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FINAL DESIGN OF SODIUM-HEATED, MODULAR,
STEAM GENERATORS FOR THE SCTI

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ABSTRACT

In July 1963, design of a 37-tube, modular-type, low-temperature evaporator and high-temperature superheater for testing in the Sodium Component Test Installation was initiated. The evaporator and superheater modules, when combined, would be capable of generating 2400-psi, 1050°F steam. The following report covers the final design of the modular steam generators.

Two other reports cover the development and test phase of the project. One pertains to the fatigue testing program, NAA-SR-9827; the other to a required operational testing program, NAA-SR-9828.

MSG SUPERHEATER MODULES (2)

SODIUM INLET

STEAM OUTLET

FEEDWATER INLET

BLOWDOWN GAS DISCHARGE

TEST PIT

SODIUM OUTLET

MSG BLOWDOWN

RUPTURE MEMBRANES

BLOWDOWN TANKS

MSG EVAPORATOR MODULES (2)

ALCO (B-L-H) STEAM GENERATOR

MSG BLOWDOWN

SODIUM DUMP TANK

ALCO and Modular Steam Generators Installation

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I. INTRODUCTION

In the nuclear energy field today, the sodium-cooled reactor holds a unique position. It may be designed as a thermal reactor when a moderator is used in the core, or it may be designed as a fast reactor when unmoderated. In both of these concepts, however, the nuclear energy generated in the core appears as high-temperature sodium exiting from the core. Since sodium has a low vapor pressure (boiling point 1621°F) and excellent heat transfer properties, this source of high-temperature sodium offers the capability for generating high-temperature steam. Since the reactor is cooled by sodium and the primary loop, intermediate heat exchanger, and secondary piping loop contain only sodium, they are relatively independent of the steam pressure developed in the steam generators. Therefore, the problems associated with steam pressure are basically limited to the steam generators.

One of the major problems associated with sodium-cooled reactor power plants has been the design of reliable and economical sodium-heated steam generators for high-quality steam (above 900 psi at 900°F). Problems encountered due to caustic stress corrosion, resulting from sodium-water reaction and chloride stress-corrosion resulting from condenser leakage have been previously circumvented by the use of expensive noncorrosive linings, double-wall-tube construction, or the reduction of steam temperatures to levels where lower strength alloys not subject to the corrosion problem, may be used. These solutions, however, do not lend themselves to lower steam-generator prices, and they do not offer the promise of fully exploiting the more advanced steam-cycle concepts which are currently being used in fossil-fired power-generating stations.

From the above considerations, the design criteria for sodium-steam generators may be outlined as follows:

- a) Detail design features must minimize the possibility of caustic stress-corrosion.
- b) Designs should allow the generation of 950 to 1050°F steam with the lowest possible cost consistent with reliable performance.
- c) The design should be adaptable to high pressures to exploit the associated lower plant heat rates.

- d) The design should allow the units to be isolated and removed from service without a major plant outage.
- e) The design should allow steam-generator fabrication-and-installation schedules which will not govern the plant construction schedule.

In order to determine optimum steam conditions for a sodium-cooled reactor power plant, based on economic considerations only, an economics study¹ was conducted considering steam conditions of 1400 psi (condensing and single reheat), 1800 psi (single reheat), 2400 psi (single reheat) and 3500 psi (single and double reheat). The results of the economics study indicated an incentive to use supercritical steam conditions.

In 1962, authorization was received to design and develop fabrication techniques for the supercritical modular steam generators and, upon completion of the work to submit an interim report. The report² summarized the engineering and design phases of the project. Three other reports covered the development and test phases of the work, one pertained to the welding development³ and the other two pertained to the experimental stress^{4,5} programs.

In 1963, the Modular Steam Generator Program was continued with the scope of work defined as follows:

"A low-temperature evaporator and a high-temperature superheater module capable of generating 2400-psi, 1050°F steam will be designed and fabricated. In addition, the necessary design work for installation of the modules in the Sodium Component Test Installation (SCTI) will be completed and the necessary equipment for installation of the modules will be ordered."

These steam generators would be for testing in the Sodium Component Test Installation. They would be considered as prototypes of steam generators to be used in ASCR power plants. The 2400-psi conditions were chosen because this pressure is a standard in the power industry and its use would result in minimum changes to the SCTI system.

This report summarizes the final design and analyses pertaining to the 2400-psi modular steam generators. Complimentary reports cover the fatigue testing program,⁶ the control system analyses,⁷ and the suggested operational testing program.⁸

II. DESIGN CRITERIA

A. STEADY STATE

The basic requirements for the present design of the sodium-heated modular steam generators is to show the capability of furnishing 2400-psi, 1050°F steam at the turbine throttle utilizing an Advanced Sodium-Cooled Reactor (ASCR) as the source of hot sodium. With the ASCR as the heat source, the design criteria for the primary and secondary sodium systems must be adhered to and be the basis for the steam generator design criteria. Within these limits, there are various alternatives for designing and operating the overall system. These alternatives have been discussed and tentative design criteria established for a supercritical system.² These basic concepts previously developed for supercritical units may be applied to the development of the design criteria for subcritical units.

The design of steam generators for pressures greater than 1400 psi usually requires that provision be made for steam reheat. With sodium as the heating fluid, it is necessary to separate sodium coming from the IHX so that a portion flows through the reheaters and the balance flows through the superheater or superheater and evaporator. It is then necessary to recombine the two sodium flows at some appropriate point downstream of the reheaters and superheater or evaporator. In order to minimize reheater surface, the sodium flow through the reheaters should be recombined with the balance of the sodium at the superheater outlet. This arrangement gives a higher temperature difference between the reheat steam and the sodium, which minimizes the reheat surface area required.

The division of the superheat and evaporation sections also allows less expensive ferritic material to be used in the evaporation section. The use of ferritic materials also improves the corrosion resistance ability of the evaporator.

The present design will be for installation and testing in the Sodium Components Test Installation (SCTI) which has no allowance for reheat steam. Under these conditions, the sodium temperature entering the evaporator modules would be higher, see Figure 1, but the operating pressures would be the same.

Figure 1. Typical Modular Steam Generator Installation
 Left = ASCR Conditions with Reheaters. Right = SCTI
 Conditions with no Reheaters.)

The sodium temperature of 1100°F entering the superheater was based on conditions typical of a sodium-cooled nuclear power plant. Similarly, a sodium ΔT of 400°F was chosen. The steam temperature of 1050°F was chosen to show the capability of the steam generator and to match other sodium heated steam generator concepts. In actuality, the steam temperature of 1050°F cannot be justified economically because of excessive heat transfer surface requirements.

The feedwater temperature of 500°F was based on economic considerations and representative conditions for a nuclear power plant. The temperature of the steam leaving the evaporator was chosen at 715°F to give about a 40°F superheat to ensure no carryover to the superheater. With this 715°F evaporator-exiting steam temperature, a sodium temperature of 973°F is required for the evaporator inlet.

In order to furnish 2400-psi steam at the turbine throttle, steam should be produced at 2500 psi to allow 100 psi for main steam line and throttle valve pressure drop. Allowing for a pressure sensing and control error of 150 psi, the maximum operating pressure should be 2650 psi. However, the design pressure based on the setting of the relief valves would be 2800 psi for the superheater modules.

The maximum operating pressure for the evaporator modules would then be 2750 which allows for a 100 psi drop in the superheater section. The design pressure based on the setting of the relief valves would be 2900 psi for the evaporator modules.

In each of the modules, the remote possibility exists that a tube becomes partially plugged at one end with the result that the tube would be subjected to the full pressure and maximum sodium temperature. For this reason, there seemed to be no alternative but to use the maximum sodium temperature as the design temperature for the tubes and tube sheets; therefore, the superheater modules were designed for 1115°F and the evaporator modules for 975°F.

The above criteria for steady-state and part-load operation are summarized in the following table.

In addition to these criteria, the modular steam generator shell sections were required to contain full steam pressure for a short period of time. This additional requirement was imposed on the design since the modules were for testing and this requirement would provide additional safety in the event of a sodium-water interaction.

The criteria for startup or shutdown, assuming 1000 cycles are:

1) Evaporator

- a) Temperature, 350°F isothermal to steady-state full-load temperature at 200°F/hr or vice versa.

OPERATION CRITERIA

Criteria	Modules			
	Superheater		Evaporator	
	Design	Operating	Design	Operating
Steam Pressure (psi)	2800	2500	2900	2600
Steam Temperature (°F)				
Tube sheet	1115	1050	975	715
Tube	1115	1095	975	960
Sodium Pressure (psi)				
Shell	300	≈75	300	≈75
Rupturable Membrane	250-300		250-300	
Sodium Temperature (°F)	1115	1100	990	975

b) Pressure, 0 to 2900 psi.

2) Superheater

a) Temperature, 350°F isothermal to steady-state full-load temperatures at 200°F/hr or vice versa.

b) Pressure, 0 to 2800 psi.

The criteria for load changes within the range 15 to 100% load will be made by changing sodium flow and sodium inlet temperatures to maintain 2500-psi, 1055°F-steam. The number of cycles are assumed to be 1,000,000 each at:

- 1) Flow-load ramp $\pm 5\%$ of full load per minute between 20 and 100%,
- 2) Short-time ramp $\pm 20\%$ of full load in one minute, and
- 3) Step change $\pm 10\%$ of full load value.

B. TRANSIENT DESIGN CRITERIA

The modular steam generators, when used in power plants for a period of 30 years, could experience various transient conditions resulting from uncontrolled variables. To ensure performance, these transients have to be anticipated. The number of times the transient may occur has to be estimated, and the steam generators designed accordingly.

The following are representative transients which might be expected during a 30-year period. The number of times these transients might occur will be highly dependent on the maintenance policy in effect at the plant, the alertness of the operator, and many other intangible items. In actuality, transients one to five listed below should not occur during the life of the plant. However, in order to arrive at a conservative design and allow for transient conditions not yet conceived, the number of times the transient may occur was assumed to be the values shown below. Therefore, these transients establish the transient design criteria for the modular steam generators to be tested in the SCTI.

1. Reactor Scrammed, Turbine Tripped, Primary Sodium Temperatures Maintained

System conditions assumed are that the turbine throttle valve is closed, the startup and dump systems fail, and the primary outlet sodium temperature is maintained.

Under these conditions, the sodium flows in the primary and secondary loops are reduced to free convection or low-flow conditions and the feedwater flow is compatible utilizing the safety valves. The temperature difference across the IHX would collapse and the sodium entering the steam generators would be essentially 1150°F, thereby generating steam at 1150°F; this is a casualty condition. Twenty such transients are assumed. Corrective action should be taken within five minutes by relieving steam pressure with the steam dump system. The conditions imposed on the steam generator units would be:

	<u>Superheater</u>	<u>Evaporator</u>
Steam Pressure (psi)	2800	2800
Steam Temperature (°F)	1150	1150
Feedwater Temperature (°F)	-	500

2. Reactor Scrammed, Turbine Tripped, Secondary Sodium Temperature Maintained

System conditions assumed are that the turbine throttle valve is closed; the normal pressure control, startup, and steam dump systems do not operate; and the secondary sodium temperature entering the steam generators is maintained.

Under these conditions, the sodium flows in the primary and secondary loops are reduced to free convection or low flow conditions; however, the sodium temperature entering the steam generator is maintained at 1100°F. The feedwater flow is compatible utilizing safety valves. Steam is produced at 1100°F and the pressure rises to 2800 psi before the safety valves are actuated and feedwater flow completely stopped; this is a casualty condition. Five such transients are assumed. Corrective action should be taken within five minutes by stopping sodium and feedwater flow in the steam generator bank affected. The conditions imposed on the steam generators would be:

	<u>Superheater</u>	<u>Evaporator</u>
Steam Pressure (psi)	2800	2800
Steam Temperature (°F)	1100	1100
Feedwater Temperature (°F)	-	500

3. Reactor Scrammed

System conditions assumed are that the primary and secondary sodium and feedwater systems continue at full flow.

