CONTRACTOR REPORT

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Molten Salt Steam Generator Subsystem Research Experiment Phase 1—Specification and Preliminary Design Final Report Volume I—Technical

Babcock & Wilcox Company
Barberton, Ohio

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MOLTEN SALT STEAM GENERATOR

SUBSYSTEM RESEARCH EXPERIMENT

Phase I: Specification and Preliminary Design

Vol. 1 - Technical

FINAL REPORT

Prepared for: Sandia National Laboratories
Livermore, California

Prepared by: The Babcock & Wilcox Company

Nuclear Equipment Division
Barberton, Ohio

and

Contract Research Division
Alliance, Ohio

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Abstract

A study was completed in which steam generator subsystem and component designs were developed for central receiver solar power applications using molten nitrate salt as the primary heat transfer medium. Designs were established for a 100 MWe stand-alone plant and for a 100 MWe fossil-fueled plant which has been 50 percent repowered by solar energy. In the course of this progam, (1) an optimum steam system arrangement was selected for both the stand-alone and repowering applications; (2) cost-effective heat exchanger designs (preheater, evaporator, superheater, and reheater) were established based on conventional fabrication processes; (3) comprehensive subsystem and component specifications were prepared; (4) a control system was designed and characterized, and the system response to selected upset transients was simulated; (5) shop fabrication and field erection plans, schedules, and cost estimates were developed; and (6) development plans intended to resolve design uncertainties and assure user confidence and acceptance were prepared. An executive summary of this report is available from TIC as SAND84-8177.

TABLE OF CONTENTS

Table of Contents (continued)

Table of Contents (continued)

LIST OF FIGURES

List of Figures
Continued

List of Figures
Continued

LIST OF TABLES

EXECUTIVE SUMMARY

The Babcock & Wilcox Company, under contract to Sandia National Laboratories, has completed a study to develop steam generator subsystem and component designs for central receiver solar power applications using molten nitrate salt as the primary heat transfer medium. Subcontractor support was provided by Martin Marietta Corporation, Black & Veatch Consulting Engineers, and the Arizona Public Service Company.

The principal objectives of the program were:

(a) to select an optimum steam system arrangement for both stand-alone and repowering applications

(b) to establish cost-effective heat exchanger designs based on conventional fabrication processes

(c) to prepare comprehensive subsystem and component specifications

(d) to develop shop fabrication and field erection plans, schedules, and cost estimates

(e) to prepare development plans intended to resolve design uncertainties and assure user confidence and acceptance

The subsystem boundary conditions upon which the designs are based are summarized in Table 1.

STEAM GENERATOR CONCEPT SELECTION

A parametric evaluation of steam system arrangements and heat exchanger designs considered most appropriate for solar power service was completed. In the following paragraphs, the candidate designs are described, and the results of pertinent trade studies are presented.

Steam System - Recirculation, once-through (or Bensen), and spill-over (or Sulzer) steam systems were considered. The principal selection criteria were overall cycle efficiency, total plant and component cost, operational requirements and control system complexity, and water chemistry and balance-of-plant material limitations. Based on this evaluation, the recirculation system was chosen as the preferred arrangement for both stand-alone and repowering applications. The recirculation steam system schematic is shown in Figure 1.

TABLE 1

Steam Generator Subsystem Boundary Conditions

	Stand-Alone (100MWe)	Repowering (50Mwe)
Thermal Rating - MWt	264.2	132.2
Salt Inlet Temp - ^{o}C (^{o}F)	566(1050)	566(1050)
Salt Outlet Temp - ^{o}C (^{o}F)	288(550)	288(550)
Salt Flow Rate - kg/sec (lb/hrx10^{-6})	603(4.78)	301(2.39)
Steam Throttle Temp - ^{o}C(^{o}F)	538(1000)	538(1000)
Steam Throttle Pres - MPa(psia)	12.5(1815)	12.5(1815)
Main Steam Flow Rate - kg/sec (lb/hr x 10^{-6})	96.3(.764)	48.2(.382)
Cold Reheat Stm Temp - ^{o}C(^{o}F)	371(700)	371(700)
Hot Reheat Stm Temp - ^{o}C(^{o}F)	538(1000)	538(1000)
Hot Reheat Stm Pres - MPa(psia)	3.4(500)	3.4(500)
Reheat Steam Flow Rate - kg/sec (lb/hr x10^{-6})	83.4(.661)	41.6(.330)
Saturation Pressure - MPa(psia)	13.8(2000)	13.8(2000)
Feedwater Temp - ^{o}C (^{o}F)	238(460)	238(460)

