(Vol. 2)

to

Volume 2
of 4 Volumes

U.S. Atomic Energy Commission
Chicago Operations Office
Lemont, Illinois

November 15, 1965

MASTER

RELEASED FOR ANNOUNCEMENT

IN NUCLEAR SCIENCE ABSTRACTS

SODIUM-HEATED STEAM GENERATOR DEVELOPMENT

AEC CONTRACT NO. AT (11-1) - 1280
B&W CONTRACT NO. 610 - 0067

THE BABCOCK & WILCOX CO.

BOILER DIVISION

Barberton, Ohio

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Report No. BAW - 1280 - 28

PRELIMINARY DESIGN REPORT
PROTOTYPE STEAM GENERATOR

Volume 2
of 4 Volumes

Vol. 2 - Stress Analysis

to

U.S. Atomic Energy Commission
Chicago Operations Office
Lemont, Illinois

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AEC Contract No. AT (11-1)-1280
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SODIUM-HEATED STEAM GENERATOR
DEVELOPMENT

Prepared By: J.E. Hocking

Approved By: P.B. Probert

Date: November 15, 1965

THE BABCOCK & WILCOX CO.
BOILER DIVISION
BARBERTON, OHIO

VOLUME - 2 - STRESS ANALYSIS

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1.0 Introduction

This preliminary design report will summarize the analytical results of the Prototype Sodium-Heated Steam Generator. This particular portion of the design report concerns itself with the structural integrity of the unit; functional considerations are not included in this section of the report.

The report is basically narrative in accordance with Paragraph 6.2.2.1 of contract #AT(11-1)-1280, Appendix C. Pg. 6, between the Babcock & Wilcox Co. and the A.E.C. The report includes index pages of the detailed calculations prepared by the B&W Co. The bulk of the detailed calculations have been omitted but concise summaries of the analytical conclusions reached to date, including pertinent assumptions and methods of solution will be given.

The primary purpose of the Prototype is to prove the structural integrity of this design under both steady-state and transient conditions so that the results can be extrapolated with confidence to the Full-Size Steam Generator.

A sodium-heated steam generator will have the same problems as any steam generator, plus the problems that may result from the high heat transfer rates and very fast temperature changes that can occur when using sodium as a heat transfer fluid. The various problems studied under Phase I of this Contract, and how these problems will be modeled in the Prototype, are discussed below under the general heading of Stress Problems.

1.1 Stress Problems

The areas which have presented stress problems are as follows:

1. Tubesheets and Thermal Sleeves:

By locating the tubesheets out of the sodium up in the inert gas space, it has been possible to protect them from rapid sodium temperature transients,

however, tubesheets for high-pressure service are a source of concern and the most sophisticated analysis techniques still leave a great deal to be desired. For this reason it was decided to model one tubesheet and thermal sleeve full size in the Prototype Steam Generator. The feedwater inlet tubesheet was selected because it has higher stresses than the boiler outlet, superheater inlet, or superheater outlet. Obviously with a 1/30-size model there will not be enough tubes to fill even one tubesheet with a full compliment of tubes. To model the deflection of the tubesheet under pressure and temperature stresses, the feedwater inlet tubesheet will be drilled with full pattern of holes of the Full-Size Steam Generator. Some of these holes will have tubes attached to them, and the rest of the holes will be blind drilled from each side of the tubesheet leaving sufficient metal at the mid-plane to prevent leaks. The thermal sleeve nozzle, attaching the tubesheet to the shell, will also be modeled full size to study the way the temperature gradients and motions of the tubesheet are carried out into the thermal sleeve nozzle. Strain gages and thermocouples will be installed and these data will be used to prove the adequacies of analytical techniques used in analyzing these thermal sleeves and nozzles.

The boiler outlet, superheater inlet, and superheater outlet tubesheets will be designed to meet the needs of the Prototype and will not necessarily model the Full-Size Steam Generator.

2, Sodium Inlet Nozzle

The stresses in the sodium inlet nozzle proved to be quite high for the Full-Size Steam Generator. This was mainly due to the high temperatures involved and the presence of the dissimilar weld (2-1/4 Croloy to Type 316 SS) in the thermal sleeve. This same nozzle, for the Prototype, has been designed to the same general configuration although reduced in diameter to accommodate the reduction in the size of the upper head and shell. The thickness of the various

parts were reduced in relationship to the decreased diameter.

For the Full-Size Steam Generator the worst stress intensity at the inside surface of the dissimilar weld area (Croloy Side) in the thermal sleeve was 31,109 psi vs. an allowable of 29,000. This allowable stress is a primary plus secondary function and the small variation was justified on the basis of the small primary pressure acting in the unit. By comparison the computed stress of the Prototype Sodium Inlet, at the inside surface of the dissimilar weld (Croloy Side) is 27,947 psi vs. an allowable of 27,600 psi. The stresses in the Prototype Sodium Inlet Nozzle compare favorably and are in the same range as the Full-Size Steam Generator Sodium Inlet Nozzle.

3. Sodium Outlet Nozzle

The sodium outlet nozzle for the Prototype Steam Generator is of the same shape and configuration as will be used in the Full-Size Steam Generator, but because the nozzle is smaller in diameter the wall thicknesses are less.

Since the center by-pass valve system has been deleted we no longer have the problems of fluctuating temperatures during operation through the nozzle due to by-passing 1140°F sodium from the top of the superheater. This nozzle will see some temperature change during the "loss of feedwater" accident; however, a check of this indicates no particular problem for the number of times (cycles) involved.

4. Flanges

The large vessel flanges are protected against rapid temperature changes by the annular inert gas space between the liner and the shell. The rate of temperature change of these flanges is slow enough that there are no significant transient stress problems. The operating pressure for the steam generator is so much lower than the design pressure that the normal operating stresses in these flanges and bolts will be extremely low. The

flanges for the Prototype Steam Generator are designed on the same basis as the Full-Size Steam Generator. Because of the slow rates of temperature change and the low operating pressure, it is felt that no significant steady-state or transient stress will be developed in these flanges.

5 Tube Flexibility Problems

Experience has shown that severe thermal expansion differences resulting in high thermal stresses can occur in sodium components. Portions of a number of sodium heat exchangers have operated beyond yield and a few have experienced tube failures due to differential expansion.

The riser and downcomer tubes of the Full-Size Steam Generator were carefully analyzed for flexibility using a sophisticated computer program to insure that the stresses in the tubes are within allowable limits.

Although the riser and downcomer tube configuration of the Prototype Steam Generator will not exactly duplicate the Full-Size Steam Generator, the same method of analysis will be used on the Prototype and its operation will demonstrate the adequacy of the method of calculation.

6 Vibration Problems

Tube failures on one Enrico Fermi steam generator because of vibration has re-emphasized the importance of this problem. The tubes and other parts of the Full-Size Steam Generator have been analyzed for vibration. The support structure has been designed to keep the natural frequency remote from any forcing function, such as Von Karman vortices or 60 cps normal plant vibration. In addition to this, the steam generator has been analyzed for vibration during shipment in a horizontal position.

The same type analysis is being applied to the Prototype Steam Generator.

Operation of the Prototype will prove out the adequacy of the method of analysis rather than modeling the vibration problems directly.

7 Heads and Shells

The heads and shells were designed to the same criteria, rules and exceptions as the 1000 MW unit. The design pressure of 300 psi was used, however, in place of the 200 psi as used on the 1000 MW unit design. The same general configuration, proportion of thickness and stress levels were maintained in the Prototype shells.

The following tabulated calculations show the stresses in the heads and shells for the units as based on Section VIII Code formulas, for the design conditions.

Heads:

Item	P	L	t	E	$S = \frac{PL + 2Pt}{2Et}$ (psi)
Upper Head 1000 MW	200	78	3	1	2580
" " 30 MW	300	36	2	1	2680
Lower Head 1000 MW	232	78	1.5	1	6000
" " 30 MW	317	36	1.0	1	5700

Shells:

Item	P	L	t	E	$S = \frac{PL + 6Pt}{Et}$ (psi)
Spool Shell 1000 MW	200	78	3	1	5050
" " 30 MW	300	36	2	1	5240
Lower Shell 1000 MW	200	78	1.5	1	10150
" " 30 MW	300	36	1.0	1	10620

The heads and shells while no particular problem from the design standpoint, because they are designed by A.S.M.E. Code, are of some concern due to the possibility of a sodium-water reaction.

Calculations were made to determine the degree of integrity or what pressure the heads and shells will sustain for allowable stresses of 0.9 yield. The following table gives these results.

Item	S	E	t	L or R	P** P.S.I.
Upper Head	18990*	1	2	36	2083
Spool Drum	"	1	2	36	1019
Shell Drum	"	1	1	36	518***
Bottom Head	"	1	1	36	1047

*S = $0.9(21,000) = 18,990$ psi. Yield taken from table N-424 A.S.M.E. Sect. III.
allow This yield is Conservative.

** $P = \frac{2SET}{L+2t}$ For Hemi-Heads

*** The inner liner would increase the integrity of this shell especially for short duration pressure pulses.

$P = \frac{SEt}{R+.6t}$ For Cylindrical Shell

Calculations were also made to determine the life expectancy of the heads and shells with a pressure of 300 psi and temperature of 1200°F. The following table gives these results.

Item	Stress	Hours to * Rupture @ 1200°F	Hours to Rupture @ 1000°F
Upper Head	2735	80,000	over 100,000
Spool Drum	5590	5,000	" "
Shell "	10499	100	" "
Bottom Head	5440	5000	" "

* The time to rupture taken from B&W Tubular Products Division Technical Bulletin 6H (For 2-1/4 Croloy) Page 69.

Calculations were also made to determine the integrity of the shells during a major sodium-water reaction. The assumption being that all 25 tubes at the top of the boiler downcomer annulus rupture and create high pressure of short time durations.

The conclusion developed from the calculations is that the liner-vessel wall complex will contain, without bursting, the dynamic pressures developed from a major sodium-water reaction.

The complete calculation, documentation and conclusions for this is included in Volume 3 of this report.

2.0 A Summary of the Analytical Results, Steam Generator

2.1 A.S.M.E. Code Calculations

All the pressure parts, including heads, shells, nozzles and flanges were sized the same as for the full size unit by using the rules set forth in Section VIII of the A.S.M.E. Code, with the following exceptions:

a) Reinforcement of openings in the circumferential plane -

The area of reinforcement supplied in the shell is figured using Para. N 452 (b) of Section III of the A.S.M.E. Code. This requires replacement of only 50% of the area removed in the circumferential plane, predicated on the fact that the longitudinal pressure stress is only 50% of the circumferential stress. This is common practice for large vessels with many penetrations around the circumference.

b) Flange Bolts - The bolts were sized using code case 1273N-7 of the A.S.M.E. Nuclear Code Cases. This allows one to go to 1/3 yield at temperature of 800F for the bolt stress.

c) Tubesheets - There is no provision in Section VIII of the A.S.M.E. Code for the design of tubesheets. The tubesheets were sized by the methods outlined in "Standards of Tubular Exchanger Manufacturers Association," better known as "TEMA". This is standard practice.

d) Allowable Stresses - The main containment shells and heads for the sodium-heated steam generator are designed to meet the following conditions.

1. For the sodium environment and possible sodium-water reaction the outer shells are designed for:

- a. Pressure 300 psi

Since this pressure is for an emergency short time condition no consideration will be taken for the environment. The allowable stress is from A.S.M.E. Section VIII at temperature.

2. The shells were also reviewed for the operating condition in a sodium environment. For this the allowable is from the curve shown on Fig. 5* at the greatest temperature in the area under consideration.

2.2 Weights

The weight of the unit dry, no sodium, is 131,000#. The weight of the unit with sodium to the normal operating level is 164,000# and the weight of the sodium itself when the unit is at normal operating level is 33,000#.

2.3 Inner Liner

The inner liner was sized using Section VIII of the A.S.M.E. Code with the pressure equal to 10.5 psi @ 650°F, the static head of sodium, the operating pressure being equalized across both sides of the liner.

2.4 Supports

The boiler and superheater support systems were not considered to be a problem area. No attempt was made to obtain the same stress levels as the loads are very small when compared to the Full-Size Unit. No attempt was made to model the supports other than to stay as nearly like the larger unit as practically possible.

a) Boiler Supports are composed of:

- 1) The coil support bars with a maximum stress of 1630 psi vs. an allowable stress of 2100 psi @ 1140°F. (2-1/4 Croloy).

*Developed Design Stresses for Croloy 2-1/4 Alloy Steel, Report BW 67-3. This curve shown in Section 5 materials Page 60.

See Page 60 of this report also.

- 2) The hanger bars with a maximum stress of 284 psi vs. an allowable stress of 2100 psi @ 1140°F. (2-1/4 Croloy).
 - 3) The main boiler support beams with a maximum stress of 2687 psi vs. an allowable stress of 6100 psi @ 1140°F. (Type 304 stainless steel).
 - 4) The boiler support shroud with a maximum stress of 1284 psi vs. an allowable stress of 6100 psi @ 1140°F. (Type 304 stainless steel).
 - 5) The boiler support rings with a maximum stress of 817 psi vs. an allowable stress of 8500 psi @ 1050°F. (Type 304 stainless steel).
- b) Superheater supports - The Type 304 stainless steel superheater supports are composed of:
- 1) The coil support bars with a maximum stress of 1763 psi vs. an allowable stress of 4500 psi @ 1200°F.
 - 2) The hanger bars with a maximum stress of 104 psi vs. an allowable stress of 4500 psi @ 1200°F.
 - 3) The main superheater support bars with a maximum stress of 1608 psi vs. an allowable stress of 4500 psi @ 1100°F.
 - 4) The superheater support shroud with a maximum stress of 730 psi vs. an allowable stress of 4500 psi @ 1200°F.
 - 5) The superheater support rings with a maximum stress of 159 psi vs. an allowable stress of 5460 psi @ 1000°F.
- 2.5 Vibration - The same general rules used for the Full-Size Unit have been followed in the Prototype Design.
- a) Tubes - The general operation of this unit in conjunction with the

turbines, pumps and other accessories could produce vibrations throughout the system. The characteristics of this system could produce vibrations of 30 or 60 cycles or some multiple of these amounts. Every attempt was made to keep the tube natural frequencies away from these excitation frequencies. In general this was done with the use of vibration snubbers or tie offs.

In the case of the helical wound portions of the tube banks, concern was given for the effect of the NA flow velocity over the tubes. Eddy frequencies based on the NA velocity were computed and care was taken to assure that the tube natural frequencies differed by at least 50% from the forcing eddy frequencies.

It is also necessary to protect the generator from resonant frequencies during shipping (should the tubular components be shipped by rail). The inherent natural frequency of a rail system is considered to be from 3 to 5 cycles per second. If the natural frequency of the tubes should fall in this range and the travel distance is great, there is a possibility of fatigue damage to the tubes. To avoid these limitations, all the natural frequencies of the tubes were kept above 41 C.P.S.

- b) The same considerations given to the design of the tubes was exercised on any other components for which vibration could possibly pose a problem.

2.6 Piping Flexibility

The stresses in the external piping from the boiler tubesheets to the superheater tubesheets have been computed in the same manner as the

Full-Size Unit by using the B&W Company's "Piping Flexibility Program." This is a sophisticated program that, given the piping size, configuration, material, temperature growth, degree of fixity, etc., will compute stresses at all pertinent points in the system. Using this program, the maximum stress obtained was 28,336 psi vs. an allowable stress of 29,524 psi @ 872°F for the 2-1/4 Croloy section and 36,481 psi vs. an allowable stress of 38,687 psi @ 872°F for Type 316 SS.

2.7 Allowable Forces & Moments

The header inlets and outlets were checked by the same method used for the Full-Size Unit for the maximum forces and moments which could be applied through the connecting piping. This also applies to the sodium inlet and outlet. The allowable stress levels are satisfactory.

2.8 Stresses in tubes due to thermal growth and dead weight

Again the B&W Company "Piping Flexibility Program" is utilized to obtain stresses due to thermal growth restraint. Allowable stresses were evaluated the same as the Full-Size Unit by using the stress range value S_A for expansion stresses from the American Standard Code for Pressure Piping Section 6. Para. 622. Stresses due to dead weight and pressure were evaluated from simple static equations and the stresses were evaluated by the allowable stress S_H (allowable stress in hot position) from the above mentioned code. Some sample stresses are given on page 115 and 116.

2.9 Stresses in boiler and superheater helical coils due to dead load

The helical coils are free to grow radially and longitudinally, thereby not inducing significant restraint due to thermal growth.

However, they must span between supports and support their own weight. Calculations are available on the stresses due to dead weight and pressure and for the boiler, the maximum dead weight and pressure stress is 3031 psi vs. an allowable of 8000 psi @ 950°F. For the superheater coils, the maximum dead weight and pressure stress is 3725 psi vs. an allowable of 8000 psi @ 1124°F. These were evaluated in similar manner to the larger unit.

