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AEC Research and
Development Report
UC-81, Reactors - Power
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**preliminary stress
and fatigue analysis
of the SM-1 primary system**

Contract No. AT[30-1]-2639
with U. S. Atomic Energy Commission
New York Operations Office

ALCO PRODUCTS, INC.
NUCLEAR POWER ENGINEERING DEPARTMENT

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Development Report
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PRELIMINARY STRESS AND FATIGUE
ANALYSIS OF THE SM-1 PRIMARY SYSTEM

By:

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Issued: March 31, 1962

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ABSTRACT

A Preliminary analysis of the SM-1 primary system showed it was safe from strain cycling damage with possible exception of steam generator tube sheet and primary pump casing. These are recommended for further analysis. Reactor vessel stress analysis is subject of separate report.

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TABLE OF CONTENTS

	<u>Page</u>
ABSTRACT	vii
SUMMARY	1
1.0 INTRODUCTION	4
2.0 ANALYTICAL APPROACH	5
3.0 CYCLIC OPERATING HISTORY	6
4.0 STRESS AND FATIGUE ANALYSIS	8
4.1 Pressurizer	8
4.1.1 Pressurizer Heater	8
4.1.2 Pressurizer Stem at Sphere Reinforcement	15
4.1.3 Stress in Pressurizer Stem Due to Baffle	22
4.1.4 Restraint of Stem by Support Bracket	26
4.2 Wye Valve	39
4.2.1 Temperature Gradient	39
4.2.2 Throttle Transient	43
4.2.3 Load Transient	45
4.2.4 Emergency Cooling	47
4.3 Primary Coolant Pump Casing	48
4.4 Steam Generator	48
4.4.1 Vaporizer Shroud	51
4.4.2 Channel Partition	51
4.4.3 Shell at Centrifix Base	56
4.4.4 Tube Sheet	58
4.4.5 Support Bracket	66
5.0 CONCLUSIONS	73
6.0 REFERENCES	74

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LIST OF ILLUSTRATIONS

<u>Figure</u>	<u>Title</u>	<u>Page</u>
1	Drawing DE-43062-1A, Pressurizer Assembly, APPR-1	9
2	Cross-Section of Pressurizer Heater, Penetration of Heater Tube Sheet.	11
3	Pressurizer Heater Jacket - Fatigue Diagram	16
4	Pressurizer Stem - Fatigue Diagram	21
5	Section through Pressurizer Stem	23
6	Wye Valve - Assembly Drawing	41
7	Wye Valve - Fatigue Diagram	44
8	Wye Valve - Rib Reinforcement	46
9	Primary Coolant Pump and Casing	49
10	Steam Generator, SM-1	50
11	Vaporizer Shroud and Channel Partition-Fatigue Diagram	52
12	Dwg. DE-43058-1A, Steam Generator Assembly	53
13	Shell, Channel, and Tube Sheet - Fatigue Diagram	59
14	Dwg. H-152400 33 Inch Tube Layout 2-pass, 3/4" OD. U-Tubes, 1" Tri Pitch	61

LIST OF TABLES

<u>Table</u>	<u>Title</u>	<u>Page</u>
1	Stress Summary of SM-1 Pressurizer, Steam Generator, and Wye Valve	2
2	SM-1 Cyclic Operating History	3
3	Stress Summary (Pressurizer Heater)	14
4	Stress Intensity Summary (Pressurizer Heater)	14
5	Pressurizer Stem Stresses (due to Baffle)	24

SUMMARY

Preliminary analysis of the SM-1 primary system showed it was safe from strain cycling damage with the possible exception of the steam generator tube sheet and the primary pump casing. These two items have been recommended for further analysis as an extension of the work covered in this report. Reactor vessel analysis is not covered in this report, but is subject of APAE Memo-302.*

The pressurizer was determined to be safe from strain cycling damage on the basis of a preliminary stress and fatigue analysis of the heaters, sphere reinforcement and stem. The steam generator analysis, including the vaporizer shroud, tube sheet, channel partition and shell, indicated these members were safe from fatigue damage, except for the tube sheet. Table 1 gives stress summary of pressurizer, steam generator and wye valve.

The steam generator tube sheet is a major load carrying member, failure of which would require immediate and prolonged shutdown of the steam generator. Although calculated stress in the tube sheet was moderate, peak stresses in the ligaments were not accounted for. The method of analysis used was a simple modification of a solid circular plate analysis and there was no assurance that the stresses obtained with it represented a worst case. The tube sheet has a complicated geometry which was not adaptable to preliminary analysis.

The wye valve, on the downstream side of the steam generator, is subjected to sudden temperature increase of approximately 180°F if the throttle on the steam generator secondary is suddenly closed as it would during a scram. The reinforcement ribs of the valve suffer the greatest thermal stress under this condition, listed as the sum of Conditions IV, VI, and VII of Table 2. Previous test data for emergency cooling conditions show the sharpest temperature drop passing through the wye valve is approximately 60°F in 5 minutes or 80°F in 16 minutes. Table 2 shows the maximum number of cycles to be expected for this condition. All values are well within the Navy Code allowables.

The primary pump casing is subjected to the same temperature transients as the wye valve. Because of its intricate cross-sectional geometry it was not possible to obtain meaningful stress values by preliminary analysis.

* Sayre, M. F., SM-1 Reactor Cover and Flange Stress Analysis, APAE Memo-302, February 19, 1962.

TABLE 1

STRESS SUMMARY OF SM-1 PRESSURIZER, STEAM GENERATOR, AND WYE VALVE

<u>Component</u>	<u>Cause of Max. Stress (See Table 2)</u>	<u>Mean Stress</u>	<u>Alternating Stress</u>	<u>Operating Cycles-20 Yr.</u>	<u>Allowable Cycles</u>
<u>Pressurizer:</u>	Condition VII	5,140	5,140	900	100,000
(a) Heaters	Condition I	4,365	2,815	480,000	>>10 ⁶
(b) Cylinder to Sphere Reinforcement	Condition VII	11,050	11,050	900	100,000
(c) Stem stress due to baffle	Condition VII	17,325	17,325	900	2,500
(d) Stem stress due to support bracket					
(1) Lower ring	Condition VII	10,400	10,400	900	100,000
(2) Upper ring	Condition VII	14,850	14,850	900	100,000
<u>Steam Generator:</u>					
(a) Vaporizer Shroud	Condition II	0	35,700	15,000	19,000
(b) Shell at Centri- fix base	Condition VII	19,845	19,845	900	2,500
(c) Channel partition	Conditions VI and VIII	17,000	17,000	2,160	2,500
(d) Tube sheet	Condition II	14,850	14,850	15,000	100,000
(e) Shell and Channel at Support bracket					
(1) Lower ring	Condition VII	5,535	5,535	900	100,000
(2) Upper ring	Condition VII	17,275	24,725	900	2,500
<u>Wye Valve:</u>					
<u>Reinforcing Rib</u>	Conditions VI and VIII	12,270	12,270	2,160	100,000
	Conditions IV, and VI and VIII	4,165	4,165	11,160	100,000
	Condition II	15,520	2,680	15,000	100,000
	Condition IX	17,780	17,780	90	2,500

TABLE 2

SM-1 CYCLIC OPERATING HISTORY

* Temperature excursions are from operating temperatures (440°F)

<u>Cyclic Condition</u>		<u>Source of Information</u>	<u>To Date</u>	<u>Cycles 20 Yr Life</u>
I	Maximum number of operations of one heater in pressurizer	Estimate by SM-1 personnel	80,000	480,000
II	Load Changes greater than 50%	Extrapolation from sample months	2,500	15,000
III	Total number of temperature reductions out of the Operating range.	Sum of IV, VI and VIII	1,900	11,160
IV	Temperature excursion* Between operating range and 410°F (Load changes and scrams)	Estimate by SM-1 personnel	1,500	9,000
V	Pressure drops of less than 200 psi (due to scrams)	Estimate by SM-1 personnel	300	1,800
VI	Temperature excursions* below 410°F but above 170°F (due to scrams)	Extrapolation from sample months	300	1,800
VII	Pressure drops greater than 200 psi (planned shutdown)	Estimate by SM-1 personnel	150	900
VIII	Temperature reductions from 440 F to below 170°F (planned shutdown)	Extrapolation from sample months	60	360
IX	Temperature drop of about 85°F (as in emergency cooling test)	Estimate by SM-1 personnel	15	90

1.0 INTRODUCTION

The SM-1 was designed basically to operate as a power plant, but through operating life has been used extensively for research and development and as a training facility for military personnel. These operations have subjected the plant to pressure and thermal cycles far in excess of those required for the original intended operation.

The purpose of this task was to determine through a preliminary conservative stress analysis, whether any areas in the SM-1 primary system might be subject to fatigue failure due to the extensive cyclic operation of the plant during training, research, and development testing. Those components found highly stressed, were to be recommended for a more detailed stress analysis. Based on the results of the detailed analysis, changes in operating procedures or replacement of components would be recommended, if found necessary, to assure safety of operation and maximum utilization of the existing plant.

As a part of this study, an estimate of the total cyclic operating history of the SM-1 during its design life was made. It was based upon a compilation of the past history of plant operation, extrapolated to cover the training programs and research and development tasks planned for the future.

The report covers the following SM-1 primary loop components: the pressurizer, steam generator, wye-valve, and primary circulating pump casings. The study was conducted as a part of Item 1.1 of the Program Plan for Engineering Support and Development of Army PWR Power Plants, AP Note 286 Add 1 Rev. 1. The reactor vessel analysis was covered under Task 6, Subtask 6.9 of this same contract.

2.0 ANALYTICAL APPROACH

The preliminary analysis consists principally of limit calculations made using the worst possible conditions of temperature and pressure. The stresses calculated are extremely conservative, because their purpose is only to locate areas of possible failure.

The critical areas selected for analysis were those of potentially high thermal or pressure discontinuity stresses. Junctions of dissimilar metals, thick flanges and walls, abrupt geometrical discontinuities, members subjected to thermal shock, and those restrained from free thermal expansion, were investigated.

The Navy Code (1) was used as a reference for analytical methods, design criteria, and materials data. The procedure followed was to determine the maximum stresses in each member for a given operating condition, then plot these stresses on a fatigue diagram to determine the allowable number of cycles of stress. If the allowable number of cycles is more than the number estimated for the design 20-year life of SM-1, the condition causing the stress was termed "safe", i.e. it would not cause failure of the member.

3.0 CYCLIC OPERATING HISTORY

The fatigue analysis of the SM-1 plant as covered by this report is based on calculated stresses and anticipated cycles which will occur during a 20-yr operating life. An operating cycle is defined as a deviation from, and return to, an operating reference point of temperature and/or pressure. Operating cycles are the result of (1) start-ups and planned shutdowns, (2) load changes, (3) unscheduled scrams, and (4) emergency operating conditions.

The expected number of cycles for a particular operating condition was obtained from the operating log sheets of the SM-1 when possible. Information not available from the log sheets and information about abnormal reactor conditions, such as loss of primary pumps, were furnished by the operating personnel. In order to cover the 20-yr design life of the primary system, a straight line extrapolation of the accumulated operating data was made in order to predict the total number of cycles for each particular operating condition.

The operating cycles are broken down into the 9 cyclic conditions shown in Table 2 (Summary).

