysis of 0.26% C, 0.83% Mn, 0.20% Si, 0.017% P, and 0.028% S. The probability is that the A 569 steel would show similar percentage changes in yield strength with strain, so the data for the room temperature variation of yield strength with strain were plotted as a percentage of the unstrained yield strength. The yield strength remains essentially constant up to 0.4% strain and then increases with further increase in strain. The curve of yield strength vs. strain is shown in Figure 8.

For this analysis it was assumed that the shape of the steel sheet transverse to the longer axis at the mid-point was that of an arc of a circle. This assumption permitted the calculation of the amount of strain from the measured deflection, and also permitted the determination of the radius of curvature corresponding to a given strain. The strain and radius of curvature were calculated for a number of different center deflections and plotted to give a curve of strain in percent vs. radius of curvature, Figure 9. The calculations are given in the appendix. The strain in a plate 11.0 in. wide was also plotted against the deflection at the center, the value "b" vs. strain in the calculations. The resulting curve is shown in Figure 10.

When pressure is applied to the plate it will strain elastically until the stress reaches the yield point. Then it will strain plastically. When the pressure is relieved only the plastic strain will remain. The limit to the elastic strain equals (yield stress)/(modulus of elasticity). Before the material has strain hardened this equals $35,000/(29 \times 10^6) = 0.00121$ or 0.121%. The total strain then equals measured plastic strain plus relieved elastic strain. The plot of yield strength vs. strain, Figure 8, utilizes total strain. In Table 1 the total strain equals plastic strain from Figure 10 plus (corrected yield strength divided by E). To be completely accurate the new value of total strain should be used to determine a new, more accurate value of yield strength, a new elastic strain, and a new total strain. However, this refinement would not add appreciably to the accuracy and was not done.

The pressure P to produce the given deflection was calculated from the hoop stress equation, $S = PR/t$ where:

S = yield strength corresponding to the amount of total strain to which the steel has been subjected.

R = radius of curvature corresponding to total strain assuming a circular arc.

t = thickness of steel sheet = 0.125.

This gives $P = 0.125 \times S_y/R$.

The calculated values of the pressure required to cause yielding of the sheet are tabulated in Table 1. These values are plotted against the permanent deflection at the center of the sheet in Figure 11. The measured test pressures are also plotted in the same figure for comparison.

At low strain, low center deflection, the picture is complicated because of edge effects at the welds and the values of pressure obtained from the simplified hoop stress relationship do not match the test values. At higher strains, however, the portions of the sheet next to the welds appear to act as plastic hinges. The effects of the edge conditions are markedly reduced as the strain increases and when the strain exceeds 2% the hoop stress analysis gives values for the pressure which agree very well with the measured values.

It can be concluded from the results of this test that when a ductile material is used to form a flat panel which has a length-to-width ratio greater than 1.6, and that panel is pressurized over one face, it can deform plastically until it is stressed as a membrane. The edges will act as plastic hinges and the originally flat panel will deform into a circular shape as measured on a section through the center of the plate transverse to the longer axis. When the plastic strain exceeds 2%, the membrane stress in the plate can be calculated with reasonable accuracy with the aid of the hoop stress relationship.

The test was conducted at room temperature. However, the same conditions should obtain at the higher temperatures to which the low-temperature liner would be subjected in the event of a sodium spill. The ductility of the steel would increase at the higher temperature, and the plastic strain before failure should be greater.

Test Plate

Figure 1

Test Setup

Figure 2

Strain Pattern After 65 psi Pressure

Figure 3

Strain Pattern After 110 psi Pressure

Figure 4

Strain Pattern After 126 psi Pressure

Figure 5

strain lines not so close here

Strain lines starting to appear in blank space at top, but still noticeable blank space at bottom

Lines starting to appear at corners

Strain Pattern After 175 psi Pressure

Figure 6

Very few lines this area

Strain lines starting to appear at both ends

More lines in corners

Strain Pattern After 325 psi Pressure

Figure 7

Yield Strength of A315 Gr. 70 Steel
vs. Strain 0.2%

Figure 8

Strain - percent

21.1 - 5 Radius Curvature 15
 Strain 2.5 2.5 2.5
 Standard Error 1.0 1.0 1.0
 Figure 9

Strain - percent

Test Points

Calculated from Hoop Stress
Formula Assuming Circular Arc

Pressure to Stress to Yield
Plotted vs. Deflection AT Center

Test Points show Measured Deflection at
Center of Plate Resulting from Application
of Indicated Pressure

Plate 0.125 in. Thick, 11" x 18" Welded All Four
Edges Pressure Applied Over Entire Area

Figure 11

Permanent Deflection at Center of Plate - in.

1	2	3	4	5	6	7	8
Measured Pressure psi	Thickness At Center in.	Plastic Deflection in.	Plastic Strain %	Total Strain %	Yield strength ksi	Radius of Curvature in.	Pressure causing (calculated) psi
0	1.18	0.00	0	0	35		
40	1.22	0.04	0.0035	0.125	35	63	69
54	1.25	0.05	0.0054	0.125	35	63	71
65	1.27	0.10	0.023	0.144	35	59	74
75	1.30	0.12	0.031	0.152	35	57	77
92	1.35	0.12	0.031	0.152	35	57	77
110	1.24	0.16	0.056	0.177	35	53	80
126	1.36	0.18	0.070	0.191	35	51	82
153	1.42	0.24	0.125	0.246	35	45	87
178	1.51	0.33	0.215	0.336	35	39	112
204	1.61	0.43	0.43	0.52	36.6	31.5	117
238	1.69	0.51	0.57	0.69	37.45	27.5	120
281	1.80	0.62	0.83	0.95	37.9	23.5	122
325	1.91	0.73	1.15	1.27	42.25	20.5	128
362	2.02	0.84	1.50	1.62	45.15	18	134
400	2.07	0.89	1.72	1.94	46.73	17	137
445	2.17	0.90	2.15	2.27	49.7	15.4	143
490	2.25	1.07	2.50	2.62	52.15	14.5	150
530	2.35	1.20	3.15	3.27	55.83	12.9	157
605	2.51	1.32	4.0	4.12	59.5	11.6	161
620	2.52	failed, leak in weld					

$$\text{Hoop stress} = \frac{PR}{t} \quad \text{or} \quad P = \frac{t \times S_y}{R} = \frac{0.125 \times 51}{R}$$

where p = pressure to cause yielding

Columns 1 and 2 are measured test data

Appendix

Difference Between Lengths of Arc and Chord for 49-in. Long Pencil

$$\theta = \sin^{-1} \frac{24.5}{175.1} = \sin^{-1} 0.12557 = 7^{\circ} 12.8' \text{ or } 7.2138^{\circ}$$