The number of coil supports were reduced to four in the Prototype from eight used in the Full-Size Unit. This was done to more nearly model the span of tubes between supports and the stresses due to the combination of dead load plus pressure stresses of the larger generator.

The following tabulation will give a comparison of these conditions and results.

FULL-SIZE STEAM GENERATOR

Tube	Dia. Outer Coil	Span Between Supports	Stress psi	Allow. Stress psi
<u>Boiler</u>				
1" x .120 MW	131"	51"	5015	15000
1" x .145 MW	"	"	4007	14508
1" x .165 MW	"	"	3357	8000
<u>Superheater</u>				
7/8" x .120 MW	90-3/8"	35"	3869	8000

PROTOTYPE STEAM GENERATOR

<u>Boiler</u>				
1" x .120 MW	45-1/2"	36"	4626	15000
1" x .145 MW	"	"	3656	14508
1" x .165 MW	"	"	3031	8000
<u>Superheater</u>				
7/8" x .120 MW	44"	35"	3752	8000

2.10 Stresses in the Boiler Support & Superheater Support Shroud

1. Boiler Support Shroud - The maximum stress is 2409 psi @ 1140°F vs. an allowable stress of 6100 psi.
2. S.H. Support Shroud - The maximum stress is 730 psi @ 1200°F vs. an allowable stress of 4500 psi.

Shroud stresses were computed using elementary statics.

2.11 Support Skirt

The support skirt was examined for dead load. Due to the large diameter and cross sectional area the dead load stress in the skirt are very low and the skirt is satisfactory.

Wind loading was not considered since this prototype vessel will be installed in a pit.

Seismic conditions were considered and seismic ties have been provided at the approximate center of gravity of the unit to handle this element. For seismic calculations see Section 9.0 - Page 150.

2.12 Primary Plus Secondary Plus Steady State Thermal Stresses

The items listed below (items 6a thru 6h) have been examined for steady state stresses. All the items, except for the flanges, the support skirt and the sodium outlet nozzle have been examined using classical discontinuity methods. A discontinuity analysis of a particular configuration such as the sodium inlet nozzle consists of the following general steps:

1. Describe the configuration in terms of a finite difference gridwork.
2. Expose the configuration to thermal heating identical to what would be expected during normal operation of the unit, using the B&W Company's temperature program. Run this program until a "steady state" thermal condition has been reached.
3. "Break" the configuration up into elementary shapes such as cylinders, rings, etc. and allow these shapes to deform freely under the action of the previously determined temperatures.
4. Describe the physical characteristics of each of the above elements (including their free thermal motions computed in the above step and the pressure loads on each element) to the B&W Company Interaction Program. Run this program and determine the redundant forces and moments necessary for compatibility between each of the elements. This program also computes the stresses due to the redundants.
5. For each element, combine the redundant stresses with the static membrane stresses and any previously determined stresses existing in the free body.

6. Compare these stresses to the allowable steady state stresses. The status of the steady state calculations for the following items is given below:

6a. Sodium Inlet Nozzle

The steady state analysis of this nozzle with the combined difficulty of high temperature (1,140°F inlet sodium) and the dissimilar weld (2-1/4 Croloy to Type 316 S.S.) in the thermal sleeves has proven acceptable. The design goal for acceptable steady state stress intensity is 3 X Sa (See Item 12 Allowable Stress Intensities) Basis for Allowable Stresses Page 129. The maximum stress intensity is 37,046 psi vs. an allowable of 39,250 psi.

6b. Boiler Inlet Header

The four types of Tubesheets and Headers are calculated in the same manner. A calculating showing this method for the steady state condition on the boiler inlet header and tubesheet follows. All other tubesheets and headers will be done in like manner and only the results or conclusions will be stated.

Discussion of Problem:

This tubesheet was run on the B&W Company "Temperature Distribution Program #030" until a steady state condition was reached for the normal operating condition of 540°F steam temperature in the header. A plot of the temperature pattern obtained is shown on Page 22 of the analysis. Having obtained the temperature pattern, the tubesheet-header-skirt-shell complex is then "broken up" into a series of elemental shapes (rings, short cylinders, etc.) as shown on Page 20. These elements are then allowed to deform

freely under the action of temperature. The "free body" motions are obtained from either well-known classical formulas for those elements with simple temperature gradients (i.e.: radial or longitudinal temperature gradients) or through the use of a series of programs available to the B&W Company which can calculate these motions in elements with complex temperature patterns. The next step is to describe each element's geometrical characteristics, (radius, thickness, etc.), material, free body temperature motions and pressure loading to the B&W Company's interaction Program #90060. This program ties all the elements together in sequential order and solves for redundant moments and forces necessary to insure compatibility between adjacent elements throughout the complex. From these redundant forces and moments, the program goes on to calculate the redundant stresses at various pertinent locations.

Now these redundant stresses do not include the basic membrane stresses due to pressure (membrane stresses being the axial and circumferential pressure stresses computed by $\frac{PR}{2t}$ & $\frac{PR}{t}$ respectively.) The redundant stresses are set up by those forces and moments necessary to insure capability within the complex. The membrane stresses are known as primary stresses and the redundant stresses are known as secondary stresses. On Pages 25 through 30 are printouts of Program #90060 showing the redundant stresses at the inside and outside of each element (Page 21 shows these locations pictorially). Penciled in on these printouts are the membrane stresses at these same locations. They are then combined algebraically to obtain the maximum stress intensities as shown on these pages. The worst steady state primary and secondary stress intensity of the boiler inlet header is 40,857 psi which is below the allowable value of 68,000 psi @ 540°F.

SIGN CONVENTION
 + DEFLECTION IN DIRECTION
 OF SHEAR
 + ROTATION IN DIRECTION
 OF MOMENT

BABCOCK & WILCOX COMPANY
 DEPARTMENT

BY 7

DATE 1/31/64

BOILER IN THE FIELD NO. 1

FIG NO 610-0067

BREAK IF

OF

PIECE #1

$$\Delta = R_m \alpha \Delta T = (11.0625)(7.09 \times 10^{-6})(533-70) = 36318 \times 10^{-6}$$

$$M = \frac{\alpha E \Delta T h^2}{12(1-\nu)} = \frac{(7.09 \times 10^{-6}) E (536-531) (2.125)^2}{12(1-.3)} = 19.057 \times 10^{-6} E$$

$$D = \frac{E h^3}{10.92} = \frac{E (2.125)^3}{10.92} = 0.879 E$$

$$\beta = \frac{4 \sqrt{3(1-\nu^2)}}{\sqrt{R_m h^2}} = \frac{4 \sqrt{3(1-.3^2)}}{\sqrt{(11.0625)^2 (2.125)^2}} = 0.265$$

$$\Delta_m = \frac{M}{2\beta^2 D} = \frac{19.057 \times 10^{-6} E}{2(.265)^2 (.879 E)} = 154 \times 10^{-6}$$

$$\theta = \frac{M}{\beta D} = \frac{19.057 \times 10^{-6} E}{(.265) (.879 E)} = \underline{81.82 \times 10^{-6}}$$

$$\Delta_{TOTAL} = 36318 \times 10^{-6} + 154 \times 10^{-6} = \underline{\underline{36472 \times 10^{-6}}}$$

PIECE #2

$\theta \approx 0$ DUE TO (APPROX.) RADIAL GRADIENT

$$\text{AT JUNC. } \frac{1}{2} \Delta = R_m \alpha \Delta T = (11.0625)(7.1 \times 10^{-6})(536-70) = \underline{36,602 \times 10^{-6}}$$

$$\text{AT JUNC. } \frac{3}{5} \Delta = R_m \alpha \Delta T = (13.5)(7.14 \times 10^{-6})(558-70) = \underline{47,038 \times 10^{-6}}$$

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY PC / AGD

DATE

BOILER INLET - STEADY STATE

JOB NO. 610-0067

FREE - BY MEMORANDUM

SHEET OF

PIECE #3

RADIAL GRADIENT $\theta = 0$

$$\Delta = R \alpha \Delta T = (11.0625)(7.1 \times 10^{-6})(536-70) = \underline{36,602 \times 10^{-6}}$$

PIECE #4

$$\Delta_{TOP} = R \alpha \Delta T = (13.5)(7.14 \times 10^{-6})(560-70) = \underline{47,231 \times 10^{-6}}$$

$$\Delta_{BOT} = R \alpha \Delta T = (13.5)(7.44 \times 10^{-6})(700-70) = \underline{63,275 \times 10^{-6}}$$

$$\theta = \frac{63,275 - 47,231}{7} \times 10^{-6} = \underline{2,292 \times 10^{-6}}$$

PIECE #5

$$\Delta_{TOP} = R \alpha \Delta T = (11.0625)(7.1 \times 10^{-6})(539-70) = \underline{36,837 \times 10^{-6}}$$

$$\Delta_{BOT} = R \alpha \Delta T = (10.7288)(7.11 \times 10^{-6})(543-70) = \underline{36,081 \times 10^{-6}}$$

$$\theta = \frac{36,837 - 36,081}{1.5625} \times 10^{-6} = \underline{484 \times 10^{-6}}$$

PIECE #7

$$\Delta = R \alpha \Delta T = (13.5)(7.47 \times 10^{-6})(710-70) = \underline{64,544 \times 10^{-6}}$$

$$\theta = 0$$

JUNCTURE NO. 1

MOMENT PER RADIAN -30573.209 IN-LB
 SHEAR FORCE PER RADIAN -3980.859 LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R 11.063 IN.
 BENDING CONC. FACTOR SCF. 1.000 IN
 BENDING CONC. FACTOR SCF. 1.473 OUT
 YOUNGS MODULUS 27800000.000 PSI

MOTIONS FOR THICK ELEMENT
 TOTAL RADIAL DISPLACEMENT -.40420062, -01 IN.
 TOTAL ANGULR DISPLACEMENT -.40375962, -03 RAD.

MOTIONS FOR THIN ELEMENT
 TOTAL RADIAL DISPLACEMENT .40420063, -01 IN.
 TOTAL ANGULR DISPLACEMENT .40376200, -03 RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

	Membrane stress	REDUNDANT STRESS DISCONTINUITY	Minimum Redundant TOTAL
INSIDE SURFACE			
MERIDIONAL STRESS BASIC	+ 6612	-3675.448 PSI	+ 2937
CIRC. STRESS BASIC	+ 13224	1107.069 PSI	+ 14,331
MERIDIONAL STRESS W/S.C.F.		$\sigma_1 = -3675.448$ PSI	
CIRC. STRESS WITH S.C.F.		$\sigma_2 = +1107.069$ PSI	
		$\sigma_3 = -2,810.00$ "	
OUTSIDE SURFACE			
MERIDIONAL STRESS BASIC	+ 6612	3675.448 PSI	+ 10,287
CIRC. STRESS BASIC	+ 13224	2925.252 PSI	+ 16,149
MERIDIONAL STRESS W/S.C.F.		$\sigma_1 = +5413.935$ PSI	
CIRC. STRESS WITH S.C.F.		$\sigma_2 = +3446.798$ PSI	
		$\sigma_3 = 0.0$	

ALLOWABLE VALUE OF PRIM. & SEC. STRESS INTENSITY FOR CRO. 2 1/4 MATL AT 535°F IS 68,000 PSI

MAX. VALUE OF PRIM. & SEC. STRESS INTENSITY FOR THIS JUNCTURE IS $14,331 - (-2810) = 17,141$ PSI

JUNCTURE NO. 2

MOMENT PER RADIAN -53264.699 IN-LB
 SHEAR FORCE PER RADIAN -2677.871 LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R 11.063 IN.
 BENDING CONC. FACTOR SCF. 1.000 IN
 BENDING CONC. FACTOR SCF. 1.750 OUT
 YOUNGS MODULUS 27800000.000 PSI

MOTIONS FOR THICK ELEMENT
 TOTAL RADIAL DISPLACEMENT -.39814919,-01 IN.
 TOTAL ANGULR DISPLACEMENT -.40376200,-03 RAD.

MOTIONS FOR THIN ELEMENT
 TOTAL RADIAL DISPLACEMENT .39814921,-01 IN.
 TOTAL ANGULR DISPLACEMENT .40375970,-03 RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

$$\sigma_r = \frac{PR}{2b} \quad \sigma_z = \frac{PR}{t}$$

INSIDE SURFACE
 MERIDIONAL STRESS BASIC
 CIRC. STRESS BASIC
 MERIDIONAL STRESS W/S.C.F.
 CIRC. STRESS WITH S.C.F.

Membrane Stress	Redundant DISCONTINUITY	Redundant + membrane TOTAL STRESS
6,612	-6397.347 PSI	+205.0
13,224	-5957.441 PSI	+7267.0
	$\sigma_1 = -6397.347$ PSI	
	$\sigma_2 = -5957.441$ PSI	
	$\sigma_3 = -2,810.0$ "	

OUTSIDE SURFACE
 MERIDIONAL STRESS BASIC
 CIRC. STRESS BASIC
 MERIDIONAL STRESS W/S.C.F.
 CIRC. STRESS WITH S.C.F.

Membrane Stress	Redundant DISCONTINUITY	Redundant + membrane TOTAL STRESS
6,612	6397.347 PSI	+13009
13,224	-1411.329 PSI	+11,813
	$\sigma_1 = +11195.358$ PSI	
	$\sigma_2 = +28.073$ PSI	
	$\sigma_3 = 0.0$	

ALLOW. VALUE OF $PR_1 + SEC.$ STRESS INTENSITY FOR CRO 2 1/4 MAT'L AT 540°F IS 68000 PSI

MAX. VALUE OF $PR_1 + SEC.$ STRESS INTENSITY FOR THIS JUNCTURE IS 11,813-0 = 11,813 PSI

JUNCTURE NO. 3

MOMENT PER RADIAN	18957.517	IN-LB
SHEAR FORCE PER RADIAN	-854.649	LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R	13.500	IN.
BENDING CONC. FACTOR SCF.	1.380	IN
BENDING CONC. FACTOR SCF.	1.000	OUT
YOUNGS MODULUS	27600000.000	PSI

MOTIONS FOR THICK ELEMENT		
TOTAL RADIAL DISPLACEMENT	-.50250919,-01	IN.
TOTAL ANGULR DISPLACEMENT	-.40376200,-03	RAD.

MOTIONS FOR THIN ELEMENT		
TOTAL RADIAL DISPLACEMENT	.50250920,-01	IN.
TOTAL ANGULR DISPLACEMENT	.40375880,-03	RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

NOTE.. LOCAL FLEXIBILITY INCLUDED

INSIDE SURFACE		
MERIDIONAL STRESS BASIC	14978.778	PSI
CIRC. STRESS BASIC	10529.217	PSI
MERIDIONAL STRESS W/S.C.F	$\sigma_1 = 20670.713$	PSI
CIRC. STRESS WITH S.C.F.	$\sigma_2 = 12236.797$	PSI
	$\sigma_3 = 0.0$	

OUTSIDE SURFACE		
MERIDIONAL STRESS BASIC	-14978.778	PSI
CIRC. STRESS BASIC	1215.702	PSI
MERIDIONAL STRESS W/S.C.F	$\sigma_1 = -14978.778$	PSI
CIRC. STRESS WITH S.C.F.	$\sigma_2 = 1215.702$	PSI
	$\sigma_3 = 0.0$	

ALLOW. VALUE OF PRI. & SEC. STRESS INTENSITY FOR CRO. 2 1/4 AT 560°F IS 67,000 PSI
 MAX. VALUE OF PRI. & SEC. STRESS INTENSITY FOR THIS JUNCTURE IS 14,979.-1215 =
 16,194 PSI

JUNCTURE NO. 4

MOMENT PER RADIAN 99836.300 IN-LB
 SHEAR FORCE PER RADIAN -81136.756 LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R 11.063 IN.
 BENDING CONC. FACTOR SCF. 1.000 IN
 BENDING CONC. FACTOR SCF. 1.000 OUT
 YOUNGS MODULUS 27800000.000 PSI

MOTIONS FOR THICK ELEMENT
 TOTAL RADIAL DISPLACEMENT -.36162259, -01 IN.
 TOTAL ANGULR DISPLACEMENT .61238300, -03 RAD.