Under these conditions, the primary sodium leaving the reactor and the secondary sodium leaving the IHX decrease in temperature at a rate of 35°F/sec. The feedwater flow continues and the steam or water leaving the steam generator would be diverted to the steam dump system and safety valves when the throttle valve closed; this is a casualty condition. Thirty such transients are assumed. The conditions imposed on the steam generator would be:

	<u>Superheater</u>	<u>Evaporator</u>
Steam Pressure (psi)	2800	2900
Sodium Temperature at 35°F/sec (°F)	1100 to 700	975 to 700

4. Loss of Secondary Sodium Flow

System conditions assumed are that the feedwater flow continues at full flow.

Under these conditions, the sodium flow in the steam generators goes to zero. The feedwater flow continues at full flow. The steam or water leaving the steam generators would be diverted to the steam dump system and safety valves when the throttle valve is closed; this is a casualty condition. Thirty

such transients are assumed. The conditions imposed on the steam generators would be:

	<u>Superheater</u>	<u>Evaporator</u>
Steam outlet temperature (°F)	1050 to 500	715 to 500
Steam-Water inlet temperature (°F)	715 to 500	500
Steam pressure (psi)	2800	2900

5. Loss of Feedwater

System conditions assumed are that the reactor and primary and secondary sodium systems continue at full power.

Under these conditions, the sodium flow in the steam generators continues at full flow while the feedwater flow goes to zero. Whatever steam is developed is relieved through the safety valves or the steam dump system; this is a casualty condition. Thirty such transients are assumed. The conditions imposed on the steam generators would be:

	<u>Superheater</u>	<u>Evaporator</u>
Steam pressure (psi)	2800	2800
Sodium outlet temperature (°F)	975 to 1100	700 to 1100
Sodium inlet temperature (°F)	1100	975 to 1100

6. Steam Generators Containing No Sodium, Preheated from Ambient Temperature to Sodium Loading Temperature

This condition will exist every time the steam generators are preheated. One hundred twenty such transients are assumed. The conditions imposed on the steam generators would be:

Water heated from 60 to 350°F at 100°F/hr with a maximum step change of 25°F.

III. MECHANICAL DESIGN

A. GENERAL

The 37-tube modular steam generators consist essentially of tubes, tube sheets, tube supports, and a shell. The tubes are straight and butt welded to tube sheets at both ends. The tube supports are investment castings, supporting 20 of the 37 tubes. The design of the tube supports is such that the majority of the sodium flows through the supports, thereby minimizing sodium cross-flow velocities. Since the cross flow is minimized and the majority of the flow is parallel to the tubes, the sodium velocity is increased to about 15 ft/sec through the supports. The shell is essentially straight pipe with piping crosses or tees for the sodium inlet and outlet nozzles. The remaining nozzles of the crosses provide for installing rupturable membranes to allow pressure relief in the event of a sodium-water interaction.

The evaporator and superheater modules are designed in conformance to the ASME Unfired Pressure Vessel Code⁹ (Section VIII). Additional requirements are in an Atomics International specification.¹⁰

The evaporator module assembly is shown in Figure 2 and the superheater module assembly in Figure 3.

The tube sheet design for the modular steam generator utilizes a hemispherical configuration on the high-pressure or steam side and a flat surface with machined nipples on the sodium side. The hemispherical configuration gives a minimum material thickness for the tube sheet which would minimize any transient temperature differences and thermal stresses. The flat surface has nipples trepanned on the sodium side which allows a crevice-free butt-welded tube-to-tube sheet joint to be made amenable to x-ray inspection. The tube sheet design is shown in Figures 4 and 5.

The basic configuration of the tube sheet was determined previously² for supercritical steam conditions and an experimental stress program was performed^{4,5} on the configuration. The previous program utilized tube sheets of 2-1/4 chrome - 1 moly; however, the data obtained was applied to the present designs with allowance made for change in stress allowables.

Figure 2
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Figure 3
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Figure 4
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Figure 5. Tube-Sheet Design

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The current tube-sheet designs specify vacuum melted material so as to reduce inclusions and voids.

An experimental fatigue stress program was also conducted on the present tube-sheet designs, this is documented in a complimentary report.⁶

For the evaporator and superheater modules, 5/8-in.-OD, 0.095-in. minimum wall tubing was specified. The tubing was ordered to an Atomics International specification¹¹ which requires seamless drawn tubes with double ultrasonic inspection in addition to the code requirements.

The tube supports are 347 SS investment castings designed to engage 20 of the 37 tubes and 3 of the 4 tube support tie rods. The supports are designed so that they may be spaced alternately opposite each other so as to engage all tubes at any two adjacent locations. Spacing of the tube supports is such as to minimize vibration from both parallel and cross flow of the sodium. The location of the tubes relative to each other is based on providing equal sodium flow adjacent to each tube. In this manner, theoretically speaking, there should be no temperature difference in the sodium across the tube bundle. Since there will be turbulence and a small degree of cross flow and mixing due to the tube supports in practice, there should be no temperature difference in the sodium across the tube bundle.

The tube support design clearances relative to the tubes were based on TEMA¹² and are 0.030 in. diametrically. The tube support design is shown in Figure 6.

The shells of the evaporator and superheater modules are essentially straight pipe, 6.357-in ID by 0.250-in. wall. There are no bellows in the shells because under steady-state and transient conditions the tension and compression loads on the tubes and shell are not excessive (discussed in Section V). Omission of the bellows permits a more reliable and economical design. The use of the nonstandard pipe size for the shells was predicated on minimizing development costs and utilizing existing development work as discussed below.

The original experimental stress program on the 37-tube tube-sheet for supercritical conditions was based on a 6-in. schedule-10 pipe which gave an ID of 6.357 in. for the shell. The spacing of the 37 tubes within the bundle was based on this shell size. This pipe size would have been used in the present

Figure 6
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design except for the request by the test facility operators that the shell contain full steam pressure for a short period of time. This condition requires a wall thickness of 0.250 in. In order to use a standard pipe size, it would have required respacing the tubes and additional experimental stress analysis of the module tube sheets. Therefore, in order to minimize costs, the decision was made to use the tube sheet design already developed with special size pipe for the shell and special reducers where the shell attached to 10-in. schedule-40 tees and crosses. In any future designs for a large plant, a standard size pipe would be used for the shell.

The rupturable membranes are designed to relieve pressure at 275 ± 25 psi. At the normal operating pressures and temperatures, the life of the membranes should be infinite. Therefore, unlike rupture disks which are set to relieve pressure near their operating pressure and where long time creep is a problem, the rupturable membranes used in the present applications should not have to be replaced during the life of the units.

B. EVAPORATOR

The choice of materials for the evaporator module was limited to the ferritic steels due to their resistance to chloride stress corrosion. With a subcritical once-through steam generator, there is the possibility that the concentration of chlorides may build up in the boiling region, since there is no blowdown as such. An additional requirement is that the ferritic material used be compatible with sodium at the design temperature.

In Figure 7, the code allowable stresses are shown for the various materials considered, namely 2-1/4 Cr - 1 Mo, 5 Cr - 1/2 Mo, 7 Cr - 1/2 Mo, and 9 Cr - 1 Mo. At the design temperature of 975°F, the order of preference based on code allowables only was 9 Cr - 1 Mo, 2-1/4 Cr - 1 Mo, 5 Cr - 1/2 Mo, and 7 Cr - 1/2 Mo. At the design temperature, the potential carbon transport from the steels to the sodium was such that the 2-1/4 Cr - 1 Mo allowable stresses would have to be reduced by about 20%. When this was considered in the economic comparison of the various materials, the 5 Cr - 1/2 Mo material gave the lowest cost. Therefore, on the basis of resistance to chloride stress-corrosion, carbon transport in sodium at the design temperature, code allowable stresses, and material costs, the choice was made to use 5 Cr - 1/2 Mo material for the evaporator.

Figure 7. Stress-Temperature Allowables, Seamless, Low-Alloy Steel Pipes and Tubes SA-213

At the sodium inlet to the shell, a 10 by 6-in. schedule-40 tee is used in conjunction with a diffuser cylinder. The diffuser is used to protect the tubes from the high inlet sodium velocity at the inlet nozzle. The sodium would flow around the diffuser and a velocity reduction occurs before it enters the bundle. The 10-in. ends of the tee are attached to the shell by reducers and the 6-in. end, to the external sodium piping.

At distances of 15 and 30 ft from the sodium outlet nozzle, 8 by 6-in. schedule-40 tees are installed. The 8-in. ends are attached to the shell by reducers; the 6-in. end contains rupturable membranes for over pressure relief.

At the sodium outlet from the shell, a 10 by 6-in. schedule-40 cross is used in conjunction with a guide cylinder. The guide cylinder maintains the sodium flow around the tubes for a distance beyond the reducers. The reducers attach the 10-in. ends of the cross to the shell. One 6-in. end of the cross

provides the sodium outlet nozzle which is attached to the external sodium piping. The other 6-in. end of the cross is for a rupturable membrane.

The complete shell assembly has a code rating of 300 psi at 975°F. However, the shell will sustain full steam pressure for approximately five minutes.

The tubes and tube sheets have a code rating of 2900 psi at 975°F.

The evaporator modules for the SCTI have nine thermocouples clamped to nine separate tubes in each module, Figure 8. The thermocouples are located in the argon gas space near the top tube sheet to detect if there is any flow instabilities on the water/steam flow. Clamping of the thermocouples to the tubes is preferred over welding since by clamping there is no possibility of jeopardizing the integrity of the tubes themselves. The maximum diametral clearance which would exist between the tubes and clamps due to differential thermal expansion would be about 0.002 in. Allowing for original looseness in the clamp, one might expect as a maximum, 0.004-in. diametral clearance which would give about a 2°F error in the thermocouple reading (discussed in Appendix A).

In the event flow instabilities should exist, the normal exiting steam temperature of 715°F would decrease to about 675°F for the case of excessive water flow or increase to about 800°F for the case of decrease in water flow. In any event, temperature changes of about 40°F or greater would be expected and the error of 2°F would be negligible.

C. SUPERHEATER

The choice of materials for the superheater module was limited to the austenitic steels due to their high strengths and compatibility with sodium at elevated temperatures.

In Figure 9, the code allowable stresses are shown for the various materials considered, namely 304H, 316H, 321H, and 347H. At the design temperature of 1100°F, the order of preference based on code allowables was 321H or 347H (same allowable), 316H and 304H. With the choice narrowed to 321H and 347H, the first impression was to use 321H since it was a more economical material. This was further substantiated by the characteristic of 347H to develop weld cracking in sections over 1 in. in thickness.¹³ The main question relative to the use of 321H was that it was a "dirty" steel. When this was presented to tube manufacturers, they felt they could meet the specifications using 321H.

Figure 8
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Figure 9. Stress-Temperature Allowables, Seamless, Stainless Steel Pipes and Tubes SA-213

For the tube sheets, the added requirements of vacuum melt material provided the cleanliness required for the tube to tube-sheet welding. Therefore, for the tubes and tube-sheets in the superheater module, 321H material was specified.

The shell section, since it had less stringent code requirements, was made of 304H material. This was the most economical.

The shell for the superheater is essentially the same as that described previously for the evaporator except that the material is 304H and there is only one rupturable membrane located opposite the sodium outlet nozzle.

The complete shell assembly has a code rating of 300 psi at 1100°F. However, the shell will sustain full steam pressure for a short period of time.

The tubes and tube sheets have a code rating of 2800 psi at 1100°F.

IV. PERFORMANCE ANALYSES

A. STEADY-STATE THERMAL PERFORMANCE

Prediction of the performance of the steam generators for the SCTI under steady-state full- and part-load conditions required extensive calculational time. In order to reduce these calculations and still allow investigation of the important items, the hand calculations were limited to the steady-state, full-load conditions. In general, the determination of the length of heat transfer surface required was obtained by dividing the steam generators into a number of axial sections, initiating the solutions from known values at the bottom, and then obtaining solutions for each axial section. The predicted full-load performance of the SCTI modular-steam-generator evaporators and superheaters are shown in Figures 10 and 11. The part load conditions were investigated in conjunction with the transient studies performed on the analog computer.⁷

Figure 10. Evaporator, Steady-State, Full-Load Temperature Profile

Figure 11. Superheater, Steady-State, Full-Load, Temperature Profile

Sodium properties were taken from Reference 14. Values for the thermal conductivity, viscosity, and Prandtl number for water were taken from Reference 15. Specific heat and enthalpy values were taken from Reference 16.