Startups and planned shutdowns are normal operations. The SM-1 Operating Manual⁽²⁾ permits a 30°/hr rate of heat-up and cooldown of the primary system. The rate of temperature change on heat-up is controlled by power level and the rate of temperature change on cooldown is controlled by dumping steam into the condenser, by-passing the turbine. The 30°/hr rate of heat-up and cooldown is strictly adhered to by the operators during startups and planned shutdowns. However, in the case of an unscheduled scram, the rate of change in temperature has run well above these limits. The actual performance depends almost entirely on the alertness of the operator and the conditions causing the scram. Cyclic conditions IV, V, and VI of Table 2 cover the cases of scrams. Cyclic conditions VII and VIII cover the cases of planned shutdowns.

In the case of complete loss of electrical power, the primary coolant pump stops, the reactor is scrammed automatically and the decay heat is removed by dumping steam into the condenser. By using the condenser as a heat sink, the primary water temperature is dropped suddenly approximately 85°F. The original testing of the SM-1 plant included an emergency cooling test. This test and results are described in Chapter VII of the Initial Operation and Testing Report⁽³⁾. The test results include a curve of reactor outlet temperature which shows a drop from 445°F to 361°F in 17 minutes at a maximum slope of 50° in 4.1 minutes or 12.2°F/min. Cyclic condition IX of Table 2 covers this case.

The SM-1 plant was designed to take a sudden loss of full load without scrambling the reactor. This is accomplished by taking advantage of the negative temperature coefficient of reactivity present in all pressurized water reactors. Load changes greater than 50% cause primary temperature changes of up to 27°F in 1 hr. Referring to the curve in the Initial Operation and Testing Report of reactor outlet temperature during a sudden loss of full load, the reactor outlet temperature rises from 448°F to 457°F in 5 min at a maximum slope of 8°F in 1.8 min or 4.44°F/min. Approximately 25 min later, the reactor outlet temperature is back to 448°F and continues to drop gradually until it stabilizes at 430°F 1 hr after the loss of load. This case is covered in cyclic condition II and IV of Table 2.

4.0 STRESS AND FATIGUE ANALYSIS

4.1 PRESSURIZER

The pressurizer is a spherical stainless steel clad vessel 45 in. ID mounted at the top of a cylindrical member or stem 16 in. ID by 54 in. long, containing immersion heaters. The primary system pressure is maintained by a steam pocket which is maintained at 1200 psia and saturation temperature, 567°F, by the heaters.

An investigation of the pressurizer, see Fig. 1 shows that the following areas may be subject to excessive stresses:

- (1) the heaters at their penetration through the heater tube sheet,
 - (2) the reinforcement where the sphere joins the stem, and (3) the stem where it joins the baffles and the support bracket.
- These areas were analyzed, and are discussed in the following paragraphs.

4.1.1 Pressurizer Heater

Information on the heater was obtained from Dwg. B-154628, C-154354 and D-151386 and from the SM-1 Operating Manual.⁽²⁾ The heater unit is rated at 3-1/3 kw in 567°F, 1200 psi water.

Since there is a greater heater capacity installed in the pressurizer than is required for normal operation, some of the heaters are not used and are subjected only to an external pressure load. The operating heaters are subjected to both pressure and thermal forces at the heater tube sheet penetration shown in Fig. 2.

The unused heaters are subjected to only pressure discontinuity and are analyzed first. From "Theory of Plates and Shells"⁽⁶⁾ and using the boundary conditions that the net edge deflection and rotation are zero,

Sign Convention

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Reference Drawing used was Alco B-154628 and D-151386.

Figure 2. Cross Section of Pressurizer Heater Penetration of Heater Tubesheet

$$w = \frac{1}{2 \beta^3 D} (\beta M_o + Q_o) = \delta$$

$$\theta = \frac{1}{2 \beta^2 D} (2 \beta M_o + Q_o) = 0$$

where $a =$ mean wall radius $= .197''$

$h =$ wall thickness $= .047''$

$p =$ pressure $= 1200$ psi

$\nu = .3$
 $E = 26.2 \times 10^6$

$$D = \frac{Eh^3}{12 (1 - \nu^2)} = \frac{26.2 \times 10^6 (.047)^3}{12 (1 - .09)} = 249.$$

$$\beta = \frac{1.285}{\sqrt{ah}} = \frac{1.285}{\sqrt{(.197) (.047)}} = 13.35$$

$$\delta = \frac{Pa^2}{Eh} (1 - \frac{\nu}{2}) = \frac{1200 (.197)^2 (.85)}{26.2 \times 10^6 (.047)} = 32.2 \times 10^{-6}$$

From the slope equation, $Q_o = -2 \beta M_o$

Substituting in the deflection equation,

$$M_o = 2 \beta^2 D \delta = 2(13.35)^2 (249) (32.2 \times 10^{-6}) = 2.85 \text{ in.lbs/in.}$$

The stresses are,

$$\sigma_1 = \pm \frac{6M}{h^2} - \frac{Pa}{2h} = \pm \frac{6 (2.85)}{(.047)^2} - \frac{1200 (.197)}{2 (.047)}$$

$$\sigma_1 = \pm 7,740 - 2,520 = 5,220 \text{ psi (outside); } -10,260 \text{ psi (inside)}$$

$$\sigma_2 = \pm \nu \frac{6M}{h^2} - \frac{Pa}{h} + \frac{E}{a} w$$

$$= \pm 2,320 - 5,040 + \frac{26.2 \times 10^6}{.197} (32.2 \times 10^{-6})$$

$$\sigma_2 = \pm 2,320 - 5,040 + 4,280$$

$$\sigma_2 = \pm 2,320 - 760 = 1560 \text{ psi (outside); } -3.080 \text{ psi (inside)}$$

$$\sigma_3 = 0 \text{ inside surface; } -1,200 \text{ outside surface}$$

The heaters which are used to maintain the system pressure are subjected to both thermal and pressure discontinuity forces since the heater jackets are hotter than the sleeves through which they pass. It is assumed that the sleeve is at water temperature and that the temperature gradient from sleeve to heater is abrupt.

The heat generation rate, q/A of the shortest heater is,

$$\frac{3333}{119} \text{ watts} \times \frac{1}{\pi \times .440 \text{ inches}} = 20.3 \text{ watts/in}^2$$

or 10,000 Btu/hr per ft^2 . The film coefficient of heat transfer, h_f , is conservatively estimated at 1000 Btu/hr ft^2 $^{\circ}\text{F}$.

$$q/A = h_f (t_{\text{jacket}} - t_{\text{sleeve}})$$

$$\Delta T = \frac{10,000}{1,000} = 10^{\circ}\text{F}$$

Therefore, jacket temperature is about 577°F .

The thermal deflection is,

$$\delta_{\text{th}} = a \alpha (\Delta T) = .197 (9.5 \times 10^{-6}) (10)$$

$$\delta_{\text{th}} = 18.7 \times 10^{-6}$$

Adding the thermal deflection to the pressure deflection and using the same method of analysis as used above,

$$\begin{aligned} M_o &= 2 \beta^2 D (\delta_p - \delta_{\text{th}}) \\ &= 2 (13.35)^2 (249) (32.2 - 18.7) \times 10^{-6} \end{aligned}$$

$$M_o = 1.195 \text{ in-lbs/in.}$$

The stress due to pressure and thermal discontinuity forces are,

$$\sigma_1 = \pm \frac{6M}{h^2} - \frac{Pa}{2h} = \pm \frac{6 (1.195)}{(.047)^2} - \frac{1200 (.197)}{2 (.047)}$$

$$\sigma_1 = \pm 3,250 - 2,520 = 730 \text{ psi (outside); } -5,770 \text{ psi (inside)}$$

$$\sigma_2 = \pm \nu \frac{6M}{h^2} - \frac{Pa}{h} \pm \frac{E}{a} W = \pm 975 - 5,040 + \frac{26.2}{.197} (13.5)$$

$$\sigma_2 = \pm 975 - 3,245 = -2,270 \text{ psi (outside); } -4,220 \text{ psi (inside)}$$

$$\sigma_3 = 0 \text{ inside; } -1,200 \text{ psi (outside)}$$

TABLE 3
STRESS SUMMARY (PRESSURIZER HEATER)

Discontinuity Stress Due To:	σ_1	<u>Inside Surface</u> σ_2	σ_3	σ_1	<u>Outside Surface</u> σ_2	σ_3
Pressure	-10,260	-3,080	0	+ 5,220	+ 1,560	-1,200
Pressure & Thermal	- 5,770	-4,220	0	+ 730	-2,270	-1,200

TABLE 4
STRESS INTENSITY SUMMARY (PRESSURIZER HEATER)

Stress Intensity	<u>Pressure Only</u>		<u>Pressure & Thermal</u>	
	<u>Inside</u>	<u>Outside</u>	<u>Inside</u>	<u>Outside</u>
S_{12}	-7,180	+ 3,660	- 1,550	+ 3,000
S_{23}	-3,080	+ 2,760	- 4,220	- 1,070
S_{31}	+ 10,260	- 6,420	+ 5,770	- 1,930

For the case where the heaters are on and off during normal operation, the maximum range of stress intensity is $S_{12} = 7,180 - 1,550 = 5,630$

$$\therefore S_r = 5,630$$

$$Salt = 5,630/2 = 2,815$$

$$S'_{mean} = 4,365$$

$$S_{mean} = 4,365$$

Plotting this case as point "a" on the fatigue diagram, Fig. 3, shows this condition to fall in fatigue class I which allows 100,000 cycles. Condition I of Table 2 is applicable to this case and shows 480,000 cycles. Since this is higher than the allowable number of cycles for class I, the method described on p. 49 of the Navy Code was used to determine the safe number of cycles for the calculated stress. It was found that the stress was below the fatigue limit so that it should be safe for an infinite number of cycles.

For the case of the unused or standby heaters, the maximum stress intensity is $S_{31} = 10,260 - 0$.

$$\begin{aligned} \therefore S_r &= 10,260 \\ \text{Salt} &= 5,140 \\ S'_{\text{mean}} &= 5,140 \\ S_{\text{mean}} &= 5,140 \end{aligned}$$

Plotting this case as point "b" on the fatigue diagram, Fig. 3 shows this condition falls in fatigue class I which allows 100,000 cycles. Condition VII of Table I which is applicable to this case of startup and shutdown of the plant shows 900 cycles. Therefore, the standby heaters are safe. It should be noted that the operating heaters and standby heaters are interchanged to keep the operating times of all the heaters equal.

4.1.2 Pressurizer Stem at Sphere Reinforcement

Assume the sphere with reinforcement completely restrains the stem. From "Theory of Plates and Shells"⁽⁶⁾ and using the boundary conditions that the net edge deflection and rotation are zero:

$$\begin{aligned} w &= -\frac{1}{2\beta^3 D} & (\beta M_o + Q_o) &= \delta \\ \theta &= \frac{1}{2\beta^2 D} & (2\beta M_o - Q_o) &= 0 \end{aligned}$$

Figure 3. Pressurizer Heater Jacket

where a = mean wall radius = 8.5 in.

h = wall thickness = 1 in.