MOTIONS FOR THIN ELEMENT
 TOTAL RADIAL DISPLACEMENT .36162254, -01 IN.
 TOTAL ANGULR DISPLACEMENT -.61237180, -03 RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

	MEMBRANE STRESS	REDUNDANT DISCON.	REDUNDANT TORSION
INSIDE SURFACE			
MERIDIONAL STRESS BASIC	6,612	11990.821 PSI	+18,602
CIRC. STRESS BASIC	13,224	-10594.897 PSI	2,630
MERIDIONAL STRESS W/S.C.F		$\sigma_1 = +11990.821$ PSI	
CIRC. STRESS WITH S.C.F.		$\sigma_2 = -10594.897$ PSI	
		$\sigma_3 = -2810.10$	
OUTSIDE SURFACE			
MERIDIONAL STRESS BASIC	6612	-11990.821 PSI	-5378
CIRC. STRESS BASIC	13224	-15302.209 PSI	-2078
MERIDIONAL STRESS W/S.C.F		$\sigma_1 = -11990.821$ PSI	
CIRC. STRESS WITH S.C.F.		$\sigma_2 = -15302.209$ PSI	
		$\sigma_3 = 0.0$	

ALLOW VALUE OF PRI & SEC. STRESS INTENSITY FOR CRO2 1/4 MAT'L. AT 540°F IS 68,000 PSI
 MAX. VALUE OF PRI & SEC. STRESS INTENSITY FOR THIS JUNCTURE IS -2810-18,602 =
 -21,412 PSI.

JUNCTURE NO. 5

MOMENT PER RADIAN 261730.310 IN-LB
 SHEAR FORCE PER RADIAN 126179.310 LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R 11.063 IN.
 BENDING CONC. FACTOR SCF. 1.380 IN
 BENDING CONC. FACTOR SCF. 1.000 OUT
 YOUNGS MODULUS 27800000.000 PSI

MOTIONS FOR THICK ELEMENT
 TOTAL RADIAL DISPLACEMENT .35559367,-01 IN.
 TOTAL ANGULR DISPLACEMENT .76754700,-04 RAD.

MOTIONS FOR THIN ELEMENT
 TOTAL RADIAL DISPLACEMENT -.35559366,-01 IN.
 TOTAL ANGULR DISPLACEMENT -.76764000,-04 RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

INSIDE SURFACE		MEMBRANE STRESS	REDUNDANT DISCONT.	TOTAL REDUNDANT + M.
MERIDIONAL STRESS BASIC	6,612		31435.073 PSI	+ 38,047
CIRC. STRESS BASIC	13,224		-2332.451 PSI	+ 10,892
MERIDIONAL STRESS W/S.C.F			$\sigma_1 = 43380.400$ PSI	
CIRC. STRESS WITH S.C.F.			$\sigma_2 = 1251.148$ PSI	
			$\sigma_3 = -2810.0$	
OUTSIDE SURFACE				
MERIDIONAL STRESS BASIC	6,612		-31435.073 PSI	- 24,823
CIRC. STRESS BASIC	13,224		-19132.027 PSI	- 5,908
MERIDIONAL STRESS W/S.C.F			$\sigma_1 = -31435.073$ PSI	
CIRC. STRESS WITH S.C.F.			$\sigma_2 = -19132.027$ PSI	
			$\sigma_3 = 0.0$	

ALLOW. VALUE OF PRI. + SEC. STRESS INTENSITY FOR CRO 2 1/4 MAT'L AT 540°F IS 68000 PSI
 MAX. VALUE OF PRI. + SEC. STRESS INTENSITY FOR THIS JUNCTURE IS -2810-38,047 =
 -40,857 PSI

JUNCTURE NO. 6

MOMENT PER RADIAN	-32945.142	IN-LB
SHEAR FORCE PER RADIAN	15288.310	LBS.

THE ABOVE VALUES MAY BE DIV. BY THE MEAN RAD. R, TO OBTAIN PER IN. CIRC. LOADS.

JUNCTURE MEAN RADIUS, R	13.500	IN.
BENDING CONC. FACTOR SCF.	1.000	IN
BENDING CONC. FACTOR SCF.	1.100	OUT
YOUNGS MODULUS	26600000.000	PSI

MOTIONS FOR THICK ELEMENT		
TOTAL RADIAL DISPLACEMENT	-.64544000,-01	IN.
TOTAL ANGULR DISPLACEMENT	.00000000, 00	RAD.

MOTIONS FOR THIN ELEMENT		
TOTAL RADIAL DISPLACEMENT	.64544000,-01	IN.
TOTAL ANGULR DISPLACEMENT	.80000000,-09	RAD.

INTERACTION STRESSES, BASED ON SMALLER JUNCTURE THICKNESS

NOTE.. LOCAL FLEXIBILITY INCLUDED

INSIDE SURFACE		
MERIDIONAL STRESS BASIC	-26030.729	PSI
CIRC. STRESS BASIC	-5540.847	PSI
MERIDIONAL STRESS W/S.C.F	$\sigma_1 = -26030.729$	PSI
CIRC. STRESS WITH S.C.F.	$\sigma_2 = -5540.847$	PSI

$\sigma_3 = 0.0$

OUTSIDE SURFACE		
MERIDIONAL STRESS BASIC	26030.729	PSI
CIRC. STRESS BASIC	9954.975	PSI
MERIDIONAL STRESS W/S.C.F	$\sigma_1 = 28633.801$	PSI
CIRC. STRESS WITH S.C.F..	$\sigma_2 = 10735.897$	PSI

$\sigma_3 = 0.0$

ALLOW. VALUE OF PRI. & SEC. STRESS INTENSITY FOR CRO 2 1/4 MAT'L AT 700°F IS 62,000 PS
 MAX VALUE OF PRI & SEC. STRESS INTENSITY FOR THIS JUNCTURE IS -26030 - (0) = 26,030 PS

6c. Boiler Outlet Header

The maximum primary plus secondary plus steady-state thermal stress intensity is 35,875 psi vs. an allowable stress of 37,400 psi @ 765°F.

6d. Flanges

No steady state discontinuity has been performed on the flanges since they are basically isothermal at steady state and a discontinuity analysis would not give stresses as severe as a Standard ASME Section VIII Flange Analysis. This analysis gives us a maximum stress of 10,437 psi vs. this allowable stress of 21,000 psi @ 900°F.

6e. Support Skirt

No steady-state discontinuity analysis was made for the prototype support skirt. This skirt is similar to the one on the Full-Size Steam Generator in which the stresses were low and acceptable.

6f. Sodium Outlet Nozzle

No steady-state discontinuity has been performed on the sodium outlet nozzle since it is basically isothermal at steady-state. During operation the fluid sodium temperature and the nozzle wall are at the same temperature. There will be pressure induced membrane stresses but they will be quite small because of the low operating pressure and can be considered insignificant.

6g. S.H. Inlet Header

The maximum primary plus secondary steady-state stress intensity is 38,794 psi vs. an allowable stress of 47,200 psi @ 770°F.

6h. S.H. Outlet Header

The maximum primary plus secondary steady-state stress intensity is 39,137 psi vs. an allowable of 40,600 psi @ 1030°F.

2.13 Thermal Transients

Thermal transient discontinuity analyses basically follow the rules for steady state discontinuities outlined in steps 1 thru 6 of Section 2.13 of this report. The differences are that for a transient analysis, the whole procedure is followed for a sequence of times during the transient. Using sound engineering judgment, it is possible to pick the times during the transient when the stress intensities at certain critical locations are either at a maximum or at a minimum. Once these maximum and minimum stress intensities are found, we can obtain the alternating components of stress which contribute to fatigue damage. The cumulative damage from all the transients are combined using Miner's Hypothesis.

Now if a component is exposed to little or no temperature excursions during the various operating and emergency transients, or if the temperature excursion is border line significant but the number of cycles of the transient is less than 100, it is obvious that a transient evaluation is unnecessary. Based on this statement, we can conclude that the flanges, and support skirt, do not require any transient evaluation.

The following items do require some transient evaluation:

1. S.H. Outlet Header.
2. S.H. Inlet Header.
3. Boiler Outlet Header.

The above items have been designed to meet the needs of the Prototype design and do not necessarily model the Full-Size Steam

Generator. These designs were analyzed using the same thermal transients as used on Full-Size unit. The tabulation of usage factors on page 35 indicates the stress values are within acceptable limits.

4. Boiler Inlet Header

This design is an exact model of the Boiler Inlet Header on the Full-Size Steam Generator, except that the Prototype has a 2" shell and the Full-Size unit has a 3" shell. The shell thickness does not enter into the calculation and the slight difference in the length of the thermal sleeve would only tend to decrease the stresses.

5. Sodium Inlet Nozzle

The size of this nozzle does not exactly model the full size unit, however the basic configuration and proportions were maintained.

The nozzle was analyzed for the same transients as the Full-Size Unit and the stress results do model the larger unit. The maximum stress reversal occurred at the same juncture location for both units and was 61,059 psi on the Prototype vs. 59,521 psi on the large unit. Also the stress reversal at the bimetallic weld was 11,216 psi on the prototype vs. 11,571 psi on the Full-Size Unit.

6. Sodium Outlet Nozzle

This nozzle does not model the large unit because of the removal of the valve and center pipe. Calculations for the transient condition of loss of feedwater by classical methods proves the design to be satisfactory.

An evaluation has been made as to the transients and the work required for each of the above items, this breakdown of the necessary items has been shown on pages 35 thru 46.

A tabulation of the transients and the usage factors for each item is shown on page 35 . This shows that the cumulative usage factors are either zero or below 1.0. The various components subjected to stress cycling conditions are satisfactory.

Tubesheet Analysis

The procedure employed in calculating the tubesheet stresses is the same as that used in the B&W Computer Program 094. This program was used in the full-size unit, but since that time it has been concluded that this did not give a complete stress picture because the program did not consider the total geometry of the header; thus the moment and shear force were obtained from the interaction analysis, and then the same procedure was followed to obtain the stresses.

Calculations were made on the Boiler Inlet because this design is identical for the full-size and the prototype units. The Superheater Inlet was also checked because it had the largest and most severe temperature gradient across the tubesheet. The usage factors from both of these analyses are considerably below the allowables.

It is felt that the worst conditions have been considered, and based upon the results obtained, it would be safe to conclude that the Boiler Outlet and S.H. Outlet tubesheets are also satisfactory; therefore, it is unnecessary to perform calculations on these tubesheets.

Item	Transients Evaluated	Location of points having usage factors	Usage factor for these points			Total Usage Factor	Comments
			trans.	point	U.F.		
a) S.H. Outlet H'd'r.	#7	H'd'r.				0.0	No usage factor at all.
b) S.H. Inlet H'd'r.	#7 #9	H'd'r.	#9 #9 #9 #9	1 2 3 4	.035 .010 .035 .012	.035 .010 .035 .012	Usage factor below 1.0 therefore design is acceptable.
c) Boiler Outlet H'd'r.	#7 #9	H'd'r.				0.0	No usage factor at all.
d) Boiler Inlet H'd'r.	#3 #4 #7 #8 #10	H'd'r.	#10 #10 #10 #10 3&4 3&4	1 2 3 4 5 6	0.60 0.25 0.50 0.50 0.10 0.18	0.60 0.25 0.50 0.50 0.10 0.18	All usage factors are below 1.0. Therefore the design is acceptable.
e) NA Inlet Nozzle	#7 #8	NA Inlet	#7 8	1 1	.025 .025	0.050	All usage factors are below 1.0. Therefore the design is acceptable.
f) NA Outlet Nozzle	#9	NA Outlet	#9	1	0.50	0.50	Usage factor below. Therefore design is acceptable.

The transient evaluation results on Items a thru f are as follows:

TEMPERATURE TRANSIENTS

The steam generator being designed under this Contract will be subjected to transient conditions. These conditions exist in both the sodium side and the water side of this unit.

The following is a list of transients to be analyzed. On the attached sheets are temperature vs. time plots for various transients. On each plot is a temperature history for the sodium inlet and outlet, and the boiler and superheater inlets and outlets.

Transient No. 1. - Cold Start-up - Number of Cycles - 1000

Cold Condition Temperature - 350F

Final Sodium Inlet Temperature - 1140F

Final Superheater Outlet Temperature - 1050F

Time Required - 48 hours

The reverse of this transient will also exist, that is a normal shut-down for 1000 cycles.

Transient No. 2 - Normal Sodium Temperature Fluctuations - Number of Cycles 10^6

The inlet sodium temperature varies between 1135F and 1140F, a 5F difference. This small fluctuation is too small to prevent any serious temperature effects on the boiler. No sketch is attached.

Transient No. 3 - Normal Power Increase - Number of Cycles - 1000

Sodium Inlet - 1070F to 1140F in 15 minutes

Sodium Outlet - 494F to 644F in 15 minutes.

Feedwater Inlet 370F to 530F in 15 minutes

Superheater Outlet - Temperature fluctuates ± 30 F around a constant superheat temperature of 1050F.

Transient No. 4 - Normal Power Decrease - Number of Cycles - 1000

Sodium Inlet - 1140F to 1070F in 15 minutes

Sodium Outlet - 644F to 494F in 15 minutes

Feedwater Inlet - 530F to 370F in 15 minutes

Superheater Outlet - Temperature fluctuates ± 30 F around a constant
superheat temperature of 1050F.

Transient No. 5 - Fast Power Change - Number of Cycles - 25

This transient as shown on the attached sketch starts at any operating point from 20% to 100% load. A $\pm 20\%$ of full load/minute in one minute transient is imposed on the boiler. The plot shows the temperature changes at 80% load. The changes in fluid temperatures will be the same for any load, but the beginning and ending temperatures will vary.

Transient No. 6 - Step Change in Power Level - Number of Cycles - 25

This transient can start at any operating point and consists of a $\pm 10\%$ of full load change in the fluids.

Change in Sodium Inlet Temperature - ± 9 F

Change in Sodium Outlet Temperature - ± 15 F

Change in Feedwater Inlet - ± 16 F

Change in Superheater Outlet - ± 15 F

No sketch for this transient is attached.

Transient No. 7 - Primary Sodium Pump Stops - Secondary sodium flow to generator
continues - Number of Cycles - 25

The water will continue to flow and remove heat from the sodium until an equilibrium temperature of 650F is reached. All temperatures, except the boiler inlet temperature will approach 650F. The boiler inlet temperature will level off at 370F.

Transient No. 8 - Sodium Flow Stops - Feedwater flow continues. Number of
Cycles - 25

This transient is similar to Transient Number 7 as far as temperature transients are concerned.

Transient No. 9 - Feedwater Flow Stops - Sodium flow continues - Number of
Cycles - 25

The boiler fluid temperatures will approach the 1140F equilibrium temperature. This is based on the assumption that the sodium by-pass fails to operate and all flow goes over the boiler tubes. Therefore, the steam and water temperatures approach 1140F at a faster rate than normal. The feedwater temperature goes from 530F to 1140F in one hour.

Transient No. 10 - Sodium Excursion - Number of Cycles - 10,000

The boiler load will drop from full load to zero load in 15 minutes. Boiler will be held at zero load temperatures until it can either be returned to full power or taken off the line.

TRANSIENT NO. 1
COLD START-UP
1000 CYCLES

TRANSIENT NO. 3
POWER INCREASE
1000 CYCLES

TRANSIENT NO. 4
POWER DECREASE
1000 CYCLES

TRANSIENT NO. 5
FAST LOAD CHANGE
25 CYCLES

TRANSIENT NO. 7
 PRIMARY N₂ PUMP STOPS
 SECONDARY N₂ FLOW AND
 FEEDWATER FLOW CONTINUES

TRANSIENT NO. 8
SODIUM FLOW STOPS
FEED WATER FLOW CONTINUES
25 CYCLES

TRANSIENT NO. 9
 FEEDWATER FLOW STOPS
 SODIUM FLOW CONTINUES
 25 CYCLES

TRANSIENT NO. 10
SODIUM EXCURSION
10,000 CYCLES

TRANSIENT ANALYSIS ON PROTOTYPE

ITEM	Steady State	TRANS. 1	TRANS. 2	TRANS. 3	TRANS. 4	TRANS. 5	TRANS. 6	TRANS. 7	TRANS. 8	TRANS. 9	TRANS. 10	
NA INLET	T	N	N	N	N	N	N	T	T	N	N	
NA OUTLET	T	N	N	N	N	N	N	N	N	T	N	
BOILER INLET TUBESHEET	T	N	N	T	T	N	N	T*	T*	N	T	
BOILER OUTLET TUBESHEET	T	N	N	N	N	N	N	T	N	T	N	
S.H. INLET TUBESHEET	T	N	N	N	N	N	N	T	N	T	N	
S.H. OUTLET TUBESHEET	T	N	N	N	N	N	N	T	N	N	N	

N - NEGLIGIBLE, NO ANALYSIS REQ'D.

T - SEE PG. FOR TRANSIENT EVALUATION RESULTS.

*TRANSIENT #7&8 MAY BE COMBINED INTO ONE TRANSIENT.

3.Q Conclusions

Stresses in the Prototype as a unit have been held at levels acceptable to the B&W Company for the steady-state and transient analysis as covered in the body of the report. Natural frequencies are within bounds set up as acceptable. The areas chosen for modeling have proven to be satisfactory in this respect. As of this report, the B&W Company feels that the structural integrity of the Prototype as a unit has been confirmed and the design goal of modeling the problem areas has been met by the unit.