$$\theta = 0.12593 \text{ radians}$$

$$\text{Arc length} = r \theta$$

$$\text{Chord length} = 2R \sin \frac{\theta}{2}$$

$$\frac{\text{Chord length}}{\text{Arc length}} = \frac{\sin \theta}{\theta} = 1 - \frac{\theta^2}{3!} + \frac{\theta^4}{5!} - \frac{\theta^6}{7!} + \dots$$

$$\frac{\sin \theta}{\theta} = 1 - \frac{\theta^2}{6} + \frac{\theta^4}{120} - \frac{\theta^6}{5040} + \dots$$

$$\text{for } \theta = 0.1259 \quad \frac{\sin \theta}{\theta} = 1 - 0.002,647 + 0.000,002,1 = 0.997,353$$

$$\frac{\theta}{\sin \theta} = 1.002,647 \text{ or arc is } 0.2647\%, \text{ longer than chord}$$

$$2a = R \cdot 2\theta$$

$$a = R \theta$$

$$\text{Chord length} = 2c = 11.0 \quad c = 5.5$$

$$c = R \sin \theta$$

$$R = \frac{c}{\sin \theta}$$

$$a = k c$$

$$k = \frac{a}{c} = \frac{\theta}{\sin \theta}$$

$$\frac{1}{k} = \frac{\sin \theta}{\theta} = 1 - \frac{\theta^2}{6} + \frac{\theta^4}{120} - \frac{\theta^6}{5040} + \dots$$

θ degrees	θ radians	k	$\sin \theta$	R in.	$\cos \theta$	b in.	Strain
1	0.017, 453	1.000, 030, 75	0.017 452	315.19	0.999, 829	0.0-7	0.005
2	0.034, 907	1.000, 263, 1	0.034 900	157.59	0.999, 091	0.096	0.020
3	0.052, 360	1.000, 457	0.052 326	105.09	0.998, 629	0.1-4	0.050
4	0.069, 813	1.000, 813	0.069 756	79.15	0.997, 56	0.192	0.10
5	0.087, 266	1.001, 270	0.087 150	63.11	0.996, 19	0.240	0.127
7	0.122, 173	1.002, 492	0.121 87	45.13	0.992, 33	0.336	0.249
9	0.157, 080	1.004, 124	0.156 43	35.16	0.987, 69	0.402	0.412
11	0.191, 986	1.006, 170	0.190 81	29.82	0.981, 63	0.529	0.617
13	0.226, 893	1.008, 602	0.224 95	24.45	0.974, 37	0.627	0.863
15	0.261, 799	1.011, 515	0.252 82	21.25	0.965, 73	0.724	1.152
20	0.349, 066	1.020, 601	0.342 02	16.08	0.939, 69	0.973	2.060
25	0.436, 332	1.032, 450	0.422 62	13.01	0.906, 01	1.219	3.245
30	0.523, 599	1.047, 123	0.500 00	11.00	0.866, 03	1.474	4.720
35	0.610, 865	1.065, 011	0.573 58	9.59	0.819, 15	1.734	6.501

$$\sin \theta = \theta - \frac{\theta^3}{6} + \frac{\theta^5}{120} - \frac{\theta^7}{5040} + \dots$$

$$\cos \theta = 1 - \frac{\theta^2}{2!} + \frac{\theta^4}{4!} - \frac{\theta^6}{6!} + \dots$$

$$b = R(1 - \cos \theta)$$

PERMEABILITY OF THE REACTOR-CAVITY WALL

Lloyd M. Polentz

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I. INTRODUCTION

It has been hypothesized that an HCDA could be followed by an occurrence in which the melted fuel could melt through the bottom of the reactor vessel and the guard vessel. Hot, liquid sodium could then flow out of the reactor vessel. It is also theoretically possible that the sodium could then leak through the skirt of the guard vessel and find its way into the reactor cavity. If such a sequence of events should occur, liquid sodium could rise to a height of 16.5 feet in the reactor cavity, to an elevation of 505.5 feet. This assumes that the sub-cavity has been filled with solid material, i.e., concrete or a refractory material. See Figures 1 and 2.

Hot liquid sodium would be in contact with some 1224 ft² of low-temperature liner which, at present is not vented. The heat from the hot sodium would flow through the steel liner plates into the concrete behind them. This heat would cause the moisture contained in the concrete to flash into steam. If the steam did not escape as rapidly as it was generated, the pressure would rise until the liner plates ruptured. It has been suggested that the steam might escape through the concrete. To check the validity of this assumption it has been proposed that a test be made of the permeability of the concrete.

It has been proposed that air under pressure be introduced into the space between a liner plate and the concrete wall behind it. By measuring the rate at which the air flows through the concrete at a particular supply pressure, an estimate can be made of the permeability of the concrete. The rate at which steam could be expected to flow out through the cavity wall at a given pressure and temperature can then be estimated.

The area of particular interest is the portion of the low-temperature liner between elevations 495'-6 and 505'-6, that portion of the low-temperature liner which is wetted by hot, liquid sodium. The location of this section of liner is shown in Figures 1 and 2, and the details of attachment are shown in Figure 1.

II. CONCLUSIONS

- A. It is estimated that the leak rate of steam through the wall of the reactor cavity would be on the order of 0.27% or less (1/372) than would be required to vent the steam at the maximum rate at which it would be produced behind the submerged portion of the low-temperature liner as a result of an HCDA and subsequent melt-through of the reactor vessel and guard vessel.
- B. A simple non-destructive test can be conducted to determine the leak rate of air through the reactor cavity wall. The low-temperature liner panels 49 in. x 49 in. will safely withstand a pressure of 3 psig.
- C. The correlation between the leak rate through concrete of air at isothermal conditions and of steam at varying temperature and moisture content is not known. This introduces an unknown factor in the application of a measured rate of air leakage to the problem of steam flow through the reactor cavity wall.
- D. Published information states that porosities of different concretes may vary by a factor of a million or more depending upon the W/C ratio, length of curing time, and other factors. This wide variation indicates that the rate of leakage of air through one portion of the reactor-cavity wall would not necessarily be a reliable indication of the rate of leakage through another part of the reactor-cavity wall.

III. DISCUSSION AND ANALYSIS

A. Permeability of Concrete

Concrete is made up primarily of cement paste and aggregate components. In addition it contains a void component composed of air voids, "under-aggregate" fissures, and small pockets of water. The result is a somewhat porous structure through which air or steam can permeate.

As the hydration of the cement proceeds, the volume of the space initially occupied by water (volume of voids) is reduced. The reduction in volume results from the fact that the hydrated "gel" has a volume which is greater than the unhydrated cement. Thus hydration reduces the size and volume of the capillary space in the paste. If the original capillary space is low (water-cement ratio of about 0.35 by weight), the volume of the gel resulting from complete hydration will be sufficient to fill the original capillary space. At higher water-cement ratios (the water-cement ratio of the concrete used in the reactor-cavity wall was approximately 0.47) the volume of the gel will not be sufficient to completely fill all of the original water space. The resulting voids and capillaries provide passageways through which air or steam can migrate through the concrete.

