ATOMICS INTERNATIONAL
A Division of North American Aviation, Inc.

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STATEMENT OF PROBLEM

To determine the structural integrity of the bottom head of the reactor vessel during normal operation and accident transient conditions.

ABSTRACT:

The maximum stress in the bottom head exists at the outermost support pad where the following values were computed:

Mechanical stress: 8,700 psi (Allowable mechanical stress: 12,750 psi)

"Elastic" thermal stress: 35,000 psi

Cumulative fatigue damage factor: 0

It was also established that the lower head will not buckle or wrinkle because of the frictional forces developed during thermal expansion of the vessel.

02

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GENERAL REMARKS

The bottom head of the vessel is 19'-0" in diameter and is fabricated from Type 304 stainless steel. It rests on a number of support pads; one under each moderator can support rod.

The mechanical loads acting on the bottom head of the reactor vessel result primarily from the transfer of the weight of: (1) the reactor vessel; (2) the sodium coolant, and (3) the vessel internals. In addition to the above weights, the lower head is subject to the pressure forces (i.e., pump pressure) in the lower plenum. However, the effect of these loads on the plate has been minimized because of the large number of support pads.

The normal operating temperature of the lower plenum is 607 F. However, the temperature used to evaluate the integrity of the reactor vessel was chosen as 1050 F which is based on a possible accident condition.

The reactor vessel lower plenum is also subject to transient temperature changes due to accident conditions and normal power swings. These temperatures and the rates of change were taken from the Intermediate Heat Exchanger Equipment Specification. 1 *

The allowable stresses for the mechanical loadings are given in the ASME Boiler Code, Section VIII, 2 while the thermal stresses were evaluated using the criteria of the "Tentative Structural Design Basis for Reactor Pressure Vessels". 3

Loading Conditions

The weights and pressures acting on the bottom head of the vessel were computed as follows:

Internal pressure (100% Flow)	Hydrostatic	11.5 psi	
	Pump	<u>16.5 psi</u>	
	Total	28 psi	(use 30 psi)
Weights and Loads	Reactor Vessel and Attachments		130,000 lb
	Bellows Reaction		30,000 lb
	(4.5 in. x 7,000 lb/in.)		
	Piping Reactions		<u>10,000 lb</u>
			170,000 lb

* Superscripts refer to references listed at end of report.

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Bottom Head 25,000 lb.

Internals

Moderator Cans
(141 cans @ 1650 lbs.) *
1650 x 141 233,000 lb.

Misc. 10,000 lb.

243,000 lb.

The transient temperature changes occurring in the reactor vessel are as follows (Reference 1):

Normal Operation

Power swings 607 F to 640 F (100% to 0% load)

No. of times 10^6

Accident Conditions

(I) Temp. change: 607 F to 945 F

No. of times: 5

Rate of change: Essentially a step change

(II) Temp. change: 639 F to 553 F

No. of times: 500

Rate of change: Essentially a step change

Plate Bending Stresses at the Outermost Support Pads

Discontinuity Stress

The discontinuity stresses at the juncture of the vessel wall and the bottom head develop because of the expansion of the vessel wall relative to the bottom plate.

The hoop stress in the vessel wall due to the 30 psi pressure is computed as:

* 1650 lbs. is the buoyant weight of a moderator can in sodium.

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$$\sigma = \frac{pr}{t} = \frac{30 \times 114}{2} = 1710 \text{ psi}$$

σ = unit hoop stress (psi)

r = radius (inches)

t = thickness (inches)

p = pressure (psi)

The discontinuity stresses can be found with the relationship given in Reference 4, page 399.

$$\text{Moment} = \frac{prt}{2 \left[3(1-\nu^2) \right]^{\frac{1}{2}}}$$

$$\sigma_B = \text{Bending stress} = \frac{6M}{t^2} = \frac{3pr}{t \left[3(1-\nu^2) \right]^{\frac{1}{2}}} = \frac{pr}{t} \left[\frac{3}{1-\nu^2} \right]^{\frac{1}{2}} = 1.815 \frac{pr}{t}$$

ν = Poisson's ratio = 0.3

$$\sigma_B = 1710 \times 1.815 = 3100 \text{ psi}$$

Stresses due to Shell Load

The weight of the reactor vessel shell and attachments form a line load which acts around the vessel circumference. Since the support pads are not directly below the vessel wall, bending stresses are developed in the plate. Inasmuch as it is not known what degree of rotational restraint exists at the support pad, the worst condition was assumed; that is, complete restraint.

Figure 1

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It is also assumed that the plate is not permitted to rotate at the vessel wall. This plate geometry and loading condition are shown in Figure 1.