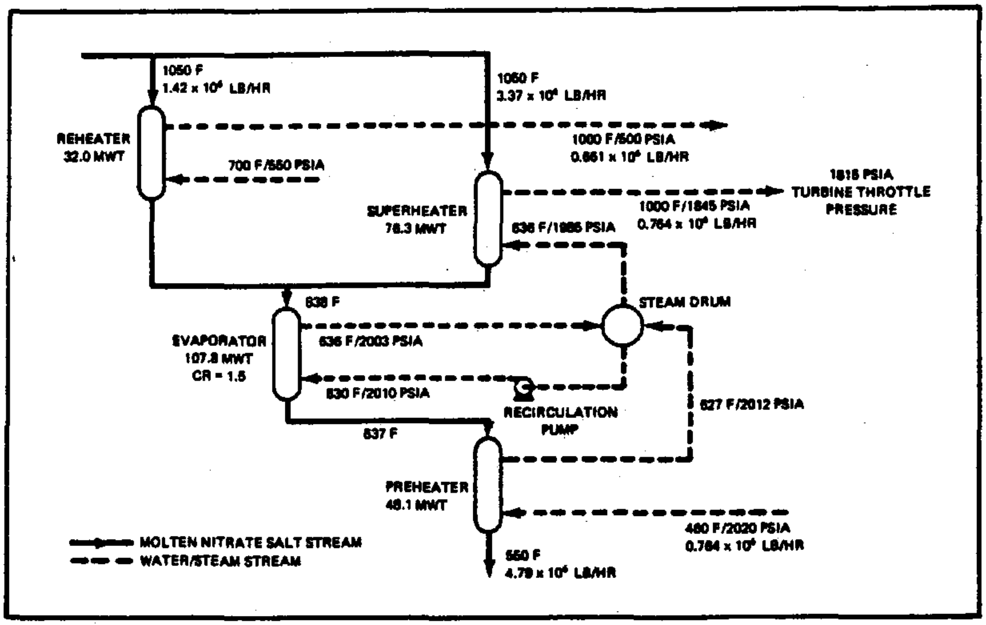


FIG. 1 RECIRCULATION STEAM SYSTEM-PROCESS FLOW SCHEMATIC

The recirculation boiler system is familiar to most users. It is used throughout the power industry for peaking service and other applications where frequent startups and load swings must be accommodated. It is thus uniquely suited to the diurnal cyclic service required in solar power applications.

A once-through system requires high-purity feedwater to avoid deposition of contaminants on evaporator surfaces. Achievement of the necessary water quality at candidate repowering sites would be prohibitively expensive as it would require substantial outlays for new water treatment equipment and high-alloy feedwater heaters. This system is unattractive for both stand-alone and repowering applications, even should adequate condensate polishing equipment be in place, because considerable time would be required to return the feedwater to once-through quality prior to each daily startup.

Inherent in the Sulzer system are high blowdown rates leading to large unrecoverable heat losses. The resultant poor cycle efficiency leads to unacceptable collector and receiver subsystem cost increases.

Heat Exchangers - Numerous heat exchanger configurations, including U-tube, straight-tube, helical coil, and serpentine tube bundles were examined for application in the recirculation steam system. These candidate designs were compared on the basis of performance characteristics, structural integrity and capability for withstanding operational transients, component cost and fabricability, reliability, and maintainability.

Based on this investigation, a horizontal U-tube bundle housed in a straight shell was chosen for the preheater and evaporator, and a U-tube bundle housed in a U-shell for the superheater and reheater. The heat exchanger designs are shown in Figure 2.