Verbeck, Reference No. 1, explains that the capillary system is composed of submicroscopic voids randomly distributed throughout the hydrated cement paste matrix. The voids are of a wide variety of sizes and shapes, and probably range in diameter from 0.000,008 to 0.013 mm (3×10^{-7} to 5×10^{-3} in.). The gel will also contain pores estimated to be from 10^{-6} to 8×10^{-6} mm in diameter (4×10^{-8} to 3×10^{-7} in.). The volume of the gel pores in the cement paste increases with continued hydration of the cement while the volume of the capillary spaces decreases.

The permeability of concrete can vary over a wide range. Verbeck estimates that there may be a millionfold difference between concrete

with a W/C ratio of 0.8 cured for 7 days and concrete with a W/C ratio of 0.4 cured for one year. The permeability can also be greatly affected by the presence of cracks in the concrete which occur as a result of curing or of tensile strain.

Tests have been made of the permeability of the concrete (References Nos. 2 and 3) and the results of these tests indicate the range of magnitude of the leak rate we might expect to obtain by applying air under pressure to the wall of the reactor cavity. One test was conducted with air and concrete slabs 6 in. and 12 in. thick. The Water/Cement ratio was about 0.6. The sample slabs were cured for 28 days and then temperature cycled before the leak rate was measured.

Small additions of Pozzolith and Plastiment were made to different sample slabs. These additives are water extenders, i.e., they increase the workability of the concrete without requiring that extra water be added. Through their use a relatively low W/C ratio can be maintained at the same time that good workability is achieved. This gives a denser, stronger concrete. Otherwise they do not affect the properties of the concrete. A Pozzolith additive was used in the reactor-cavity wall concrete.

The leak rate was checked by applying differential pressures of 0.25 and 5.0 psi across the slabs and measuring the rate of an air flow through those slabs. The measured rates of leakage of 75°F air in $\text{ft}^3/\text{hr-ft}^2$ were reported as follows:

Slab No.*	Differential Pressure	
	<u>0.25 psi</u>	<u>5.0 psi</u>
1	0.0049	0.048 $\text{ft}^3/\text{hr-ft}^2$
2	0.0047	0.062
3	0.035	0.822
9*	0.119	5.78
2a	(no leak)	0.0198
2b	0.00084	0.0181
3a	0.0107	0.334

- * Slabs Nos. 1, 2, 3, 2a and 3a were 6.0 in. thick, slab No. 2b was 12.0 in. thick. Slab No. 9 was an old piece of concrete about 8 years old and was roughly 3.5 in. thick, composition and W/C not known.

The investigator of this test concluded that flow of the air through the concrete was in the transition region between laminar and molecular flow.

The leakage rates of air through 4.0-in. thick slabs of concrete with 2 ounces of Plastiment admixture per sack and of different W/C ratios were reported in Reference No. 2. These investigators concluded that the rate of flow of air through the concrete conformed to the relationship--

$$\text{cfm/ft}^2/\text{in.} = AP \text{ where } P = \text{pressure in inches of water.}$$

Two values of the Constant A were determined:

<u>W/C</u>	<u>A</u>
0.35	2.8×10^{-6} cfm/ft ² -inch of water/inch thickness of concrete
0.42	2.0×10^{-5} cfm/ft ² -inch of water/inch thickness of concrete

The test slabs were 4.0 in. thick. The curing time is assumed to have been 28 days.

These investigators concluded that the flow of air through the concrete was laminar and that the measured flow rate constant is applicable over the range of differential pressure from 0 to 20 psi.

The results of the two investigations can be compared by converting the flow rates to cfm/ft²-psi/ft by assuming that the laminar flow relationships hold, i.e., flow is proportional to the differential pressure and inversely proportional to the thickness of the concrete.

The data for the two tests of each sample are averaged.

For the 0.25 psi tests of 6-in. thick slabs the measured flow rate in ft³/hr would be multiplied by 4.0 to convert to 1.0 psi, divided by 2.0 for an equivalent concrete thickness of one foot, and divided by 60 to convert from ft³/hr to CFM: $4/(2 \times 60) = 0.0333$.

For the 5.0 psi test results of the 6-in. thick slabs the multiplying factor would equal $1/(5 \times 2 \times 60) = 0.00167$.

For the leak rates reported in Reference No. 2 the multiplying factor would equal $27.69(\text{in. of water/psi}) \times 1/12 (\text{in./ft of thickness}) = 2.308$.

The leakage rates reported for the different samples convert to the values in the table below.

<u>Leakage Rates in $\text{cfm/ft}^2\text{-psi/ft}$</u>		
<u>Sample</u>	<u>Reference No.</u>	<u>Leakage Rate ($\text{cfm/ft}^2\text{-psi/ft}$) $\times 10^6$</u>
1		122
2		130
3		1,269
9		3,967
2a		16.5
2b		58
3a		457
W/C - 0.35	2	6.46
W/C - 0.42	2	46.2

The widest range of variation occurs between the leakage rate of $3.967 \times 10^{-6} \text{ cfm/ft}^2\text{-psi/ft}$ for the slab of unknown properties and the leakage rate of $6.46 \times 10^{-6} \text{ cfm/ft}^2\text{-psi/ft}$ for the slab with the W/C ratio of 0.35. The larger measured leakage rate equals $3,967/6.46 = 614$ times the least leakage rate.

The markedly greater leakage rate through sample no. 9 in spite of the fact that the hydration of the cement in that sample would have been complete, indicates that there must have been cracks or fissures in that slab or perhaps the W/C ratio was excessively large. The greatest variation between the rates of leakage through sample slabs of known composition made under controlled conditions equals $1269/6.46 = 196$.

Using the largest leakage rate for the concrete slab prepared under controlled conditions, the expected leakage rate of air through

the wall of the reactor cavity in the area behind a low-temperature panel for the assumed test pressure of 3 psi would equal--

$$Q = (1,269 \times 10^{-6} \text{ cfm/ft}^2 \text{ psi/ft}) \times (3 \text{ psi}) \times (1/4\text{-ft thickness}) = 0.000,951 \text{ cfm/ft}^2$$

where the flow rate of the air is measured at 70°F and one atmosphere.

III-B Comparison of Steam Flow and Air Flow Through Concrete

The leakage of air through concrete would take place through the pores of the concrete and whatever cracks that might be present. The reactor cavity wall was poured and cured under controlled conditions and the concrete was vibrated into place. The wall is not subject to tensile stress under ordinary conditions. It is, therefore, doubtful if there would be any large number of cracks or an appreciable flow area through cracks in the wall.