This case corresponds to Case 20, page 200, of Reference 5.

@ the inner edge $\nabla_r = \frac{3W}{2\pi t^2} \left[1 - \frac{2a^2}{a^2-b^2} \log_e \frac{a}{b} \right]$

$$\nabla_\phi = \nu \nabla_r$$

@ the outer edge $\nabla_r = \frac{3W}{2\pi t^2} \left[1 - \frac{2b^2}{a^2-b^2} \log_e \frac{a}{b} \right]$

$$\nabla_\phi = \nu \nabla_r$$

where: a = outer radius = 114 inches

b = inner radius = 100 inches

W = total weight = 170,000 lbs.

t = plate thickness = 2 inches

Evaluating the above expressions, the following stresses were obtained:

@ $r = b$; $\nabla_r = 2800$ psi, $\nabla_\phi = 840$ psi

@ $r = a$; $\nabla_r = 2350$ psi, $\nabla_\phi = 705$ psi

Stresses due to Pressure

The internal pressure of 30 psi which acts on the lower head of the vessel also results in bending stresses in the plate. Again it was assumed that no rotation is permitted at the vessel wall and at the support pads. This assumption gives conservative results.

The case analysed is shown in Figure 2.

Figure 2

The stresses can be evaluated from the relationships given in Reference 6.

• the inner radius

$$\sigma_r = \frac{3p}{4t^2} \left[(b^2 - 3a^2) + \frac{4a^4}{a^2 - b^2} \log_e \frac{a}{b} \right]$$

$$\sigma_\phi = \nu \sigma_r$$

• the outer radius

$$\sigma_r = \frac{3p}{4t^2} \left[-(a^2 - b^2) + \frac{4a^2 b^2}{a^2 - b^2} \log_e \frac{a}{b} \right]$$

$$\sigma_\phi = \nu \sigma_r$$

Substituting the proper values into the above expressions, the following stresses result:

• $r = b$; $\sigma_r = 2810 \text{ psi}$, $\sigma_\phi = 843 \text{ psi}$

• $r = a$; $\sigma_r = 1690 \text{ psi}$, $\sigma_\phi = 815 \text{ psi}$

Plate Bending Stresses at an Interior Point

As was stated in the General Remarks Section, the large number of support pads minimize the bending stresses in the bottom head. However, the plate must span the distance between the support pads without becoming highly stressed. This problem was considered by analyzing that section of the bottom plate which spans the greatest distance. In this case, it was found

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that the distance between support pads was 27.7 in. in one direction and 16 in. in the other. For purposes of conservative analysis, it was assumed that this segment of the head was simply supported on all four sides and loaded with the 30 psi internal pressure.

By using the tables of Reference 4, page 133, the maximum stresses can be computed by the following:

$$\text{Ratio of sides } \frac{b}{a} = \frac{27.7}{16} = 1.73$$

Maximum moments

$$M_x = 0.0920pa^2 = 0.0920 \times 30 \times 256 = 706 \text{ lb-in.}$$

$$M_y = 0.0483pa^2 = 0.0483 \times 30 \times 256 = 371 \text{ lb-in.}$$

Maximum stresses

$$\sigma_x = \frac{6M_y}{t^2} = \frac{6 \times 371}{4} = 1060 \text{ psi}$$

$$\sigma_y = \frac{6M_x}{t^2} = \frac{6 \times 706}{4} = 560 \text{ psi}$$

Frictional Loads and Plate Buckling

When the temperature changes in the lower plenum, the reactor vessel responds by expanding or contracting. The expansion or contraction is partially restrained by frictional forces acting on the bottom plate. An investigation is made to determine the possibility of buckling the bottom plate by these frictional forces.

The total load on the lower pads is approximately 450,000 lbs. Using a coefficient of friction of 0.5, the radial force acting on the bottom head is approximately 225,000 lbs. There is little likelihood that general buckling of the plate will occur. Local buckling was investigated for a rectangular plate element. Again the element chosen was that with the largest free span. The dimensions of the plate are 27.7 in. x 16 in.

Figure 3

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The stress distribution which may cause buckling is biaxial. This case is analyzed in Reference 7, page 333. The critical stress is given by the relationship

$$\sigma_{cr} = \frac{\pi^2 D}{a^2 h} \left(1 + \frac{a^2}{b^2} \right) = \frac{\pi^2 D}{a^2 h} (1 + 2.85) = \frac{3.85 \pi^2 D}{a^2 h}$$

$$\text{where } D = \frac{Et^3}{12(1-\nu^2)} = \frac{20 \times 10^6 (2)^3}{12(1-\nu^2)} = 14.7 \times 10^6$$

$$\sigma_{cr} = \frac{2.85 \pi^2 \times 14.7 \times 10^6}{(27.7)^2 (2)}$$

$$\sigma_{cr} = 3.64 \times 10^5 \text{ psi}$$

This value of the stress is not correct, since this stress is considerably above the elastic limit. An estimate of the direct stress in the plate due to the radial shear forces was made by assuming that the plate is subject to radial force distribution which varies linearly, and is equal to zero at the outside edge with the maximum occurring at the plate center. When this maximum plate force is used, and is distributed over a one inch strip of the plate, the friction stress is less than 3000 psi. Therefore, it is very unlikely that buckling of the lower head will occur.