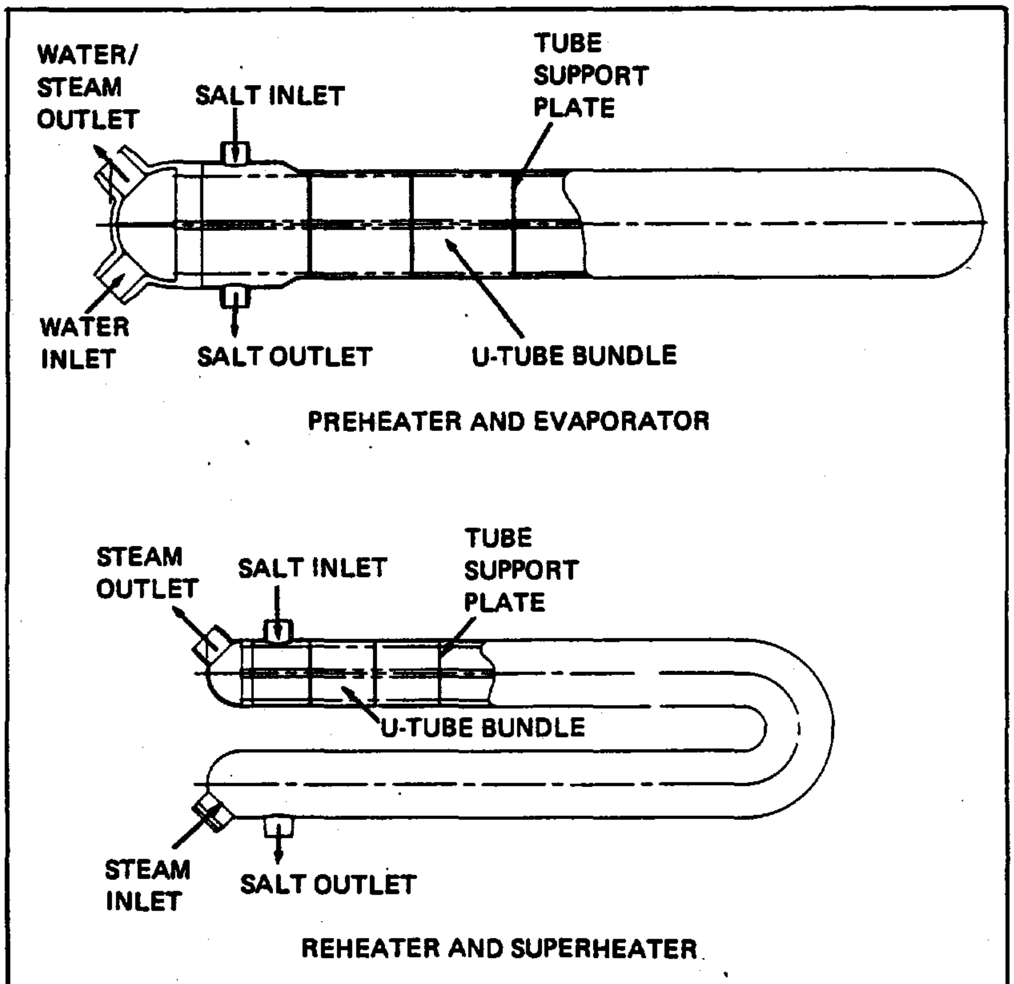


FIG. 2 STEAM GENERATOR COMPONENTS

Both arrangements offer compact tube bundles with efficient counterflow heat transfer. The inherent flexibility of the U-tubes readily accommodates the tube/shell and tube/tube differential thermal expansion characteristic of high-temperature, cyclic solar service. The straight shell configuration is a relatively low-cost assembly used reliably in many applications. However, it is not a suitable arrangement for the superheater and reheater in which the inlet-to-outlet tubeside terminal temperature difference is large, and in which unacceptable thermal stresses will be developed unless the tubesheets are independent. Thus, a U-shell assembly is preferred for these components, although it does present fabrication cost penalties associated with the complex shell closure weld in the bend region.

Straight-tube arrangements also offer compact tube bundles and efficient counterflow heat transfer. The cost of fabrication is typically low. However, when straight tubes are pinned between fixed tubesheets, thermal stresses produced by tube/shell and tube/tube differential thermal expansion are often unacceptable, particularly in high-temperature, cyclic service.

Use of an expansion bellows within the pressure boundary structure is one means of absorbing thermal motion. However, a bellows assembly is considered undesirable because it (a) requires rigorous analytical and/or experimental qualification, (b) requires lateral support, and (c) demands periodic in-service inspection and possibly replacement. It fails to alleviate thermal stresses produced by tube/tube differential expansion, which may occur in superheaters, reheaters, or heat exchangers having a significant amount of surface in subcooled heat transfer. Expansion bends within straight tubes, such as sine-wave bends, may also be used to absorb thermal motion. However, the complexity of the tubing process and tube support arrangement serves to substantially increase fabrication cost.

Helical coil and serpentine tube bundle configurations are superior technical designs, but not cost-effective for solar power service. Their major shortcomings are (a) high fabrication cost associated with the complex tube bundle and tube support assembly, and (b) a high material weight-to performance ratio for applications in which plant thermal ratings are small.

STEAM GENERATOR DESIGN

Hot molten salt, the primary heat source, is delivered to the steam generator from the high-temperature thermal storage tank. The stream is split and apportioned between the superheater and reheater, re-mixed, and delivered in sequence to the evaporator and preheater. The cold salt is then returned to the thermal storage subsystem.