The mean free path of air molecules at standard conditions is on the order of 4×10^{-6} inch. Molecular flow occurs when the diameter of the tube through which the gas is flowing is about equal to or less than the mean free path. The flow area which would result from such small diameter capillaries would be inadequate to handle the rate of flow that would be necessary to vent the steam produced from the moisture in the reactor cavity wall. Moreover, some portion of the steam would condense in the concrete when the temperature of the concrete lowered below 212°F. This would produce moisture in the pores. Blanks, Reference 4, states that pores less than four microns (157×10^{-6} inch) in diameter will drain effectively only at pressures high enough to cause failure of some rocks and concrete in tension. This implies that the pressure which would be required to force the condensed steam out of the pores would be much greater than was available. In effect, then, the pores in the outer foot or two of the reactor cavity wall would quickly become plugged with water. There would be two results of the plugging: 1. The flow area through the concrete for the steam would be less than that for the air, and, 2. The pores through which flow would be in the molecular-flow region would be plugged in the case of steam flow. Thus, steam flow could be expected to be laminar.

Laminar flow occurs when the value of the Reynolds number is less than about 2000. It is assumed that the flow of air or steam which might take place through the reactor cavity wall which would be at a rate to be of interest in venting the steam would be in the laminar region.

The relationship for laminar flow through cylindrical tubes is represented by the equation -- $h_L = f(L/D)(v^2/2g)$ where --

h_L = frictional head loss, ft. of fluid flowing

f = friction factor = $64/\text{Reynolds number} = 64/R_e$ as used in the Darcy-Weisbach equation

L = length of flow path, ft.

D = diameter of circular cross-section tube, ft. For non-circular sections $D = 4 \times \text{hydraulic radius} = 4 \times R_H$

R_H = hydraulic radius = cross-sectional area of flow divided by wetted perimeter, ft

v = velocity of flow (average), ft/sec

g = acceleration of gravity, ft/sec²

$R_e = \rho Dv/\mu$

ρ = density of fluid flowing, slugs/ft³

D and v as described above

μ = absolute viscosity, lb sec/ft² = centipoise/47,800

The flow rates of the two fluids (air and steam) can be compared using the equations for laminar flow.

Buss, Reference No. 3, in calculating the rate of flow through cracks in concrete used the following form of relationship for isothermal flow:

$$\frac{P_1^2 - P_2^2}{2 P_1} = f \frac{L v_1^2 w_1}{D \cdot 2g}$$

w = specific weight, lb/ft³

P = pressure, lb/ft²

The subscript 1 indicates conditions at the beginning of the crack.

The relationship reduces to

$$\frac{P_1 - P_2}{w_1} \times \frac{P_{ave}}{P_1} = f (L/D)(v_1^2/2g)$$

The term to the left of the equal sign equals $(P_1 - P_2)/w_{ave.}$ or the head loss in terms of feet of fluid flowing at the average pressure. The equation is the same as the relationship given previously for laminar flow. However, Buss uses a value of $f = 1200/Re$ derived from test data for Re between 0.01 and 100. This indicates a much higher frictional resistance through concrete than through circular cross-section tubes or through slots (see below).

Grinnell, Reference 5, investigated the flow of gases through thin slots. He developed the following relationship for isothermal flow:

$$W = \frac{bt^3}{24\mu RTL} (P_1^2 - P_2^2)$$

where b = width of slot, ft

t = thickness of slot, ft

R = gas constant for the particular gas = 1544/molecular wt (approx.)

T = temperature, °R

W = flow rate, lb/sec

This relationship can be modified to give--

$$W = v \times w \times A = \frac{A \times t^2}{12\mu L} \frac{P_{ave}}{RT} (P_1 - P_2)$$

where A = flow area, $ft^2 = b \times t$

$$P_{ave} = (P_1 + P_2)/2$$

$$w = \text{average specific weight, lb/ft}^3 = P_{ave}/RT$$

The relationship can then be further modified to give--

$$\frac{P_1 - P_2}{\rho g} = \frac{12\mu L v}{\rho g t^2}$$

$$Re = \rho D v / \mu, \text{ for a thin slot } D = 4xR_H = 4 \times \frac{bt}{2b} = 2t$$

$$\text{which gives } Re = \rho \times 2t \times v / \mu$$

$$h_L = 96 \times \frac{\mu}{\rho \times 2t \times v} \times \frac{L}{2t} \times \frac{v^2}{2g} = (96/Re) \times (L/D) \times (v^2/2g)$$

or the friction factor developed by Grinnell for compressible fluid flow through a thin passage equals $96/Re$.

$$\text{From Grinnell's relationship, } v = \frac{(P_1 - P_2)t^2}{12\mu L}$$

The foregoing reports all concur that, for a given system, the velocity of isothermal flow is proportional to the pressure drop and is inversely proportional to the absolute viscosity. The relationships given are all approximate for compressible flow, but the error due to the approximation is small.

For the case under consideration, the flow of air will be isothermal, assuming that the concrete wall is at the same temperature throughout. The flow of steam through the wall as an aftermath of an HCDA followed by a fuel melt-through would not be isothermal. However, the rate of steam flow can be estimated for the conditions assumed by taking a mean temperature.

In addition to the reduction in the temperature of the steam as it flows outward through the wall, part of the steam will condense as it nears the outside of the wall. However, in order for the steam to condense it must give up the heat of vaporization. The concrete will only be able to absorb a small amount of this, so it is probable that most of the water that permeates the wall of the reactor cavity will exit in the form of vapor.

The same relationship will hold for the flow of a liquid as for a gas.

$$v = \frac{(P_1 - P_2) t^2}{12\mu L}$$

$$W = vWA = \frac{(P_1 - P_2) t^2}{12\mu L} \times WA$$

It is assumed that the flow passages are essentially the same through the thickness of the reactor cavity wall. Then, for a given system, the values for A, t, and L will remain constant. The equation for flow can be written as follows:

$$W \text{ lb/sec} = K (P_1 - P_2) w/\mu \text{ where } K \text{ is a constant.}$$

Using this relationship we can calculate the relative pressure drops through a given thickness of concrete for steam and water.

The temperature in the reactor-cavity wall behind that portion of the low-temperature liner which is in contact with liquid sodium will vary from 900°F at the inner surface to 120°F at the outer surface.