Reactor Vessel Support Pads and Shoes

The support pads located on the bottom head of the reactor vessel must transfer the entire weight of the reactor vessel and core into the containment vessel bottom head. In order to insure that galling does not occur as a result of the thermal expansions of the reactor vessel during operation, it is necessary that the bearing stresses be at a minimum. The most highly loaded pads exist at the outer periphery. These pads carry the load due to the reactor vessel shell, bellows reaction and piping reactions. Also, they share the weight of the reactor vessel internals. This results in a load of approximately 11,000 lbs on each pad. The cross-sectional area which will resist this load is approximately 54 square inches resulting in a bearing stress which is approximately 200 psi. The inner pads carry the weight of the internals of the reactor. The load applied to these pads is approximately 3700 lbs and is resisted by an area of approximately 12 square inches. This yields a maximum direct bearing stress of approximately 300 psi. These bearing stresses are sufficiently low so that sliding without galling is expected.

The support pads are connected to the shoes by a screw. The purpose of this screw is only to hold the shoe to the pad at the time that the reactor vessel is positioned in the cavity. There is approximately 3/16" gap between the screw and the hole in the shoe. When expansion or contraction of the vessel occurs, movement of the shoe slides freely without imposing a load on the screw and no shearing stress is developed.

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Thermal Stresses

Thermal Stresses Resulting from Normal Power Swings

During normal operation of the reactor, the temperature of the lower plenum changes from 607 F to 640 F. At 0% load, the temperature is 607 F, while at 100% load the plenum temperature is 640 F. It has been assumed that this temperature change occurs one million times. Obviously, this implies that the resulting stresses must be below the endurance limit.

If it is assumed that this temperature change of 33 F occurs instantaneously and that the film coefficient is infinite, a step change in the surface temperature of the plate will occur. The unit stresses in the plate can be computed from

$$\sigma = \frac{E\alpha(\Delta T)}{1 - \nu}$$

where $E\alpha = 250 \text{ psi/F}$

$$\Delta T = 33 \text{ F}$$

$$\nu = 0.3$$

$$\sigma = 11,800 \text{ psi}$$

This particular computed stress is much higher than the actual stress because of the conservative assumptions.

Accident Transient Conditions

Transient condition (I) given in the Loading Conditions Section arises from the condition that the secondary sodium pump or line fails and the primary system continues to function. It is assumed that this occurs at full flow with the further assumption that the time required to change the temperature of the lower plenum is 25 seconds.

The following thermal properties of 304 stainless steel are used in the subsequent calculations:

$$h = \text{film coefficient} = 100 \text{ Btu/ft}^2\text{-hr-F (Reference 8)}$$

$$k = \text{thermal conductivity} = 12 \text{ Btu/ft-hr-F}$$

$$d = \text{diffusivity} = 0.2 \text{ ft}^2\text{/hr}$$

$$E = \text{modulus of elasticity} = 23 \times 10^6 \text{ psi}$$

$$\alpha = \text{coefficient of thermal expansion} = 10.5 \times 10^{-6} \text{ F}^{-1}$$

$$\nu = \text{Poisson's ratio} = 0.3$$

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Computing the Fourier modulus,

$$N_F = \text{Fourier modulus} = \frac{d\tau}{L^2} = \frac{(0.2)(25/3600)}{(1/6)^2} = 0.05$$

where τ = time in hours

L = thickness in ft

By using Figure 8 of Reference 9, a stress coefficient of 0.3 is determined.

The thermal stress can be computed from

$$\sigma = \frac{kE\alpha(\Delta T)}{(1-\nu)}$$

where $\Delta T = 945 \text{ F} - 607 \text{ F} = 338 \text{ F}$

k = 0.3

$$\sigma = \frac{0.3 \times 23 \times 10^6 \times 10.5 \times 10^{-6} \times 338}{0.7}$$

= 35,000 psi

This condition is expected to occur 5 times.