P = pressure = 1200 psi

ν = .3

E = 26.2×10^6

$$D = \frac{E h^3}{12 (1 - \nu^2)} = \frac{26.2 \times 10^6 (1)^3}{12 (.91)} = 2.4 \times 10^6$$

$$\beta = \frac{1.285}{\sqrt{ah}} = \frac{1.285}{\sqrt{8.5 \times 1}} = 0.442$$

$$\delta = \frac{P a^2}{E h} \left(1 - \frac{\nu}{2}\right) = \frac{1200 (8.5)^2}{26.2 \times 10^6 (1)} (.85) = 2810 \times 10^{-6}$$

From the second equation, $Q_0 = -2 \beta M_0$. Substituting in the first equation,

$$M_0 = 2 \beta^2 \delta D = \frac{P}{2 \beta^2} \left(1 - \frac{\nu}{2}\right) = .258 \text{ pah}$$

$$M_0 = 2,640 \text{ in, lbs/in.}$$

The stresses are:

$$\sigma_1 = \pm \frac{6 M}{h^2} + \frac{P a}{2 h} = \frac{6 (2640)}{(1)^2} + \frac{1200 (8.5)}{2 (1)}$$

$$= \pm 15,800 + 5,100$$

$$\sigma_2 = \pm \nu \frac{6 M}{h^2} + \frac{P a}{h} - \frac{E \nu}{a} = .3 (15,800) + \frac{1200 (8.5)}{1} - \frac{26.2 (28.0)}{8.5}$$

$$= \pm 4,740 + 10,200 - 8.670$$

To determine if thermal stresses in the stem are significant the average temperature of the stem and reinforcement was determined as follows:

(1) The heat flow through the film at the inside surface of the spherical shell is,

$$q_1 = h_{f_i} A_i (t_w - t_s)$$

where h_{f_i} = film coef. = $.13 \text{ KY}^{1/3} \Delta t^{1/3} (N_{Pr})^{1/3}$ from
Heat Transfer⁽⁷⁾ by Giedt

A_i = film area

t_w = water temperature = 567°F

t_s = shell surface temperature

(2) The heat flow through the shell and reinforcement is,

$$q_2 = K_1 A_{m_1} \frac{t_s - t_a}{X_1}$$

where: K_1 = thermal conductivity of shell and reinforcement

A_{m_1} = area through which heat flows

X_1 = thickness of shell and reinforcement = 2-5/8 in.

t_a = average temperature of shell and reinforcement

(3) The heat flow through the insulation is,

$$q_3 = K_2 A_{m_2} \frac{t_a - t_{s_o}}{X_2}$$

where: K_2 = thermal conductivity of insulation

A_{m_2} = area through which heat flows

X_1 = Insulated thickness = 3 in.

t_{s_o} = outside surface temperature of insulation

(4) The heat loss to the atmosphere is,

$$q_4 = h_{f_o} A_o (t_{s_o} - t_{air})$$

where h_{f_o} = film coefficient = $.38 (\Delta T)^{.25}$ from

Heat Transfer and Insulation⁽⁸⁾

A_o = surface area of the insulation covering the shell and reinforcement

t_{air} = ambient temperature of the air

The boundary conditions are:

$$q_1 = q_2 = q_3 = q_4$$

The heat transfer equations were simplified by reducing all the known terms to a constant.

$$\begin{aligned} (1) \quad q_1 &= 260 \left\{ \frac{.785 [(28.5)^2 - (18)^2]}{144} \right\} (t_w - t_s) = 692 (t_w - t_s) \\ (2) \quad q_2 &= \frac{324}{2.625} \left\{ \frac{.785 [(28.5)^2 - (18)^2]}{144} \right\} (t_s - t_a) = 328 (t_s - t_a) \\ (3) \quad q_3 &= \frac{.40}{3} \left\{ \frac{.785 [(28.5)^2 - (18)^2]}{144} \right\} (t_a - t_{s_o}) = .354 (t_a - t_{s_o}) \\ (4) \quad q_4 &= 1 \left\{ \frac{.785 [(28.5)^2 - (18)^2]}{144} \right\} (t_{s_o} - t_{\text{air}}) = 2.66 (t_{s_o} - t_{\text{air}}) \end{aligned}$$

The ambient temperature of the air was assumed to be 100°F. By trial and error calculation, it was found that the following temperatures satisfied the boundary conditions.

$$t_s = 566.8$$

$$t_a = 566.3$$

$$t_{s_o} = 155^\circ\text{F}$$

Therefore, the thermal stresses will be negligible since the approximate temperature drop through the shell and reinforcement is less than one degree.

The stresses in the pressurizer stem due to the constraint of the reinforcement are summarized below, (+ inside surface, - outside surface).

Inner Surface				Outer Surface		
1	2	3		1	2	3
20,900	6,270	-1,200	=	-10,700	-3,210	0

By inspection the maximum stress intensity occurs at the inner surface and is $S_{31} = 22,100$ psi. The minimum stress intensity is zero

$$S_r = 22,100$$

$$S_{alt} = 22,100/2 = 11,050$$

$$S'_{mean} = 11,050$$

$$S_{mean} = S'_{mean} = 11,050$$

Plotting this condition as point "a" on the fatigue diagram, Fig. 4, shows that it falls in the fatigue class I which allows 100,000 cycles. Condition VII of Table 2 is the type of cycle that causes this stress and shows a total of 900. Therefore, this condition is safe.

Figure 4. Pressurizer Stem - Fatigue Diagram

4.1.3 Stress in Pressurizer Stem Due to Baffle

The pressurizer material is carbon steel SA-212 Gr. B. The cladding and baffle material is stainless steel type 304. From Fig. 5, it can be seen that the difference in thermal expansion between the baffle and the stem will increase the stresses in both members. The analysis was directed at the stresses in the stem because it is a major pressure-containing member. Failure of the baffle would cause no hazard to the operation of the pressurizer.

The water in the pressurizer is maintained at the saturation temperature corresponding to the system pressure. At the 1200 psia operating pressure, the pressurizer water is at 567°F. The baffle is 3/16 in thick and the temperature through its thickness is constant. The stem is 1 in thick and is insulated so that temperature will be the same as the baffle.

To determine the loading or stress in the pressurizer stem wall due to the difference in thermal expansion, a 1 in strip of baffle was analyzed as a column. The critical load of a column with fixed ends is

$$P_{cr} = \frac{4\pi^2 EI}{l^2} = \frac{4\pi^2 (26 \times 10^6) (5.5 \times 10^{-4})}{(12.5)^2} = 3,610 \text{ lb.}$$

This load is the maximum load that the baffle can exert on the pressurizer wall, since additional load will only cause the baffle to buckle. The ambient temperature in the vapor container is about 100°F. Then,

$$\text{Stem expansion } (\epsilon) = \alpha \Delta T = (567 - 100) (7.2 \times 10^{-6}) = .00336 \text{ in/in.}$$

$$\text{Baffle expansion } (\epsilon) = \alpha \Delta T = (567 - 100) (9.57 \times 10^{-6}) = .00446 \text{ in/in.}$$

Differential thermal expansion = .0011 in/in. Assuming the strain is taken entirely by the thin baffle, the baffle stress and load are

$$S = \epsilon E = .0011 (26 \times 10^6) = 28,600 \text{ psi}$$

$$S = P/A \text{ or } P = SA$$

$$P = 28,600 (.1875 \times 1) = 5,360 \text{ lb.}$$

Since this is above the critical load of the baffle, the baffle will buckle and maximum force that can be exerted by the baffle on the wall due to the differential thermal expansion is limited to 3,610 lb.

In order to calculate the stress of the wall due to the critical load of the baffle, use case 8, Table VIII of Roark.⁽⁴⁾

$$M = \frac{AT}{WR} \theta \text{ and at } \phi, \left[.3183 (n\phi + e - s\theta - c - us^2 + un^2) - n + s \right]$$

$$\theta = 38.7^\circ \text{ or } 38^\circ 40'$$

$$\phi = 141.3^\circ \text{ or } 141^\circ 20'$$

Figure 5. Section Through Pressurizer Stem

Where:

M = moment
 W = Applied Load = 3610 lb
 R = Radius = 8 in.
 n = $\sin \phi$ = .625
 e = $\cos \phi$ = .781
 s = $\sin \theta$ = .625
 c = $\cos \theta$ = .781
 u = $\cos \chi$ = $\cos \theta$ = .781

From Fig. 5:

θ = 38.7°
 ϕ = 141.3°
 $\cos \theta$ = .781
 $\cos \phi$ = -.781
 $\sin \theta$ = .625
 $\sin \phi$ = .625

M = - 4,070 in-lbs.

Stress = $\frac{\pm 6M}{h^2} = \pm \frac{6(4070)}{(1)^2} = \pm 24,450$ psi (+outside, -inside)

The pressure stress (S_p) = $\frac{PR}{t}$

$S_p = \frac{1200(8.5)}{1} = 10,200$ psi

The stresses in the pressurizer stem are summarized below (Table 5) in order to determine the maximum range of stress intensity, S_r .

TABLE 5
PRESSURIZER STEM STRESSES (DUE TO BAFFLE)

Type of Stress	<u>Inner Surface</u>			<u>Outer Surface</u>		
	σ_1	σ_2	σ_3	σ_1	σ_2	σ_3
Bending	0	-24,450	0	0	+24,450	0
Pressure	+5,100	+10,200	-1,200	+5,100	+10,200	0
Total	+5,100	-14,450	-1,200	+5,100	+34,650	0

where:

σ_1 = axial stress

σ_2 = circumferential stress

σ_3 = radial stress

Therefore, the maximum $S_r = S_{23} = +34,650$

$$S_{alt} = 34,650/2 = 17,325 \text{ psi}$$

$$S'_{mean} = 17,325$$

$$S_{alt} + S'_{mean} < S_b$$

$$\text{Therefore, } S_{mean} = S'_{mean} = 17,325$$

Plotting this condition as point "b" on the fatigue diagram, Fig. 4, it is found to fall under class II allowing 2,500 cycles. Cyclic condition VII of Table 2 shows a possible 900 cycles for the life of the plant. Therefore, the pressurizer stem wall is safe from fatigue failure due to heat-ups and cooldowns. For pressure fluxuations of less than 200 psi, cyclic condition V of Table 2, the water temperature fluxuates between 567° and 545°F. This small change in temperature has negligible effect on the differential thermal expansion and the load on the pressurizer stem wall is essentially unchanged.

4.1.4 Restraint of Stem by Support Bracket

The support bracket is composed of an upper ring welded to the pressurizer stem and a lower ring welded to the closure flange both of which are connected by gussets. Neglecting the effects of the gussets, the effects of the rings on the pressurizer stem wall was analyzed to obtain thermal, pressure and discontinuity forces.

(1) LOWER RING, FLANGE and STEM

The lower ring and flange were analyzed first. The sketch below shows the free body diagram.

A discontinuity analysis was made to determine interaction of the three members. Because of the size of the ring and flange relative to the stem wall and the stiffening effect of the gussets, it was assumed that there was no rotation of the assembly. The approach used was to match the deflections of the ring, flange and stem due to temperature differences, pressure and discontinuity forces. The unknown forces were found from the two equations which resulted.

Prior to the discontinuity analysis it was necessary to determine the average temperatures of the flange and ring. The sketch below was used as an analytical model for determining these temperatures.