For the purpose of estimating the weight rates of flow for a given pressure drop, assume that the mean steam temperature is 550°F and the mean pressure is 10 psig. Assume the mean water temperature is 180°F. At these conditions the fluids have the following properties:

Steam @ 550°F and 25 psia $V = 22.74 \text{ ft}^3/\text{lb}$ $\mu = 0.020$ centipoise

Water @ 180°F $V = 0.01651 \text{ ft}^3/\text{lb}$ $\mu = 0.35$ centipoise

$$W = K (P_2 - P_1) w/\mu = K (P_2 - P_1)/(V\mu)$$

or the weight rate of flow is inversely proportional to $V\mu$

Steam @ 550°F & 45 psia $V\mu = 0.455$

Water @ 180°F $V\mu = 0.00578$

For water to flow through an equal distance of concrete at the same weight rate as steam, $W_W = W_S$

$$K (P_1 - P_2)_W/(V\mu)_W = K (P_1 - P_2)_S/(V\mu)_S$$

$$(P_1 - P_2)_W = (P_1 - P_2)_S \times (V\mu)_W/(V\mu)_S = 0.0127 \times (P_1 - P_2)_S$$

The pressure differential required to force water through a foot of concrete is thus only about 1.3% of the pressure differential required to force steam through the same thickness of concrete at the same weight rate of flow, assuming laminar flow. Thus it is conservative to assume that the water will remain in the form of steam throughout its passage through the cavity wall, i.e., the rate of flow of steam through the reactor cavity wall for a given pressure differential will be less if no condensation takes place except for the loss of the flow area of the smaller voids (4 microns and smaller) as previously discussed.

Compare the rate of steam flow to the rate of air flow.

Assume a differential pressure for steam of 20 psi with $V = 22.74 \text{ ft}^3/\text{lb}$ and $\mu = 0.020$ centipoise.

Assume a differential pressure for air of 3 psi with $V = 11.88 \text{ ft}^3/\text{lb}$ (av. pressure = 16.2 psia and $T = 60^\circ\text{F}$) and $\mu = 0.018$ centipoise.

Required flow ratio of air to steam--

$$\frac{W_A}{W_S} = \frac{K \times 3}{11.88 \times 0.018} \times \frac{22.74 \times 0.020}{K \times 20} = 0.319$$

It is estimated that the maximum rate of steam release behind the low temperature liner will equal 5 lb/hr-ft², Reference No. 7. This would produce a required steam flow rate of $5 \times 22.74/60 = 1.90$ ft³/min.-ft² or 0.0833 lb/min.-ft². The steam would have to flow through the concrete wall at this rate under the influence of a 20 psig back pressure.

The required flow rate of air under a 3 psi differential pressure corresponding to a flow rate of steam under a 20 psi differential pressure at the assumed conditions equals $W_A = 0.319 \times W_S$. So the measured air flow would have to equal

$$0.0833 \times 0.319 = 0.0266 \text{ lb/min. ft}^2 \text{ or}$$

$$0.0266/0.0750 = 0.354 \text{ cfm/ft}^2 \text{ at one atmosphere and } 70^\circ\text{F}$$

or $0.0266/0.0902 = 0.295 \text{ cfm/ft}^2$ at the flow conditions of 3 psig and 70°F.

The required flow rate of 0.354 cfm/ft² is more than 372 times the maximum expected flow rate of 0.000951 cfm/ft² based on reported test results.

III-C. Air Test Considerations

The low-temperature liner panels below the 499 ft. elevation have a maximum size of 30 in. x 49 in. The low-temperature liner panels at the 505 ft. elevation are approximately 49 in. x 49 in. The larger panels are more critical stresswise.

Assuming a flat, square plate with edges fixed and a uniform load over the surface, from Roark p. 225, Case No. 34--

$$S_{\text{Max}} = 0.308 w a^2/t^2$$

w = pressure loading, psi

a = 49 in.

t = thickness = 0.25 in.

$$S = 0.308 \times p \times 49^2/0.25^2 = 11,832 \times p$$

The panels are made of A516 Grade 55 steel (low-temperature liner by Bechtel). This steel has a yield strength at room temperature of 30,000 psi. The pressure to produce a stress of 30,000 psi equals 2.54 psi. However, the curvature will supply added strength.

For a curved panel under uniform radial pressure, curved edges free, straight edges clamped (Roark p. 354, Case No. 34) --

$$p = \frac{E t^3 (k^2 - 1)}{12 r^3 (1 - \nu^2)}$$

p = critical pressure (for instability)

E = 30×10^6 psi

t = thickness = 0.25 in.

r = radius of curvature = 216 in.

ν = Poisson's ratio = 0.3

k = constant where $k \tan \alpha \cot k\alpha = 1$

α = half central angle

For 49 inch arc $\alpha = 49/(2 \times 216) = 0.1134$ radians or 6.5°

k = 39.5 ($39.5 \times 0.11394 \times \cot 256.75^\circ = 1.06$)

$$p = \frac{30 \times 10^6 \times 0.0156 (1560-1)}{12 \times 10.03 \times 10^6 \times 0.91} = 6.63 \text{ psi}$$

For the case where the curved edges are free and only the straight edges are welded, the panel would withstand a pressure differential of 6.63 psi before buckling. With the curved edges held (welded) the panel would withstand an even greater pressure without buckling.

Based on the above analysis it is concluded that the 49-in. square, curved panels should be able to safely withstand a pressure differential of 3.0 psi. This is the recommended test pressure for air at ambient temperature.

The maximum safe back pressure for the steam at the sodium-spill conditions can be determined by means of a failure analysis.

When the pressure behind a panel rises to the point that the stress in the panel exceeds the yield strength the panel will deform and the center of the panel will belly out. The shape of the curve would be essentially circular, and the stress in the panel can be

estimated using the hoop-stress formula. Referring to Figure 3, for a given strain, $R = L/\sin \alpha$. The hoop stress --

$$S_H = PR/t \quad \text{The panel thickness is 0.25 in.}$$

$$S = P(L/\sin \alpha)/0.25. \quad \text{For a 49 in. x 49 in. panel, } L = 24.5$$

$$S = 98 p/\sin \alpha.$$

The estimated tensile strength of the panel steel (A516 Grade 55) at the spill temperature of 1600°F is 3960 psi. Then

the maximum allowable short-term pressure across a 49-in. square panel would equal

$$p = (3960/98) \sin \alpha = 40.41 \sin \alpha.$$

From Figure 3 --

Strain percent	α radians	$\sin \alpha$	Pressure to Produce a Membrane Stress of 3960 psi
5	0.54	0.514	20.8 psi
6	0.59	0.556	22.5
7	0.63	0.589	23.8
8	0.675	0.625	25.3
9	0.715	0.656	26.5
10	0.75	0.682	27.6

The panel steel has an estimated total elongation capability of more than 85% at 1600°F. However, some portion of the weld would undoubtedly fail before this amount of elongation took place. There is another complication here and that is that the strain would be biaxial. Tests have shown that metals subjected to biaxial tensile strain demonstrate unstable plastic flow. The condition of instability is at a maximum when the transverse strains are equal, Reference 6, as they would be for the square, low-temperature panels. The result of the biaxial strain is a premature necking-down and failure at a considerably lower strain than would cause failure in the same metal as the result of uniaxial strain. Moreover, the localized strain can be considerably greater than the average strain. An approximate relationship (Reference 6) for the maximum strain at the crown of a bellied-out section is given by the relationship n^2/L^2 (quantities as defined in Figure 2). For the case of 5% elongation, $h/L = 0.275$ and the maximum strain equals 7.6%.