The following pages titled "Index to Calculations Performed," lists under their various heading the calculations made and from which this report was gathered. Copies of these calculations will be available on request.

4.0

INDEX TO CALCULATIONS PERFORMED

4.1 A.S.M.E. Calculations

DESIGN DATA & MATERIALS.

HEADS & SHELLS.

1. Pressure Thickness Req'd. for Vessel Top Head
2. Pressure Thickness Req'd. for Spool Drum & Shell Drum
3. Pressure Thickness Req'd. for Vessel Bottom Head

SODIUM INLET NOZZLE.

1. Pressure Thickness Req'd. for Pipe Connector.
2. Pressure Thickness Req'd. for Nozzle
3. Pressure Thickness Req'd. for Nozzle Attachment to Shell
(Thermal Sleeve)
4. Area of Reinforcement Req'd. at Nozzle on Shell
5. Reinforcement Limits
6. Reinforcement Available

BOILER INLET HEADER.

1. Pressure Thickness Req'd. for Header Shell
2. Pressure Thickness Req'd. for Header Hemispherical Head
3. TEMA - Tube Sheet Thickness for Pressure
4. Thermal Sleeve Thickness.
5. Distance from Flange to Header
6. Reinforcement Required in Shell & Nozzle for Header Penetration
7. Reinforcement in Plane Normal to Center Line of Vessel

8. Pressure Thickness of Pipe Connection to Header Nozzle
9. Reinforcement Calculations for Inlet Nozzle & Handhole Openings
10. Pressure Thickness for Blind Holes in Tube Sheet
11. Pressure Thickness of Piping into Header
12. Pressure Thickness of Drain Connection
13. Handhole Plugs

BOILER OUTLET HEADER.

1. Pressure Thickness Req'd. for Header Shell
2. Pressure Thickness Req'd. for Header Hemispherical Head
3. TEMA - Tube Sheet Thickness for Pressure
4. Thermal Sleeve Thickness
5. Distance from Flange to Header
6. Reinforcement required in Shell & Nozzle for Header Penetration
7. Pressure Thickness of Pipe Connection to Header Nozzle
8. Reinforcement for outlet nozzle and handhole openings
9. Pressure Thickness for Blind holes in Tube Sheet
10. Pressure Thickness of Drain Connections
11. Handhole Plugs

SUPERHEATER INLET HEADER.

1. Pressure Thickness Req'd. for Header Shell
2. Pressure Thickness Req'd. for Header Hemispherical Head
3. TEMA - tube sheet thickness for pressure
4. Thermal Sleeve Thickness

FLANGES.

SODIUM OUTLET NOZZLE.

1. Pressure Thickness of Pipe Connection ($2\frac{1}{4}$ Groloy section)
2. Pressure Thickness of pipe connection (stainless steel section)
3. Pressure Thickness nozzle section
4. Reinforcement required in the lower head
5. Pressure thickness of outlet drain

SAFETY HEAD AND RUPTURE DISK NOZZLE.

1. Pressure Thickness of Nozzle
2. Area of reinforcement req'd.
3. Area of reinforcement available

SAFETY VALVE NOZZLE.

1. Pressure Thickness of Nozzle
2. Area of Reinforcement req'd.
3. Area of Reinforcement available

TUBES.

1. Pressure Thk. 1" O.D.x.120" MW. Boiler tubes
2. Pressure Thk. 1" O.D.x.145" MW. Boiler tubes
3. Pressure Thk. 1" O.D.x.165" MW Boiler tubes
4. Pressure Thk. 7/8" O.D.X.120" MW, S.H. tubes

5. Pressure Thk. 5/8" O.D.x.076" MW. Boiler tubes

HYDROSTATIC TEST PRESSURES FOR BOILER SECT.

HYDROSTATIC TEST PRESSURE FOR S.H. SECTION.

HYDROSTATIC TEST PRESSURE FOR VESSEL SHELLS & HEADS.

PNEUMATIC TEST PRESSURE FOR VESSEL SHELLS & HEADS.

4.2 WEIGHTS.

1. Shell section
2. Boiler section
3. Superheater section
4. Water in tubes
5. Sodium

4.3 MAIN SHELL & HEADS.

1. Req'd. thickness for operating condition
2. " " " allowable stress of 0.9 yield.
3. Life expectancy of shells & heads for 300 psi @ 1200°F.

4.4 INNER LINER.

1. Thickness of Head
2. " " Shell

3. Thickness of Head & Shell for Emergency Temp. of 1140°F.

4.5 SUPPORTS.

1. Boiler Supports
2. Superheater Supports

4.6 VIBRATION.

1. Boiler Inlet Tubes (Downcomers)
2. " Outlet Tubes (Risers)
3. S.H. Inlet Tubes (Downcomers)
4. " Outlet Tubes (Inlet)
5. Boiler Helical Coils due to Sodium Flow
6. S.H. " " " " " "
7. Sodium Distribution System

4.7 INTERCONNECTING PIPE - BOILER OUTLET TO S.H. INLET.

4.8 ALLOWABLE FORCES & MOMENTS.

1. Boiler Inlet Header
2. " Outlet "
3. S.H. Inlet "
4. S.H. Outlet "
5. Sodium Inlet
6. " Outlet

4.9 STRESSES IN TUBES DUE TO EXPANSION AND DEAD LOAD.

1. Boiler Downcomer
2. " Outlet
3. S.H. Downcomers
4. S.H. Outlet

4.10 STRESSES IN HELICAL COILS DUE TO DEAD LOAD AND LONGITUDINAL PRESSURE.

1. Boiler
 - 1" O.D.x.165 thickness
 - 1" O.D.x.145 "
 - 1" O.D.x.120 "
2. Superheater

4.11 SHROUDS.

1. Boiler Support
2. S.H. "
3. Sodium Inlet

4.12 SUPPORT SKIRT & SUPPORT BASE.

4.13 SEISMIC LOADING AND SNUBBERS.

4.14 STEADY-STATE CALCULATIONS.

1. Sodium Inlet Nozzle
2. Boiler Inlet Header

3. Boiler Outlet Header

4. S.H. Inlet Header

5. S.H. Outlet "

4.15 THERMAL TRANSIENTS.

1. Sodium Inlet Nozzle.

2. Boiler Inlet Header

3. Boiler Outlet Header

4. S.H. Inlet Header

5. S.H. Outlet Header

6. Sodium Inlet Nozzle

5.0 Materials.

The materials being used in the Prototype Steam Generator are exactly the same for all modeled areas as used in the Full-Size Steam Generator. Croloy 2-1/4 material was selected for the lower temperature portions of the steam generator (boiler tubes, boiler supports, shells, heads, boiler inlet and outlet headers). At the temperatures above the use limit of 2-1/4 Croloy, Type 316 Stainless Steel was selected because of its superior strength at the elevated temperature portions of the steam generator (superheater tubes, sodium inlet nozzle, superheater inlet and outlet headers). This material selection was analyzed and justified in The Babcock & Wilcox Company Report No. BW 67-3. "The Combined Effects of a Sodium Environment and Extended Life on Type 316 Stainless Steel and Croloy 2-1/4 Alloy Steel Design Stresses."

For the prototype the general exception to the use of these aforementioned material has been in the substitution of Type 304 Stainless Steel for Type 316 Stainless in the supports and shrouds. This was done for an economic reason based on the following facts.

1. The relatively small number of tubes in the Prototype Steam Generator (25 boiler, 45 superheater compared to the Full-Size Steam Generator (676 boiler, 1220 superheater) give very light loads (Prototype, boiler 11,000#, superheater 4,400# as compared with the larger unit (boiler 167,000#, superheater 70,000#).

2. The short term test life of the unit combined with the light loads reduces the probability of creep being a factor in the design.

3. While no direct modeling of the stresses in the supports and shrouds has been attempted or deemed necessary, the general configurations have been maintained and the status of the Prototype has been kept as nearly like the Full-Size Unit as possible. The limits of practical design, assembly and manufacturing set sizes for the smaller unit which in general produce stresses in the supports system greatly below those in the larger unit. This makes it economically more feasible to use the Type 304 stainless steel with its lower allowable stress level without sacrificing the integrity of the unit in any respect.

The two curves on the following pages were taken from The Babcock and Wilcox Co. Report No. BW 67-3 titled "The Combined Effects of a Sodium Environment and Extended Life on Type 316 Stainless Steel and Croloy 2-1/4 alloy steel design stresses." They have been noted in various places in this report and are included for reference.

DEVELOPED DESIGN STRESSES FOR
CROLOY 2 1/4 ALLOY STEEL IN A
SODIUM ENVIRONMENT AND A 30
YEAR LIFE

DEVELOPED DESIGN STRESSES FOR TYPE 316
STAINLESS STEEL IN A SODIUM ENVIRONMENT
AND A 30 YEAR LIFE

6.0 Summations of Stresses Obtained in the Various Component
of the Steam Generator.

The following pages cover individual stresses in the various components giving the required thicknesses, allowable stresses, the basis for the allowables, etc.

Shells

Item	Mat'l.	Temp.F	Press. psi	Thickness Req'd. (in.)	Thickness Actual (in.)	Allowable Stress psi	Basis for Allowable Stress Level *
<hr/>							
(Design)							
Top Head	SA-387-GrD	900	300	0.4763	2	13,100	7. 1
Top Shell Drum	"	900	300	0.8999	2	13,100	7. 1
Spool Drum	"	900	300	0.8999	2	13,100	7. 1
Shell Drum	"	900	300	0.8999	1	13,100	7. 1
Bottom Head	"	900	317	0.499	1	13,100	7. 1
<hr/>							
(Operating)							
Top Head	"	900	20	0.093	2	11,750	7. 2
Top Shell Drum	"	900	20	0.124	2	11,750	7. 2
Spool Drum	"	900	20	0.124	2	11,750	7. 2
Shell Drum	"	600	20	0.1106	1	15,000	7. 2
Bottom Head	"	600	37	0.150	1	15,000	7. 2
<hr/>							
(Operating)							
Drum Liner	Plate Croloy 2 $\frac{1}{4}$ (Min. Tensile at ambient temp. 60,000 psi)	650	9.0	0.0196	$\frac{1}{2}$ "	15,000	7. 3
Head Liner	"	650	10.5	0.0114	$\frac{1}{2}$ "	15,000	7. 3
<hr/>							
(Emergency)							
Drum Liner	"	1140	8	0.1153	$\frac{1}{2}$ "	2268	7. 4
Head Liner	"	1140	9	0.0648	$\frac{1}{2}$ "	2268	7. 4

* The notation "1", "2", "3", etc., in this column refers to the explanation establishing the criteria for the allowable stresses. For these items see Pages 126 thru 137. Wherever possible the explanation is given in the column.

SODIUM OUTLET NOZZLE

BABCOCK & WILCOX COMPANY
DEPARTMENT

BY N.G.D.

DATE 7-1-64

30 MWIT PROTOTYPE

JOB NO.

SODIUM OUTLET NOZZLE

SHEET OF

Sodium Outlet.

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress psi	Basis for Allowable Stress Level
(Design. Outlet Nozzle	SA182-F22	900	300	0.2117	1"	14,000	A.S.M.E. Section VIII Code Allowable at Temperatures.
Drain Nozzle	Bar 2 $\frac{1}{4}$ Croloy Min. Tensile * 70,000 psi.	900	300	0.0785	.20	14,000	"
Pipe Conn. Extension	SA312-Type 316	900	300	0.1127	1/2"	16,000	"

(Operating)

Outlet Nozzle	SA182-F22	650	20	0.0728	1"	17,500	"
Pipe Conn. Ext.	SA312-Type 316		20	0.0107	1/2"	17,050	"

* Min. Tensile @ Ambient Temp.

Allowable Force and Moment, Sodium Outlet

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N-2 SODIUM OUTLET, ALLOWABLE FORCES AND MOMENTS

FIG. N 2-1 EXPANDED CURVE OF SODIUM OUTLET ALLOWABLE FORCES AND

MOMENTS

SHEET _____ OF _____

Boiler Inlet Header

A. Design Conditions

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress	Basis
Shell	SA182-F22	530	2825	1.861	2-1/8	17,500	7.7
Hemi Head	"	"	"	0.888	2-1/8	"	7.7
Tube Sheet	"						
a) Bending		530	2825	3.236	3-1/2	17,500	7.7
b) Shear				1.422	3-1/2	17,500	7.7
Thermal Sleeve	"	900	300	.3498	3/4	14,000	7.7
Handhole Plugs	"	530	Safe to 4180 psi				7.19

B. Operating Conditions

See following pages for steady state stress intensities and tabulation of maximum stress reversals and usage factors for the transient conditions.

STRESS INTENSITIES

σ_1 = LONGITUDINAL STRESS
 σ_2 = CIRCUMFERENTIAL "
 σ_3 = RADIAL "

LANCOK & WILCOX COMPANY

REPORT

BY E.W.

DATE

BOILER - LANCOK & WILCOX COMPANY

JOB NO. 610-0067

STEADY STATE (550°)

SHEET OF

JUNCTURE	TRANSIENT	N	MAX. STRESS REVERSAL		N		U.F.	
			OUTSIDE	INSIDE	OUT	IN	OUT	IN
1	10	10,000	$\bar{V}_{2-3} = 25,615$	$\bar{V}_{1-2} = 33,229$	40×10^3	15×10^3	.25	.60
	7½8	50	$\bar{V}_{3-1} = 6,535$	$\bar{V}_{3-1} = 4,437$	∞	∞	0	0
	3¾4	2,000	$\bar{V}_{1-2} = 10,168$	$\bar{V}_{2-3} = 5,914$	∞	∞	0	0
	TOTAL						.25	.60
2	10	10,000	$\bar{V}_{3-1} = 30,634$	$\bar{V}_{3-1} = 17,505$	20×10^3	20×10^4	.50	~0
	7½8	50	$\bar{V}_{3-1} = 10,146$	$\bar{V}_{1-2} = 5,924$	∞	∞	0	0
	3¾4	2,000	$\bar{V}_{1-2} = 9,152$	$\bar{V}_{2-3} = 7,435$	∞	∞	0	0
	TOTAL						.50	0
3	10	10,000	$\bar{V}_{3-1} = 18,667$	$\bar{V}_{1-2} = 29,490$	∞	20×10^3	0	.50
	7½8	50	$\bar{V}_{2-3} = 7,566$	$\bar{V}_{1-2} = 9,083$	∞	∞	0	∞
	3¾4	2,000	$\bar{V}_{1-2} = 14,003$	$\bar{V}_{1-2} = 23,632$	∞	∞	0	∞
	TOTAL						0	.50
4	10	10,000	$\bar{V}_{3-1} = 6,840$	$\bar{V}_{3-1} = 6,840$	∞	∞	0	0
	7½8	50	$\bar{V}_{3-1} = 5,846$	$\bar{V}_{3-1} = 5,811$	0	0	0	0
	3¾4	2,000	$\bar{V}_{3-1} = 4,744$	$\bar{V}_{1-2} = 22,048$	0	76×10^3	0	~0
	TOTAL						0	0
5	10	10,000	$\bar{V}_{2-3} = 5,559$	$\bar{V}_{1-2} = 9,287$	∞	∞	0	0
	7½8	50	$\bar{V}_{1-2} = 11,338$	$\bar{V}_{2-3} = 11,290$	∞	∞	0	0
	3¾4	2,000	$\bar{V}_{3-1} = 29,975$	$\bar{V}_{3-1} = 5,884$	2×10^4	∞	.10	0
	TOTAL						.10	0
6	10	10,000	$\bar{V}_{3-1} = 6,461$	$\bar{V}_{3-1} = 5,873$	∞	∞	0	0
	7½8	50	$\bar{V}_{3-1} = 2,678$	$\bar{V}_{1-2} = 2,998$	∞	∞	0	0
	3¾4	2,000	$\bar{V}_{3-1} = 3,583$	$\bar{V}_{3-1} = 5,221$	∞	∞	0	0
	TOTAL						0	0

Allowable Force and Moment, Boiler Inlet Header

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N-4 BOILER INLET ALLOWABLE FORCES AND MOMENTS

FIG. N 4 -- EXPANDED CURVE FOR BOILER INLET ALLOWABLE FORCES AND
MOMENTS

BADCOCK & WILCOX COMPANY
DEPARTMENT

BY N. G. D.

DATE 6-23-64

PROTOTYPE

Gut

JOB NO.