From this sketch the heat flow paths were defined as follows:

1. The heat flow through the film at the inside surface of the flange is,

$$q_{T1} = h_{fi} A_i (t_w - t_s)$$

where: h_{fi} = film coefficient = $.13 \text{ KY}^{1/3} \Delta t^{1/3} (N_{Pr})^{1/3}$
from Heat Transfer by Giedt

A_i = film area

t_w = Water temperature = 567°F

t_s = Flange surface temperature

2. The heat flow through the flange is,

$$q_{T2} = K_F A_{mF} \frac{t_s - t_{a1}}{\Delta X_F}$$

where: K_F = thermal conductivity of the flange

A_{mF} = area at mean diameter of the flange

ΔX_F = Flange width = 4-7/8 in

t_{a1} = Average temperature of the flange.

3. The heat flow through the insulation of the flange is,

$$q_1 = K_1 A_1 \frac{t_{a1} - t_{s01}}{\Delta X_1}$$

where K_1 = thermal conductivity of the flange insulation

A_1 = area through which heat flows

ΔX_1 = Insulation thickness = 3 in

t_{s01} = Outside surface temperature of the insulation

4. The heat loss to the atmosphere from the surface of the flange insulation is,

$$q_2 = h_{fo} A_{o1} (t_{s01} - t_{air})$$

where: h_{fo} = film coefficient = $.38 (\Delta T)^{.25}$
from Heat Transfer and Insulation

A_{o1} = Surface area of the flange insulation

t_{air} = Ambient temperature of the air

5. The heat flow through the ring is,

$$q_3 = K_R A_{mR} \frac{(t_{a1} - t_{a2})}{\Delta X_R}$$

where: K_R = thermal conductivity of the ring

A_{mR} = Area at the mean ring dia.

ΔX_R = Ring width = 12 in.

t_{a2} = Average temperature of the ring

6. The heat flow through the insulation of the ring is,

$$q_4 = K_2 A_2 \frac{(t_{a2} - t_{s_{o2}})}{\Delta X_2}$$

where: K_2 = thermal conductivity of ring insulation

A_2 = Area through which heat flows

ΔX_2 = Insulation thickness

$t_{s_{o2}}$ = Outside surface temperature of ring insulation

7. The heat lost to the atmosphere from the surface of the ring insulation is,

$$q_5 = h_{f_o} A_{o2} (t_{s_{o2}} - t_{air})$$

where: A_{o2} = Surface area of the ring insulation

h_{f_o} = film coefficient = $.38 (\Delta T)^{.25}$

From the boundary conditions,

$$q_{T1} = q_{T2}$$

$$q_1 = q_2$$

$$q_3 = q_4 = q_5$$

$$q_{T1} = q_1 + q_3$$

The heat transfer equations were simplified by reducing all the known terms to a constant.

$$(1) \quad q_{T_1} = 260 \left[\frac{\pi (16) (4.5)}{144} \right] (t_w - t_s) = 408 (t_w - t_s)$$

$$(2) \quad q_{T_2} = \frac{324}{4.875} \left[\frac{\pi (20.875) (4.5)}{144} \right] (t_s - t_{a1}) = 136 (t_s - t_{a1})$$

$$(3) \quad q_1 = \frac{2 (.40)}{3} \left\{ \frac{.785 [(25.75)^2 - (18)^2]}{144} \right\} (t_{a1} - t_{s_{o1}}) \\ = .492 (t_{a1} - t_{s_{o1}})$$

$$(4) \quad q_2 = (1)(2) \left\{ \frac{.785 [(25.75)^2 - (18)^2]}{144} \right\} (t_{s_{o1}} - t_{air}) \\ = 3.68 (t_{s_{o1}} - t_{air})$$

$$(5) \quad q_3 = \frac{324}{12} \left[\frac{\pi (37.75) (2.25)}{144} \right] (t_{a1} - t_{a2}) = 50 (t_{a1} - t_{a2})$$

$$(6) \quad q_4 = \frac{2 (.40)}{3} \left\{ \frac{.785 [(49.75)^2]}{144} \right\} (t_{a2} - t_{s_o}) = 2.64 (t_{a2} - t_{s_{o2}})$$

$$(7) \quad q_5 = (1)(2) \left\{ \frac{.785 [(49.75)^2 - (25.75)^2]}{144} \right\} (t_{s_{o2}} - t_{air}) \\ = 19.8 (t_{s_{o2}} - t_{air})$$

The ambient temperature of the air was assumed to be 100°F.

Using the relaxation method, the following temperatures were found to satisfy the boundary conditions.

$$t_s = 564^\circ\text{F}$$

$$t_{a1} = 555^\circ\text{F}$$

$$t_{s_{o1}} = 153^\circ\text{F}$$

$$t_{a2} = 535^\circ\text{F}$$

$$t_{s_{o2}} = 153^\circ\text{F}$$

Therefore, the average temperature of the flange is 555°F and the average temperature of the ring is 535°F.

Deflection Equations for Stem, Flange and Ring:

The deflection at the edge of the stem wall is the result of thermal, pressure and discontinuity forces. From the Theory of Plates and Shells, (6) the deflection and rotation of the edge due to discontinuity forces are

$$w = \frac{1}{2\beta^3 D} (\beta M_o + Q_o) \text{ (outward deflection is positive)}$$

$$\theta = -\frac{1}{2\beta^2 D} (2\beta M_o + Q_o) = 0$$

where: w = deflection

θ = rotation

D = Flexural Rigidity = $\frac{Eh^3}{12(1-\nu^2)} = 2.4 \times 10^6$

$\beta = \frac{1.285}{\sqrt{ah}}$ @ $\nu = .3$ or $\beta = .441$

a = mean radius of stem = 8.5"

h = wall thickness = 1"

E @ 567°F = 26.2×10^6

Deflection:

$$w = \frac{1}{2(.441)^3(2.4 \times 10^6)} (.441 M_o + Q_o)$$

$$w = 1.07 \times 10^{-6} M_o + 2.44 \times 10^{-6} Q_o$$

Rotation:

$$\theta = -\frac{1}{2\beta^2 D} (2\beta M_o + Q_o) = 0$$

$$\therefore M_o = \frac{-Q_o}{2\beta} = \frac{-Q_o}{.882}$$

Substituting the value for M_o into the deflection equation,

$$w = 1.07 \times 10^{-6} \left(-\frac{Q_o}{.882}\right) + 2.44 \times 10^{-6} Q_o$$

$$w = 1.22 \times 10^{-6} Q_o \text{ inches}$$

The pressure deflection is,

$$\delta_p = \frac{P a^2}{Eh} \left(1 - \frac{\nu}{2}\right) = \frac{1200 (8)^2}{26.2 \times 10^6 (1)} \left(1 - \frac{.3}{2}\right)$$

$$= .0025 \text{ inches}$$

The thermal deflection is,

$$\delta_t = \alpha \Delta T r = 7.2 \times 10^{-6} (563 - 70) 8 = .0284 \text{ inches}$$

Therefore, the net deflection of the stem is,

$$\delta_{\text{stem}} = 1.22 \times 10^{-6} Q_0 = .0025 + .0284 \text{ ----- (1.)}$$

The deflection at the inner and outer surfaces of the flange are due to pressure, Q_0 and F . Since rotation is ignored, M_0 is omitted from the calculations. The following free body diagram will be analyzed.

where:

$$P'' = \frac{F}{l}$$

$$P' = 1200 - \frac{Q_0}{l}$$

$$A_1 = 8''$$

$$b_1 = 12 \frac{7}{8}''$$

$$l = 4 \frac{1}{2}''$$

$$E = 26.8 \times 10^6$$

The deflection of the inner surface is given by,

$$\delta_{\text{inner}} = \frac{P' a_1}{E} \left(\frac{b_1^2 - a_1^2}{b_1^2 - a_1^2} + \nu \right) - \frac{P'' a_1}{E} \left(\frac{2 b_1^2}{b_1^2 - a_1^2} \right)$$

$$\delta_{\text{inner}} = (.765 \times 10^{-6}) P' - (.893 \times 10^{-6}) P''$$

Substituting values for P' and P''

$$\delta_{\text{inner}} = (-.17 \times 10^{-6}) Q_0 - (.1985 \times 10^{-6}) F + 918. \times 10^{-6}$$

The outer surface deflection is,

$$\delta_{\text{outer}} = \frac{P' b_1}{E} \left(\frac{2 a_1^2}{b_1^2 - a_1^2} \right) - \frac{P'' b_1}{E} \left(\frac{a_1^2 + b_1^2}{b_1^2 - a_1^2} - \nu \right)$$

$$\delta_{\text{outer}} = (.602 \times 10^{-6}) P' - .94 \times 10^{-6} P''$$

Substituting for P' and P''

$$\delta_{\text{outer}} = (-.134 \times 10^{-6}) Q_0 - .209 \times 10^{-6} F + 722. \times 10^{-6}$$

The deflection of the flange due to temperature change is,

$$\delta_{T \text{ inner}} = \alpha \Delta T a_1 = 7.2 \times 10^{-6} (555 - 70) 8 = .0279 \text{ inches}$$

$$\delta_{T \text{ outer}} = \alpha \Delta T b_1 = 7.2 \times 10^{-6} (555 - 70) (12.875) = .0449 \text{ in}$$

Adding the thermal deflections, the net deflections are,

$$\delta_{a_1} = (-.17 \times 10^{-6}) Q_0 - (.1985 \times 10^{-6}) F + .0279 + 918 \times 10^{-6} \quad (2)$$

$$\delta_{b_1} = (-.134 \times 10^{-6}) Q_0 - (.209 \times 10^{-6}) F + .0449 + 722 \times 10^{-6} \quad (3)$$

The ring was analyzed by the same approach used on the flange. The following free body diagram was analyzed.

where: $p''' = \frac{F}{l_2}$

$$A_2 = b_1 = 12 \frac{7}{8}''$$

$$b_2 = 24 \frac{7}{8}''$$

$$l_2 = 2 \frac{1}{4}''$$

$$E = 26.8 \times 10^6$$

The deflection of the inner surface of the ring due to pressure is,

$$\delta_{\text{inner}} = \frac{p''' a_2}{E} \left(\frac{b_2^2 + a_2^2}{b_2^2 - a_2^2} + \nu \right)$$

$$\delta_{\text{inner}} = .433 \times 10^{-6} F$$

The thermal deflection is

$$\delta_T = \alpha \Delta T a_2 = 7.2 \times 10^{-6} (535 - 70) 12.875 = .0431 \text{ in}$$

Adding the thermal deflection, the net deflection of the inner surface of the ring is,

$$\delta_{a_2} = .433 \times 10^{-6} F + .0431 \quad (4)$$

The boundary conditions require that the deflections be compatible.

$$\delta_{\text{stem}} = \delta_{a_1}$$

$$\delta_{b_1} = \delta_{a_2}$$

Substituting the values of the deflections in to the boundary conditions:

$$\begin{aligned}\delta_{stem} &= 1.22 \times 10^{-6} Q_0 + .0025 + .0284 \\ &= 1.22 \times 10^{-6} Q_0 + 30,900 \times 10^{-6} \\ \delta_{a_1} &= -0.17 \times 10^{-6} Q_0 - 0.1985 \times 10^{-6} F \\ &\quad + 918 \times 10^{-6} + 27,900 \times 10^{-6} \\ \delta_{b_1} &= -0.134 \times 10^{-6} Q_0 - 0.209 \times 10^{-6} F \\ &\quad + 722 \times 10^{-6} + 44,900 \times 10^{-6} \\ \delta_{a_2} &= 0.433 \times 10^{-6} F + 43,100 \times 10^{-6}\end{aligned}$$