1. Evaporator

Since the present design generates 2500 psi steam (subcritical), the complexities associated with the water-steam phase changes must be considered. The problem, therefore, was one of defining heat-transfer coefficients for the water and sodium sides, defining the tube-wall thermal resistance, defining a waterside fouling factor, and incorporating these into the energy equation.

Considering a unit length ΔX , the following energy balance may be made:

then

$$w_w C_{pw} t'_{wi} + UP(\Delta X)(t'_n - t'_w) = w_w C_{pw} t'_{wo} , \quad \dots(1)$$

and

$$w_n C_{pn} t'_{ni} = w_n C_{pn} t'_{no} + UP(\Delta X)(t'_n - t'_w) , \quad \dots(2)$$

where

$$t'_n = \frac{t'_{ni} + t'_{no}}{2} ,$$

and

$$t'_w = \frac{t'_{wi} + t'_{wo}}{2} .$$

The overall heat transfer coefficient (U) based on the outside tube surface area is:

$$U = \frac{1}{\frac{D_o}{D_i h_w} + \frac{1.05(D_o) \ln \frac{D_o}{D_i}}{2k_t} + \frac{1}{h_n} + \frac{D_o}{D_i h_{ff}}} . \quad \dots(3)$$

The sodium-side film coefficient may be defined by the Lubarsky Equation¹⁷ with the shell side diameter being defined by the equivalent diameter concept.

$$h_n = 0.625 \left(\frac{k_n}{D_e} \right)^{0.6} \left(\frac{G_n}{C_{pn}} \right)^{0.4} \quad \dots(4)$$

The use of the equivalent diameter concept in the Lubarsky equation for non-circular geometries may possibly be somewhat optimistic. However, since the sodium film coefficient represents such a small factor in the overall coefficient with any small deviation being essentially negligible, no correction factor was applied for the present design.

The fouling factor for the present design is discussed in Appendix B and the recommended value of 2.5×10^{-4} obtained. This gives a fouling-factor, equivalent-film coefficient of

$$h_{ff} = \frac{1}{2.5 \times 10^{-4}} = 4,000 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F} \quad .$$

The waterside film coefficient, as stated previously, must allow for the complexities associated with the water-steam phase changes. These changes however, appear only in the evaporator. In the superheater, one may consider only steam with no phase change.

In the evaporator, four regions are allowed for, namely: preheat, nucleate boiling, film boiling, and superheat. The preheat region is considered to be from the water inlet to the point where the inner wall temperature (clean or fouled) equals the saturation temperature of the bulk water. The waterside film coefficient is that for forced convection which uses the well-known Colburn¹⁸ equation

$$h_w = \frac{k_w}{D_i} (0.023) \left(\frac{D_i G_w}{\mu_w} \right)^{0.8} \left(\frac{C_{pw} \mu_w}{k_w} \right)^{1/3} \quad \dots(5)$$

From the point where the inner wall temperature (clean or fouled) equals the saturation temperature of the bulk water to the point where a steam quality of 20% is obtained, the heat transfer mechanism is assumed to be subcooled nucleate boiling. To define the subcooled, nucleate-boiling, heat-transfer coefficient, the Jens and Lottes¹⁸ equation was used,

$$\Delta t_{\text{sat}} = t_w - t_{\text{sat}} = \frac{1.9 \left(\frac{q}{A} \right)^{1/4}}{\frac{P}{e \cdot 900}} \quad \dots (6)$$

The cutoff point of 20% steam quality for the end of the nucleate boiling region was based on the work of Swensen, Carver, and Kakarala.¹⁹

From the 20% steam-quality point to the 100% steam-quality point, the heat-transfer mechanism was assumed to be film boiling. To define the film-boiling, heat-transfer coefficient, the equation of Swensen, Carver, and Szoeké²⁰ was used.

$$h_w = \frac{k_w}{D_i} (0.050) \left(\frac{v D_i \rho_w}{\mu_w} \right)^{0.8} \left(\frac{C_{pw} \mu_w}{k_w} \right)^{0.4} \quad \dots (7)$$

The fluid properties were evaluated at the surface temperature and the velocity v was based on the bulk velocity assuming a slip ratio of 1.0.

From the 100% steam-quality point to the point where the steam leaves the evaporator, the heat-transfer mechanism was assumed to be forced convection to superheated steam. To define the heat-transfer film coefficient, a modified Dittus-Boelter¹⁸ equation was used,

$$h_w = \frac{k}{D_i} (0.023) \left(\frac{D_i G}{\mu} \right)^{0.8} \left(\frac{C_{pw} \mu_w}{k_w} \right)^{0.4} \quad \dots (8)$$

where the properties of the steam were based on the film temperature.

The temperature profile through the evaporator, calculated as described above, is shown in Figure 10.

2. Superheater

In the superheater, the steamside, heat-transfer coefficient was defined by using the equation previously discussed for the superheat region of the evaporator.

Evaluation of the equivalent heat-transfer coefficient for the tube wall included a correction factor of 1.05 to allow for an increase in wall thickness. The design specified a minimum wall (thickness variation) of $-0 + 10\%$;^{*} therefore, on an average, the wall thickness was assumed to be 5% thicker which gave the correction factor of 1.05.

The temperature profile through the superheat modules is shown in Figure 11.

3. Flow Stability

In any steam generator configuration, the question always arises as to stability of the system under various load conditions. The present configuration using vertical tubes was no exception. Therefore, in order to investigate the stability problem, a subcritical-steam-generator code was utilized to compute the steam-generator performance at various loads. The code was similar to the AI FUGUE code²¹ and used the Richardson two-phase momentum equation development and the Martinelli and Nelson liquid fraction relation.

The physical characteristics of the steam generator (evaporator module) used in this study were:

<u>Tubes:</u>	Number	37
	Length (ft)	43.75
	Outside diameter (in.)	0.625
	Wall thickness (in.)	0.095

(continued)

*This was later revised to $-0 + 16\%$ for the evaporator and $-0 + 20\%$ for the superheater. It was determined that these changes resulted in an overall cost reduction. Since the tube length had been determined for a correction factor of 1.05, no change in the correction factor was made. It was felt that this was acceptable since the SCTI steam generators will test the overall concept and the small error would not be serious. However, in any future design, the correction factor should be 1.08 and 1.10.

<u>Sodium:</u>	<u>100%</u>	<u>50%</u>	<u>20%</u>
Inlet temperature (°F)	975	975	975
Flowrate (lb/hr)	3.06×10^5	1.53×10^5	0.613×10^5
Outlet temperature (°F)	700	700	700

Water:

Inlet temperature (°F)	500	450	400
Inlet pressure (psia)	2650	2650	2650
Flowrate (lb/hr)	3.67×10^4	1.835×10^4	0.734×10^4

The results of the study are shown in Figure 12. It is seen that good stability exists at the 20, 50, and 100% power conditions. This is indicated by the strongly positive slopes of the pressure drop curves.

Figure 12. Flow Stability for Evaporator at 2600 psia

B. TRANSIENT THERMAL ANALYSES

The evaporator and superheater module shells utilize pipe having a wall thickness of 1/4 in. The thermal response of the 1/4-in. shell thickness and the 0.095-in. thickness tube wall, under the most severe temperature transients, results in calculated mean temperature differences (averaged over the module length) which are not excessive. On this basis, bellows in the shell are omitted, resulting in a more conservative and economical design.

1. Evaporator, Hot-Shock

The most severe transient which the evaporator module will be subjected to is that caused by the feedwater being suddenly lost and the sodium flow continuing at full flow (all safety controls are lost). Under these circumstances, the sodium temperature in the evaporator will rise to 1100°F. The analyses for this transient assumed that:

- a) The sodium inlet temperature rises from an initial 975°F at 37.6 °F/sec,
- b) The sodium outlet temperature rises from an initial 700°F at 57.2°F/sec,
- c) The sodium travels through the modules at 12 ft/sec, full flow velocity,
- d) The water flow stops instantaneously,
- e) The inner tube surface is adiabatic, and
- f) The outer shell surface is adiabatic.

The method of analysis was to calculate the transient response of the shell and tube walls when subjected to sodium temperature ramps. As the sodium passes the shell and tubes, the sodium temperature will be decreased while the shell and tube temperatures are increased. The basic equations are obtained in the following manner.

then,

$$-w_n C_{pn} \frac{dt_n}{dX} = \rho_n A_n C_{pn} \frac{dt_n}{d\theta} + h'_{ns} P_s (t_n - t_s) + h'_{nt} P_t (t_n - t_t), \quad \dots(9)$$

$$h'_{ns} P_s (t_n - t_s) = \rho_s A_s C_{ps} \frac{dt_s}{d\theta}, \quad \text{and} \quad \dots(10)$$

$$h'_{nt} P_t (t_n - t_t) = \rho_t A_t C_{pt} \frac{dt_t}{d\theta}. \quad \dots(11)$$

Solving Equations 10 and 11 and assuming

$$t_n = t_{no} + R' \theta$$

gives

$$t_{no} + R' \theta - t_s = [t_{no} - t_s(o)] e^{-\alpha_s \theta} + \frac{R'}{\alpha_s} \left[1 - e^{-\alpha_s \theta} \right] \quad \dots(12)$$

and

$$t_{no} + R' \theta - t_t = [t_{no} - t_t(o)] e^{-\alpha_t \theta} + \frac{R'}{\alpha_t} \left[1 - e^{-\alpha_t \theta} \right] \quad \dots(13)$$

where

$$\alpha_s = \frac{h'_{ns} P_s}{\rho_s A_s C_{ps}}$$

and

$$\alpha_t = \frac{h'_{nt} P_t}{\rho_t A_t C_{pt}} \quad .$$

Substituting Equations 12 and 13 into 9 gives

$$\begin{aligned} -w_n C_{pn} \frac{dt_n}{dX} = \rho_n A_n C_{pn} \frac{dt_n}{d\theta} + h'_{ns} P_s \left[(t_{no} - t_{so}) e^{-\alpha_s \theta} + \frac{R'}{\alpha_s} \left(1 - e^{-\alpha_s \theta} \right) \right] \\ + h'_{nt} P_t \left[(t_{no} - t_{to}) e^{-\alpha_t \theta} + \frac{R'}{\alpha_t} \left(1 - e^{-\alpha_t \theta} \right) \right] \quad . \quad \dots (14) \end{aligned}$$

Assuming

$$t_{so} = t_{no} = t_{to} \quad ;$$

then,

$$-w_n C_{pn} \frac{dt_n}{dX} = \rho_n A_n C_{pn} \frac{dt_n}{d\theta} + \rho_s A_s C_{ps} R' \left(1 - e^{-\alpha_s \theta} \right) + \rho_t A_t C_{pt} R' \left(1 - e^{-\alpha_t \theta} \right) \quad \dots (15)$$

Considering the term

$$w_n C_{pn} \frac{dt_n}{dX} \quad ;$$

this may be rewritten

$$\rho_n A_n C_{pn} v_n \frac{dt_n}{dX}$$

where

$$w_n = \rho_n A_n v_n \quad .$$

Since

$$\frac{dt_n}{dX}$$

represents the axial temperature gradient which (under the assumption of no heat transfer to the tube and shell walls) would be the axial gradient applied at the sodium velocity v_n , one may consider the term

$$v_n \frac{dt_n}{dX}$$

as the applied temperature ramp R . In addition, the term

$$\frac{dt_n}{d\theta} = R'$$

since

$$t_n = t_{no} + R' \theta \quad .$$

Then Equation 15 reduces to

$$R' = \frac{R}{1 + \left(\frac{\rho_s A_s C_{ps}}{\rho_n A_n C_{pn}} \right) \left(1 - e^{-\alpha_s \theta} \right) + \left(\frac{\rho_t A_t C_{pt}}{\rho_n A_n C_{pn}} \right) \left(1 - e^{-\alpha_t \theta} \right)}, \quad \dots (16)$$

which enables one to compute the corrected temperature ramp R' from the applied ramp R allowing for the heat transfer to the tube and shell walls.