BOILER OUTLET HEADER

SHEET _____ OF _____

Boiler Outlet Header

A. Design Conditions

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress psi	Basis
Shell	SA 182-F22	800	2725	1.338	2-1/4	17,500	7.7
Hemi Head	"	"	"	.650	2-1/4	"	7.7
Tube Sheet	"	"	"				
a) Bending				2.351	3-1/4	17,500	7.7
b) Shear				1.787	3-1/4	17,500	7.7
Thermal Sleeve	"	900	300	0.2945	3/4	14,000	7.7
Handhole Plugs	"	800	Safe to 4180				7.19

B. Operating Conditions

See following pages for steady-state intensities and tabulation of maximum stress reversals and usage factors for the transient conditions.

STRESS INTENSITIES

σ_1 = LONGITUDINAL STRESS
 σ_2 = CIRCUMFERENTIAL "
 σ_3 = RADIAL "

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY

E.W.

DATE

BOILER OUTLET HEADER - PROTOTYPE

JOB NO. 610-0067

STEEL STATE 750°

SHEET OF

JUNCTURE	A	MAX. STRESS REVERSAL		N		U. F.		TRANSIENT
		OUTSIDE	INSIDE	OUT	IN	OUT	IN	
1 700° 900°	25	$\tau_{1-2} = 9681$	$\tau_{2-3} = 8071$	∞	∞	0	0	7
	25	$\tau_{1-2} = 9510$	$\tau_{2-3} = 7781$	∞	∞	0	0	9
						0	0	TOTAL
2 700° 900°	25	$\tau_{1-2} = 14,528$	$\tau_{2-3} = 14,664$	∞	∞	0	0	7
	25	$\tau_{1-2} = 14,737$	$\tau_{2-3} = 7,043$	7×10^4	∞	0	0	9
						0	0	TOTAL
3 750° 900°	25	$\tau_{1-2} = 15,124$	$\tau_{1-2} = 20,837$	2×10^5	7×10^4	0	0	7
	25	$\tau_{1-2} = 28,866$	$\tau_{1-2} = 43,501$	4000	800	0	.03	9
						0	.03	TOTAL
4 700° 900°	25	$\tau_{1-2} = 16,104$	$\tau_{2-3} = 14,749$	∞	∞	0	0	7
	25	$\tau_{2-3} = 2962$	$\tau_{1-2} = 4315$	∞	∞	0	0	9
						0	0	TOTAL
5 700° 900°	25	$\tau_{1-2} = 29,841$	$\tau_{2-3} = 18,991$	5×10^3	105	0	0	7
	25	$\tau_{2-3} = 19,273$	$\tau_{1-2} = 15,507$	∞	6×10^4	0	0	9
						0	0	TOTAL
6 900° 900°	25	$\tau_{1-2} = 8721$	$\tau_{1-2} = 7502$	∞	∞	0	0	7
	25	$\tau_{1-2} = 15,340$	$\tau_{1-2} = 11,985$	6×10^4	∞	0	0	9
						0	0	TOTAL

TABLE I

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY

E.W.

DATE

BOILER OUTLET HEAD - PROTOTYPE

JOB NO. 610-0067

USING FACTORS

SHEET OF

Allowable Force and Moment, Boiler Outlet

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N 5 BOILER OUTLET ALLOWABLE FORCES AND MOMENTS

FIG. N-5.1 EXPANDED CURVE OF BOILER OUTLET ALLOWABLE FORCES AND
MOMENTS

SUPERHEATER INLET HEADER.

Superheater Inlet Header

A. Design Conditions

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress psi	Basi
Shell	SA 182-F316	800	2675	1.405	2-1/4	15,650	7.7
Hemi Head	"	"	"	0.641	2-1/4	"	7.7
Tube Sheet	"	"	"				
a) Bending				2.437	3-1/4	"	7.7
b) Shear				1.440	3-1/4	"	7.7
Thermal Sleeve							
a) Croloy Sect.	SA 182-F22	900	300	.2945	3/4	14,000	7.7
b) St. Stl. Sect.	SA 182-F316	900	300	.216	3/4	14,950	7.7
Handhole Fittings	SA 182-F316	800	2675	0.410	5/8	15,650	7.7

B. Operating Conditions

See Following pages for steady state intensities and tabulation of

· maximum stress reversals and usage factors for the transient conditions.

STRESS INTENSITIES

σ_1 = LONGITUDINAL STRESS
 σ_2 = CIRCUMFERENTIAL "
 σ_3 = RADIAL "

JUNCTURE	m	MAX. STRESS REVERSAL		N		U.F.		TRANSIENT	
		OUTSIDE	INSIDE	OUT	IN	OUT	IN		
1	800	25	$\sigma_{1-2} = 15,429$	$\sigma_{2-3} = 12,329$	2×10^5	∞	0	0	7
	900	25	$\sigma_{1-2} = 26,190$	$\sigma_{2-3} = 16,427$	5×10^3	5×10^4	0	0	9
							0	0	TOTAL
2	800	25	$\sigma_{3-1} = 11,357$	$\sigma_{1-2} = 6,248$	∞	∞	0	0	7
	900	25	$\sigma_{1-2} = 22,171$	$\sigma_{2-3} = 12,582$	10^4	3×10^5	0	0	9
							0	0	TOTAL
3	900	25	$\sigma_{3-1} = 31,836$	$\sigma_{1-2} = 51,077$	2.5×10^3	400	.01	.06	7
	900	25	$\sigma_{1-2} = 16,062$	$\sigma_{2-3} = 29,902$	5×10^3	10^4	0	0	9
							.01	.06	TOTAL
4	800	25	$\sigma_{1-2} = 8,365$	$\sigma_{3-1} = 8,365$	∞	∞	0	0	7
	900	25	$\sigma_{3-1} = 8,233$	$\sigma_{1-2} = 11,294$	∞	∞	0	0	9
							0	0	TOTAL
5	800	25	$\sigma_{2-1} = 8,320$	$\sigma_{3-1} = 12,481$	∞	∞	0	0	7
	900	25	$\sigma_{2-3} = 11,540$	$\sigma_{1-2} = 20,292$	∞	1.5×10^4	0	0	9
							0	0	TOTAL
6	900	25	$\sigma_{3-1} = 9074$	$\sigma_{3-1} = 9074$	∞	∞	0	0	7
	900	25	$\sigma_{1-2} = 20,356$	$\sigma_{2-3} = 19,408$	1.5×10^4	1.5×10^4	0	0	9
							0	0	TOTAL
7	900	25	$\sigma_{2-1} = 12,681$	$\sigma_{2-1} = 9907$	∞	∞	0	0	7
	CROLOY 900	25	$\sigma_{1-2} = 49,807$	$\sigma_{2-3} = 35,006$	2.5×10^3	6×10^3	.01	.01	9
							.01	.01	TOTAL

TABLE I

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY

E. W.

DATE

SUPERHEATER INLET - PROTOTYPE

JOB NO. 610-0067

USAGE FACTORS & CALCULATED DATA

SHEET 1 OF

Allowable Force and Moment Superheater Inlet

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N-6 SUPERHEATER INLET ALLOWABLE FORCES AND MOMENTS

FIG. N 6 1 EXPANDED CURVE OF S.H. INLET ALLOWABLE FORCES AND
MOMENTS

STAINLESS STEEL
DISSIMILAR WELD
CROLOY 2 1/4

DISSIMILAR WELD

CROLOY $2\frac{1}{4}$

90

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY N.G.D.

DATE 6-16-64

30 MWIT PROTOTYPE

Cv

JOB NO.

SUPER HEATER OUTLET HEADER

SHEET _____ OF _____

Superheater Outlet Header

A. Design Conditions

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress psi	Basis
Shell	SA 182-F316	1090	2625	2.10	2-1/4	10,760	7.7
Hemi Head	"	"	"	.922	2-1/4	10,760	7.7
Tube Sheet	"	"	"				
a) Bending				2.915	3-1/4	10,760	7.7
b) Shear				2.055	3-1/4	10,760	7.7
Thermal Sleeve							
a) Croloy Sect.	SA 182-F22	950	300	.3588	3/4	11,000	7.7
b) St. Stl. Sect.	SA 182-F316	1050	300	0.266	3/4	12,200	7.7
Handhole Plugs	SA 182-F316	1090		.616	5/8	10,760	7.7

B. Operating Conditions

See following pages for steady state intensities and tabulation of maximum stress reversals and usage factors for the transient conditions.

STRESS INTENSITIES

σ_1 = LONGITUDINAL STRESS

σ_2 = CIRCUMFERENTIAL STRESS

σ_3 = RADIAL STRESS

BABCOCK & WILCOX COMPANY

DESIGN

BY E.W.

DATE

SUPERHEATER OUTLET (PROTOTYPE)

JOB NO. 610-0067

STEADY STATE 1000 F

SHEET 1 OF 1

JUNCTION	IN	MAX. STRESS REVERSAL		N		U. F.		TRANSIENT
		OUTSIDE	INSIDE	OUT	IN	OUT	IN	
1	25	$\sigma_{2-1} = 17,613$	$\sigma_{2-3} = 16,265$	∞	∞	0	0	7
2	25	$\sigma_{3-1} = 22,299$	$\sigma_{3-1} = 10,616$	1.2×10^5	∞	0	0	7
3	25	$\sigma_{3-1} = 26,332$	$\sigma_{1-2} = 40,834$	3.5×10^4	4×10^3	0	0	7
4	25	$\sigma_{1-2} = 8,735$	$\sigma_{2-3} = 10,288$	∞	∞	0	0	7
5	25	$\sigma_{2-3} = 12,824$	$\sigma_{1-2} = 17,205$	∞	∞	0	0	7
6	25	$\sigma_{1-2} = 22,384$	$\sigma_{2-3} = 19,769$	10^5	4×10^5	0	0	7
7 CELOY	25	$\sigma_{3-1} = 7,980$	$\sigma_{3-1} = 6,236$	∞	∞	0	0	7

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY E. W.

DATE

SUPERHEATER OUTLET (PROTOTYPE)

JOB NO. 610-0067

USING FACTOR OF CALCULATION

SHEET OF

Allowable Force and Moment, Superheater Outlet

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N-3 SUPERHEATER OUTLET ALLOWABLE FORCES AND MOMENTS

FIG. N 1 EXPANDED CURVE OF S.H. OUTLET ALLOWABLE FORCES AND
MOMENTS

SEE SK 1522

BABCOCK & WILCOX COMPANY		BY <u>N.G.D.</u>	DATE <u>9-10-64</u>
DEPARTMENT			
PROTOTYPE		<u>NEW</u>	JOB NO.
FLANGES			SHEET <u>2</u> OF <u>2</u>

SEE SK 1522

BABCOCK & WILCOX COMPANY
DEPARTMENT

BY

N.G.D.

DATE 9-9-64

PROTOTYPE

JOB NO.

FLANGES

SHEET

OF

Flanges.

Item	Mat'l.	Temp. F	Press. psi	Long. Hub Stress SH psi	Radial FLG SR psi	Tang. FLG ST psi	.5(SH+SR) or .5(SH+ST) psi	Allow Stress psi	Basis for Allowable Stress
Top Shell Flange	SA132-F22	900	300	9,909	5,579	1,450	7,714	SH=21,000 Others= 14,000	7.5
Top Spool Flange	"	"	"	10,437	5,813	1,510	8,125	SH=21,000 Others= 14,000	7.5
Lower Spool	"	"	"	9,909	5,579	1,450	7,714	SH=21,000 Others= 14,000	7.5
Shell Flange	"	"	"	9,823	5,500	1,552	7,662	SH=21,000 Others= 14,000	7.5

	Press. psi	Temp. F	Hydrostatic End Force	Bolt Area Req'd. In. ²	Actual Bolt Area	Bolt SpacingIn.	Stress psi	Allow Stress	Basis for Allowable Stress
Flange Bolts 68-1 1/2 X 8 THD Bolt Area=.929 in ² Min. Spcg. = 2-13/16 Max. " = Mat'l - SA-193-B14	300	800	1,700,590	61.393	63.172	3.811	26,920	27,700	6

Flanges (continued)

Item	Mat'l.	Temp. F	Press. psi	Req'd Thick in.	Actual Thick in.	Actual Stress P.S.I.	Allow. Stress	Allow. Press. P.S.I.	Basis for Allowable Pressure
Flange Joint Seal 2" O.D. Tube	SA210	800	300	0.08	0.119			800	FIG. UG 31 - Tubes under External Pressure A.S.M.E. Section VIII.

SODIUM INLET NOZZLE

Sodium Inlet, Nozzle and Distribution System.

Item	Mat'l.	Temp. F	Press. psi	Thick. Req'd. in.	Thick. Actual in.	Allow. Stress psi	Basis for Allow. Stress Level
<u>Design</u>							
Nozzle (Conn. to Inlet Pipe).	SA 182-F316	1200	300	0.233	1/2	6800	A.S.M.E. Sect. VIII Code Allowable @ Temperature
Nozzle Thermal Sleeve		1200	300	0.233	.843	6800	
a. St. Stl.							
Sect. SA 182-F316		1200	300	0.3414	1-1/2	6800	" "
b. 2 1/4 Cr. SA 182-F22 Sect.		1000	300	0.3632	1-1/2	7800	" "
<u>(Operating)</u>							
Nozzle	SA 182-F316	1140	20	0.147	.843	7300	Allowables from Report BW 67-3
Thermal sleeve							
(a) St. Stl.							
Sect. SA 182-F316		1140	20	0.0216	1-1/2	7300	" "
(b) 2 1/4 Cr.							
Sect. SA 182-F22		950	20	0.0825	1-1/2	8000	" "

*Min. Tensile @ Ambient Temp.

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY NGD.

DATE

STRESS INTENSITIES - MA / INCHJOB NO. 610-0067STEADY STATESHEET OF

Allowable Force and Moment

The curves on the following pages show the limits of forces and moments that can be applied at the connection. Any combination of a force and a moment within the shaded area is satisfactory.

Each element that makes up the connection to the shell has been considered in evaluating this condition and the allowable force and moment has been based on the weaker section.

FIG. N-1 SODIUM INLET ALLOWABLE FORCES AND MOMENTS

FIG. N-1-1 EXPANDED CURVE OF SODIUM INLET ALLOWABLE FORCES AND
MOMENTS

Boiler Support System.

Item	Mat'l.	Temp F	Press. psi	Type Stress	Actual Stress psi	Allowable Stress Level psi	Basis for Allowable St Level
Hanger Bars	Plate Croloy 2-1/4 Min. Tensile *60,000 psi	1140	—	Bending Shear	284 166	2100 1050	7.8
Coil Support Bars & Clamping Bars	Plate Croloy 2-1/4 Min. Tensile *60,000 psi	1140	—	Tensile	1630	2100	7.9
Boiler Bundle Support Beam	Plate Type 304 SS Min. Tens. *75,000 psi	1140	—	Bending Shear	2687 229	6100 3050	7.10
5/8 Dia. Pin (Hanger Bar to Coil Support Bar)	Bar Croloy 2-1/4 Min. Tensile *60,000 SS	1140	—	Shear	649	1050	7.8
1" Dia. Pin(Hanger- Bar to Clevis)	Bar Type 304 SS Min. Tensile *75,000	1140	—	Shear	1082	3050	7.10
1" Dia. Support Rods	Bar Type 304 SS Min. Tensile *75,000	1140	—	Tensile	3085	6100	7.10
Clevis	Forging Type 304 SS Min. Tensile *70,000						
Spherical Washers	Bar Type 304 SS Min. Tensile *75,000						

Superheater Support System.

Item	Mat'l.	Temp. F	Press. psi	Type Stress	Actual Stress psi	Allow. Stress psi	Basis for Allowabl Stress Level
Hanger Bars	Plate Type 304 St. Stl. Min. Tensile *75,000 psi	1200	—	Bending	104	4500	7.11
				Shear	54	2250	
Coil Support Bars & Clamping Bars	" "	1200	—	Tensile	896	4500	7.11
S.H. Bundle Support Bars	" "	1200	—	Bending	1608	4500	
				Shear	216	2250	
1/2" Dia. Pin (Hanger Bar to Coil Support Bar)	Bar Type 304 St. Stl. Min. Tensile *75,000 psi	1200	—	Shear	448	2250	7.11
3/4" Dia. Pin (Hanger Bar to Clevis)	" "	1200	—	Shear	855	2250	7.11
3/4" Dia. Supt. Rod	" "	1200	—	Tensile	2490	4500	7.11
Clevis	Forging Type 304 St. Stl. Min. Tensile *70,000 psi						
Spherical Washers	Bar. Type 304 St. Stl. Min. Tensile. *75,000 psi						

* Min. Tensile at Ambient Temp.

Vibration of Tubes

Item	Temp. F	Length Between Supports (in.)	Frequency CPS	Basis for Settling Lengths Between Supports
Boiler Inlet	530	13	326	7.18
		14	280	
		16	215	
		18	166	
		20	134	
		23	102	
		25	88	
		26	79	
		27	74	
Boiler Outlet	870	24	145	"
		33	77	
		35	68	
S.H. Inlet	870	30	81	"
		32	71	
		39	48	
		42	41	
S.H. Outlet	1100	37	51	"
		40	44	
Sodium Distribution Pipes	1200	41	173	"
		48	126	
		45 (Cantilever)	51	

Vibration Due to Forced Vibration of Tubes Due to Fluid Flow at Operating Temp.