Feedwater is preheated to near saturation temperature and delivered to the steam drum. In the drum, it is mixed with recirculated water and pumped to the evaporator, where a high-quality steam/water mixture is produced. This steam/water mixture is returned to the drum, and the steam and water phases are separated. The saturated steam is then superheated and sent to the high-pressure turbine. A portion of the high-pressure turbine exhaust steam is reheated for expansion in the intermediate and low-pressure stages of the turbine.

The principal geometric characteristics of the steam generator components for the stand-alone (100 MWe) and repowering (50 MWe) applications are summarized in Tables 2 and 3. Material selection and important aspects of the thermal-hydraulic and structural design are discussed in the following paragraphs.

Material Selection - Component materials were selected to provide adequate strength and corrosion resistance in the operating environments. These materials are summarized below.

Component Materials

Component	Material	Maximum Operating Temperature $^oC(^oF)$	
		Salt-Side	Water-Side
Preheater	Carbon Steel	336(637)	330(627)
Evaporator	2 1/4 Cr-1 Mo	448(838)	336(636)
Superheater	304 Stainless Steel	566(1050)	538(1000)
Reheater	304 Stainless Steel	566(1050)	538(1000)
Steam Drum	Carbon Steel	-	336(636)

TABLE 2

Principal Geometric Characteristics

100 MWe Components

Component	Main Shell O.D. m(in.)	Total Length m(ft)	Tube Dimensions O.D. X Wall mm(in)	No. Of Tubes	Avg. Active Tube Length m(ft)	Primary Surface Area $m^2(ft^2)$
Preheater	1.422 (56.0)	10.448 (34.28)	12.700 X 1.473 (.500 X .058)	2461	18.288 (60.0)	1803 (19410)
Evaporator	1.588 (62.5)	15.225 (49.95)	22.225 X 3.759 (.875 X .148)	1236	27.218 (89.3)	2349 (25280)
Superheater	0.610 (24.0)	7.373 (24.19)	12.700 X 1.651 (.500 X .065)	774	13.868 (45.5)	428 (4610)
Reheater	0.762 (30.0)	8.251 (27.07)	15.875 X .889 (.625 X .035)	870	15.027 (49.3)	652 (7020)