The temperature ramp which will be applied to the evaporator inlet will be the mixed mean corrected temperature ramp from the superheater outlet. The applied ramp to the superheater outlet is

$$R = \frac{(1100 - 975)(12)}{(0 - 40)} = -37.5^{\circ}\text{F/sec.},$$

where:

1100°F is the superheater outlet sodium temperature,

975°F is the superheater inlet sodium temperature,

40 ft is the superheater length, and

12 ft/sec is superheater sodium velocity.

The conservative assumption is then made that the applied ramp is continued until the corrected sodium temperature reaches 1100°F. In actuality, the applied ramp of -37.5°F/sec would only be applied for $40/12 = 3.33$ seconds with the ramp then decreasing as the superheater gradually reached the isothermal temperature of 1100°F.

Since there are basically two regions to the sodium flow, the outer tube region which includes the outer tubes and shell, and the center tube region which considers only the tubes, the applied transient of -37.5°F/sec was applied to the two regions separately in 1-sec increments. The heat capacities for the tubes, shell, and sodium are:

$$\text{tube, } \rho_t A_t C_{pt} = \frac{(490)(\pi)(0.530)(0.095)(0.12)}{144} = 0.0645 \text{ Btu/}^{\circ}\text{F-ft};$$

$$\text{shell, } \rho_s A_s C_{ps} = \frac{(490)(\pi)(6.607)(0.250)(0.12)}{144(18)} = 0.1175 \text{ Btu/}^{\circ}\text{F-ft}; \text{ and}$$

$$\text{sodium, } \rho_n A_n C_{pn} = \frac{(50)(0.550)(0.30)}{144} = 0.0573 \text{ Btu/}^{\circ}\text{F-ft}.$$

The corrected heat transfer film coefficients, allowing for the conduction resistance to the center of the tube or shell wall, for the superheater are:

$$h'_{ns} = \frac{1}{\frac{1}{5600} + \frac{0.250}{2(12)(12)}} = 954 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

and

$$h'_{nt} = \frac{1}{\frac{1}{5600} + \frac{0.095}{2(12)(12)}} = 1965 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

Calculation of the time constants with the previous data gives:

$$\frac{h'_{ns} P_s}{\rho_s A_s C_{ps}} = \frac{(954)(\pi)(6.357)}{(12)(18)(0.1175)} = 755 \text{ hr}^{-1} = 0.209 \text{ sec}^{-1}$$

and

$$\frac{h'_{nt} P_t}{\rho_t A_t C_{pt}} = \frac{(1965)(\pi)(0.625)}{(12)(0.0645)} = 4,990 \text{ hr}^{-1} = 1.385 \text{ sec}^{-1}.$$

With the above data and Equation 16, the corrected temperature ramps for the outer tube region and shell, and the center tube region were calculated. Since there are 18 tubes in the outer row and 19 tubes in the central region, the mixed mean temperature ramp was then applied to the evaporator inlet, and the transient tube and shell mean temperatures were computed by use of a digital computer program.²²

The results of the calculations for the shell and tube mean temperatures at the evaporator inlet are shown in Figure 13.

For the evaporator outlet, the initial applied temperature ramp was computed to be:

$$R = \frac{(975 - 700)(12)}{(0 - 43.75)} = -75.5^\circ\text{F/sec}.$$

This ramp was applied for $43.75/12 = 3.64 \text{ sec}$.

Figure 13. Evaporator, Hot-Shock Temperatures. Water Flow Stops, Sodium Flow Continues at Full Flow (12 ft/sec)

After 3.64 sec, the corrected ramp from the evaporator inlet was considered as the applied ramp for an additional 7.5 sec.

The heat capacities for the evaporator were the same as those for the superheater; however, the corrected film coefficients changed due to the different materials used:

$$h'_{ns} = \frac{1}{\frac{1}{5600} + \frac{0.250}{2(14)(12)}} = 1,080 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

and

$$h'_{nt} = \frac{1}{\frac{1}{5600} + \frac{0.095}{2(14)(12)}} = 2,170 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

The time constants for the superheater are:

$$\frac{h'_{ns} P_s}{\rho_s A_s C_{ps}} = 0.236 \text{ sec}^{-1}$$

and

$$\frac{h_{nt} P_t}{\rho_t A_t C_{pt}} = 1.525 \text{ sec}^{-1} .$$

With the above data for the evaporator outlet and Equation 9, the corrected temperature ramps for the outer tube and shell region and the central tube region were calculated. With these corrected temperature ramps, the transient tube and shell mean temperatures were computed by the digital, transient-heat-conduction code.²²

The results of the calculations for the shell and tube mean temperatures at the evaporator outlet are shown in Figure 13.

From Figure 13, the shell and tube mean temperature difference averaged over the evaporator length for the central tube and outer tube regions were determined. These temperature differences are shown in Figure 14.

The analyses for the evaporator hot-shock condition assumed the water flow stopping instantaneously and the inner tube surface being adiabatic. These assumptions are very conservative, since in the actual case, the water flow would continue for a short period, approximately 3 to 5 sec due to pump and fluid inertia, and there would be heat transfer across the inner tube surface as well. In both cases (the water flow continuing and heat transfer across the inner surface), the tubes would have additional cooling which would reduce the final temperature differences.

The division of the module cross section into an outer tube region (outer row of tubes and shell) and a central tube region (19 central tubes) assumed there was no mixing of the sodium between the regions as the sodium traveled down the module. In actuality, there will be some mixing due to the tube supports, tees, and crosses. Any mixing which would take place should again reduce the final temperature differences below these calculated.

Figure 14. Evaporator, Hot-Shock Temperature Differences

2. Superheater, Cold Shock

The most severe transient which the superheater module could be subjected to is that of the sodium flow being suddenly lost and the water flow continuing at full flow (again all safety controls being lost). The analyses for this transient assumed:

- a) The water flow travels through the modules at full flow velocity, 5.34 ft/sec,
- b) The sodium flow stops instantaneously, and
- c) The outer shell surface is adiabatic.

The method of analyses was to calculate the transient response of the shell and tube walls when subjected to the steam/water temperature ramps. As the steam/water passes through the tubes, the steam/water temperature will be increased while the tube wall, sodium, and shell (when applicable) temperatures are decreased. The basic equation was derived in a manner similar to that for the evaporator hot transient. The equation for the outer tube region is

$$R' = \frac{R}{1 + \left(\frac{\rho_t A_t C_{pt}}{\rho_w A_w C_{pw}} \right) \left(1 - e^{-\alpha_t \theta} \right) + \left(\frac{\rho_n A_n C_{pn}}{\rho_w A_w C_{pw}} \right) \left(1 - e^{-\alpha_t \theta} \right) \left(1 - e^{-\alpha_n \theta} \right)} + \frac{\left(\frac{\rho_s A_s C_{ps}}{\rho_w A_w C_{pw}} \right) \left(1 - e^{-\alpha_t \theta} \right) \left(1 - e^{-\alpha_n \theta} \right) \left(1 - e^{-\alpha_s \theta} \right)}{\dots (17)}$$

For the central tube region, the above equation applies; but the term

$$\left(\frac{\rho_s A_s C_{ps}}{\rho_w A_w C_{pw}} \right),$$

which allows for the heat capacity of the shell, is made zero.

The above equation enables one to compute the corrected temperature ramp R' from the applied ramp R allowing for heat transfer from the tube wall, sodium, and shell.

The temperature ramp which will be applied to the superheater will result from the water level in the evaporator rising into the superheater. However, as the water level rises and encounters a tube wall temperature above the water saturation temperature, boiling will take place and cool the wall. Under these conditions, the amount of water which would be transformed into steam is directly related to the amount of heat which the tube walls, sodium, and shell lose to the water.

At the superheater inlet, the conservative assumptions were made that the applied temperature ramp on the steam or water side was that resulting from the evaporator outlet; that is,

- a) Steam temperature decreasing from 715 to 668°F in 1.2 sec
(R = 38.8°F/sec),
- b) Steam/water temperature remaining constant at 668°F for 3.7 sec,
and
- c) Water temperature decreasing from 668 to 500°F in 3.3 sec
(R = 51°F/sec),

Heat capacities for the tubes, sodium, and shell for the evaporator have been previously determined where for the steam/water, there will be two values dependent on the region being considered:

Steam in superheat region

$$\rho_w A_w C_{pw} = 50(1.03 \times 10^{-3})(2.34) = 0.1205 \text{ Btu/}^\circ\text{F-ft, and}$$

Water in preheat region

$$\rho_w A_w C_{pw} = 50(1.03 \times 10^{-3})(1.45) = 0.0746 \text{ Btu/}^\circ\text{F-ft.}$$

The corrected, heat-transfer film coefficients, allowing for the conduction resistance to the center of the tube wall, sodium annulus, or shell wall, are:

Steam in superheat region

$$h'_{wt} = \frac{1}{\frac{1}{900} + \frac{0.095}{2(14)(12)}} = 717 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F, ,}$$

Water in preheat region

$$h'_{wt} = \frac{1}{\frac{1}{2400} + \frac{0.095}{2(14)(12)}} = 1430 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F ,}$$

Sodium

$$h'_n = \frac{40(12)}{(0.4175) \ln \frac{0.4175}{0.3125}} = 3960 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F, and}$$

Shell

$$h'_s = \frac{2(14)(12)}{0.250} = 1340 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

Calculation of the time constants by use of the previous data gives:

Steam in superheat region

$$\frac{h'_{wt} P_t}{\rho_t A_t C_{pt}} = \frac{717(\pi)(0.435)}{12(0.0645)} = 1265 \text{ hr}^{-1} = 0.352 \text{ sec}^{-1},$$

Water in preheat region

$$\frac{h'_{wt} P_t}{\rho_t A_t C_{pt}} = \frac{1430(\pi)(0.435)}{12(0.0645)} = 2520 \text{ hr}^{-1} = 0.700 \text{ sec}^{-1},$$

Sodium

$$\frac{h'_n P_n}{\rho_n A_n C_{pn}} = \frac{3960(\pi)(2)(0.4175)}{12(0.0573)} = 15,150 \text{ hr}^{-1} = 4.21 \text{ sec}^{-1},$$

Shell

$$\frac{h'_s P_s}{\rho_s A_s C_{ps}} = \frac{1340(\pi)(6.607)}{12(0.1175)(18)} = 1095 \text{ hr}^{-1} = 0.304 \text{ sec}^{-1}.$$

With the above data and Equation 17, the corrected temperature ramps for the outer tube region and the central tube region of the evaporator were calculated. From these data, the mixed mean temperature ramp was determined, based on 18 tubes in the outer tube region and 19 tubes in the central tube region. This corrected mean temperature ramp was then applied to the superheater inlet and the transient tube and shell mean temperatures were computed by use of a digital computer program.²² The heat transfer coefficients used in the digital computer program were

Superheat region,

$$h_w = 900 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}$$

and Boiling and preheat region,

$$h_w = 2400 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

The results of the calculations for the shell and tube mean temperatures at the superheater inlet are shown in Figure 15.

Figure 15. Superheater, Cold-Shock Temperatures. Sodium Flow Stops, Water Flow Continues at Full Flow (5.34 ft/sec)

For the superheater outlet in the superheat region, allowance was made for the amount of water which was boiled off due to cooling of the tube wall, sodium, and shell. By use of the steam-side film coefficient, various corrected steam-side temperature ramps were tried in the digital computer program until the amount of heat lost by the tube wall, sodium, and shell wall were equal to the amount of heat which went into boiling off the water.

a. Central Tube Region

The mean tube temperature at the superheater outlet changes from 1090 to 790°F and the mean tube temperature at the superheater inlet changes from 920 to 700°F. Therefore, the average change in temperature of the tube and sodium is 260°F. The heat removed is

$$Q = (0.0645 + 0.0573)(40.0)(260) = 1265 \text{ Btu,}$$

where:

0.0645 Btu/°F-ft = heat capacity of tube wall,

0.0573 Btu/°F-ft = heat capacity of sodium,

40.0 ft = effective length of superheater, and

260°F = temperature change of tube wall and sodium.