Item	Tube Size	Temp. F	Max. Natural Freq. (Inner Coil) (Cps	Min. Nat. Freq. (Outer Coil) (Cps	Sodium Flow FT/SEC.	Eddy Freq. (Cps f_2	Nat. Freq. must exceed $1.5 \times f_2$	Basis for Design
Boiler Tube Coils	1" O.D.x.120 MW	700	136	69	1.0	2.641	3.96	7.18
	1" O.D.x.145 MW	800	135	68	1.0	2.641	3.96	7.18
	1" O.D.x.165 MW	900	130	66	1.0	2.641	3.96	7.18
S.H. Tube Coils	7/8" O.D.x.120 MW	989	111	58	1.28	3.863	5.79	7.18

Natural Frequency of Helical Coils at Ambient Temp.

Item	Tube Size	Temp. F	Max. Nat. Freq.	Min. Nat. Freq.	Basis for Design
			(Inner Coil) CPS	(Outer Coil) CPS	
Boiler Coils	1" O.D.x.120 MW	70	149	76	7.18 These frequencies are well above the inherent natural frequency of a rail system, which is considered to be from 3 to 5 cycles/second.
	1" O.D.x.145 MW	"	147	74	
	1" O.D.x.165 MW	"	144	73	
Superheater	7/8" O.D.x.120 MW	"	122	65	

Natural Frequency of Coil Ends at Top of Boiler Bundle (From Last
Coil Support To Vertical Riser Legs.)

Tube No.	Distance To First Clamped Support	C.P.S.	Basis for Design
1	19.6"	216	7.18
2	22.4	165	
3	28.0	106*	
4	30.8	87*	
5	20.0	208	
6	23.2	153	
7	32.3	79*	
8	35.2	67*	
9	38.3	56*	
10	23.5	150	
11	26.8	115	
12	33.7	73*	
13	37.0	61*	
14	40.5	50*	
15	23.8	146	
16	27.5	110	
17	38.4	57*	
18	42.0	47*	
19	45.7	40*	
20	23.6	149	
21	27.4	109	
22	31.5	84	
23	39.2	54*	
24	43.3	44*	
25	47.2	37*	

Tube Marked with * are further snubbed and the frequency
will be higher.

Natural Frequency of Coil Ends at Top of Superheater Bundle.
(From Last Coil Support to Vertical Riser Legs.

Tube No.	Distance to First Clamped Support	C.P.S.	Basis for Design
5	22.3	139	7.18
6	20.9	158	
7	19.5	180	
8	18.1	211	
13	24.4	106	
14	22.9	132	
15	21.4	151	
16	19.8	176	
17	18.3	206	
22	26.5	99	
23	24.8	112	
24	23.2	129	
25	21.5	149	
26	19.9	175	
31	28.6	85	
32	26.8	97	
33	25.0	110	
34	23.2	128	
35	21.5	150	
41	30.7	73	
42	28.8	84	
43	26.9	96	
44	24.9	111	
45	23.0	130	

The remainder of the 45 tubes are all above the frequencies listed.

The natural frequency at the top of both the boiler and superheater coils are all greater than required for the eddy frequency created by the flow of sodium over the tubes. (2.64 cps for boiler, 3.863 cps for superheater). The frequencies are all greater than the 3 to 5 cps expected during shipping. The unsupported coil ends are well away from the 30 and 60 cycles range.

Boiler & Superheater (Downcomers & Risers Tubes) Expansion Stresses, Dead Load & Pressure Stresses

Tube NO. & Description	Expansion Stresses (psi)	Allowable Stress Range S _A (psi)	Sum of Dead Load & Press. Stress (psi)	Allow. Stress in Hot Cond. S _H psi	Basis for Allowable Stress
#4 S.H. Riser	3,926	30,489	4,199	9,000	7.15
8 " "	5,063	28,793	5,895	"	
36 " "	2,684	28,771	5,917	"	
45 " "	5,435	28,480	6,208	"	
#4 S.H. Downcomer	16,601	38,186	5,502	16,200	7.15
8 " "	32,298	38,546	5,142	"	
36 " "	12,079	36,222	7,466	"	
45 " "	36,481	38,687	5,001	"	
#1 Boiler Risers	28,336	29,524	5,726	13,200	7.15
3	21,759	30,078	5,172	"	
9	13,097	29,617	5,633	"	
20	24,021	29,707	5,543	"	
22	19,638	29,981	5,269	"	
23	15,567	29,652	5,598	"	
#1 Boiler Downcomer	15,668	30,807	6,693	15,000	7.15
3 " "	20,738	30,807	6,693	"	
9 " "	23,165	31,872	5,628	"	

Boiler & Superheater (Downcomers & Risers Tubes) Expansion Stresses, Dead Load & Pressure Stresses (Continued)

Tube NO. & Description	Expansion Stresses (psi)	Allowable Stress Range S _A (psi)	Sum of Dead Load & Press. Stress (psi)	Allow. Stress in Hot Cond. S _H psi	Basis for Allowable Stress
#20 Boiler Downcomer	10,658	28,077	9,423	15,000	7.15
22 " "	13,919	29,008	8,492	"	
23 " "	19,201	30,157	7,343	"	

Boiler & S.H. Tubes in the Coil Bundle

Item	Tube	Temp.	Dead Load Stress (psi)	Long Press. Stress (psi)	Stress psi	Allow. Stress psi	Basis for Allow. Stress
Boiler Sect.	1" O.D. X .120 MW	730	272	4354	4626	15,000	7.17
	1" O.D. X .145 MW	841	290	3366	3656	14,508	7.17
	1" O.D. X .165 MW	950	265	2766	3031	8,000	7.17
S.H.Sect.	7/8" O.D. X .120 MW	870	279	3539	3818	16,200	7.16
	" "	1124	279	3473	3752	8,000	7.16

Boiler Shroud

Item	Mat'l.	Temp. F	Max. Stress psi	Location or Type Stress	Allowable Stress Level (psi)	Basis for Allowable Stress Level
Shroud	Plate Type 304 S.S. Min. Tensile = 75,000 psi	1140	352	C-C Tensile	6100	A.S.M.E. Section VIII
Shroud	" "	1140	1284	A-A Tensile	6100	" "
Shroud	" "	1140	1065	B-B Tensile	6100	" "
Beam Seat	Forging Type 304 S.S. Min. Tensile 70,000 psi	1140	2644 660	Bending Shear	6100 3050	" "
Transition	" "	1140	2409	Bending	6100	" "
Shroud Support Ring	Plate Type 304 S.S. Min. Tensile = 75,000 psi	1050	817	Bending	8500	" "
Shroud Support Seat	SA 182-F22	900	1305	Bending	14,000	" "

SHROUD SUPPT.
RING

120

BABCOCK & WILCOX COMPANY

DEPARTMENT

BY N. G. D.

DATE 3-19-64

PROTOTYPE

JOB NO.

S. H. SUPPORT SHROUD

SHEET

OF

Superheater Support Shroud

Item	Mat'l.	Temp. F	Max. Stress	Location or Type Stress	Allowable Stress Level	Basis for Allowable Stress Level
Shroud	Plate Type 304 S.S. Min. Tensile = 75,000 psi	1200	730	A-A	4500	A.S.M.E. Sect. VIII
"	" "	1200	1000	Shear @ Bolt Holes	4500	" "
7/8" Dia. Bolts	SA 193-B8	1050	1730	Shear	1/2 (6800) = 3400	" "
Shroud Sup- port Ring	Plate 2-1/4 Croloy Min. Tensile= 60,000 psi	1000	159	Tensile	5750	Report # BW 67-3

BABCOCK & WILCOX COMPANY
DEPARTMENT

BY N.G.D.

DATE 8-7-04

PROTO TYPE

JOB NO.

BOILER SUPPORT SHROUD & RING

SHEET

OF 3

Boiler Downcomer Support

Item	Mat'l.	Temp. F	Max. Stress psi	Type Stress	Allowable Stress Level	Basis for Allowable Stress
Downcomer Shroud Suppt. Plate	Plate Type 304 S.S. Min. Tensile = 75,000 psi	1050	2700	Bending	8500	A.S.M.E. Sect. VIII
1" Stud	SA 193 B8	950	1035	Shear	3400	" "
Hanger Rod For Downcomer Shroud	Bar Type 304 S.S. Min. Tensile = 75,000	1050	938	Tensile	8500	
Hanger Rod For Downcomer Tubes	" "	1050	1126	Tensile	8500	" "
Beam Seat (Part of Flange Forging)	SA 182-F22	900	1305	Bending	14000	" "

BABCOCK & WILCOX COMPANY

DEPARTMENT _____

BY _____

DATE _____

SUPPORT SKIRT.

JOB NO. _____

SHEET _____ OF _____

Support Skirt

Item	Location	Dead Load Stress psi	Allow. Stress psi	Basis For Allowable Stress
Complete Unit	A-A	716	7033	7.14
	B-B	128	7000	7.14
	C-C	678	8033	7.14

7.0 Basis for Allowable Stresses (or Design)

The following pages give an explanation for the basis of design, choice of condition, and allowable stresses as indicated by the notation given under the appropriate heading on the preceding pages under Section 6.0.

Basis for Allowable Stress
or Design.

Vol. 2

7.1. Main Containment Shells and Heads (Design)

Designed for sodium environment and possible sodium-water reaction for:

- a) Pressure = 300 psi
- b) Temperature = 900 F.

This pressure is for an emergency short time condition. The allowable stress is from A.S.M.E. Section VIII, Unfired Pressure Vessels at temperature.

7.2. Main Containment Shells and Heads (Operating)

Designed for sodium environment. For this the allowable is from curve figure 5 report *BW 67-3 for the 2-1/4 Croloy material. Operating pressure is 20 psi. The additional pressure on the bottom head is due to the sodium weight.

7.3. Drum Liner and Head Liner (Operating)

Designed for the operating temperature and a pressure due to the static head of sodium. The operating and emergency pressure is balanced across the inside and outside with only the sodium weight affecting the plate. Section VIII Code allowable at temperature.

7.4. Drum Liner and Head Liner (Design)

Designed to the emergency condition where the sodium temperature might increase to 1140°F. For this the allowable is from Curve Figure 5* Report BW#67-3. The pressure decreases due to density change in sodium.

*Developed design stresses for Croloy 2 1/4 Alloy Steel.

Report No. B&W 67-3.

7.5. Flanges.

Designed by ASME Rules for Bolted Flange Connections as calculated by the Taylor--Forge Analysis. ASME Section VIII Code Allowables.

7.6. Flange Bolts.

The flanges are to be uninsulated or partially insulated to maintain the operating temperature of the bolts at 800°F or less.

The allowable stress in the bolts is then based on the Nuclear Code Case Interpretation 1273 N-7. These stresses must not exceed 1/3 of yield strength at temperature.

Yield for SA 193 B 14 material is 105,000 psi.

Allowable Stress at 800F = 27,700 psi.

7.7. Headers. Boiler Inlet, Boiler Outlet, S.H. Inlet & S.H. Outlet.

- a) The shells, hemispherical heads and thermal sleeves are designed in accordance with Section VIII, for code allowable at temperature.
- b) The tube sheets are designed by methods outlined in "Standards for Tubular Exchanger Manufacturers Association (TEMA)". for code allowables at temperature.

7.8. Boiler Bundle Hanger Bars & Pins.

The design allowable stress for the 2-1/4 material at 1140°F is 2100 psi. This allowable taken from BW Report 67-3 is the basis for choosing 2,100 psi for consideration of the short time emergency conditions.

The long time operation is based on a temperature of 1050°F. The allowable stress from the BW Report 67-3 is 4,300 psi.

The short time (say 6 hours) emergency is based on 1140F. The allowable stress for this is 2,100 psi.

$$\text{Allowable, } S_{\text{Bending}} = 2,100 \text{ psi}$$

$$\text{Allowable, } S_{\text{Shear}} = 1,050, \text{ Based on } 1/2 S_{\text{Bending}}$$

7. 9. Coil Support Bar.

The allowable stress in the 2-1/4 Croloy Coil Support Bars in tension will be based on the emergency temperature of 1140°F from the BW Report 67-3, which gives an allowable stress of 2,100 psi.

7. 10. Boiler Bundle Support Beams, Pins, Rods, Etc.

The design stress will be 6,100 psi. The allowable for the Type 304 SS is taken directly from the ASME Section VIII. The operating temperature of the sodium is 1050°F with an allowable stress of 10,000 psi. The short time emergency condition of 1140°F with an allowable stress of 6,100 psi is used as a design condition.

7. 11. Superheater Bundle Support Beams, Hanger, Bars, Coil Support, Pins, Rod, Etc.

The design stress for the stainless steel in the superheater section will be 4,500 psi. The basis for this stress is from the ASME Section VIII, Unfired Pressure Vessels, for a temperature of 1200°F.

$$\text{Allowable Stress } S_B = 4,500 \text{ psi}$$

$$\text{Allowable Stress Shear} = 1/2 S_B = 2,250 \text{ psi}$$

7. 12. Allowable Steady State Primary & Secondary Stress Intensities

The allowable values of steady state primary and secondary stress intensities for Croloy 2-1/4 up to 700°F and Type 316 stainless steel up to 800°F will be limited to $3 \times S_m$ per the rules set forth in Section III of the ASME Boiler and Pressure Vessel Code.

BABCOCK & WILCOX COMPANY		BY <u>G.A.P.</u>	DATE <u>4/4/64</u>
DEPARTMENT			
PLOT OF ALLOWABLE STEADY STATE PRIMARY + SECONDARY			JOB NO. <u>610-0067</u>
STRESS INTENSITY VS. TEMPERATURE FOR CROLOY 2 1/4 & TYPE 316 S.S.			SHEET <u> </u> OF <u> </u>

(5) Except as provided in paragraph (6) below, the allowable value of primary-plus-secondary stress intensity (N-414.4) shall be three times the allowable amplitude of fatigue stress at 10^6 cycles for the metal temperature involved, as determined in (7) below.

(7) The design fatigue curves of Section III (Figs. N-415 (a) and (b) are not applicable at temperatures above the limits of N-202 (a), and the curves of Figs. 1 and 2 of this Case shall be used in their place. Values for temperatures intermediate between the curves shown may be interpolated. The fatigue curve to be used shall be that corresponding to the maximum metal temperature occurring during the cycle at the point on the vessel being analyzed.

In order to facilitate the selection of allowable stress intensities for different materials @ various temperatures, a plot of code values of $3 \times S_m$ vs. temperature was prepared and is shown on page 130. Throughout our steady state analysis of the various components of the steam generator, we have referred to this plot for our allowable stress intensities levels. Note that the strict interpretation of the Code case allowable primary and secondary stress intensities for Type 316 stainless steel would allow us to use values of 78,000 psi @ 800°F, 55,000 psi @ 900°F, etc. We do not care to take advantage of this quirk in the curve. Rather, we will predicate our allowable stress values on the dotted curve between 800°F and 1100°F.

7.13. Fatigue Analysis For Components Subjected to Cyclic Loading

The fatigue life of any component will be determined using Figs. 1 and 2 of the publication mentioned in - 12 -.

The allowable value of steady state primary and secondary stress intensities for Croloy 2-1/4 up to 700°F and Type 316 stainless steel beyond 800°F will be based on a Code case (see appendix "C" of the minutes of a meeting on 1/6/64 by the ASME Subcommittee on Nuclear Power) ruling made by the ASME Subcommittee on Nuclear Vessels.

The Code case inquiry states "Under what rules shall nuclear vessels be constructed whose design temperatures exceed those for which allowable stress values are given in Section III." Paragraphs (5) and (7) of this Code case state the following:

7.14. Support Skirt.

The support skirt and vessel have been checked for compressive loading imposed by the dead load of the unit. Since the vessel is located fully within a pit wind loading has not been considered. Seismic loading has been considered and snubbers have been provided to relieve this loading on the support skirt.

The allowable safe compressive stress that can be imposed without failure by wrinkling is expressed by the following which is taken from investigation by Wilson and Newmark and others.

$$\text{Allowable} = 1.5 \times 10^6 \frac{t}{r} \text{ or } 1/3 \text{ yield point.}$$

Yield point for 2-1/4 Croloy of which the vessel shell is constructed is: $S_y @ 900^\circ\text{F} = 21,100$

$$\text{Allowable} = 1.5 \times 10^6 \frac{(1)}{(36)} \text{ or } \frac{21,100}{3}$$

$$49,500 \text{ or } 7,033 \text{ psi}$$

Allowable in Compression for the 2-1/4 Croloy is 7,033 psi.