Transient condition II subjects the head to a ΔT of 86 F (639 F to 553 F). In this case the stress is computed to be

$$\sigma = \frac{0.3 \times 23 \times 10^6 \times 10.5 \times 10^{-6} \times 86}{0.7}$$

= 8,900

This condition is expected to occur 500 times.

Stresses due to the Transient Deflection of the Lower Head

An investigation was made to determine the deflection behavior of a circular plate subjected to a transient temperature change.¹⁰ This particular problem occurs when the lower plenum is shocked by a high temperature transient. When this temperature change occurs, there is a tendency for the bottom head to bow upward and if it bows upward there is a possibility that additional loads would be developed in the moderator can support rods. This investigation is necessary to insure that the support rods do not become overloaded because of this phenomenon.

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The sample problem presented in Reference 10 was made using the design condition and dimensions of the Reactor vessel lower head. In this case, a film coefficient of 150 Btu/ft²-hr-F⁸ was used. It was found that the bottom head, if it were simply supported along the circumference and completely free of any vertical load, would bow upward to a maximum deflection of 6.76 inches. To determine whether any deflection or increased stresses would occur, it is necessary to consider the boundary conditions and the applied loads. As a measure, the amount of uniform pressure that would be necessary to negate the 6.76 inches was computed. Using the formula given in Reference 4, page 62, the equation relating the uniform pressure to the deflection can be written as

$$w_{\max} = \frac{(5 + \nu)}{64(1 + \nu)} \frac{p_2 a^4}{D}$$

$$\text{where } D = \frac{Et^3}{12(1 - \nu^2)} = 14.6 \times 10^6 \text{ lb-in.}$$

$$a = 114 \text{ inches}$$

$$\nu = 0.3$$

$$\text{If, } w_{\max} = 6.76 \text{ inches; then } p_2 = 9.2 \text{ psi.}$$

This means that a pressure of 9.2 psi acting on the bottom head is sufficient to completely negate the maximum deflection.

It will be noted that this is approximately the hydrostatic pressure on the bottom head; also, it will be noted that the assumed plate support conditions are simply supported. There is restraint at this boundary in the real case, which reduces the uniform pressure requirement. Consequently, there appears to be little likelihood that bowing will occur with the subsequent increase in the loads in the moderator can support rods.

In order to determine the response of the lower head when the sodium flow patterns are such as to raise the film coefficient to a higher value, a film coefficient of 1000 Btu/hr-ft²-F was used. In this case it was found that 11.75 psi will negate the maximum deflection at the center of the plate. The hydrostatic sodium pressure is 11.5 psi. Consequently, there appears to be little possibility that additional loads will be placed on the moderator can support rods.

Analysis of Mechanical and Thermal Stresses

In order to make an analysis of the stress conditions, it is necessary to evaluate the total stress states.

The mechanical stresses are summarized as:

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@ edge, $r = b$	Discontinuity stress	3100 psi
	Line load bending stress	2800 psi
	Uniform pressure bending stress	<u>2800 psi</u>
	Total	8700 psi

bending
The ASME Boiler Code allowable stress for 304 stainless steel at 1050 F is 12,750 psi.

Investigating points within the boundary of the plate, it was found that the mechanical stress due to the uniform hydrostatic pressure is approximately 1060 psi. This is considerably less than the Code allowable.

The thermal stresses can be summarized as:

Due to normal power swings: $\sigma = 11,800$ psi

Accident transient conditions, I: $\sigma = 35,000$ psi

II: $\sigma = 8,900$ psi

Assuming that the yield strength stress for 304 stainless steel is 15,000 psi¹¹, the following two cyclic conditions are analyzed: (1) Normal Power Swings, and (2) Transient Conditions I.

An "elastic" stress diagram showing the stresses due to normal power swings is shown in Figure 4.

Normal Power Swings

Figure 4

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For the case of normal power swings, S'_{mean} is 14,800 psi with $S_{alt} = 5900$ psi. Since $S_{alt} + S'_{mean} > S_y$, and $S_{alt} < S_y$; $S_{mean} = 15,000 - 9,100$. The fatigue diagram for the above condition is shown in Figure 5.

Figure 5

The S_{alt} used with the fatigue data of Reference 12 is 15,000 psi. This is below the endurance limit (20,000 psi) and the plate can therefore be subject to the temperature change associated with normal power swings in indefinite number of times without failure.

The stress which results from the mechanical and thermal loads at the time that the accident transient condition I occurs is shown in Figure 6.

Figure 6

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In this case S'_{mean} is equal to 26,400 psi with $S_{alt} = 17,500$ psi; also, $S_{alt} > S_y$ ($S_y = 15,000$ psi).

Therefore, $S_{mean} = 0$

Since the alternating stress component is less than the endurance limit, the fatigue damage factor due to the above condition is zero.

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