The heat absorbed by the water is

$$Q = 50(1.03 \times 10^{-3})(5.340)[335 + 2.34(191)],$$

where:

50(lb/ft³) = water density,

1.03 x 10⁻³(ft²) = tube area,

5.34(ft/sec) = water velocity in tube,

θ(sec) = time,

335(Btu/lb) = heat of vaporization for water,

2.34(Btu/lb) = heat capacity of steam, and

191(°F) = change in steam temperature.

Equating the two equations gives

$\theta = 6.0$ sec which represents the additional time it takes for the water level to rise to the superheater outlet. This, when combined with the transport time of 9.0 sec, gives a corrected transport time of 15.0 sec for the water to travel through the superheater.

b. Outer Tube Regions

The mean tube and sodium temperature changes 252°F and the shell temperature changes 237°F. The heat removed is then

$$Q = (0.0645 + 0.0573)(40)(252) + (0.1175)(40)(237) = 2343 \text{ Btu.}$$

Since the equation describing the heat absorbed by the water is the same as that for the central tube region, $\theta = 11.0$ sec; this, when combined with the transport time of 9.0 sec, gives a corrected transport time of 20 sec for the water to travel through the superheater.

Determination of the transient tube and shell mean temperatures at the superheater outlet under the boiling and preheat conditions were made with the digital computer program.²² The corrected temperature ramps at the superheater outlet were determined from Equation 17 where the applied temperature ramp was the corrected temperature ramp at the superheater inlet.

The results of the calculations for the shell and tube mean temperatures at the superheater outlet are shown in Figure 15.

From Figure 15, the shell and tube mean temperature difference averaged over the superheater length for the center tube and outer tube regions were determined. These temperature differences are shown in Figure 16.

The analyses for the superheater cold-shock condition assumed the sodium flow stopping instantaneously. This assumption is conservative since in the actual case the sodium flow would continue for about 5 to 7 sec due to pump and fluid inertia. With the sodium flow continuing, the tubes would not be cooled as quickly, which would reduce the final temperature differences between the tubes and shell.

The division of the module cross section into an outer tube region and a central tube region assumed there was no mixing or conduction of the

Figure 16. Superheater, Cold-Shock Temperature Differences sodium between the two regions. In fact, there will be some mixing and conduction which would again reduce the final temperature differences over what had been calculated.

3. Tube Sheets

The upper tube sheet of the superheater module will have an argon gas space between it and the sodium level; therefore, considering the 37 penetrations through the tube sheet for the steam, there should be no axial temperature gradients, even under transient conditions.

The lower tube sheet of the superheater module will have about 18 in. of sodium between it and the lower edge of the sodium exit nozzle. The steady-state temperature gradient through the tube sheet was calculated on the assumption of the tube sheet being a 1-3/4-in. plate with 6 in. of stagnant sodium above it. The 6 in. of stagnant sodium is conservative, since there are two tube supports about 12 in. away which restrict any sodium flow; from Reference 23, the maximum penetration depth of a jet is about five diameters. At the tube support, the equivalent diameter is about 3/4 in. which would give a maximum

jet penetration of about 4 inches. Analyses, assuming a fin geometry having the sink temperature as the inlet steam temperature, gave a maximum axial temperature gradient of 5°F . For fatigue analysis, a gradient of 10°F was used.

For transient conditions, it was assumed that the interface of the sodium at 6 in. above the tube sheet took a step temperature change from 975 to 1100°F . Under this condition, a maximum temperature difference across the tube sheet of 85°F was computed.

The evaporator module was analyzed in a similar manner. For the upper tube sheet with the argon gas blanket, there should be essentially no axial gradient. The lower tube sheet was calculated to have an axial gradient of 5°F , but 10°F was used in the fatigue analyses. The transient analysis assumed that the sodium interface 6 in. above the lower tube sheet took a step temperature change from 500 to 1100°F ; this gave a maximum temperature difference across the tube sheet of 130°F .

V. STRESS ANALYSES

A. TUBES

1. Pressure Stresses

a. Superheater Module

The calculation of the required tube thickness was based on Reference 9, paragraph UG-27, where

$$t = \frac{Pr_i}{SE - 0.6P} \quad \dots(18)$$

Since

$$E = 1; S = 9,350 \text{ psi at } 1115^\circ\text{F for 321H material;}$$

then

$$t = \frac{2800 \times 0.2175}{9350 - 0.6 \times 2800} = \frac{610}{7670} = 0.0795 \text{ in.}$$

Since a minimum wall of 0.095 in. is specified, the margin of safety (MS) is

$$MS = \frac{0.095}{0.0795} - 1 = 0.195$$

b. Evaporator Module

Using the same equation as in "a," with $E = 1$, and $S = 8,650 \text{ psi at } 975^\circ\text{F for 5 chrome - 1/2 moly material,}$

$$t = \frac{2900 \times 0.2175}{8650 - 0.6(2900)} = \frac{630}{6910} = 0.0912 \text{ in.}$$

Again, a minimum wall of 0.095 in. is specified and the margin of safety is:

$$MS = \frac{0.095}{0.0912} - 1 = 0.04 \quad .$$

2. Vibration Analyses

With the sodium on the shell side of the modules, the sodium flow presents two possible modes for tube vibration: (a) cross flow, which causes vibration according to the approximate relationship $f = 0.21v/D$, and (b) parallel flow, which causes the tubes to vibrate at their natural frequency.

With cross flow of the sodium, it is necessary to make sure that the exciting frequency does not approach the natural frequency of the tubes. For parallel flow, an estimate of the amplitude of vibration is made in order to check the alternating stress in the tubes.

a. Superheater

(1) Natural Frequency

Based on the equation for the natural frequency of a simply supported beam

$$f_n = \frac{1.57}{\ell^2} \sqrt{\frac{EIg}{w}} \quad , \quad \dots(19)$$

the natural frequency of the tubes is calculated where $L = 28$ in., $E = 25 \times 10^6$ psi, $I = 0.00575$ in.⁴, and $w = 0.0447$ lb/in.; then

$$f_n = \frac{1.57}{(28)^2} \sqrt{\frac{(25 \times 10^6)(0.00575)(386)}{0.0447}}$$

or

$$f_n = 70.5 \text{ cps.}$$

From the above value for the natural frequency, the approximate crossflow velocity that will cause resonance is:

$$v = \frac{f_n D_o}{0.21} = \frac{(70.5)(0.625)}{0.21} = 210 \text{ in./sec} ,$$

or

$$v = 17.5 \text{ ft/sec}$$

In the present design, the maximum crossflow velocity in the high temperature module shell should not exceed 3 ft/sec. Therefore, no excessive tube vibration stresses due to crossflow of the sodium are expected.

(2) Vibration Due to Parallel Flow

From experimental data,²⁵ the amplitude of vibration is determined from a curve of δ/D_o vs $K\Omega\Gamma^{1/2}$ where

$$\Gamma = \frac{\rho_n v^2 \ell^4}{gEI} \quad \dots(20)$$

Hence

$$\rho = 0.0307 \text{ lb/in.}^3$$

$$v = 15 \text{ ft/sec}$$

$$= 180 \text{ in./sec,}$$

then

$$\begin{aligned} \Gamma &= \frac{0.0307 \times 3.23 \times 10^4 \times 6.1 \times 10^5}{386 \times 25 \times 10^6 \times 0.00575} \\ &= 10.9 \end{aligned}$$

or

$$\Gamma^{1/2} = 3.3;$$

and

$$\Omega = \frac{\rho_n v^2}{\mu_n^f} = \frac{0.0307 \times 3.23 \times 10^4}{1.41 \times 10^{-5} \times 2\pi \times 70.5} = 1.59 \times 10^5 \quad \dots(21)$$

where

$$\mu = 1.41 \times 10^{-5} \text{ lb/in. -sec}$$

and

$$f_n = 70.5 \text{ cycles/sec;}$$

then,

$$K \Omega \Gamma^{1/2} = (5)(1.59 \times 10^5)(3.3) = 2.63 \times 10^6 .$$

Then, from the curve of Reference 25,

$$\frac{\delta}{D_o} = 1.1 \times 10^{-3},$$

where:

$$D_o = 0.625 \text{ in.}$$

then,

$$\delta = 1.1 \times 10^{-3} \times 0.625 = 6.89 \times 10^{-4} \text{ in.}$$

Knowing the amplitude of vibration, the maximum stress can be calculated by assuming that the tube is a simply supported beam. Therefore,

$$\sigma = \frac{MC}{I} \quad \dots(22)$$

where

$$M = \frac{W\ell}{8}$$

and

$$W = \frac{384EI\delta}{5L^3}$$

from the deflection equation for a simply supported beam. Then

$$\sigma = \frac{384E\delta C}{40\ell^2}$$

where

$$C = 0.3125 \text{ in.} \quad \dots(23)$$

then

$$\sigma = 656 \text{ psi.}$$

With the above computed stress, no problems associated with vibration due to parallel flow are expected.

b. Evaporator

(1) Natural Frequency

Applying the same equations as described in "a (1)"

$$f_n = \frac{1.57}{(27.4)^2} \sqrt{\frac{(25 \times 10^6)(0.00575)(386)}{0.0449}}$$
$$= 73.7 \text{ cps} \quad .$$

The crossflow velocity required to resonate the tubes is then

$$v = \frac{(73.7)(0.625)}{0.21} = 220 \text{ in./sec} = 28.9 \text{ ft/sec} \quad .$$

The maximum crossflow velocity in the evaporator module shell should not exceed 3 ft/sec; therefore, no excessive tube vibrations are expected due to crossflow of the sodium.

(2) Vibration Due to Parallel Flow

Applying the same equations as described in "a (2)"

$$\Gamma = 10.9 \left(\frac{27.4}{28} \right)^4 = 10.0 \quad ,$$

$$\Gamma^{1/2} = 3.16 \quad ,$$

$$\Omega = 1.59 \times 10^5 \left(\frac{70.5}{73.7} \right) = 1.52 \times 10^5 \quad ,$$

$$K\Omega\Gamma^{1/2} = 5 \times 1.52 \times 10^5 \times 3.16 = 2.40 \times 10^6 \quad ,$$

$$\frac{\delta}{D_o} = 1.5 \times 10^3 \quad ,$$

and

$$\delta = 0.625 \times 1.5 \times 10^{-3} = 0.938 \times 10^{-3} \text{ in.}$$

The stress due to vibration caused by parallel flow would then be

$$\sigma = \frac{0.938}{0.96} \left(\frac{28}{27.4} \right)^2 \times 920 = 938 \text{ psi} .$$

Again, due to the low computed stress, no problems associated with vibration due to parallel flow are expected.

3. Fatigue Analyses

a. Superheater

1) The longitudinal stress due to steam pressure and sodium pressure is calculated in the following manner.

The end loads on the tubes and shell caused by steam pressure is found from the equation

$$\begin{aligned} F_1 &= P_s A_t = 2800 \left(\frac{\pi}{4} \right) (0.435)^2 (37) \\ &= 15,300 \text{ lb} . \end{aligned}$$

The end loads caused by sodium pressure is found from the equation

$$\begin{aligned} F_2 &= P_n A_s = 300 \left(\frac{\pi}{4} \right) \left([6.357]^2 - 37 (0.625)^2 \right) \\ &= 6,120 \text{ lb} , \end{aligned}$$

and the stress in the shell and tubes is

$$\sigma = \frac{F_1 + F_2}{A_t + A_s} = \frac{21,420}{11.0} = 1,950 \text{ psi (tension)} .$$

2) Steady-state thermal stress due to the operating ΔT across the tube walls is calculated in the following manner.