For the support skirt the allowable is 8,033 psi as follows:

$$1.5 \times 10^6 \frac{(1)}{(38)} = \frac{24,100}{3}$$

$$39,000 = 8,033$$

7.15. Boiler Inlet and Outlet, Superheat Inlet and Outlet Leg Tubes.

Allowable stress levels are in accordance with the requirements of the Code for Pressure Piping.

- a. Stress in the tubes, of a cyclic nature, due to thermal growth must not exceed the allowable stress range, S_A where:

$$S_A = \chi(1.25 S_C + 0.25 S_H)$$

S_C = Allowable stress in cold condition.

S_H = Allowable stress in hot condition.

χ = Stress range reduction factor for cyclic

condition. Taken as 1.0 for 7000 cycles or less.

Stress reduction factor of 1.0 was used. This gives a total of 7000 full temperature cycles over the expected life and exceeds the specified 1000 ramp transients from 200F to 1140F.

- b. The sum of the dead load and longitudinal pressure stresses must not exceed S_H (allow. code stress in hot condition) (Piping code stresses)
- c. The allowable stresses for the superheater riser downcomers, boiler risers and downcomers were taken directly from Fig. 4 and Fig. 5 "Developed Design Stresses in a Sodium Environment and a 30 Year Life," for Type 316 Stainless Steel and Croloy 2-1/4 Alloy Steel, respectively. These are contained in B&W Company report No. BW 67-3. These curves are included in this report for ready reference (See pages 60 & 61).

7.16. S.H. Helical Coil Tubes.

For dead load and longitudinal pressure stresses - (Maximum Span between supports). The allowable stresses are from the B&W Report No. 67-3.

7.17. Boiler Helical Coil Tubes.

For dead load and longitudinal pressure stresses - (Maximum Span between supports). The allowable stresses are from the B&W Report No. 67-3.

7. 18, Vibration.

The operating characteristics of the system may produce vibrations in the 30 and 60 cps range.

Based on this the lengths between vibration supports was set to avoid these ranges.

The distance between supports was also varied to take advantage of the effect of varied spacing on vibration damping.

"It has been experimentally determined that a relatively small variation in support plate spacing can have a large effect on vibration damping." This is from a letter from R.C. Baird, Engineer and Consultant to M.W. Peterson dated 5-20-1958.

The forced vibration of the tubes due to fluid flow has been calculated by using the Von Karman effect.

$$t_1 = k \left(\frac{v}{d} \right)$$

The natural frequency f_1 should differ from the forcing frequency f_2 by at least 50%.

7. 19. Handhole Fittings.

a) Croloy 2-1/4 Plugs

The B&W Company uses standard handhole designs as shown on B&W dwgs. 71450-13.

Boiler Inlet Header

The 4" handhole plug of Croloy 2-1/4 material is good for 4180 psi @ 800°F. Thus it is satisfactory for the design conditions of 2825 psi and 530°F.

Boiler Outlet Header

The 4" handhole plug of Creloy 2-1/4 material is good for 4180 psi @ 800°F. Thus it is satisfactory for the design conditions of 2725 psi and 800°F.

8.0 Structural Support Base Calculations

The following pages contain the complete calculations for the base required to support the prototype unit in the "pit" at the S.C.T.I. installation.

Nuclear and Special Products Engineering
The Babcock & Wilcox Company
Barberton, Ohio

Structural Support Base
For
30 MWT Prototype Sodium Heated Steam Generator

Customer: Atomic Energy Commission

B&W Contract No. 610-0067-45

Prepared By: *Paul Oppenhegger*
John E. Stocking
Checked By: *GA Murphy*
Approved By: *R. M. Douglas*

April 1, 1965

John F. Harvey, 4/8/65

BABCOCK & WILCOX COMPANY		DATE
DEPARTMENT	BY P.O.N. CHECK SM.	JOB NO. 610-0067-95
SUPPORT STRUCTURE FOR 30 MWT. SODIUM HEATED STEAM GENERATOR		SHEET 1 OF

1. REFERENCES.

- 1.1 STEEL CONSTRUCTION MANUAL OF THE A.I.S.C., SIXTH EDITION.
- 1.2 PROCESS EQUIPMENT DESIGN - BROWNELL & YOUNG.
- 1.3 A.S.M.E. SECT III, TABLE N 424
- 1.4 DESIGN OF WELDED STRUCTURAL CONNECTIONS. BY BLODGETT & SCALZI; PUBLISHED BY THE JAME F. LINCOLN ARC WELDING FOUNDATION,

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BABCOCK & WILCOX COMPANY

DEPARTMENT

BY JH

DATE

4-2-65

SUPPORT STRUCTURE FOR 30 MWT

JOB NO. 610-0067-45

SODIUM HEATED STEAM GENERATOR

SHEET 2 OF

2. COMPRESSIVE DEAD LOAD STRESS IN SHELL SUPPORT SKIRT.

$$A = \frac{\pi}{4} (d_o^2 - d_i^2) = \frac{\pi}{4} (78^2 - 76^2) = 241.9 \text{ in}^2$$

$$f_a = \frac{170,880}{241.9} = 706.4 \text{ psi}$$

ALLOWABLE SAFE COMPRESSIVE STRESS THAT CAN BE IMPOSED WITHOUT FAILURE BY WRINKLING FOR THE SUPPORT SKIRT.

$$\text{REF. 1.2} \quad f_{\text{ALLOW}} = 1.5 \times 10^6 \left(\frac{t}{r} \right) \leq \frac{Y.P.}{3}$$

$$\text{MATL - SA 212 GR B} \\ \text{Y.P. @ 900}^\circ\text{F} = 24,100 \text{ psi}$$

REF 1.3

$$f_{\text{ALLOW}} = 1.5 \times 10^6 \left(\frac{1}{38} \right) \text{ or } \frac{24,100}{3} \\ = 39,450 \text{ or } 8,033$$

$$f_{\text{ALLOW}} = 8033 > f_a = 706.4$$

BABCOCK & WILCOX COMPANY		DATE <u>4-2-65</u> JOB NO. <u>610-0067-45</u> SHEET <u>3</u> OF <u> </u>
DEPARTMENT	BY <u>JH</u>	
SUPPORT STRUCTURE FOR 30 MWT		
SODIUM HEATED STEAM GENERATOR		

3. VESSEL SUPPORT FRAME.

NOTE. THE FOUR ELEVATIONS OF THE SUPPORT FRAME ARE THE SAME.

3.1 BEAMS.

$$M_{MAX} = 21,360 \times 42 - 6(1780 \times 12) = 768,960 \text{ IN LBS}$$

TRY 10 WF 77

$$f_B = \frac{M_{MAX}}{S} = \frac{768,960}{86.1} = 8,931 \text{ psi}$$

$$\frac{ld}{bt} = \frac{84 \times 10.62}{10.195 \times 8.68} = 100.8 < 600$$

$$F = 20,000 > f_a$$

USE 10 WF 89

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BABCOCK & WILCOX COMPANY

DEPARTMENT

BY P.O.
CHECKED G.M.

DATE 4-2-65

SUPPORT STRUCTURE FOR 30 MW

JOB NO. 610-006745

SODIUM HEATED STEAM GENERATOR

SHEET 4 OF

3.2 COLUMNS.

TRY 10 WF 49

$$f_a = \frac{P}{A} = \frac{42,720}{14.4} = 2,967 \text{ psi}$$

$$F_a = 17,000 - 0.485 \left(\frac{L}{r} \right)^2 = 17,000 - 0.485 \left(\frac{58}{2.54} \right)^2$$

$$= 16,747 \text{ psi}$$

$$F_a > f_a$$

USE 10WF 49

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SUPPORT STRUCTURE FOR 30MW

JOB NO. 610-0067-45

SODIUM HEATED STEAM GENERATOR

SHEET 5 OF

4. COLUMN BASE PLATES.

THE DESIGN PROCEDURE IS FROM AISC "STEEL
CONSTRUCTION" SIXTH EDITION PG 3-75

$P = 42,720^*$

$$f_c' = 3000 \text{ psi}$$

$$f_{fp} = .25 f'_c = 750 \text{ psi}$$

$$F_0 = 25,000 \text{ psi}$$

$$A = \frac{P}{F_p} = \frac{42,720}{750} = 57 \text{ in}^2$$

$$B = 16''$$

$$C = 18''$$

$$B \times C = 16 \times 18 = 288 > 57$$

$$m = (C - 95d)/2 = (18 - .95(10))/2 = 4.25''$$

$$n = (B - 806)/2 = (16 - .8(10))/2 = 4.00''$$

$$F_p = \frac{43,726}{288} = 149$$

$$t = \sqrt{\frac{3 \times F_m \cdot l}{F_s}} = \sqrt{\frac{3 \times 149 \times 4.25^2}{25,000}} = (32)^{\frac{1}{2}} = .57''$$

USE $\frac{1}{2}$ 16" x 18" x 1" 145

BACOCK & WILCOX COMPANY		DATE <u>4-2-65</u>
DEPARTMENT	BY <u>PO</u>	
SUPPORT STRUCTURE TO MWT		JOB NO. <u>610-0067-45</u>
SODIUM HEATED STEAM GENERATOR		
		SHEET <u>6</u> OF <u> </u>

G. COLUMN TO BEAM CONNECTION

$$R = 21,360 \text{ lb}$$

$$f_v = \frac{R}{2A} = \frac{R}{2 \times t \times l}$$

$$t = \frac{R}{2 \times f_v \times l} = \frac{21,360}{2 \times 13,000 \times 7.875} = .104 \text{\"}$$

USE $3 \times 2\frac{1}{2} \times \frac{7}{16}$ ANGLES $\times 7\frac{7}{8}$ LG

CONNECTIONS C.T'D

"DESIGN OF WELDED STRUCTURAL CONNECTIONS" pg 39

ASSUME WELD IS A LINE REF. 1.4

ANGLE TO BEAM WELD.

$$\bar{x} = \frac{2 \times 2.5 \times 1.25 + 7.875 \times 2.5}{2 \times 2.5 + 7.875} = 2.01$$

$$J_w = I_x + I_y = 2 \times 2.5 \times 3.938^2 + \frac{7.875^3}{12} + \frac{2 \times 2.5^3}{12} + 2 \times 2.5 \times .76^2 + 7.875 \times .49^2$$

$$J_w = 125.624 \text{ in}^2$$

$$f_v = \frac{V}{A_w} = \frac{21,360}{2[2 \times 2.5 + 7.875]} = 830 \text{ #/in}$$

$$f_v' = \frac{T_c}{J_w} = \frac{21,360 \times 2.51 \times 2.01}{2 \times 125.624} = 429 \text{ #/in}$$

$$f_H = \frac{T_c}{J_w} = \frac{21,360 \times 2.51 \times 7.875}{2 \times 125.624} = 841 \text{ #/in}$$

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BABCOCK & WILCOX COMPANY

DEPARTMENT

BY

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DATE

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SUPPORT STRUCTURE 30 MW

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610-0067-45

SODIUM HEATED STEAM GENERATOR

SHEET

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OF

CONNECTIONS CONT'D

$$f_r = \sqrt{(f_v + f_v')^2 + f_h^2} = \sqrt{(830 + 429)^2 + 841^2}$$

$$f_r = \sqrt{2.292 \times 10^6} = 1514$$

$$W = \frac{1514}{\frac{4}{3} \times 600} = 1.90 \sim 2 \quad \text{USE } 3/8" \text{ WELD}$$

ANGLE TO COLUMN WELD

THIS LOADING CONDITION THE SAME AS THE ANGLE TO BEAM WELD CONDITION.

USE 3/8" WELD

THE THICKNESS OF ANGLE WILL DEPEND ON THE SIZE OF THE WELD.

$$t = \frac{3}{8} + \frac{1}{16} = \frac{7}{16}$$

$$t = 7/16 > .160$$

USE $\angle 3 \times 2\frac{1}{2} \times 7/16$

WEB CRIPPLING 10WF77 BEAMS

$$F_g = \frac{P}{.75t(N+2K)} = \frac{42,720}{(.75)(.368)(24 + 2 \times 1.375)}$$

$$F_g = 2,453 < F_{all} = 25,000 \text{ psi}$$

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DEPARTMENT

BY

P.O.

CHK OF GEN.

DATE

4-2-65

JOB NO.

610-0067-93

SHEET

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OF

SUPPORT STRUCTURE 30 MW.

SODIUM HEATED STEAM GENERATOR

9.0 Seismic Support Frame Calculations.

The following pages contain the complete calculation for earthquake loadings and for the design of the seismic framing, snubber and connections.

Nuclear and Special Products Engineering
The Babcock & Wilcox Company
Barberton, Ohio

Seismic Support Frame

For

30 MWT Prototype Sodium Heated Steam Generator

Customer: Atomic Energy Commission

B&W Contract No. 610-OC67-45

Prepared By: *Paul Oppenheuser*

Checked By: *JA Murphy*

Approved By: *RB Douglas*

April 1, 1965

John F. Hurvey, 4/8/65

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Seismic Tie Loading.	13
Size Of Beams.	17
Connections.	20

Introduction

This report covers the design of the seismic support frame located at the center of gravity of the vessel and includes stress analysis calculations of the seismic snubber.

A value of earthquake loading was calculated by the method described in the "City of Los Angeles Building Code," 1960 Edition. However, a method for determining earthquake loadings is outlined in Bromell and Young "Process Equipment Design," Wiley & Sons, 1959. Using this method, higher earthquake loading values were obtained. These higher earthquake loading values were used in this report. An allowable stress of $4/3 F_a$ was used for design of structural members; this is in accordance with Los Angeles Building Code. The structural members were designed by AISC "Specifications for the Design Fabrication & Erection of Structural Steel Buildings," adopted April 17, 1963, for ASTM A7 steel.

Since the vessel is operating at 900°F and Los Angeles Building Code allowable stress limits are designated for atmospheric temperatures, the yield stress of 21,100 PSI was used as the limiting stress for pressure combined with earthquake stress. The yield stress is obtained from ASME Boiler and Pressure Vessel Code, Section III, "Nuclear Vessels," 1963 Edition. This method of stress analysis is in accordance with accepted pressure vessel design used throughout industry.

Conclusion

The calculations indicate that the design requirements are met for seismic support frame and vessel.

Design Data

1. Operating Pressure = 20 PSI
2. Operating Temperature = 900°
3. Material and Allowable Stress

Shell & Snubber - SA 387 GR D

$F_y = 21,100 \text{ PSI}$ (ASME Boiler and Pressure Vessel Code,
Section III, "Nuclear Vessels," Table
N-424)

Structural Steel - ASTM A7

$F_a = 20,000 \text{ PSI}$ Tension & Compression

Structural Bolting - ASTM A 307

$F_v = 10,000 \text{ PSI}$

Fillet Weld - $F_v = 13,600 \text{ PSI}$

EARTHQUAKE LOADING

1. "CITY OF LOS ANGELES BUILDING" CODE PG 109
1960 EDITION

$$T = \frac{0.05 H}{\sqrt{D}} = \frac{0.05 \times 49.287}{\sqrt{6.5}} = 0.966'$$

$$C = \frac{0.05}{\sqrt[3]{T}} = \frac{0.05}{\sqrt[3]{0.966}} = 0.0507$$

$$K = 1.5 \quad \text{FROM TABLE 23-C PG 111}$$

$$V = KCW = 1.5 \times 0.0507 \times 170,880 = 12,990\#'$$

2. "PROCESS EQUIPMENT DESIGN" BY
BROWNELL & YOUNG PG 167

$$T = 2.65 \times 10^{-5} \left(\frac{H}{D} \right)^2 \left(\frac{WD}{t} \right)^{1/2}$$

$$T = 2.65 \times 10^{-5} \left(\frac{49.287}{6.17'} \right)^2 \left(\frac{4,017 \times 6.17}{1} \right)^{1/2}$$

$$T = 0.266 < 0.4$$

$$C = 0.20 \quad \underline{\text{ZONE \#3}} \quad \text{TABLE 9.3 RECOMMENDED COEFFICIENTS}$$

$$F = CW = 0.2 \times 170,880 = 34,176\#$$

* USE 34,176\# EQ LOAD

LAYOUT OF SEISMIC TIE

SEE DETAILS:

SNUBBER — PG 5

BEAM CONNECTION — PG 17

BASE PLATE — PG 20

BRACE PG 22

BRACE CONNECT TO WALL PG 23

BRACE CONNECT. TO BEAM PG 24

C.G. 21'-0" FROM BOTTOM OF SKIRT FLANGE.