Because of these two complicating factors it is desirable to limit the apparent strain in the low-temperature panels to 5%. This in turn limits the pressure across the panel to 20 psi when the temperature is 1600°F.

The maximum permissible pressure due to the steam at the sodium spill conditions would thus equal 20 psig. The purpose of the test is then to estimate the rate at which steam (at the sodium spill conditions) would permeate the wall under the influence of a 20 psi differential pressure with the aid of the measured rate of airflow through the wall with a 3 psi pressure differential. The relationship for the relative flow rates has been developed in a previous section.

The flow rate of air through the reactor cavity wall due to a 3 psi pressure differential would have to equal 0.354 cfm/ft^2 to guarantee that the steam would flow through the wall behind the low-temperature liner at the maximum rate at which it would be produced following an HCDA and a melt-through. This assumes that the laminar flow relationships apply. The rate of airflow behind a 49 in. x 49 in. panel would have to equal $0.354 \times 49 \times 49/144 = 5.90 \text{ cfm}$ (measured at 70°F and one atmosphere).

III-D. Test Setup

Each panel in the area of maximum steam generation is welded to I-beam embedments on all four edges, see Figure 1. The length of the flow path around a 3-inch I-beam embedment is approximately one foot, while the flow path through the wall is four feet. In addition, it is possible that the concrete drew away from the embedments a small amount as it set. Thus, the path of least resistance for airflow would be around the embedments to which the low-temperature panels are welded. To prevent the flow of air around the embedments from the space behind the test panel to the spaces behind the adjacent panels it is only necessary to eliminate the pressure drop across the flow paths around the embedments. This can be accomplished most easily by pressurizing the spaces behind the panels which abut the test panel to the same pressure as that behind the test panel. The recommended test arrangement is shown in Figure 4.

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support
Ledge — Elev. 531' 1¹³/₁₆

← Radius 18'-0

EFCO
Liner

↑
— Elev. 513'-4
↓
Bechtel
Liner

Sodium Level
505' 6 Elev.

Section Through Reactor Cavity
Wall showing Locations of Liners

— Elev. 499'-0

Low-Temperature
Liner

↑
— Elev. 495'-6
↓
High-Temperature
Liner

← Radius 16'-3

— Elev. 489'-0

Reactor Cavity Floor

Figure 2

Test Arrangement For Determining Permeability
of Reactor Cavity Wall To Air

Figure 4

2-3-75

LMP

From: Reactor Safety
Date: October 2, 1974
Subject: EFFECT OF 103,000 LB SPILL OF 1050°F
SODIUM ON THE LOW-TEMPERATURE LINER
IN THE REACTOR CAVITY

Hanford Engineering Development Laboratory

To: R. K. Brown

cc: JP Hale DE Simpson CC Steele
 DI Hulbert RJ Slember AB Webb
 RD Peak DD Stepniewski SA Weber LMP File/LB

References:

1. "Some Problems in Unstable Plastic Flow under Biaxial Tension," W. T. Lankford and Edward Saibel, Trans. AIME, Vol. 171, 1947.

It is hypothesized that 108,000 lb of 1050°F sodium could spill into the reactor cavity. Some of the heat contained in the sodium would flow into the liner and walls of the reactor cavity. The resulting increased temperatures would cause a reduction in the strength of the liner and could cause the moisture contained in the concrete wall to flash into steam. If the steam pressure were high enough, and the liner weak enough, failure of the liner could result. The purpose of the attached analysis is to determine whether or not there is any danger of failure of the liner as a result of such a hypothesized spill.

The low-temperature liner would safely withstand the effects of the increased temperature and pressure resulting from the hypothesized 108,000 lb spill of 1050°F sodium as determined by a CACECO calculation by R. D. Peak of HEDL. The individual panels of the low-temperature liner would be stressed beyond their yield point, but the plastic strain would be quite low, probably less than 1.0%. The low-temperature liners are made of ASTM A516 Grade 55 steel. This steel has a minimum total uniaxial elongation of 16% at 300°F.

Based on the calculation, the temperature of the concrete behind the low-temperature liner will level off at a maximum which is slightly less than 250°F. The temperature of the low-temperature liner will have dropped to less than 300°F when the temperature of the concrete wall behind it has reached 212°F, see Figure 1.

R. K. Brown

-2-

October 2, 1974

The vapor pressure of water at 250°F is 29.8 psia. Even if the air originally trapped in the space behind the low-temperature liner remains in that space and increases in pressure at constant volume from an assumed 70°F at construction to the 300°F liner temperature, the total pressure would only equal 50.9 psia or 36.2 psig. This is well below the maximum pressure the low-temperature liner can withstand without failure.

A detailed analysis is attached.

L. M. Polentz
L. M. Polentz

LMP:10

Attachment

ANALYSIS OF EFFECT OF A 108,000 LB SPILL OF 1050°F SODIUM ON THE LOW-TEMPERATURE LINER

Lloyd M. Polentz

After the hypothesized spill of 1050°F sodium, the sodium would rise to a height of 2.6 ft above the floor of the reactor cavity, Reference 3. Heat from the sodium would be absorbed by the walls, liners, and other portions of the reactor cavity structure. Due to the transferal of heat, the temperature of the sodium would reduce while the temperatures of the different portions of the reactor cavity structure would rise. A plot of the temperatures of the sodium pool, the low-temperature liner, and the outer face of the concrete wall behind the liner vs. time from spill is shown in Figure 1. The data for the plot was obtained from a CACECO code calculation by R. D. Peak of HEDL.

From Figure 1 it can be seen that the temperatures of the liner and the concrete behind it are converging and will equalize at some temperature below 250°F. The maximum temperature of the concrete can then be conservatively taken as 250°F. Similarly, the temperature of the liner will be below 300°F when the temperature of the concrete rises to the point where the pressure of the steam produced from the moisture in the concrete exceeds one atmosphere. It is then conservative to assume that the temperature of the liner is 300°F.

Taking the extreme case of 250°F moisture temperature, the vapor pressure of the steam will equal 29.8 psia. It can further be assumed that, in the extreme, the air enclosed behind the liner panels when the concrete was poured remained trapped in the small space. If the original temperature was 70°F and the temperature of the air rises to 300°F, the partial pressure of the air would rise to $(760/530) \times 14.7 = 21.1$ psia. The total pressure behind the liner could then rise as high as 50.9 psia, or 36.2 psig. Actually the liner would strain as it was stressed and the volume behind it would increase. The increase in volume would reduce the partial pressure of the air to perhaps 10 psia or below. However, the ability of the liner plates to withstand the total pressure differential of 36.2 psig will be determined.