From the equation for the thermal stress caused by a linear temperature gradient across a wall, the stress in the tube is

$$\sigma = \frac{E\alpha \Delta t}{2(1 - \nu)} = \frac{(25 \times 10^6)(10 \times 10^{-6})(78)}{2(1 - 0.3)} = 13,900 \text{ psi.} \quad \dots(24)$$

(Tension inside surface and
compression outside)

3) Steady-state stress during operation due to ΔT between the shell and tubes is found from the following equation derived in Appendix C:

$$\sigma_s = E\alpha \left[\frac{\Delta t_{it} A_{it} + \Delta t_{ot} A_{ot}}{A_s + A_{it} + A_{ot}} \right] ,$$

Assuming

$$\Delta t_{it} = \Delta t_{ot} = 40^\circ \text{F},$$

$$A_{it} + A_{ot} = 2.98 + 2.83 = 5.81 \text{ in.}^2 ,$$

and

$$A_s = 5.19 \text{ .}$$

$$\sigma_s = 25 \times 10^6 \times 10^{-5} \left[\frac{40 \times 5.81}{11} \right] = 5,300 \text{ psi (compression) .}$$

$$\sigma_t = 25 \times 10^6 \times 10^{-5} \times 40 \times \frac{5.19}{11} = 4,730 \text{ psi (tension) .}$$

4) Total Steady-State Stress

From Equations (11) and (12),

$$\sigma_{ss} = 13,900 + 1,950 + 4,730 = 20,580 \text{ psi (tension) .}$$

5) Transient Stress Due to Cold Shock

In Appendix C, the transient thermal stress caused by a cold shock is calculated, giving, $\sigma_{it} = 33,600 \text{ psi (tension)}$.

6) Transient Stress Due to Hot-Shock

In Appendix D, the transient thermal stress for a hot-shock is calculated, giving, $\sigma_{it} = 108,600$ psi (compression).

7) Fatigue Damage

(a) Transient Conditions

The method outlined in Reference 27 for calculating fatigue damage is used in this section. First the stress range is computed as follows.

$$\text{Stress range} = (\sigma_{\min}) + (\sigma_{\max}) = 33,600 + 108,600 = 142,200 \text{ psi} ;$$

then the alternating stress and mean stress is calculated:

$$\sigma_{\text{alt}} = \frac{1}{2} (\sigma_{\text{range}}) = \frac{1}{2} (142,200) = 71,100 \text{ psi}$$

$$\sigma'_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} = \frac{108,600 - 33,600}{2} = 37,500 \text{ psi} .$$

The actual mean stress depends on whether or not the yield stress has been exceeded. When the alternating stress exceeds the yield stress, such as it does in this case, the mean stress is zero.

Hence, from curve 2, Reference 27, for $\sigma_{\text{alt}} = 71,100$ psi at 1000°F ; $N = 300$ cycles.

With $N = 300$ cycles, the damage factor is computed:

$$\text{Damage Factor} = \frac{n}{N} = \frac{30}{300} = 0.1 .$$

(b) Steady-State Conditions:

Stress range = 20,580 psi,

$$\sigma_{\text{alt}} = \frac{1}{2} (20,580) = 10,290 \text{ psi}$$

$$\sigma'_m = 10,290 \text{ psi},$$

$$\sigma_m = \sigma_{\text{yield}} - \sigma_{\text{alt}} = 18,900 - 10,290 = 8,610 \text{ psi},$$

$$n = 11,000 \text{ cycles};$$

$$N = 125,000 \text{ cycles, and}$$

$$\text{Damage factor} = \frac{11,000}{125,000} = 0.088.$$

(c) Accumulative Fatigue Damage

$$\text{Accumulative Damage} = 0.188.$$

The permissible design damage factor²⁷ is 0.8.

b. Evaporator Module

(1) Tube Buckling

The evaporator module is analyzed for hot-shock only since cold-shock in this module will not be severe enough to cause the tubes to yield.

From Appendix C, Equations (38), (30), and (40), and Figure 13,

$$\begin{aligned} \sigma_s &= \frac{(25 \times 10^6)(7 \times 10^{-6})(49)(2.98)(29)(2.83)}{11} \\ &= 3,630 \text{ psi (tension)} \end{aligned}$$

$$\sigma_{ot} = 29(175) - 3,630 = 1,440 \text{ psi (compression)}$$

and

$$\sigma_{it} = 49(175) - 3,630 = 4,970 \text{ psi (compression)}.$$

The critical buckling stress for the tubes is then

$$\sigma_{cr} = \frac{C\pi^2 E}{\left(\frac{\ell}{T}\right)^2} = \frac{C(9.86)(25 \times 10^6)(0.0367)}{(27.4)^2} = 12,000 \text{ C} \quad \dots(25)$$

For pinned ends, $C = 1$, hence $\sigma_{cr} = 12,00$ psi; therefore, the margin of safety is

$$MS = \frac{12,000}{4,970} - 1 = 1.42 \quad .$$

(2) Steady-State Stresses for the Evaporator Module

These will be calculated using the same equation used for the superheater module. The end loads on the tubes caused by steam pressure is

$$F_1 = 2900 \left(\frac{\pi}{4}\right)(0.435)^2(37) = 15,900 \text{ lb.}$$

The end load caused by sodium pressure is

$$F_2 = 300 \left(\frac{\pi}{4}\right) \left[(6.357)^2 - 37 (0.625)^2 \right] = 6,120 \text{ psi.}$$

The stress in the tubes is:

$$\sigma = \frac{22,020}{11} = 2,000 \text{ psi.}$$

(a) Steady-state stress due to operating Δt across the tube wall, is

$$\sigma = \frac{E\alpha\Delta T}{2(1-\nu)} = \frac{(25 \times 10^6)(10 \times 10^{-6})(88)}{2(1-0.3)} = 15,700 \text{ psi} \quad .$$

(b) Steady-state stress due to operating the ΔT between the shell and tubes, is

$$\sigma_s = 25 \times 10^6 \times 10^{-5} \left[\frac{70 \times 5.81}{11} \right] = 9,300 \text{ psi}$$

$$\sigma_T = 25 \times 10^6 \times 10^{-5} \times 70 \times \frac{5.17}{11} = 8,300 \text{ psi} .$$

(c) Total steady-state stress is

$$\sigma_{ss} = 15,700 + 2,000 + 8,300 = 26,000 \text{ psi} .$$

(d) Fatigue Damage:

Stress range = 26,000 psi,

$$\sigma_{alt} = 1/2 (26,000) = 13,000 \text{ psi},$$

$$\sigma'_m = 13,000 \text{ psi},$$

$$\sigma_m = 18,380 - 13,000 = 5,380 \text{ psi},$$

$$n = 11,000 \text{ cycles},$$

$$N = \infty,$$

and

Fatigue Damage = 0.

c. Weld Between Tube and Tube Sheet

Assume the weld temperature is 2600°F over a region of 1/2 inch. In cooling down to room temperature then, the elongation in the tube is

$$\Delta l = X \alpha \Delta t$$

$$\Delta l = 1/2 \times 10^{-5} \times (2600 - 70) = 0.01270 \text{ in. /in.}$$

The residual stress in the tube is then

$$\sigma = E \frac{\Delta l}{l} = \frac{25 \times 10^6 \times 0.01270}{551} = 575 \text{ psi} .$$

This is a conservative value since the weld would yield at the higher temperatures.

B. SHELL

1. Pressure Stresses

a. Superheater

(1) Shell Reducer Section

From Reference 9* the required shell thickness is calculated using the equation

$$t = \frac{PD_i}{2 \cos \alpha (SE - 0.6P)} , \quad \dots(26)$$

*Paragraph UG-32g

where

$$S = 7,000 \text{ psi at } 1115^\circ \text{F}$$

$$E = 1$$

and

$$t = \frac{(300)(10.02)}{(0.944)(2)(7000 - 180)} = 0.238 \text{ in.}$$

The margin of safety is

$$MS = \frac{0.365}{0.238} - 1 = 0.535$$

(2) Shell Pressure Stress

From Reference 9, Paragraph UG-27, the required shell thickness is calculated from the equation

$$t = \frac{Pr_i}{SE - 0.6P} = \frac{(300)(3.178)}{7000 - 180} \quad \dots(27)$$

$$= \frac{955}{6820} = 0.140 \text{ in.}$$

The margin of safety is

$$MS = \frac{0.25}{0.14} - 1 = 0.79$$

b. Evaporator

(1) Shell Reducer Section

The dimensions of the reducer section of the evaporator module are the same as those of the superheater; it is then only necessary to account for the difference in temperature and material. Hence, using the equation from "a (1)", the required shell thickness is:

$$t = \frac{(300)(10.02)}{(0.944)(2)(7800 - 180)} = 0.213 \text{ in.}$$

where

$$S = 7,800 \text{ psi at } 990^\circ\text{F}$$

$$E = 1.0.$$

The margin of safety is

$$MS = \frac{0.365}{0.213} - 1 = 0.715$$

(2) Shell Pressure Stress

Using the equations from "a (2)" the required shell thickness is

$$t = \frac{P r_i}{SE - 0.6P} = \frac{(300)(3.178)}{7800 - 180}$$

$$t = \frac{955}{7620} = 0.125 \text{ in.}$$

The margin of safety is:

$$MS = \frac{0.25}{0.125} - 1 = 1.00.$$

2. Nozzles

The maximum reactions on the sodium side of the superheater nozzles are on the 10 by 6-in. sodium inlet tee; hence, this is the only nozzle that will be analyzed for the superheater. The maximum loads on the evaporator sodium side nozzles are on the lower 8 by 6-in. nozzle. The water side nozzles for both modules have no significant loads on them.

a. Superheater

The reactions on the superheater 10 by 6-in. tee are given on the following page.

$$M_x = -465 \text{ in. -lb,}$$

$$M_y = 3472 \text{ in. -lb,}$$

$$M_z = -24,748 \text{ in. -lb,}$$

$$F_x = -625 \text{ lb,}$$

$$F_y = -592 \text{ lb, and}$$

$$F_z = -145 \text{ lb.}$$

The method of analysis that will be used in this section is outlined in Reference 24.

1) Longitudinal Moment

$$M_\ell = M_z + F_y \times 6.625 = 24,748 + 592 \times 6.625$$

$$M_\ell = 28,668 \text{ in. -lb.}$$

2) Circumferential Moment

$$M_c = M_y + F_z \times 6.625 = 3,472 - 145 \times 6.625$$

$$M_c = 2,512 \text{ in. -lb.}$$

3) Torsional Moment

$$T = M_x = 465 \text{ in. -lb.}$$

4) Bending Moment

$$M_b = \sqrt{M_c^2 + M_\ell^2} = \sqrt{(2,512)^2 + (28,668)^2}$$

$$= 28,800 \text{ in. -lb.}$$

5) Torsional Shear

$$\sigma_T = \frac{T}{2Z} = \frac{465}{2(8.95)} = 26 \text{ psi (negligible).}$$

6) Bending Stress

$$\sigma_b = \frac{M_b}{Z} = \frac{28,800}{8.95} = 3,220 \text{ psi.}$$

7) Stress Concentration

$$i = \frac{(0.9)}{h^{2/3}} \quad \dots(28)$$

$$= \frac{0.9}{0.468} = 1.92$$

8) Expansion Stress

$$\sigma_E = i\sigma_b = (1.92)(3,220) = 6,180 \text{ psi}$$

$$MS = \frac{7200}{6180} - 1 = 0.17$$

9) Membrane Stress

$$\sigma_\ell = \frac{Pr_i}{2t} = \frac{(300)(6.06)}{2(0.280)(2)} = 1620 \text{ psi}$$

$$\sigma_c = 2\sigma_\ell = 3240 \text{ psi}$$

$$\sigma_{\text{eff}} = \sqrt{(3240)^2 + (1620)^2} = 3580 \text{ psi}$$

$$MS = \frac{7200}{3580} - 1 = 1.01$$

b. Evaporator

The method of analysis for this tee is the same as that in the previous section.

$$M_z = 2,550 \text{ ft-lb}$$

$$F_y = 565 \text{ lb}$$

1) Moment

$$M_b = M_z + F_y \times 6.625 = 30,500 + 565 \times 6.625$$

$$M_b = 30,900 \text{ in. -lb}$$

2) Bending Stress

$$\sigma_b = \frac{M_b}{Z} = \frac{30,900}{8.95} = 3,450 \text{ psi}$$

3) Stress Concentration

$$i = \frac{0.9}{h^{2/3}} = \frac{0.9}{0.444} = 2.03$$

4) Expansion Stress

$$\sigma_E = i\sigma_b = 2.03 \times 3,450$$

$$\sigma_E = 7,000 \text{ psi}$$

$$MS = \frac{7,800}{7,000} - 1 = 0.115$$

5) Membrane Stress

Same as for "hi-temp" module tee

$$\sigma_{\text{eff}} = \frac{7800}{3580} - 1 = 1.08$$

3. Mounting Pad

The mounting pad is shown below

The forces on the mounting pad are as follows:

F_1 is the support load of each lug and F_2 is the frictional force caused by differential thermal expansion between the shell and support ring. This will be assumed to be at its maximum value. F_3 , F_4 , and M are resultants of F_1 and F_2 .