But $\delta_{stem} = \delta_{a_1} ; \delta_{b_1} = \delta_{a_2}$

$$\begin{aligned}(1.22 + 0.17) Q_0 + 0.1985 F &= 30,900 + 28,820 \\ -0.134 Q_0 + (-0.209 - 0.433)F &= 43,100 - 45,622\end{aligned}$$

and

Simplifying, we get

$$1.39 Q_0 + 0.1985 F = -2080 \quad (5)$$

$$-0.134 Q_0 - 0.642 F = -2522 \quad (6)$$

Solving by determinants,

$$D = -.892 + .0266 = -.8654$$

$$D_{Q_0} = +1335 + 501 = 1836$$

$$Q_0 = \frac{1836}{-.8654} = 2,120 \text{ \#/in.}$$

$$D_F = -3510 - 279 = -3789$$

$$F = \frac{-3789}{-.8654} = 4380 \text{ \#/in.}$$

Check:

$$1.39 (-2,120) + .1985 (4380) = -2081 \text{ vs. } -2080 \quad \text{OK}$$

$$-.134 (-2,120) - .642 (4380) = -2526 \text{ vs. } -2522 \quad \text{OK}$$

$$M_o = \frac{-Q_o}{.88} = \frac{-2120}{.88} = 2410 \text{ in. \#/in.}$$

$$W_o = 1.22 \times 10^{-6} (-2120) = -.00259" \text{ (inward)}$$

$$\sigma_1 = \frac{P a}{2 h} + \frac{6 M_o}{h^2} = \frac{1200 (8.5)}{2 (1)} + \frac{6 (2410)}{(1)^2} = 5100 + 14,500$$

$$\sigma_1 = 19,600 \text{ psi. inside; } -9,400 \text{ psi. outside}$$

$$\sigma_2 = \frac{P a}{h} + \frac{E W_o}{a} + \frac{\nu 6 M}{h^2} = 10,200 + \frac{26.2 (-2590)}{8.5} + 4350$$

$$\sigma_z = 10,200 - 7,990 + 4350 = -2140 \text{ psi. inside; } 6,560 \text{ psi. outside}$$

$$\sigma_3 = -1200 \text{ psi. inside; } 0 \text{ outside}$$

$$S_{\max} = S_{13} \text{ inside} = 20,800 \text{ psi.}$$

$$S_{\min} = 0$$

$$S_r = 20,800$$

$$S_{alt} = 20,800/2 = 10,400$$

$$S'_{mean} = 10,400$$

$$S_{mean} = S'_{mean} = 10,400$$

Plotting S_{alt} and S_{mean} as point "c" on the fatigue diagram, Fig. 4, shows that this condition falls in the fatigue class I which allows 100,000 cycles. Condition VII of Table 2 is the type of cycle producing the above effect on the stem and shows 900 cycles for the 20 year life. Hence, this condition is safe.

(2) Upper Ring and Stem

The upper ring of the pressurizer support bracket was analyzed using The Theory of Plates and Shells⁽⁶⁾ as a reference.

$$W = \frac{F a^2 \beta}{2 E h} ; M_{\max.} = \frac{F}{4 \beta}$$

$$\delta = \frac{P a^2}{E h} \left(1 - \frac{\nu}{2}\right) + \Delta \delta_{th}$$

where:

F = Force on pressurizer stem wall from the upper ring

$2a$ = Stem O.D. and ring I.D.

$$\beta = \frac{1.285}{\sqrt{ah}} = .428$$

$$E = 26.8 \times 10^6$$

h = Stem wall thickness = 1"

$$\Delta \delta_{th} = a \left[(\alpha \Delta T)_{stem} - (\alpha \Delta T)_{ring} \right]$$

$$\alpha = 7.2 \times 10^{-6}$$

The sketch below shows the free body diagram of the stem and ring that was analyzed.

The boundary conditions are such that, $\delta + W = 0$

Therefore:

$$\frac{F (a)^2 \beta}{2 E h} = - \frac{P a^2}{E h} \left(1 - \frac{\nu}{2} \right) - \Delta \delta_{th}$$

From the temperature analysis of the lower ring, it can be seen that the temperature difference between the stem and upper ring is not greater than 20°F.

Substituting in values,

$$\frac{F (9)^2 .428}{2(26.8 \times 10^6)(1)} = \frac{1200(9)^2}{26.8 \times 10^6 (1)} \left(1 - \frac{3}{2}\right) - 9$$

$$\times [7.2 \times 10^{-6} (20)]$$

$$\frac{F (34.7)}{53.6 \times 10^6} = \frac{(97,200)(.85)}{26.8 \times 10^6} - 1300 \times 10^{-6}$$

$$F = \frac{-3,080 - 1,300}{.648} = \frac{-4,380}{.648} = -6,750 \text{ \#/in.}$$

$$M_{\max.} = \frac{-6750}{4(.428)} = -3850 \text{ in.-lbs/in.}$$

$$\sigma_1 = \pm \frac{6 M}{h^2} + \frac{P a}{2 h} \quad \begin{array}{l} (+\text{inner surface}) \\ (-\text{outer surface}) \end{array}$$

$$\sigma_2 = \pm \frac{6 M}{h^2} + \frac{P a}{h} - \frac{E}{a} (W)$$

$$\sigma_3 = 0 \text{ and } -p$$

$$\sigma_1 = \pm \frac{6 (3850)}{(1)^2} + \frac{1200 (9)}{2 (1)}$$

$$\sigma_1 = \pm (23,100) + 5400$$

$$\sigma_2 = \pm (.3) (23,100) + 10,800 \left(\frac{26.8}{9}\right) (-4380)$$

$$\sigma_2 = \pm (6,940) + 2,250$$

Inner Surface				Outer Surface			
σ_1	σ_2	σ_3		σ_1	σ_2	σ_3	
28,500	9,190	-1,200		-17,700	-4,690	0	

$S_{31} = 29,700$ at the inner surface and is the maximum stress intensity,

therefore, $S_r = 29,700$

$$S_{alt} = 29,700/2 = 14,850$$

$$S'_{mean} = 14,850$$

$$S_{alt} + S'_{mean} < S_b \text{ therefore}$$

$$S_{mean} = S'_{mean} = 14,850$$

Plotting S_{alt} and S_{mean} as " σ_2 " on the fatigue diagram,

Fig. 4. shows that this condition falls in the fatigue class I which allows 100,000 cycles. Condition VII of Table 2 covers this type of cycle and shows 900 cycles for the 20 year life. Hence, this condition is safe.

4.2 WYE VALVE - SEE FIG. 6 (DWG. D9-47-1007)

4.2.1 Temperature Gradient

The wye valve (or two-way valve) is analyzed for pressure and thermal stresses in the Design Analysis⁽⁹⁾ of the SM-1A plant. Since the SM-1A and the SM-1 Wye valves are alike, and their primary system temperatures and pressures are similar, the analysis of the SM-1A valve is applicable to the SM-1 as well. The SM-1A analysis was used because it contained a more detailed thermal stress analysis than was done for the SM-1 valve. The SM-1 Operating Manual states that the maximum temperature difference across the wye valve reinforcing rib should not exceed 45°. This limitation permits rate of change of temperature of approximately 30 degrees/hr. during heat-up and cooldown. Page 72 of the SM-1A Design Analysis indicates that Case II corresponds to a conservative 35 degrees/hr. heating rate. Page 79 summarizes the stress in the outside fibre of a reinforcing rib for Case II as:

Pressure	14,780
Thermal	<u>13,440</u>
Combined	28,220 psi

The pressure stress was figured at the design pressure of 1600 psi. Changing to working pressure of 1200 psi:

pressure	=	$\frac{1200}{1600}$	x	14,780	=	11,100
thermal						<u>13,440</u>
				Combined		24,540 psi

Figure 6. Wye Valve - Assembly Drawing

The maximum principle stress is 24,540 psi and the minimum one is 0. The maximum range of stress intensity, $S_r = 24,540$.

$$S_{alt} = \frac{24,540}{2} = 12,270. \quad S'_{mean} = 12,270 \text{ and } S_{mean} = 12,270.$$

Plotting this point on the fatigue diagram in Fig. 7 as point "a", it is found to lie in the fatigue class I which allows 100,000 cycles. The maximum number of operating cycles for the life of the plant is 2,160 from the sum of cyclic conditions VI and VIII of Table 2. This condition is, therefore, considered safe.

4.2.2 Wye Valve - Throttle Transient

The wye valve is on the downstream side of the steam generator. If the throttle is suddenly closed, as in a scram, the outlet primary water from steam generator and inlet to the wye valve rises rapidly in temperature from 431.6°F to 450°F as described on pages 70 and 71 of the Operating Manual. This results in a mild thermal shock to the wye valve.

The unreinforced section of the first rib, 7-1/2" deep, is treated as though it is a vessel wall 7-1/2" thick and analyzed by the Navy Code paragraph A.3.3. It is assumed that the temperature change is instantaneous.

$$\sigma_{inside} = \frac{E \alpha (\Delta T_f)}{1 - \nu} \times \eta_i$$

$$\sigma_{outside} = \frac{E \alpha (\Delta T_f)}{1 - \nu} \times \eta_o$$

$$\Delta T_f = 450 - 431.6 = 18.4^\circ F, \quad \theta \approx 0, \quad t = 7.5"$$

$$h = 10,000$$

$$k = 10.5, \quad \epsilon = .006, \quad E \alpha = 249, \quad \nu = .3$$

$$\lambda = \frac{\epsilon \theta}{t^2} = \frac{.006 \times 0}{(7.5)^2} = 0$$

$$\frac{12 k}{ht} = \frac{12 \times 10.5}{10,000 \times 7.5} = .00168$$

$$\text{From curve A.3-5 Navy Code, } \eta_i = 1$$

$$\sigma_{inside} = \frac{249 \times 18.4 \times 1}{.7} = 6,550 \text{ psi compression}$$

$$\text{From curve A.3-6 Navy Code, } \eta_o < .06$$

$$\sigma_{outside} = \frac{249 \times 18.4 \times .06}{.7} = 392 \text{ psi tension}$$

Figure 7. WYE Valve Stresses

In this case the thermal stress at the inside of the middle of the straight part of the rib would be about 6,550 psi compression. This then, combines with the pressure stress listed on page 79 of the Design Analysis. The pressure stress, adjusted from design pressure to working pressure is,

$$\frac{1200}{1600} \times -3970 = -2,980 \text{ psi}$$

The combined stress is $-2,980 - 6,550 = -9,530$ psi, with compression on the inside. This occurs any time the throttle is suddenly closed during full power operation.

The maximum principal stress is $-9,530$ and the minimum stress is $-1,200$ psi.

$$S_r = 8,330$$

$$S_{alt} = 8,330/2 = 4,165$$

$$S'_{mean} = 4,165 = S_{mean}$$

Plotting S_{alt} versus S_{mean} on the fatigue diagram in Fig. 7 as point "b", we find that this point falls under Class I or 100,000 cycles. The maximum number of cycles that can be expected is 11,160 per cyclic conditions IV, VI, and VII of Table 2 Hence, this condition is safe.

The increase in stress at the outer surface of the rib is so low that the fatigue analysis for the steady state temperature gradient is still valid.