ALL MEMBERS
ARE 10 WF 49

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OF

DETAIL OF SEISMIC SNUBBER

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STRESS ANALYSIS OF SEISMIC SNUBBER

a. SHELL

MOMENTS AND AXIAL FORCES CALCULATED IN ACCORDANCE WITH BILLAARD'S PAPER "STRESSES FROM RADIAL LOADS AND EXTERNAL MOMENTS IN CYLINDRICAL PRESSURE VESSEL'S" FOUND IN ASME "PRESSURE VESSEL AND PIPING DESIGN". DUE TO DIFFERENTIAL THERMAL EXPANSION OF VESSEL TO FIXED SUPPORT, THE SNUBBER IS ASSUMED TO BE 8" LONG IN THE ANALYSIS.

$$M = 17,088 \times 2.5 = 42,720 \text{ '}\#$$

$$\alpha = \frac{L}{a} = \frac{320}{36.5} = 8.89 > 1.0$$

$$\gamma = \frac{a}{t} = \frac{36.5}{1} = 36.5$$

$$\beta_1 = \frac{c_1}{a} = \frac{1}{36.5} = 0.0274$$

$$\beta_2 = \frac{c_2}{a} = \frac{4}{36.5} = 0.110$$

$$\beta_1/\beta_2 = 0.0274/0.110 = 1/4$$

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SEISMIC SNUBBER CONT'D

β = EQUIVALENT COEF. REDUCING A RECTANGULAR
LOADING SURFACE TO A SQUARE SURFACE

FOR $\frac{\beta_1}{\beta_2} = \frac{1}{4}$ AND $\gamma = 36$

CONSTANTS

FOR $M\phi$

$K_c = 1.28$ (GRAPH I)

$\beta = K_c (\beta_1^2 \beta_2)^{1/3} = 1.28 (.0274^2 \times .110)^{1/3} = 1.28 \times 0.04355$

$\beta = 0.0555$

$\frac{M\phi}{M/a\beta} = 0.11$ (REFER BIJLAARD'S PAPER, FIG. 4)

FOR M_x

$K_c = 1.68$ (GRAPH I)

$\beta = K_c (\beta_1^2 \beta_2)^{1/3} = 1.68 \times 0.04355 = 0.07316$

$\frac{M_x}{M/a\beta} = 0.06$ (REFER BIJLAARD'S PAPER, FIG. 4)

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SEISMIC SNUBBER CONT'D

GRAPH I

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SEISMIC ROLLER CONT'D GRAPH II

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SEISMIC ANALYSIS - TEST CONT'DFOR N_b

$$C_c = 0.25 \text{ (GRAPH I)}$$

$$\beta = (\beta_1^2 \beta_2)^{1/3} = 0.04355$$

$$\frac{N_b}{M/a^2\beta} = 0.4 \text{ (GRAPH II)}$$

$$\frac{C_c \cdot N_b}{M/a^2\beta} = 0.25 \times 0.4 = 0.100$$

FOR N_x

$$C_c = 0.47 \text{ (GRAPH I)}$$

$$\beta = 0.04355$$

$$\frac{N_x}{M/a^2\beta} = 0.45 \text{ (GRAPH II)}$$

$$\frac{C_c \cdot N_x}{M/a^2\beta} = .47 \times 0.45 = .21$$

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SEISMIC SHAKER CONT'DSTRESSES

$$\frac{GM\phi}{t^2} = \frac{M\phi}{M/a\beta} \times \frac{GM}{a\beta t^2} = 0.11 \times \frac{6 \times 42,720}{36.5 \times 0.0555 \times 1^2}$$

$$\frac{GM\phi}{t^2} = \pm 13,917 \text{ PSI}$$

$$\frac{N\phi}{t} = \frac{C_\phi N_\phi}{M/a^2\beta} \times \frac{M}{ta^2\beta} = 0.10 \times \frac{42,720}{1 \times 36.5^2 \times 0.04355}$$

$$\frac{N\phi}{t} = 74 \text{ PSI}$$

$$\sigma_\phi = \frac{N\phi}{t} \pm \frac{GM\phi}{t^2} = 74 \pm 13,917$$

$$\sigma_\phi = \begin{cases} \pm 13,991 \text{ PSI} - \text{OUTSIDE RADIUS} \\ \pm 12,843 \text{ PSI} - \text{INSIDE RADIUS} \end{cases}$$

$$\frac{GMx}{t^2} = \frac{Mx}{M/a\beta} \times \frac{GM}{a\beta t^2} = 0.06 \times \frac{6 \times 42,720}{36.5 \times 0.07316 \times 1^2}$$

$$\frac{GMx}{t^2} = 5,760 \text{ PSI}$$

$$\frac{Nx}{t} = \frac{C_x N_x}{M/a^2\beta} \times \frac{M}{ta^2\beta} = 0.21 \times \frac{42,720}{1 \times 36.5^2 \times 0.04355}$$

$$\frac{Nx}{t} = 155 \text{ PSI}$$

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SEISMIC SNUBBER CONT'D

$$\sigma_x = \frac{N_x}{t} \pm \frac{6M_x}{t^2} = 155 \pm 5,760$$

$$\sigma_x = \begin{cases} \pm 5915 \text{ PSI} - \text{OUTSIDE RADII} \\ \pm 5605 \text{ PSI} - \text{INSIDE RADII} \end{cases}$$

PRESSURE STRESS

$$\sigma_H = \frac{PR}{t} = \frac{20 \times 36}{1} = 720 \text{ PSI}$$

$$\sigma_a = \frac{PR}{2t} = \frac{20 \times 36}{2 \times 1} = 360 \text{ PSI}$$

STRESS SUMMARY

$$\sigma_1 = \sigma_H + \sigma_\phi = 720 + 13,991 = 14,711 \text{ PSI} < \text{FALL}$$

$$\sigma_2 = \sigma_a + \sigma_x = 360 + 5915 = 6,275 \text{ PSI} < \text{FALL}$$

$$\sigma_3 = \sigma_R = -20 \text{ PSI} < \text{FALL}$$

$$F_y = 21,100 \text{ PSI}$$

b. SNUBBER

$$f_B = \frac{M}{S} = \frac{42,720}{5.333} = 8,010 < 21,100 \text{ PSI}$$

$$\text{WHERE: } S = \frac{I}{c} = \frac{bh^3}{12 \times 1} = \frac{8 \times 2^3}{1 \times 12} = 5.333 \text{ IN}^3$$

SEISMIC TIE LOADING

SIDE TO SIDE EARTHQUAKE LOADING

$$M = 17,088 \times 6.75 = 115,344 \text{ inch-kips}$$

FORE AND AFT EARTHQUAKE LOADING

SEISMIC TIE CONT'D

BEAM ②, ③ S.T.O. - LOADING

$$\sum M_{RL} = 0 = 99.5 R_R \pm 115,344$$

$$R_R = \pm 1,159 \text{ #}$$

$$\sum M_{RR} = 0 \rightarrow R_L = \pm 1,159 \text{ #}$$

BEAM ④

S.T.O S. LOADING

BEAM ① NO LOAD S.T.O.S

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SEISMIC TEST CONT'D

BEAM ① F&A LOADING

$$\sum M_{R_R} = 0 = 91 R_L \pm 115,344$$

$$R_R = \pm 1,267.5\#$$

$$\sum M_{R_L} = 0 \rightarrow R_L = \pm 1,267.5\#$$

BEAM ④ F&A LOADING

SEE FOLLOWING PAGE FOR MOMENT DISTR.

SEISMIC T. W.D.

BEAM (4) C.T.D.

115,349"±

.53 .47

0	0	+28,836	-28,836
0	+15,783	-13,552	+28,836
-7,642	0	-16,418	+6,776
+7,642	-7,642	+6,776	-6,776
+3821	-3821	+3388	-3388
-3821	+3821	-3388	+3388

0	+7,641	+7642	0	TOTAL
---	--------	-------	---	-------

MOMENT DIAGRAM REACTIONS

↓ 95#	95# ↓ 84#	84# ↓
	↓ 1268#	1268# ↓
↓ 95#	↓ 1089#	1184# TOTAL

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SEISMIC TIE CONT'D

BEAM (2) F&A LOADING

BEAM (3) F&A LOADING

SIZE OF BEAMS

DESIGNED BY USING THE AISC
"SPEC. FOR THE DESIGN, FABRICATION
& ERECTION OF STRUCTURAL STEEL FOR
BUILDINGS" FOR ASTM A7 STEEL.

BEAM (1) MAXIMUM LOADING. CONDITION
IS UNDER F&A EQ.

TRY 10WF49

$$f_a = \frac{P}{A} = \frac{8,544}{14.40} = 593 \text{ psi}$$

$$f_b = \frac{M}{S} = \frac{57,671}{54.6} = 1,056 \text{ psi}$$

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SIZE OF BEAMS CONT'D

$$\frac{L}{r} = \frac{45.5}{2.54} = 17.9 \approx 18 \quad F_a = 19,030 \text{ psi}$$

$$F_b = 20,000 \text{ psi}$$

$$\frac{f_a}{F_a} = \frac{593}{19,030} = .031 < 0.15$$

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1.0 \times \frac{4}{3} \quad \text{INCREASE FOR EARTHQUAKE CONDITIONS}$$

$$0.031 + \frac{1,056}{20,000} = .0838 < 1.33 \quad \text{OK}$$

USE 10 WF 49BEAM (E)TRY 10 WF 49

S. TO S. LOADING

$$f_a = \frac{P}{A} = \frac{17,088}{14.40} = 1,186 \text{ psi}$$

$$f_b = \frac{M}{S} = \frac{63,586}{54.6} = 1,146 \text{ psi}$$

$$\frac{L}{r} = \frac{54}{2.54} = 21.3 \approx 22 \quad F_a = 18,820$$

$$F_b = 20,000 \text{ psi}$$

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SHEET 18 OF

SIZE OF BEAMS CONT'D

$$\frac{f_a}{F_a} = \frac{1,186}{18,320} = 0.063 < 0.15$$

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1.33$$

$$0.063 + \frac{1,146}{20,000} = 0.120 < 1.33 - \text{OK}$$

F & A LOADING LESS CRITICAL
THAN STDS.

USE 10 WF 49

BEAM (3)

USE 10 WF 49

BEAM (4) F & A LOADING

$$f_a = \frac{P}{A} = \frac{17,088}{14.40} = 1,186 \text{ psi}$$

$$f_b = \frac{M}{S} = \frac{61,472}{54.6} = 1,126 \text{ psi}$$

$$\frac{L}{r} = \frac{80.5}{2.54} = 32 \quad F_a = 18,240 \text{ psi}$$

$$F_b = 20,000 \text{ psi}$$

$$\frac{f_a}{F_a} = \frac{1,186}{18,240} = 0.065 < 0.15$$

$$\frac{f_a}{F_a} + \frac{f_b}{F_b} = 0.065 + \frac{1,126}{20,000} = 0.121 < 1.33$$

USE 10 WF 49

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SEISMIC SNURFER

a. PLATE TO 10WF49 BEAM (SEE PG 3,4)

$$M = 1.75 \times 17,088 = 29,904 \text{ #}$$

$$S = \frac{I}{c} = \frac{bh^3/12}{h/2} = \frac{8 \times 1}{12 \times \frac{1}{2}} = 1.333 \text{ in}^3$$

$$f_b = \frac{M}{S} = \frac{29,904}{1.333} = 22,433 < \frac{4}{3} F_a = 26,666 \text{ psi}$$

b. WELD

$$F = \frac{M}{d} = \frac{1.75 \times 17,088}{1} = 29,904; \quad w = \frac{29,904}{\frac{4}{3} \times 600 \times 8} = 4.672 \sim 5$$

USE 5/16" WELD

CONNECTIONS

DETAIL OF BEAM
CONNECTION.

USE ANGLES ON BOTH SIDES
1/4" x 4" x 5/16"

USE 3/4" BOLTS

SEE FOLLOWING PAGES FOR SIZING
CALCULATIONS OF WELD AND BOLTS

CONNECTIONS CONT'D

SIDE TO SIDE
EARTHQUAKE
LOADING

FORE & AFT EARTHQUAKE LOADING

CONNECTIONS CONT'D

a. BOLTS (BEAM CONNECTION)

MAXIMUM LOADING

$$P_A = \text{AXIAL LOAD} = \pm 8,544 \#$$

$$P_s = \text{SHEAR LOAD} = \pm 1,159 \#$$

ALLOWABLE LOAD FOR ASTM A7
BOLTS $3/4 \phi$

$$F_A = 4 \times 6,190 = 24,760 > P_A$$

$$F_s = 4 \times 4,420 = 17,680 > P_s$$

USE $3/4 \phi$ BOLTS

b. WELD (BEAM CONNECTION)

MAXIMUM LOADING

$$P_A = \pm 8,544$$

TRY $1/4$ " WELD

$$w = \frac{8,544}{2 \times 600 \times 6} = 1.2$$

$$\text{WELD } 1.2/16 = 1/8" < 1/4"$$

USE $1/4$ " WELD

CONNECTIONS CONT'D

DETAIL OF BASE PLATE

2. BASE PLATE

DESIGN PROCEDURE FROM AISC
 "STEEL CONSTRUCTION" SIXTH EDITION
 PG. 3-75.

$$P = 17,088 \#$$

$$F_b = 25,000 \text{ psi}$$

$$F_p = 750 \text{ psi}$$

$$f_c = 3000 \text{ psi}$$

$$A = \frac{P}{F_p} = \frac{17,088}{750} = 23 \text{ in}^2$$

$$B = 12" \quad C = 12"$$

$$B \times C = 12 \times 12 = 144 \text{ in}^2$$

$$m = (C - 9.5d)/2 = [12 - 9.5(10)]/2 = 1.25$$

$$n = (B - 8.0d)/2 = [12 - 8(10)]/2 = 2.00$$

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CONNECTIONS CONT'D

a. BASE PLATE CONT'D

$$F_p = \frac{17,088}{144} = 119 \text{ PSI}$$

$$t = \sqrt{\frac{3 F_p n^2}{F_b}} = \sqrt{\frac{3 \times 119 \times 2^2}{25,000}}$$

$$t = 0.239" < \frac{1}{2}"$$

USE 1/2" PLATE

b. WELD

$$P_s = (17,088^2 + 1,159^2)^{1/2} = \pm 17,128\#$$

USE 1/4" WELD

$$L = \frac{17,128}{\frac{4}{3} \times 2400} = 5.35"$$

WELD BOTH SIDES OF COLUMN
TO PLATE 20" > 5.35"

c. ANCHOR BOLTS

$$\text{CASE I } P_A = \pm 17,088\# \quad P_S = \pm 1,159\#$$

$$\text{CASE II } P_A = \pm 2,355\# \quad P_S = \pm 8,544$$

TRY 3/4" ϕ BOLTS

CASE I

$$f_v = \frac{P_s}{A} = \frac{1,159}{2(-.442)} = 1312 < \frac{4}{3} F_v = \frac{4}{3} \times 15,000$$

CONNECTIONS CONT'D

$$F_t = 28,000 - 1.6 f_u = 28,000 - 1.6 (1,312) = 25,900 \text{ psi}$$

$$F_t = \frac{4}{3} \times 20,000 = 26,666 \text{ psi}$$

$$f_t = \frac{P}{A} = \frac{17,088}{2(1.4418)} = 19,330 \text{ psi} < F_t$$

CASE II

$$f_u = \frac{P_u}{A} = \frac{8544}{2(1.442)} = 9,665 \text{ psi} < \frac{4}{3} F_u$$

$$F_t = 28,000 - 1.6 f_u = 28,000 - 1.6 (9,665) = 12,536$$

$$F_t = \frac{4}{3} \times 12,536 = 16,718 \text{ psi}$$

$$f_t = \frac{P}{A} = \frac{2355}{2(1.4418)} = 2,665 < F_t$$

USE 3/4" ANCHOR BOLTS

BRACE

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CONNECTIONS CONT'DBRACE CONT'DTRY JL 3x2 1/2 x 1/4 LONG LESS BACK TO BACK

$$\frac{L}{r} = \frac{175}{0.95} = 184 \quad F_a = \frac{4}{3} \times 4,410 = 5,878 \text{ psi}$$

$$f_a = \frac{P}{A} = \frac{1253}{2.62} = 478 \text{ psi} < F_a$$

USE JL 3x2 1/2 x 1/4 LLB TO BDETAIL BRACE CONNECTION TO WALL

a. BOLTS

TRY 3/8" ϕ BOLTS

$$P = 1253 \#$$

$$P_{ALL} = n \times A_k \times (2F_u) = 2 \times 0.068 \times 2 \times 10,000$$

$$P_{ALL} = 2,720 \# > P$$

USE 5/8" ϕ BOLTS

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OF

CONNECTIONS CONT'D

b. WELD

$$w = \frac{1253}{2 \times 12 \times 600} = 0.087 < \frac{1}{4}''$$

USE $\frac{1}{4}''$ WELD

DETAIL OF BRACE TO BEAM

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