The stresses in the shell will be calculated by use of the methods set up by P. P. Bijlaard as outlined in Reference 26. It is necessary to find an equivalent rectangle which has the same moment of inertia as that of the weld geometry shown below.

The moment of inertia about axis y-y

$$I_{y-y} = 107/64;$$

The moment of inertia about x-x

$$I_{x-x} = 93/192.$$

Then, for an equivalent rectangle x by y ,

$$I_{x-x} = \frac{yx^3}{12} = 107/64$$

and

$$I_{y-y} = \frac{yx^3}{12} = 93/192 \quad .$$

Solving the above two equations where $y = 1.33$ in., $x = 2.47$ in., and using Bijlaard's notation:

$$C_1 = \frac{x}{2} = \frac{2.47}{2} = 1.24 \text{ in.},$$

$$C_2 = \frac{y}{2} = \frac{1.33}{2} = 0.667 \text{ in.},$$

$$B_1 = \frac{C_1}{a} = \frac{1.24}{3.303} = 0.3756,$$

$$B_2 = \frac{C_2}{a} = \frac{0.667}{3.303} = 0.202,$$

$$\gamma = \frac{a}{t} = \frac{3.303}{1.24} = 13.212,$$

and

$$B_2/B_1 = \frac{C_2}{C_1} = \frac{0.667}{1.24} = 0.538.$$

Using the curves in Reference 26, we obtain for the moments and membrane forces caused by the moment M :

$$M_\phi = 0.0398 M \text{ per in.},$$

$$M_x = 0.0563 M \text{ per in.},$$

$$N_\phi = 0.565 M \text{ per in.},$$

and

$$N_x = 0.1682 M \text{ per in.}$$

a) The moments and membrane forces caused by F_4 are:

$$M_{\phi} = 0.078 F_4 \text{ per in.},$$

$$M_x = 0.048 F_4 \text{ per in.},$$

$$N_{\phi} = 0.545 F_4 \text{ per in.},$$

and

$$N_x = 0.696 F_4 \text{ per in.}$$

Performing a force balance on the mounting pad gives

$$F_4 = F_2 = \mu F_1,$$

$$F_1 = F_3,$$

and

$$3/4 F_4 + 3/8 F_1 - M = 0.$$

Assuming $\mu = 0.2$, the moment is calculated:

$$M = \frac{3}{4} (0.2) F_1 + \frac{3}{8} F_1 = \frac{21}{40} F_1.$$

b) Stresses: Superimposing all the axial stresses gives the equation

$$\sigma_X = \frac{Pr_i}{t} + \frac{6M_X}{t^2} + \frac{N_X}{t} \quad \dots (29)$$

and in the circumferential direction,

$$\sigma_{\phi} = \frac{Pr_i}{2t} + \frac{6M_{\phi}}{t^2} + \frac{N_{\phi}}{t} \quad \dots (30)$$

Solving these equations in terms of F_1 where $P = 300$ psi,

$$\sigma_X = 1982 + 4.19 F_1,$$

and

$$\sigma_{\phi} = 3964 + 5.19 F_1.$$

The ASME boiler code allowable is 5750 psi; specifications require that the peak stress be less than 1.5 times code stress. Therefore, calculating the peak stress using the equation

$$\sigma_{\text{eff}}^2 = \sigma_X^2 + \sigma_\phi^2 - \sigma_X \sigma_\phi \quad \dots(31)$$

gives

$$\sigma_{\text{eff}} = 1.5(5750) = 8610 \text{ psi.}$$

Solving for F_1 ,

$$F_1^2 + 1.376 \times 10^3 F_1 - 2.77 \times 10^6 = 0 \quad ;$$

$F_1 = 1100 \text{ lb}$ which is the capability of each lug. The design load on each lug is $W = 400 \text{ lb}$.

Then the margin of safety is

$$MS = \frac{1100}{400} - 1 = 1.75.$$

c) Stress in the weld: The shear stress in the weld is calculated from the equation

$$\sigma = \frac{0.707 F_1}{hl} \quad \dots(32)$$

$$\sigma = \frac{0.707(1100)}{0.5(1.0)} = 1550 \text{ psi} \quad .$$

The margin of safety is

$$MS = \frac{5750}{1550} - 1 = 2.7.$$

4. Buckling of Shell

The maximum compressive stress in the shell occurs during a cold shock in the superheater module. This value, calculated in Appendix C, is 13,200 psi.

Assuming the shell to be pinned at both ends, the critical buckling stress can be found from the Euler formula

$$\sigma_{cr} = \frac{\pi^2 E}{\left(\frac{\ell}{r}\right)^2} = \frac{\pi^2 EI^2}{\ell^2 A^2} \quad , \quad \dots(33)$$

where

$$\ell = 480 \text{ in.} ,$$

$$I = 21.9 \text{ in.}^4 ,$$

$$A = 5.18 \text{ in.}^2 ,$$

$$E = 25 \times 10^6 \text{ psi} ,$$

and

$$\sigma_{cr} = 19,100 \text{ psi} .$$

The margin of safety is

$$MS = \frac{19,100}{13,200} - 1 = 0.45 .$$

5. Sodium-Water Interaction Pressure Stress

In the event that the steam generator tubes should burst, allowing the sodium and water to interact, it will be assumed that the shell must be capable of holding the steam pressure for a period of five minutes without exceeding the ultimate strength of the shell material.

a. Superheater

From the previous calculations in Section B-1, it was shown that the weakest shell area is in the reducer sections; hence, from Equation 26, the minimum shell thickness is:

$$t = \frac{PD_i}{2 \cos \alpha (SE - 0.6P)} \quad ,$$

where

$$S = \sigma_{alt} = 48,000 \text{ psi for 321 H stainless steel at } 1115^\circ\text{F} ;$$

then,

$$t = \frac{2800 \times 10.02}{1.888(48,000 - 1680)}$$

$$t = 0.327 \text{ inches.}$$

The margin of safety is

$$MS = \frac{0.365}{0.327} - 1 = 0.115.$$

b. Evaporator

Using the same equation as in "a",

$$t = \frac{2900 \times 10.02}{1.888(50,500 - 1740)} = \frac{15,700}{48,760} = 0.323 \text{ in.,}$$

$$MS = \frac{0.365}{0.323} - 1 = 0.13.$$

C. TUBE SHEETS

The stress analysis pertaining to the tube sheets were based on experimental studies. These studies are documented in Reference 4, 5, and 6.

VI. APPENDICES

APPENDIX A

EVAPORATOR MODULE TUBES – THERMOCOUPLE TUBES

The attachment of the thermocouples on the tubes of the evaporator module utilized the design shown in Figure 8. This type of attachment does not jeopardize the integrity of the tubes by welding, but it does provide good thermal contact between the tube and thermocouple.

Under the most severe type of operation, the maximum diametral clearance between the tube and clamp due to differential expansion would be:

$$\Delta D = 0.625(10.3 - 7.2)(10^{-6})(900) = 0.00174 \text{ inches.}$$

Allowing for original looseness in the clamp, one might expect as a maximum, 0.004-in. diametral clearance. On this basis, the conduction coefficient across the radial argon gas gap of 0.002 in. would be

$$\frac{0.021(12)}{0.002} = 125 \text{ Btu/hr-ft}^2\text{-}^\circ\text{F}.$$

The maximum temperature difference between the tube wall and the surrounding argon gas at the sodium level would be about 130°F. This is based on the shell being at 975°F and the tube being at 715°F with the gas temperature midway between. In actuality, since the shell is acting like a fin with its base at 975°F and is losing heat by conduction up the shell and by convection and radiation to the gas, the shell temperature in the region of the thermocouple clamps should be much less than the 975°F. This would result in a convection current being set up within the argon gas due to the cooler gas being at the top. The net results of the lower shell temperature and the convection currents in the gas would be a lowering of the temperature difference between the gas and the tube wall particularly in the vicinity of the thermocouples. Therefore, one may consider the temperature difference of 130°F as a maximum value.

This, when compared to a free convection film coefficient for argon gas of about 2 Btu/hr-ft²-°F based on a temperature difference of $\frac{975-715}{2} = 130^\circ\text{F}$, gives an error of about $130 \left(\frac{2}{127}\right) = 2.0^\circ\text{F}$. In the practical case, the temperature difference will be less than 130°F and the radial gas gap will not be 0.002 in; so the 2°F represents a maximum error.

APPENDIX B

STEAM SIDE FOULING FACTOR

The feedwater to the evaporator units is assumed to have a purity of 100 ppb total dissolved solids at a flowrate of 36,700 lb/hr through one evaporator and superheater. The steam side heat transfer area is 184 and 168 ft² for the evaporator and superheater modules respectively.

Assuming all the dissolved solids are deposited uniformly in the evaporator and superheater, the amount deposited in one full power year would be approximately:

$$\begin{aligned}W_s &= 36,700 (8750)(100 \times 10^{-9}) \\&= 32.1 \text{ lb} \quad .\end{aligned}$$

Assuming a density of 100 lb/ft³ for the material deposited, the volume deposited would then be

$$V_s = 0.321 \text{ ft}^3 \quad .$$

The thickness of the deposit, assuming uniform thickness, would be

$$\Delta r_s = \frac{0.321}{184 + 168} = 9.1 \times 10^{-4} \text{ ft} \quad .$$

Based on previous calculations² the thermal conductivity of the deposit would be about 5 Btu/hr-ft-°F. With this value of the thermal conductivity, the fouling factor would be

$$R_{ff} = \frac{9.1 \times 10^{-4}}{5} = 1.82 \times 10^{-4} \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}} \quad .$$

If one were to assume that all the solids were deposited in the evaporator alone,

$$R_{ff} = \frac{17.4 \times 10^{-4}}{5} = 3.49 \times 10^{-4} \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}} \quad .$$

Since the assumption of depositing all the solids is rather severe and the probability of maintaining water purity below 100 ppb is rather good, a fouling factor between the two values calculated was used: that is,

$$R_{ff} = 2.5 \times 10^{-4} \frac{\text{hr-ft}^2\text{-}^\circ\text{F}}{\text{Btu}}$$

was used for design purposes.

It is hoped that the testing program for the Modular Steam Generators will enable determination of an experimental fouling factor.

APPENDIX C

SUPERHEATER COLD SHOCK

The temperature history of the superheater module for the cold shock condition is shown in Figure 15. The center region consists of 19 tubes; the outer region of 18 tubes and the shell.