Calculations of the ratio of the height of rise of the center of a flat plate to half the width for a range of strains are given on the attached calculation sheet. The analysis assumes a circular deformation of the plate. A plot of the

ratio h/L and strain in the plate vs. half angle of curvature is given in Figure 2.

The critical panel size of the low-temperature liner is 30" X 41", which gives $L = 15"$. The thickness is 0.25 inch.

For ASTM A516 Grade 55 steel at 300°F--

Yield strength = $0.93 \times 30,000 = 27,900$ psi

Ultimate tensile strength = 55,000 psi

Total elongation = 16%.

Hoop stress for a thin-walled circular section = PR/t .

For a flat plate deformed plastically due to pressure acting on one side, the stress due to the pressure is approximately equal to the hoop stress if the plastic strain equals 2% or more.

The pressure required to produce a hoop stress equal to the yield strength of the panel steel has been calculated for a number of different strains. The values are tabulated on the second calculation sheet. The calculations show that for a 2% strain in the shorter dimension, a pressure differential across the plate equal to 157 psi would be required to stress the panel to its yield point. For this condition the ratio of transverse strain to longitudinal strain equals 1.87, or almost 2, so the effect of biaxial strain would be negligible, see Reference 1.

Calculation of the Ratio h/L for Different Strains in a Flat Piston

C-1

LVP

$$R \sin \alpha = L$$

$$R = \frac{L}{\sin \alpha}$$

$$a = R \alpha = L + \Delta L$$

$$h = R - B = R(1 - \cos \alpha)$$

$$\text{Strain} = \text{percent Elongation} = \Delta L / L$$

$$\Delta L / L = (R \alpha - L) / L = R(\alpha - \sin \alpha) / R \sin \alpha$$

$$\Delta L / L = (\alpha / \sin \alpha) - 1 = \frac{1}{(\sin \alpha / \alpha)} - 1$$

$$h / L = \frac{R(1 - \cos \alpha)}{R \sin \alpha} = (1 - \cos \alpha) / \sin \alpha$$

$$\frac{\sin \alpha}{\alpha} = 1 - \frac{\alpha^2}{3!} - \frac{\alpha^4}{5!} - \frac{\alpha^6}{7!} - \dots = 1 - \frac{\alpha^2}{6} - \frac{\alpha^4}{120} - \frac{\alpha^6}{5040} - \dots$$

α radians	$\frac{\sin \alpha}{\alpha}$	Strain %	α degrees	$\cos \alpha$	$\sin \alpha$	$\frac{h}{L}$	
0.01	0.999,983	0.00167	0.573	0.999,95	0.01000	0.00500	
0.03	0.999,850	0.01500	1.719	0.999,55	0.03000	0.01500	
0.05	0.999,583	0.04117	2.865	0.999,75	0.049,98	0.02501	
0.07	0.999,184	0.0817	4.011	0.997,55	0.069,94	0.02503	
0.10	0.998,334	0.167	5.730	0.995,00	0.099,83	0.03509	0.1
0.15	0.996,254	0.376	8.594	0.989,77	0.149,44	0.07515	0.15
0.2	0.993,347	0.670	11.459	0.980,07	0.198,67	0.10032	0.2
0.25	0.989,616	1.049	14.324	0.968,01	0.247,40	0.12567	0.25
0.3	0.985,068	1.516	17.189	0.955,34	0.295,52	0.15112	0.3
0.4	0.973,546	2.717	22.918	0.921,06	0.389,42	0.20271	0.4
0.5	0.958,954	4.291	28.648	0.877,58	0.479,43	0.25534	0.5
0.6	0.941,071	6.262	34.377	0.825,34	0.564,64	0.30933	0.6
0.7	0.920,311	8.659	40.107	0.764,34	0.644,22	0.36503	0.7
0.8	0.896,695	11.521	45.837	0.696,71	0.717,36	0.42279	
0.9	0.870,362	14.894	51.566	0.621,61	0.783,33	0.48305	
1.0	0.841,468	18.840	57.296	0.540,20	0.841,47	0.54631	
0.74	0.911,124	9.754		0.738,47	0.674,29	0.38786	0.74

For Low-Temperature Liner, Panel Size 30" x 41"

Transverse strain ϵ_t	α radians	R in.	p psi	Longitudinal strain ϵ_l	Transverse $\frac{h}{L}$	Longitudinal $\frac{b}{L}$
0.01500	0.03	500	19.35		0.0150	0.011
0.04117	0.05	300	23.24		0.0250	0.018
0.0817	0.07	214.5	32.5		0.0350	0.026
0.2	0.11	127.9	50.6		0.055	0.040
0.3	0.135	111.4	62.6	0.167	0.068	0.050
0.5	0.175	86.1	81.0	0.27	0.086	0.063
1	0.245	61.3	113	0.38	0.102	0.075
2	0.345	44.4	157	1.07	0.173	0.127
3	0.42	36.3	170	1.60	0.210	0.154
4	0.48	32.5	215	2.13	0.245	0.179
5	0.54	27.8	251	2.75	0.276	0.202

For $S = 27,900$

$S = pR/t$

$p = 27,900 \cdot 1.25 / R = 6975 / R$

Case 462

108,000 Pound Spill of 1050°F Sodium Into Reactor Cavity
Plot of Temperatures Vs. Time from Spill

Figure 1

h/L

Plot of $\frac{h}{L}$ and strain
vs. Angle α

RECEIVED

APR 15 1975

D. D. STEPNEWSKI

From: Reactor Safety
Date: April 15, 1975
Subject: REACTOR-CAVITY LINER SELF-VENTING
AIR-FLOW EQUIVALENCY TEST

Hanford Engineering Development Laboratory

To: R. K. Greenwell

cc: JC Chandler DE Simpson
JP Hale RJ Slember
DI Hulbert DD Stepnewski
HD Lenkersdorfer LMP File/LB

In response to your request I have calculated the required rate of air flow past an embedment assuming that the EFCO liners are vented. The required rate of air flow at 3.0 psi differential pressure across an embedment is estimated to equal 9.92 SCFH per linear foot of embedment. It should be emphasized that this is an estimate based on a number of simplifying assumptions. If the result of the proposed air test indicates that self-venting of the reactor cavity liner is a possibility a more detailed and sophisticated analysis will be required.

The estimated rate of steam generation behind the liner panels is taken from CACECO Code Case No. 502, April 9, 1975, by R. D. Peak. Tabulated data from this run are attached. The maximum indicated rate of steam generation occurs during the first increment of time and equals 2.27 lb/hr-ft² for heat structure no. 11. To allow for indeterminacies, etc., this rate was doubled to give the rate at which steam would have to be vented. At the time considered no steam was generated behind the reactor-cavity liner at elevations above the top of the sodium pool. The rates at which the steam is generated behind the panels for a one-foot-wide strip are shown in the attached figure.