4.2.3 Wye Valve Load Transient

Page 140 of the Operating Manual states that the maximum allowable rate of increase or decrease in load is 200 KW per minute.

If the load is increased from zero to full load (2000 KW) or decreased from full load to zero at 200 KW per minute it

will take $\frac{2000}{200} \times 60 = 600$ seconds. The change in tem-

perature of the primary water entering the wye valve, from no load to full load, is 18.4°F .

Analyzing this situation by the Navy Code paragraph A.3.5: $\Delta T_f = 18.4$, $\theta = 600$, $t = 7.5$, $h = 10,000$,

$$\epsilon = .006, \quad E\alpha = 249, \quad \nu = .3, \quad K = 10.5$$

$$\lambda = \frac{\epsilon \theta}{t^2} = .064$$

$$\frac{12 K}{ht} = .00168 \quad \left(\begin{array}{l} \text{using these values in Navy} \\ \text{Code curve } \eta_1 = .82 \end{array} \right)$$

$$\sigma_{\text{inside}} = \frac{E \alpha (\Delta T_f)}{1 - \nu} \times \eta_i = 5,360 \text{ psi}$$

It can be either tension or compression, depending on whether the electrical load is being increased or decreased. Tension, or increasing electrical load, is more severe because it adds to the pressure stress at point $\theta = 30^\circ$ as shown on the Fig. 8 below.

Figure 8 Wye Valve - Rib Reinforcement

The maximum pressure stress at this point is normally 10,430 plus not over 5,120 tension direct stress or 15,550. Converting from design to operating pressure this stress will be

$$\frac{1200}{1600} \times 15,550 = 11,660 \text{ psi.}$$

$$11,660 + 5,360 = 17,020 \text{ psi for the combined stress.}$$

The minimum principle stress is 1200 psi compression due to internal pressure.

The maximum stress difference = $17,000 + 1200 = 18,200$ psi. It will fluctuate approximately through the range of 5,360 psi.

$$S_{\text{alt}} = \frac{5360}{2} = 2680 \text{ psi}$$

$$S'_{\text{mean}} = 18,200 - 2,680 = 15,520 \text{ psi}$$

$$S_{\text{alt}} + S'_{\text{mean}} = 18,200 \text{ which is } < S_b (36,000) \text{ so}$$

$$S'_{\text{mean}} = S_{\text{mean}} = 15,520$$

See Fatigue Diagram - Fig. 7 point "c"

Point "c" is in fatigue class I allowing 100,000 cycles. The maximum number of loading cycles is 15,000 for the life of the plant, from Cyclic condition II, Table 2 (Summary). The condition is therefore considered safe.

4.2.4 Wye Valve - Emergency Cooling

Initial Operation and Testing Fig. VII-2 gives a reactor outlet temperature plot. The same temperature time relationship was assumed for the water passing through the wye valve.

The most severe gradient is 440° to 380°F in 5 minutes. The total drop is 445° to 361° in 16 minutes.

For the worst gradient

$$\Delta T_f = 440 - 380 = 60^\circ\text{F}$$

$$\theta = 5 \times 60 = 300 \text{ seconds}$$

$$\lambda = \frac{\epsilon \theta}{t^2} = \frac{.006 \times 300}{(7.5)^2} = .0322$$

$$\frac{12 k}{ht} = \frac{12 \times 10.5}{10,000 \times 7.5} = .00168$$

$$\eta_i \text{ from curve} = .89$$

$$\begin{aligned} \text{The stress inside} &= \frac{E (\alpha T_f)}{1 - \nu} \times \eta_i = \frac{249 \times 60}{.7} \times .89 \\ &= 19,000 \text{ psi tension} \end{aligned}$$

For the total drop,

$$\Delta T_f = 445 - 361 = 84^\circ$$

$$\theta = 16 \times 60 = 960 \text{ seconds}$$

$$\lambda = \frac{\epsilon \theta}{t^2} = \frac{.006 \times 960}{(7.5)^2} = .103$$

$$\frac{12 k}{ht} = \frac{12 \times 10.5}{10,000 \times 7.5} = .00168$$

$$\eta_i \text{ from curve} = .76$$

$$\text{The stress inside} = \frac{249 \times 84}{.7} \times .76 = 22,700 \text{ psi tension}$$

This can be combined with 11,660 psi tension due to pressure stress at operating pressure as derived in paragraph 5.3.3 above.

$$11,660 + 22,700 = 34,360 \text{ psi combined.}$$

The minimum principle stress is 1200 psi compression due to internal pressure.

$$\text{Maximum stress difference} = 34,360 + 1200 = 35,560$$

$$\text{Then, } S_r = 35,560 \quad S_{alt} = \frac{35,560}{2} = 17,780 \text{ psi}$$

$$S'_{mean} = 17,780$$

$$S_{alt} + S'_{mean} = 35,660 < S_b \text{ (36,000) so}$$

$$S'_{mean} = S_{mean} = 17,780$$

This point, plotted on fatigue diagram (Fig. 7) as point "d", falls within fatigue Class II which allows 2500 cycles.

Cyclic condition IX of Table 2 indicates a maximum of 90 cycles for the life of the plant. This condition is therefore considered safe.

4.3 PRIMARY COOLANT PUMP CASING

From Fig. 9 it can be seen that the primary coolant pump casing has a highly complicated cross-sectional geometry. A preliminary analysis of such a member would require so many simplifying assumptions that meaningful stress values could not be obtained. For this reason, a preliminary analysis of the casing has been omitted from this report and a detailed analysis has been recommended under Task 1.1 of the FY '62 contract work program.

4.4 STEAM GENERATOR

The steam generator, shown in Fig. 10, is a vertical unit 16-1/2 ft. high and 45-5/8 in. in diameter. The vaporizing section consists of a "U" tube steam generating bundle surrounded by a shroud. Inside the same shell, separated from the steam generating section by a baffle plate, is a "u" tube superheating bundle. Primary water enters the heavy channel at the bottom and passes through the inside of the steam generating and superheating tubes. Secondary water enters the shell through the feedwater inlet about half way up on the steam generating section and is converted to steam by contact with the tube bundle. The steam collects at the top of the shell, passes through a moisture separator and then passes downward over the superheater section to the steam outlet.

Figure 9. Primary Coolant Pump and Casing

Figure 10 Steam Generator.

The areas which appeared to offer potential strain-cycling problems were: the vaporizer shroud, the channel partition, the shell at the centrifix base, the tube sheet, and the support brackets.

4.4.1 Vaporizer Shroud

From Fig. 10 it can be seen that the inlet feedwater can impinge on the shroud, causing a local cold spot. With the feedwater at about 250°F, the shroud at 450°F, and no mixing of feedwater and secondary water, the shroud is subjected to a local gradient of 200°. Assuming the shroud retains its cylindrical shape the maximum stress is as follows:

$$S = \frac{E \alpha \Delta T}{1 - \nu} \quad \frac{249 (450 - 250)}{.7}$$

$$S = 71,400 \text{ psi}$$

The minimum principal stress which is at the shroud surface in the direction of the thickness of the plate is zero. Therefore:

$$S_r = 71,400$$

$$S_{alt} = 71,400/2 = 35,700$$

$$S'_{mean} = 35,700$$

$$S_{alt} + S'_{mean} = 71,400 \text{ which is } > S_b$$

$$S_{alt} \text{ is also } > S_b, \text{ therefore } S_{mean} = 0$$

Plotting S_{alt} versus S_{mean} on the fatigue diagram, Fig. 11 as point "a" shows that this point falls in fatigue class II which permits 2500 cycles. It is difficult to determine how many thermal cycles the shroud may receive, however, most of them would probably occur during load changes of 50% or more. In this case, cyclic condition II of Table 2 shows 15,000 cycles. The only stress the shroud is subjected to is that due to thermal cycling. Using the method on P. 49 of the Navy Code, the actual allowable number of cycles for this stress was found to be 19,000.

4.4.2 Channel Partition

From Fig. 12, it can be seen that a 304 stainless steel partition is welded to the 5-1/2 in. thick carbon steel tube sheet and to the 4-3/4 in. thick carbon steel channel wall. The 1/2 in. thick partition is thin in comparison to the channel wall, so that it is reasonable to assume that all the differential expansion is taken by the partition.

Figure 11. Steam Generator Vaporizer Shroud and Channel Partition Stresses

At full load, the temperature on one side of the partition is 450°F and 430°F on the other. The average temperature is 440°F.

The carbon steel is SA-105 which is not listed in the Navy Code. The values used are those for A-212 steel which is similar.

At 440°F: Subscripts c and ss refer to carbon steel and stainless steel respectively.

$$E_c = 27.2 \times 10^6$$

$$\alpha_c = 6.9 \times 10^{-6}$$

$$E_{ss} = 26.8 \times 10^6$$

$$\alpha_{ss} = 9.3 \times 10^{-6}$$

Since the stress is two dimensional,

$$S = \frac{\epsilon E}{(1-\nu)}$$

Assume the partition is stress free at 70°F. Then ϵ at 440°F = $(9.3 - 6.9) \times 10^{-6} \times (440 - 70) = 888 \times 10^{-6}$

The maximum stress in the partition = $\frac{(888 \times 10^{-6})(26.8 \times 10^6)}{1 - .3} = 34,000 \text{ psi}$

The stress in the direction of the thickness of the partition is zero at both surfaces so that the maximum principal stress is also the maximum stress intensity. The intensity stress is cycled between 34,000 and 0 between cold conditions and full load. Therefore:

$$S_r = 34,000$$

$$S_{alt} = 17,000$$

$$S'_{mean} = 17,000$$

$$S'_{mean} + S_{alt} = 34,000 < S_b$$

Therefore, $S_{mean} = S'_{mean} = 17,000$

Plotting S_{mean} versus S_{alt} on the fatigue diagram, Fig. 11 as point "b", shows that this condition falls in fatigue class II which allows 2500 cycles. Cyclic condition VI and VIII of Table 2 shows a total of 2,160 cycles. This is below the allowable number of cycles so the partition is not subject to strain cycling failure.

4.4.3 Shell at Centrifix Base

The centrifix base is a horizontal baffle located inside the upper end of the steam generator shell. It separates the dry steam from the wet steam. The centrifix base is stainless steel type 304 and is welded to the carbon steel shell. Since the differential thermal expansion could cause a large stress in the shell, this area was analyzed.

Assume the centrifix base and shell to be stress free at 70°F and zero pressure. From the Operating Manual, the following information was obtained:

<u>Loading</u>	<u>Steam Outlet Temp.(°F)</u>	<u>Shell Side Pressure(psi)</u>
No load	450	422
Full load	407	200

The worst case, which was no load, was analyzed. It was assumed that the steam generator cycled between this condition and a cold shutdown.