Performing a force balance on the tubes and shell during a transient gives,

$$F_s = F_{it} + F_{ot} \quad \dots (34)$$

Equating relative strains,

$$\epsilon_s = \epsilon_{it} = \epsilon_{ot} \quad \dots (35)$$

which in the elastic range, gives

$$\frac{\sigma_s}{E} = \alpha \Delta t_{it} - \frac{\sigma_{it}}{E} = \alpha \Delta t_{ot} - \frac{\sigma_{ot}}{E} \quad ; \quad \dots (36)$$

also,

$$F = \sigma A \quad \dots (37)$$

Hence, solving these equations gives

$$\sigma_s = E\alpha \frac{\Delta t_{it} A_{it} + \Delta t_{ot} A_{ot}}{A_s + A_{it} + A_{ot}} \quad \dots (38)$$

$$\sigma_{it} = E\alpha \Delta t_{it} - \sigma_s \quad \dots (39)$$

and

$$\sigma_{ot} = E\alpha \Delta t_{ot} - \sigma_s \quad \dots (40)$$

When either the shell or tubes yield, the equations above do not apply. To analyze this, it will be assumed that after either a tube or the shell yields, the stress in it is constant.

For the cold-shock, the center tubes will yield first; hence, for the incremental stress,

$$\Delta\sigma_s = \frac{\Delta\sigma_{ot} A_{ot}}{A_s} \quad \dots(41)$$

and

$$\Delta\epsilon_s = \Delta\epsilon_{ot} \quad ;$$

or,

$$\frac{\Delta\sigma_s}{E} = \alpha \Delta(\Delta t_{ot}) - \frac{\Delta\sigma_{ot}}{E} \quad .$$

Solving these equations gives

$$\Delta\sigma_s = \frac{E\alpha \Delta(\Delta t_{ot})}{\frac{A_s}{A_{ot}} + 1} \quad \dots(42)$$

and

$$\Delta\sigma_{ot} = E\alpha \Delta(\Delta t_{ot}) - \Delta\sigma_s \quad \dots(43)$$

To find the Δt 's, from Figure 15, which initiate yielding of the center tubes requires an iteration process; this will not be shown here, but the final calculation will be shown.

After 7.3 sec,

$$\Delta t_{it} = 130^\circ\text{F}$$

$$\sigma_{\text{yield}(321 \text{ SS})} = 20,000 \text{ psi},$$

$$\Delta t_{ot} = 88^\circ\text{F}, \text{ and}$$

$$t_{it} = 965^\circ\text{F} \quad .$$

From Equations 38, 39, and 40,

$$\sigma_s = 25 \times 10^6 \times 10 \times 10^{-6} \left(\frac{130 \times 2.98 + 88 \times 2.83}{11} \right) ,$$

$$\sigma_s = 14,400 \text{ psi (compression)} \quad ,$$

$$\sigma_{it} = 250 \times 130 - 14,400 = 18,100 \text{ psi (tension)} \quad ,$$

and

$$\sigma_{ot} = 250 \times 88 - 14,400 = 7,600 \text{ psi (tension) } .$$

Superimposing the pressure stresses calculated in Section A-3 gives

$$\sigma_s = 14,400 - 1950 = 12,450 \text{ psi (compression) } ,$$

$$\sigma_{ot} = 7,600 + 2000 = 9,600 \text{ psi (tension) } ,$$

and

$$\sigma_{it} = 18,100 + 2000 = 20,100 \text{ psi (tension) } .$$

At this point, the center tubes yield, but the Δt continues to increase; hence, to calculate the incremental stresses in the outer tubes and shell, Equations 42 and 43 are used giving

$$\Delta\sigma_s = \frac{250 (97 - 88)}{\frac{5.18}{2.83} + 1} = 800 \text{ psi (compression)}$$

and

$$\Delta\sigma_{ot} = 250 (9) - 800 = 1,450 \text{ psi (tension) } .$$

The incremental strain in tubes and shell

$$\Delta\epsilon = \frac{1,450}{25 \times 10^6} = 5.82 \times 10^{-5} \text{ in./in. } .$$

The apparent incremental strain in the inner tubes is due to $\Delta(\Delta t_{in})$:

$$\Delta\epsilon_{it} = \alpha \Delta(\Delta t_{in}) = 10^{-5} (190 - 130) = 60 \times 10^{-5} \text{ in./in. } .$$

The apparent elastic stress in the inner tubes is

$$\Delta\sigma_{it} = (60 - 5.82) \times 10^{-5} \times 25 \times 10^6 = 13,600 \text{ psi (tension) } .$$

The total stress is

$$\sigma_s = 12,450 + 800 = 13,250 \text{ psi (compression) } ,$$

$$\sigma_{ot} = 9,550 + 1,450 = 11,000 \text{ psi (tension) } ,$$

and

$$\sigma_{it} = 20,000 + 13,600 = 33,600 \text{ psi (tension) } .$$

Isothermal conditions:

From Equations 38, 39, and 40,

$$\sigma_s = 22.7(190 \times 2.98 + 97 \times 2.83) \quad ,$$

$$\sigma_s = 19,100 \text{ psi (tension)} \quad ,$$

$$\sigma_{ot} = 97 \times 250 - 19,100 = 5200 \text{ psi (compression)} \quad ,$$

and

$$\sigma_{it} = 190 \times 250 - 19,100 = 28,400 \text{ psi (compression)} \quad ;$$

Therefore, when the steam module has reached isothermal conditions, the residual stresses are:

$$\sigma_s = 19,100 - 13,250 = 5850 \text{ psi (tension)} \quad ,$$

$$\sigma_{ot} = 11,000 - 5200 = 5800 \text{ psi (tension)} \quad ,$$

and

$$\sigma_{it} = 28,400 - 20,000 = 8400 \text{ psi (compression)} \quad .$$

APPENDIX D SUPERHEATER HOT SHOCK

The temperature history of the evaporator module for the hot-shock is shown in Figure 13. The gradients in the superheater module will not be this severe; but for conservatism, this curve will be used. These stresses will be superimposed upon those calculated in Appendix C for the isothermal condition.

Using Equation 25, the critical buckling stress for the tubes is 11,500 psi. Since in the isothermal condition, the compressive stress in the inner tubes is 8400 psi, the incremental compressive stress required to buckle the tubes is $\Delta\sigma_{it} = 11,500 - 8400 = 3100$ psi.

From curve 13, after 1.8 sec,

$$\Delta t_{it} = 26^\circ\text{F} \quad , \text{ and}$$

$$\Delta t_{ot} = 24^\circ\text{F} \quad .$$

From Equation 38, 39, and 40,

$$\Delta\sigma_s = 22.7 (26 \times 2.98 + 24 \times 2.83) \quad ,$$

$$\Delta\sigma_s = 3300 \text{ psi (tension)} \quad ,$$

$$\Delta\sigma_{ot} = 24 \times 250 - 3300 = 2700 \text{ psi (compression)} \quad ,$$

and

$$\Delta\sigma_{it} = 26 \times 250 - 3300 = 3200 \text{ psi (compression)} \quad .$$

At this time, the center tubes will buckle. The stress in these tubes will remain constant as the Δt 's continue to increase.

The peak stress in the shell and tubes caused by the hot-shock occurs at approximately 4 seconds; at this time the Δt 's are

$$\Delta t_{ot} = 36^\circ\text{F} \quad ,$$

$$\Delta t_{it} = 52^\circ\text{F} \quad ,$$

$$\Delta\sigma_s = \frac{250 (36 - 24)}{2.84} = 1050 \text{ psi (tension)} \quad ,$$

and

$$\Delta\sigma_{ot} = 250 \times 12 - 1050 = 1950 \text{ psi (compression)} \quad .$$

The incremental strain in tubes and shell is

$$\Delta\epsilon = \frac{1,950}{25 \times 10^6} = 7.8 \times 10^{-5} \text{ in./in.}$$

The apparent strain in inner tubes due to $\Delta(\Delta t_{in})$,

$$\Delta\epsilon_{it} = 10^{-5} \times (52 - 26) = 26 \times 10^{-5} \text{ in./in.}$$

The total strain in inner tubes that must be taken up in buckling of the center tubes,

$$\epsilon = (26 - 7.8) \times 10^{-5} = 18.20 \times 10^{-5} \text{ in./in.}$$

From Appendix E, the bending moment caused by this buckling is

$$\begin{aligned} M &= \frac{\sigma_{cr} A}{\pi} \sqrt{4\ell \Delta\ell} \\ &= \frac{(11,500)(0.157)}{\pi} \sqrt{(4)(27.4)(0.0874)} \\ &= 1,780 \text{ in.-lb} \end{aligned}$$

$$\sigma_b = \frac{MC}{I} = \frac{(1,780)(5)}{(0.00575)(16)} = 97,100 \text{ psi}$$

The total stress is

$$\sigma_s = 3300 + 1050 + 5850 = 10,200 \text{ psi (tension)}$$

$$\sigma_{ot} = 5800 - 1950 - 2700 = 1150 \text{ psi (tension)}$$

and

$$\sigma_{it} = 11,500 + 97,100 = 108,600 \text{ psi (compression)}$$

APPENDIX E

BUCKLING OF TUBES WITHOUT COMPLETE COLLAPSE

Assuming that a tube buckles, the following conditions apply:

Assuming that the total thermal growth of a tube is taken up in bending between two successive spacers, then for a given Δt ,

$$\Delta l = l \alpha \Delta t \quad \dots (44)$$

and

$$S = \Delta l + l \quad \dots (45)$$

where

$$S = \int_0^l \sqrt{1 + \left(\frac{ds}{dx}\right)^2} dx, \quad \dots (46)$$

$$\delta = \delta_{\max} \sin \frac{\pi X}{l},$$

and

$$\frac{d\delta}{dX} = \frac{\pi \delta_{\max}}{l} \cos \frac{\pi X}{l}$$

From the binomial expansion,

$$\int_0^l \sqrt{1 + \left(\frac{d\delta}{dX}\right)^2} dX = l + \frac{\pi^2 \delta_{\max}^2}{4l} - \frac{\pi^4 \delta_{\max}^4}{32l^3} + \dots \quad \dots (47)$$

Keeping only the first two terms on the right gives

$$S = \Delta\ell + \ell = \ell + \frac{\pi^2 \delta_{\max}^2}{4\ell}$$

or

$$\Delta\ell = \frac{\pi^2 \delta_{\max}^2}{4\ell}$$

Then, solving for the bending Moment at the center of the span gives:

$$M = F\delta_{\max} = \sigma_{cr} A \delta_{\max} = \frac{\sigma_{cr} A}{\pi} \sqrt{4\ell \Delta\ell} \quad \dots (48)$$

which is the equation used in Appendix D.

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NOMENCLATURE

A = Area
 C = Distance from neutral axis to outermost fiber, or constant
 C_p = Specific heat
 D = Diameter
 E = Weld efficiency, Modulus of elasticity
 F = Force
 f = Frequency
 G = Mass flowrate per unit area
 g = Gravitational acceleration
 h = Heat transfer coefficient, Parameter defined in Ref. 24
 I = Moment of inertia
 i = Stress concentration factor
 K = Defined in Ref. 25
 k = Thermal conductivity
 l = Length
 M = Moment
 MS = Margin of safety
 N = Allowable cycles
 n = Applied cycles
 P = Perimeter, Pressure
 Q = Heat
 q = Heat flux

R = Temperature ramp, Thermal resistance
 r = Radius, Radius of gyration
 S = Allowable stress, Arc length
 T = Torsional moment
 t = Temperature, thickness
 U = Overall heat transfer coefficient
 V = Volume
 v = Velocity
 W = Concentrated load, or weight
 w = Flowrate, Weight per unit length
 X = Axial distance
 Z = Section modulus
 α = Time constant, Coefficient of expansion, Angle
 Γ = Dimensionless parameter
 δ = Deflection
 ϵ = Strain
 θ = Time
 μ = Viscosity, Poissons ratio, coefficient of friction
 ρ = Density
 σ = Calculated stress
 Ω = Dimensionless parameter

Subscripts

b = Bending
 c = Circumferential
 cr = Critical
 E = Expansion
 e = Equivalent
 ff = Fouling factor
 i = Inlet, Inside
 l = Longitudinal
 m = Mean
 n = Sodium, Natural
 o = Outlet, outside
 s = Shell, Solids
 ss = Steady state
 T = Torsion
 t = Tubes
 w = Water or steam
 alt = Alternating
 eff = Effective
 min = Minimum
 max = Maximum
 sat = Saturated
 ult = Ultimate

Superscripts

$'$ = Modified value