A number of assumptions were made to simplify the analysis. The assumptions were as follows:

- Steam flow is laminar and can be represented by the laminar-flow relationship -- $h_L = (\text{constant}/\text{Re})(L/D)(v^2/2g)$ -- which, for this case, for flow across each embedment, reduces to $\Delta P = \text{Constant} \times W$ lb/sec flow rate.
- The properties of the steam, including specific volume, were assumed constant over the flow path. Actually, the specific volume would change considerably over the flow path as the pressure reduced from the maximum of 34.7 psia and the effects will have to be checked before a final, favorable, decision can be made regarding the feasibility of self-venting.
- The flow paths past all embedments were assumed to be identical.

April 15, 1975

The maximum tolerable pressure behind a panel is taken as 20 psig. The pressure is assumed to drop to 1.0 psig at the EFCO panel vent line and to 6.0 psig in the space behind the high-temperature liner. Calculations show that part of the steam produced behind the liner between elevations 499'-6 and 503'-1 would vent upward and part would vent downward. The steam flow distribution and the calculated pressure drops across the different embedments are shown in the attached figure. The result of the analysis is that a flow rate of 0.00136 lb/sec would be required across each embedment per psi of pressure drop per linear foot of embedment.

An equivalent flow rate of air can be determined with the aid of the relationship for flow of gas through a thin slot. For small pressure drops this relationship can be written--

$$h_L = \frac{96\mu}{\rho D V} \times \frac{L}{D} \times \frac{v^2}{2g} = \frac{48\mu L V}{W D^2}$$

The passage is the same for steam flow as for air flow, so L and D are constant. W, the flow rate in lb/sec, equals $v \times A \times w$ (velocity x area x specific weight).

$$h_L \times A \times w = k\mu(vAw)/w \quad \text{where } k = \text{a constant}$$

$$h_L \times w = \Delta P \quad \text{so } \Delta P \times A = kWu/w$$

$$\text{or } W \text{ lb/sec} = \text{constant} \times W\Delta P/\mu$$

$$\text{For the same pressure drop } \frac{W_{\text{air}}}{W_{\text{steam}}} = \frac{W_{\text{air}}}{\mu_{\text{air}}} \times \frac{\mu_{\text{steam}}}{W_{\text{steam}}}$$

Assume steam at the mean pressure of 10 psig and 300°F.

$$\text{Then } w_{\text{steam}} = P/RT = 0.0329 \text{ lb/ft}^3 \quad \mu_{\text{steam}} = 0.024 \text{ centipoise.}$$

$$\text{For air at } 70^\circ\text{F} \quad w_{\text{air}} = 0.075 \text{ lb/ft}^3 \quad \mu_{\text{air}} = 0.018 \text{ centipoise}$$

$$\frac{W_{\text{air}}}{W_{\text{steam}}} = \frac{0.075}{0.018} \times \frac{0.024}{0.0329} = 3.040$$

For the same pressure drop across an embedment the weight rate of air flow would be 3.04 times the weight rate of flow of steam. For a one psi drop the flow rate of air corresponding to the required flow rate of steam of 0.00136 lb/sec would equal 0.00413 lb/sec or 0.248 lb/min. This would increase to 0.744 lb/min for a 3 psi drop for laminar flow.

$$(0.744 \text{ lb/min})/(0.075 \text{ lb/ft}^3) = 9.92 \text{ SCFM.}$$

L. M. Polentz

*Steam Flow Per Circumferential Foot of Reactor Cavity Liner
At 0.2031 Hour*

Flow Rate of Steam

Past Embedments = 0.00136 lb/sec-psi per Foot of Embedment

Rate of Generation of Steam In Reactor-Cavity Wall

Flow Rate and Pressure Drop of Steam Past Embedments

Rate of Evolution of Water from Reactor Cavity Structure
Following an HCDA and Fuel Melt-Through
CACECO Code Case No. 502 4-9-75

Time hrs.	Avg Rate of Steam Evolution g/hr-ft ²						
	Heat Structure Number						
	7	9	11	13	15		
0.2031			2.271	2.293	2.308		
0.3051			2.042	2.035	2.040		
0.3995			1.791	1.614	1.652		
0.5023			1.570	1.571	1.526		
0.6023			1.503	1.503	1.560		
0.6063			1.495	1.485	1.471		
0.7103			1.452	1.373	1.403		
0.8063		0.360	1.293	1.297	1.360		
0.8143		0.367	1.257	1.226	1.230		
0.9153		0.179	1.279	1.271	1.276		
0.9927	0.357	0.316	1.172	1.166	1.189		
1.202	0.210	0.310	1.171	1.075	1.100		
1.406	0.273	0.307	1.038	1.024	1.036		
1.502	0.260	0.235	0.971	0.977	0.977		
1.610	0.286	0.319	0.977	0.996	0.992		
1.814	0.254	0.321	0.978	0.859	0.901		
2.006	0.286	0.319	0.878	0.895	0.897		
2.018	0.284	0.320	0.874	0.891	0.892		
2.222	0.283	0.332	0.822	0.820	0.821		
2.426	0.280	0.307	0.839	0.754	0.774		
2.510	0.270	0.294	0.820	0.821	0.826		
2.630	0.280	0.316	0.792	0.790	0.790		
2.834	0.270	0.264	0.747	0.740	0.741		
3.014	0.240	0.300	0.703	0.702	0.702		
3.039	0.265	0.303	0.711	0.697	0.697		
3.242	0.273	0.302	0.726	0.724	0.725		
3.416	0.254	0.291	0.670	0.670	0.682		
3.518	0.250	0.277	0.677	0.674	0.676		
3.610	0.275	0.310	0.654	0.655	0.655		
3.854	0.257	0.283	0.611	0.637	0.642		
4.022	0.250	0.273	0.600	0.655	0.655		
4.057	0.251	0.276	0.651	0.650	0.650		
4.207	0.262	0.266	0.600	0.673	0.670		

SWELLING RATES AND DEFECT PRODUCTION RATES (NORMALIZED) FOR AT TYPE 316 SS

SOURCE	EXPOSURE RATE, cpa/s	TEMP, °C	V/V ₀ % PLR cpa	DEFECTS/cpa	
				TOTAL	MOBILE
NEUTRONS E 0.05	1x10 ⁻⁶	520	1.0	1.0	1.0
NEUTRONS 1 MeV	2x10 ⁻²	525	0.5	1.1	1.5
PROTONS 0.2 MeV	2x10 ⁻⁴	500	12 (9.6)	1.2 (1.6)	5.2 (6.7)
ELECTRONS 1 MeV	2x10 ⁻³	580	5.3	1.6	7.6

1000 100 10 1

FIGURE 24 -- Comparison of swelling rates at peak swelling temperatures with simulated defect production. Values in parentheses are somewhat more consistent with the values for electrons.