From The Theory of Plates and Shells: (6)

$$w = \frac{F a^2 \beta}{2 E h} \quad M_{\max.} = \frac{F}{4 \beta}$$

$$\delta = \frac{P a^2}{E h} \left(1 - \frac{\nu}{2}\right) - \Delta \delta_{th}$$

$$\Delta \delta_{th} = a (\alpha_{ss} - \alpha_c) \Delta T$$

where: a = mean shell radius = 22-13/32 in.

p = pressure in shell

E = 27.2×10^6

h = Shell wall thickness

$$\beta = \frac{1.285}{\sqrt{a h}}$$

$$\nu = .3$$

F = Discontinuity force

$$\alpha_{ss} = 9.3 \times 10^{-6}$$

$$\alpha_c = 6.9 \times 10^{-6}$$

The boundary condition is,

$$\delta + w = 0$$

Substituting in values:

$$\frac{F (22.4)^2 (.302)}{2(27.2 \times 10^6)(.8125)} = - \frac{422 (22.4)^2}{27.2 \times 10^6 (.8125)} \left(1 - \frac{.3}{2}\right) + 22.4(9.3 - 6.9) \times 10^{-6} (450 - 70)$$

$$F = \frac{12,280}{3.42} = 3,590 \text{ lb/in.}$$

$$M_{\max} = \frac{3590}{4(.302)} = 2,980 \text{ in lb/in.}$$

$$\sigma_1 = \pm \frac{6 M}{h^2} + \frac{P a}{2 h} = \pm \left[\frac{6(2980)}{(.8125)^2} \right] + \frac{422 (22.4)}{2 (.8125)}$$

$$\sigma_2 = \pm \nu \frac{6 M}{h^2} + \frac{P a}{h} - \frac{E}{a} w$$

$$= \pm .3 (27100) + \frac{422 (22.4)}{(.8125)} - \frac{27.2}{22.4} (-12,280)$$

$$\sigma_3 = 0, -p$$

where - is inner surface and + is outer surface for the bending stress.

Inner Surface			Outer Surface		
σ_1	σ_2	σ_3	σ_1	σ_2	σ_3
-21,290	18,400	-422	32,910	34,640	0

where: σ_1 = axial stress
 σ_2 = circumferential stress
 σ_3 = radial stress

$S_{12} = 39,690$ for the inner surface is the maximum stress intensity.

$$S_r = 39,690$$

$$S_{alt} = 39,690/2 = 19,845$$

$$S'_{mean} = 19,845$$

$$S_{alt} = S'_{mean} < S_b \text{ therefore,}$$

$$S_{mean} = S'_{mean} = 19,845$$

Plotting this point as "a" on the fatigue diagram, Fig.13 shows that this condition falls in the fatigue class II which signifies 2,500 allowable cycles. Cyclic condition VII of Table 2 indicates 900 cycles of this type for the 20 yr life. Therefore, the shell is safe from strain cycling damage caused by the centrifix base.

4.4.4 Tube Sheet

The tube sheet is carbon steel with a stainless steel Type 304 overlay and is integral with the steam generator channel. The tube sheet is 5-1/2 in. thick while the channel wall is 4-3/4 in. thick. The tube holes are 3/4 in. and are of a 1 in. pitch triangular pattern. (Ref. Dwg. H-152400, Fig. 14)

The feedwater inlet temperature is approximately 250°F and the tube sheet is approximately that of the primary water or 450°F. The colder feedwater drops and intermixes with the secondary water. Hence, the tube sheet is washed by cool water and is, therefore, subjected to a thermal "skin effect" which causes an increase in surface or membrane stresses. The secondary water temperature washing the tube sheet is assumed to be the same as that of the secondary blowdown, since the blowdown is taken off through a penetration of the tube sheet. The blowdown is measured to be approximately 387°F. The thermal "skin effect" stress is,

Figure 13. Steam Generator Shell, Channel and Tube Sheet Fatigue Diagram

NOTE:
 DRILL & REAM TUBES FOR TUBES.
 HOLES MUST BE DRILLED BUT DO NOT REAM UNTIL
 ADVISED OF THE "D" OF THE MIN. DIA. TUBE IN
 BUNDLE. HOLES SHALL THEN BE REAMED TO .009" T.O.D.
 LARGER IN DIA. THAN THE SMALLEST O.D. TUBE IN THE
 BUNDLE.

12 TUBE HOLES MARKED THIS TO BE PLUGGED PER C-152367.

94 HOLES SHOWN THIS ON MARKED-UP
 BLUEPRINT OF DWG. D-146646 ARE TO BE
 FILLED WITH WELD METAL BEFORE ANY
 ADDITIONAL DRILLING IN TUBE SHEET IS
 DONE.

TOTAL OF 370 - 3/4" O.D. U-TUBES

ALCO		ALCO PRODUCTS, INCORPORATED	
PLANT No. 3		DUNKIRK, NEW YORK	
33" TUBE LAYOUT			
2-PASS 3/4" O.D. U-TUBES 1" TRI. PITCH			
DESIGNED	SCALE	MADE	CHECKED
43058	6"-1" O	ALP.	M.
17/7/71			
DATE		DRAWING NO.	
5/6/60		H-152400	

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

$$\sigma_{\max} = \frac{27.2 \times 10^6 (6.9 \times 10^{-6}) (450 - 387)}{1 - .3}$$

$$\sigma_{\max} = 16,900 \text{ psi}$$

The following analysis from a paper by Yu⁽⁵⁾ utilizes a classical stress analysis of a solid circular plate with a uniformly applied load, then modifies the results with a deflection efficiency in order to account for the change in flexural rigidity due to the perforations in the tube sheet. The radial moment in the plate is given by

$$M_r = \frac{3 + \nu}{16} q r^2 + D(1 + \nu) \frac{C_1}{r}$$

where C_1 is an integration constant which is defined from the edge conditions as

$$C_1 = \frac{2(K_h \theta_h - K_s \theta_s)}{D(1 + \nu) + a(K_s + K_h + K_f)} - \frac{qa^2}{8D} \left[\frac{D(3 + \nu) + a(K_s + K_h + K_f)}{D(1 + \nu) + a(K_s + K_h + K_f)} \right]$$

and

$$D = \frac{Eh^3 \eta_w}{12(1 - \nu^2)}$$

$$\eta_w = .5(1 - .952 \frac{d}{p})$$

$$\theta_s = \frac{1.285 P_s}{E_s} \left(\frac{a'}{t_s} \right)^{1.5}$$

$$\theta_h = \frac{1.285 P_h}{E_h} \left(\frac{a''}{t_h} \right)^{1.5}$$

$$K_s = \frac{E_s (t_s)^{2.5}}{4.25(a')^{.5}}$$

$$K_h = \frac{E_h (t_h)^{2.5}}{4.25(a'')^{.5}}$$

where: E = Mod. of Elasticity

h = Tube sheet thickness

η_w = Deflection efficiency

q = Net pressure = $(P_h - P_s)$

a = Tube sheet radius

a' = Mean shell radius

a'' = Mean head radius

D = Flexural rigidity

K = Rotation stiffness

θ = Edge Rotation

d = Tube O.D.

P = Tube pitch

P_s = Shell side pressure

P_h = Head (or channel) side pressure

Subscripts s and h refer to shell and head respectively.

The worst case is the full secondary load condition meaning 407°F and 200 psi on the secondary side and 440°F and 1200 psi on the primary side.

Substituting values in the equations:

$$\eta_w = .5 \left[(1 - .952) \left(\frac{.75}{1} \right) \right] = .143$$

$$D = \frac{27.2 \times 10^6 (5.5)^3 (.143)}{12(1 - .09)} = 59.3 \times 10^6$$

$$\theta_s = \frac{1.285 (200)}{27.2 \times 10^6} \left(\frac{22.8125^{1.5}}{.8125} \right) = 1420 \times 10^{-6}$$

$$\theta_h = \frac{1.285 (1200)}{27.2 \times 10^6} \left(\frac{18.875^{1.5}}{4.75} \right) = 595 \times 10^{-6}$$

$$K_s = \frac{27.2 \times 10^6 (.8125)^{2.5}}{4.25 (22.8125)^{.5}} = .8 \times 10^6$$

$$K_h = \frac{27.2 \times 10^6 (4.75)^{2.5}}{4.25 (18.875)^{.5}} = 66 \times 10^6$$

$$C_1 = \frac{2 \left[(66 \times 10^6) (595 \times 10^{-6}) - (.8 \times 10^6) (1420 \times 10^{-6}) \right]}{59.3 \times 10^6 (1.3) + 18.875 (.8 \times 10^6 + 66 \times 10^6)} \\ - \frac{1000 (18.875)^2}{8 (59.3 \times 10^6)} \left[\frac{59.3 \times 10^6 (3.3) + 18.875 (.8 \times 10^6 + 66 \times 10^6)}{59.3 \times 10^6 (1.3) + 18.875 (.8 \times 10^6 + 66 \times 10^6)} \right]$$

$$C_1 = -764 \times 10^{-6}$$

At the tube Sheet Center ($r = 0$):

$$M_r = 59.3 \times 10^6 (1.3) \frac{(-764 \times 10^6)}{2} = -29,500 \text{ in.-lbs.}$$

At the tube Sheet Edge ($r = a$)

$$M_r = \frac{3.3 (1000) (18.875)^2}{16} - 29,500 = 44,800 \text{ in.-lbs.}$$

$$\sigma_{\max} = \pm \frac{6 M_r}{h^2 \eta_\sigma} = \pm \frac{6 (44,800)}{(5.5)^2 \eta_\sigma} = \frac{8900}{\eta_\sigma}$$

where η_σ is a ligament efficiency,

$$\eta_\sigma = 1 - \frac{3}{\pi} \sin^{-1} \left(\frac{\sqrt{2} d}{2 p} \right) = .466$$

Therefore, the maximum stress at the edge of the tube sheet is,

$$\sigma_{\max} = \frac{8900}{.466} = 19,100 \text{ psi}$$

and the maximum stress at the center is,

$$\sigma_{\max} = \frac{6 (29,500)}{(5.5)^2 .466} = 12,600 \text{ Psi}$$

Stress Summary - secondary side

Type of Stress	Edge	Center
Pressure	- 19,100	+ 12,600
Thermal	+ 16,900	+ 16,900
Total	- 2,200	+ 29,500

The maximum principal stress is at the center of the tube sheet and is 29,500 psi.

The minimum principal stress is -200; therefore,

$$S_r = 29,700$$

$$S_{alt} = 29,700/2 = 14,850$$

$$S'_{\text{mean}} = 14,850$$

$$S_{\text{mean}} = S'_{\text{mean}} = 14,850$$

Plotting S_{alt} and S_{mean} as point "b" on the fatigue diagram, Fig. 13, shows that this condition falls in the fatigue class I which allows 100,000 cycles. It is difficult to estimate the number of cycles; however, it is assumed to be as cyclic condition II of Table 2 predicts which is 15,000 cycles of load changes greater than 50%.

Even though the computed stress is not severe, the tube sheet is recommended for further, more detailed analysis. The analysis used here was not refined. It gives only an estimate of stress and does not necessarily give a "worst case" or highest possible stress value. At best, computed stresses are only the average value per ligament, not the higher "peak" values, which could easily raise the stress to the 2,500 allowable cycle class. For these reasons the tube sheet should be given a more sophisticated analysis. It is a highly complex structural member, and its failure would result in a lengthy shutdown of the plant.

4.4.5 Restraint of Shell and Channel by Support Bracket

The support bracket, shown in Dwg. D-150231*, is composed of an upper ring welded to the shell, and a lower ring welded to the channel with interconnecting gussets. Neglecting the axial effect of the gussets, the effect of the rings on the shell and channel was analyzed to obtain thermal, pressure and discontinuity stresses.

(1) Lower Ring and Channel

The lower ring and channel were analyzed first. The sketch below shows the free body diagram.

* Not included in this report.

Because of the thickness of the channel wall and the stiffening effect of the gussets, it was assumed that there was no rotation of the assembly. The approach used was to match the deflections of the channel and ring to obtain the discontinuity force and moment. Based on the temperature calculations of the pressurizer support bracket, the average temperatures of the steam generator support bracket were estimated to be as follows:

Average temperature of channel = 10° less than primary water
 Average temperature of lower ring = 30° less than channel

From The Theory of Plates and Shells, ⁽⁶⁾ the deflection and rotation of the channel edge due to discontinuity forces are,

$$w = \frac{1}{2\beta^3 D} (\beta M_0 + Q_0) \quad (\text{outward deflection is positive})$$

$$\theta = \frac{1}{2\beta^2 D} (2\beta M_0 + Q_0) = 0$$

where: w = deflection

θ = rotation

$$D = \text{Flexural Rigidity} = \frac{Eh^3}{12(1-\nu^2)} = 263.5 \times 10^6$$

$$\beta = \frac{1.285}{\sqrt{ah}} \quad @ \quad \nu = .3 \quad \text{or} \quad \beta = .136$$

a = mean radius of stem = 18.875"

h = wall thickness = 4.75"

$E @ 450^{\circ}\text{F} = 26.8 \times 10^6$

Deflection:

$$w = \frac{1}{2(.136)^3 (263.5 \times 10^6)} (.136 M_0 + Q_0)$$

$$w = .1021 \times 10^{-6} M_0 + .753 \times 10^{-6} Q_0$$

Rotation:

$$\theta = - \frac{1}{2\beta^2 D} (2\beta M_0 + Q_0) = 0$$

$$\therefore M_0 = \frac{-Q_0}{2\beta} = - \frac{Q_0}{.272}$$

Substituting the value of M_0 into the deflection equation

$$w = .1021 \times 10^{-6} \left(-\frac{Q_0}{.272} \right) + .753 \times 10^{-6} Q_0$$

$$w = + .377 \times 10^{-6} Q_0 \text{ inches}$$

The pressure deflection is

$$\delta_p = \frac{P a_1}{E} \left(\frac{b^2 + a_1^2}{b^2 - a_1^2} + \nu \right) = \frac{1200 (16.5)}{26.8 \times 10^6} \left[\frac{(21.25)^2 + (16.5)^2}{(21.25)^2 - (16.5)^2} + .3 \right]$$

where a_1 = channel inner radius

$$\delta_p = 3190 \times 10^{-6} \text{ inches}$$

The thermal deflection is,

$$\delta_t = \alpha \Delta T a_1 = 6.9 \times 10^{-6} (430 - 70)(16.5)$$

$$\delta_t = 42100 \times 10^{-6}$$

Therefore, the net deflection of the channel is,

$$\delta_{\text{channel}} = + .377 \times 10^{-6} Q_0 + 45,290 \times 10^{-6}$$

The deflection of the ring is due to pressure and Q_0 alone, since rotation is assumed zero. The following free body diagram is analyzed.

where:

$$P' = 1200 - \frac{Q_0}{l_2}$$

$$a_2 = a_1 = 16.5 \text{ in.}$$

$$b_2 = 34.25''$$

$$l_2 = 3 \text{ in.}$$

$$E = 26.8 \times 10^6$$

The deflection of the ring inner surface is given by,

$$\delta_R = \frac{p' a_2}{E} \left(\frac{b_2^2 + a_2^2}{b_2^2 - a_2^2} + \nu \right)$$

$$\delta_R = 1.17 \times 10^{-6} p'$$

Substituting values of p'

$$\delta_R = -.39 \times 10^{-6} Q_0 + 1400 \times 10^{-6}$$

The deflection of the flange due to temperature change is,

$$\delta_t = \alpha \Delta T a_2 = 6.8 \times 10^{-6} (400 - 70) 16.5$$

$$\delta_t = 38,000 \times 10^{-6}$$

Adding the thermal deflection, the net deflection of the inner surface of the ring is,

$$\delta_{\text{ring}} = -.39 \times 10^{-6} Q_0 + 39,400 \times 10^{-6}$$

From the boundary conditions,

$$\delta_{\text{ring}} = \delta_{\text{channel}}$$

Therefore,

$$.377 \times 10^{-6} Q_0 + 45,290 \times 10^{-6} = -.39 \times 10^{-6} Q_0 + 39,400 \times 10^{-6}$$

$$.767 \times 10^{-6} Q_0 = -5,890 \times 10^{-6}$$

$$Q_0 = -7,670 \text{ lbs/in.}$$

$$M_0 = -\frac{Q_0}{.272} = +\frac{7,670}{.272} = 28,200 \text{ in.lbs/in.}$$

$$w = +.377 \times 10^{-6} (-7,670) = -.0029" \text{ (inward)}$$

$$\sigma_1 = \frac{P a}{2 h} + \frac{6 M_0}{h^2} = \frac{1200 (18.875)}{2 (4.75)} + \frac{6 (28,200)}{(4.75)^2}$$

$$\sigma_1 = 2390 \pm 7,480$$

$$\sigma_1 = 9,870 \text{ psi (inside); } -5,090 \text{ (outside)}$$

$$\sigma_2 = \frac{P a}{h} + \frac{E w}{a} + \frac{\nu 6 M}{h^2} = 4,780 + \frac{26.8 (-2900)}{18.875} \pm 2,240$$

$$\begin{aligned}\sigma_2 &= 4,780 - 4,120 \pm 2,240 = +660 \pm 2,240 \\ \sigma_2 &= +2,900 \text{ psi (inside); } -1,580 \text{ psi (outside)} \\ \sigma_3 &= -1200 \text{ psi (inside); } 0 \text{ (outside)}\end{aligned}$$

Maximum stress intensity = S_{31} inside = 11,070

$$S_r = 11,070$$

$$S_{alt} = 11,070/2 = 5,535$$

$$S'_{mean} = 5,535$$

$$S_{mean} = S'_{mean} = 5,535$$

Plotting S_{alt} versus S_{mean} as point "c₁" on the fatigue diagram, Fig. 13 shows that this condition falls in fatigue class I which allows 100,000 cycles. This condition is represented by cyclic condition VII of Table 2 and shows 900 cycles. The channel is therefore safe.

(2) Upper Ring and Shell

The upper ring of the steam generator support bracket was analyzed from the free body diagram below. Temperatures estimated from the temperature calculations of the pressurizer support bracket were:

$$\begin{aligned}\text{Shell} &= 5^\circ \text{ less than secondary water} \\ \text{Ring} &= 10^\circ \text{ less than shell}\end{aligned}$$

$$w = \frac{F a^2 \beta}{2 E h} \quad ; \quad M_{\max} = \frac{F}{4 \beta}$$

$$\delta = \frac{P a^2}{E h} \left(1 - \frac{\nu}{2}\right) + \Delta \delta_{th}$$

where: F = Force on shell due to upper ring

$2a$ = Shell O.D. and Ring I.D. = 45.625"

$$\beta = \frac{1.285}{\sqrt{a h}} = .211$$

$$E = 26.8 \times 10^6$$

h = Shell wall thickness = 13/16"

$$\nu = .3$$

$$\Delta \delta_{th} = a \left[(\alpha \Delta T)_{shell} - (\alpha \Delta T)_{ring} \right]$$

$$\alpha = 6.8 \times 10^{-6}$$

The boundary conditions are such that

$$\delta + w = 0$$

Therefore,

$$\frac{F (a^2) \beta}{2 E h} = - \frac{P a^2}{E h} \left(1 - \frac{\nu}{2}\right) - \Delta \delta_{th}$$

$$\frac{F (22.813)^2 (.211)}{2 (26.8 \times 10^6) (.8125)} = - \frac{422 (22.813)^2}{26.8 \times 10^6 (.8125)} \left(1 - \frac{.3}{2}\right) - 22.813 \left[6.8 \times 10^{-6} (10) \right]$$

$$F = -4,000 \text{ lb/in.}$$

$$M_{\max} = \frac{-4000}{4 (.211)} = -4,740 \text{ in.lbs/in.}$$

$$\sigma_1 = \pm \frac{6 M}{h^2} + \frac{P a}{2 h} \quad \left\{ \begin{array}{l} + \text{ inner surface} \\ - \text{ outer surface} \end{array} \right.$$

$$\sigma_2 = \pm \nu \frac{6 M}{h^2} + \frac{P a}{h} - \frac{E}{a} w$$

$$\sigma_3 = 0 \text{ outer surface; } -p \text{ at inner surface}$$

$$\sigma_1 = \pm \frac{6 (4,740)}{(.8125)^2} + \frac{422 (22.813)}{2 (.8125)}$$

$$\sigma_1 = \pm (43,100) + 5925$$

$$\sigma_1 = 49,025 \text{ psi inner surface; } -37,175 \text{ psi outer surface}$$

$$\sigma_2 = \pm .3 (43,100) + 11,850 - \frac{26.8 \times 10^6}{22.813} (-10,090 \times 10^{-6})$$

$$\sigma_2 = \pm (12,900) + 11,850 + 11,840 = \pm (12,900) + 23,690$$

$$\sigma_2 = 36,590 \text{ psi inner surface; } 10,790 \text{ psi outer surface}$$

STRESS SUMMARY

Inner Surface			Outer Surface		
σ_1	σ_2	σ_3	σ_1	σ_2	σ_3
49,025	36,590	-422	-37,175	10,790	0

$S_{31} = 49,450$ of the inner surface is the maximum stress intensity
therefore,

$$S_r = 49,450$$

$$S_{alt} = 49,025/2 = 24,725$$

$$S'_{mean} = 24,725$$

$$S_{alt} + S'_{mean} > S_b \text{ and } S_{alt} < S_b$$

$$\therefore S_{mean} = S_b - S_{alt} = 42,000 - 24,725$$

$$S_{mean} = 17,275$$

Plotting S_{alt} versus S_{mean} as point "c₂" on the fatigue diagram, Fig. 13, shows that this condition falls in fatigue class II which allows 2,500 cycles. Condition VII of Table 2 covers the condition of a total shutdown of the plant and shows 900 cycles. Therefore, this condition is safe.

5.0 CONCLUSIONS AND RECOMMENDATIONS

On the basis of this preliminary analysis the SM-1 primary system was found safe from fatigue failure with possible exception of two components, the steam generator tube sheet and the primary pump casing. These could not be sufficiently analyzed because of their geometrical complexity, to assure that the simplified methods used to analyze the other parts of the system would produce "worst case" stresses in these members. Because of their major importance they should be analyzed in detail during subtask 1.1 of the FY '62 Contract Work Program.

Three factors contributed to the capacity of the SM-1 primary system to withstand the unusual cyclic operation imposed by training and research and development testing:

1. Good thermal insulation was applied to heavy supporting structures on the pressurizer and steam generator. The rings and gussets of the support brackets might have caused severe thermal stresses if they had not been well insulated to minimize thermal gradients.
2. The large ratio (4 to 3) of design pressure to operating pressure. This resulted in operating pressure stresses only 75% of code design values, leaving a large margin to accommodate thermal stresses.
3. Relatively mild thermal transients of, at most, $30^{\circ}/\text{hr}$ have been experienced in normal operation. These were not severe enough to cause high thermal stresses. Other temperature changes experienced by the system either were few or of small magnitude, or occurred over a sufficiently long period of time so that thermal stresses again were not severe.

6.0 REFERENCES

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