TABLE 3

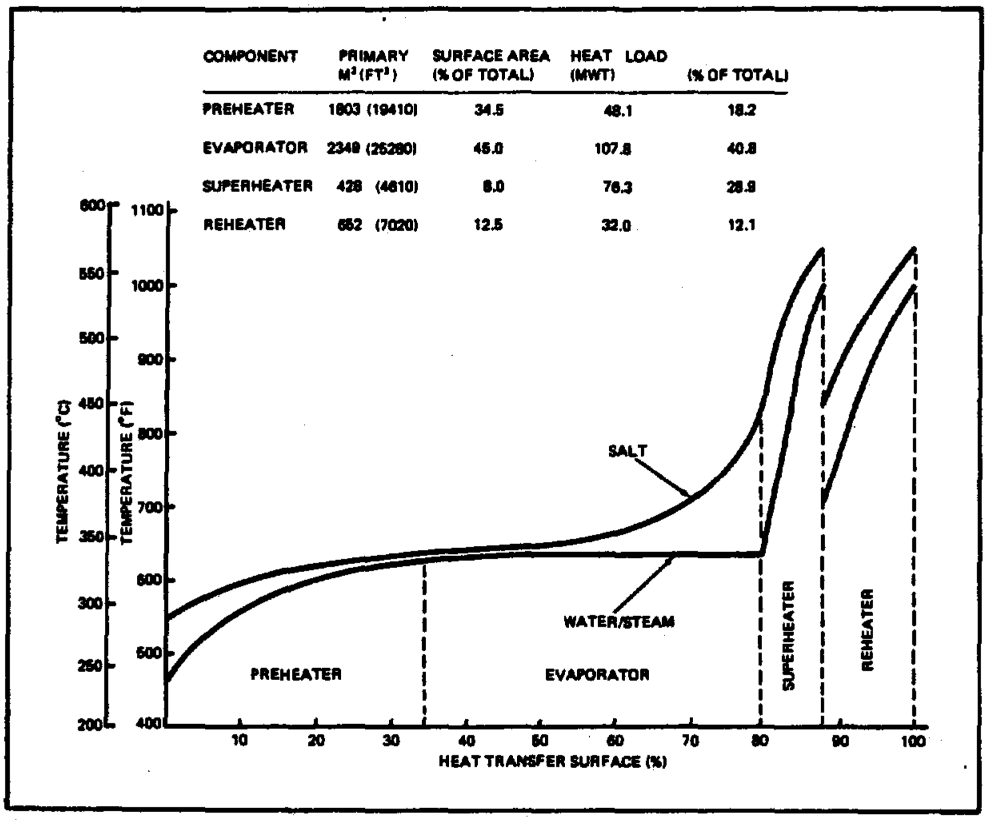
Principal Geometric Characteristics

50 MWe Components

Component	Main Shell O.D. m(in.)	Total Length m(ft)	Tube Dimensions O.D. X Wall mm(in)	No. Of Tubes	Avg. Active Tube Length m(ft)	Primary Surface Area $m^2(ft^2)$
Preheater	1.257 (49.5)	9.836 (32.27)	12.700 X 1.473 (.500 X .058)	1710	17.404 (57.1)	1187 (12780)
Evaporator	1.308 (51.5)	15.057 (49.40)	22.225 X 3.759 (.875 X .148)	775	27.157 (89.1)	1470 (15820)
Superheater	.457 (18.0)	7.705 (25.28)	12.700 X 1.651 (.500 X .065)	396	14.417 (47.3)	228 (2452)
Reheater	.560 (22.0)	8.019 (26.3)	15.875 X .889 (.625 X .035)	442	15.301 (50.2)	337 (3630)

Corrosion allowances were determined based on best available data. However, reported data pertaining to the corrosion resistance of the selected alloys, particularly the low-chromium alloys, in molten nitrate salt is very limited and often exhibits wide scatter. Thus, it will be prudent to re-assess these material choices based on the results of on-going and future test programs. The need for further corrosion testing to support design efforts is discussed later.

<u>Thermal-Hydraulic Design</u> - Predicted fluid temperature profiles throughout the steam generator subsystem, are shown in Figure 3. Design features intended to preclude DNB (departure from nucleate boiling) in the evaporator and assure proper flow distribution in all components are described in the following paragraphs.



COMPONENT	PRIMARY M^2 (FT^2)	SURFACE AREA (% OF TOTAL)	HEAT LOAD (MWT)	(% OF TOTAL)
PREHEATER	1803 (19410)	34.5	48.1	18.2
EVAPORATOR	2349 (25280)	45.0	107.8	40.8
SUPERHEATER	428 (4610)	8.0	76.3	28.9
REHEATER	652 (7020)	12.5	32.0	12.1

FIG. 3 FLUID TEMPERATURE PROFILES

- Ribbed tubing has been used for the final 32 feet of the upper, or steam/water outlet, leg of the evaporator. This represents the zone of highest heat flux and quality, and is consequently most susceptible to DNB and associated under-deposit corrosion which may occur when DNB exists in the presence of porous water-side deposits.

  The ribbed tube construction produces a swirling flow and centrifugal forces that keep the tube surface wetted and maintain nucleate boiling to a higher quality for a given pressure, heat flux, and mass flow rate than with a smooth-bore tube of the same general dimensions. Thus, nucleate boiling is maintained in all circuits without the high circulation rates that lead to increased pumping power requirements and reduced cycle efficiency.

- Since the thermal conductivity of the molten salt is relatively low, the overall resistance to heat transfer is dominated by the shell-side heat transfer resistance. Therefore, salt velocity has been maximized to promote efficient heat transfer, but within pressure drop constraints and within limits necessary to preclude tube vibration.

  Salt inlet flow distribution baffles control tube impingement velocities and limit excitation frequencies. Preferentially broached tube support plates promote crossflow to enhance heat transfer and inhibit salt stratification.

Structural Design - All component pressure boundaries were sized in accordance with Section VIII, Division 1, of the ASME code. As a supplement to this work, thermal-mechanical analyses, using Section VIII, Division 2, as a guide, and elevated temperature evaluations derived from Code Case N-47 were selectively made to assure the structural adequacy of the vessels at critical locations. The analytical work is summarized in Table 4. All calculated stresses were found to be within allowable limits.

TABLE 4

Structural Design Evaluation

Mechanical Stress

- Pressure boundary code calculations were made to establish vessel wall thicknesses.
- Pressure discontinuity analyses were made to assure acceptable tubesheet ligament and shell-tubesheet-head discontinuity stress levels.
- Dead weight and seismic loads were considered to design the vessel saddle supports.
- Allowable piping loads were determined.

Thermal Stress

- Finite element analyses were made to determine the response of the evaporator shell and tubes to imposed longitudinal thermal gradients. It was found that the shell deflections must be restrained to prevent interference with tube bundle growth.
- A finite element analysis of the preheater tubesheet was made to determine secondary stresses resulting from the divider lane temperature gradient.
- Discontinuity stresses resulting from longitudinal thermal gradients in the shell-tubesheet-head junctures of all components were determined.
- Fatigue analyses of the preheater tubesheet ligaments and evaporator shell-tubesheet juncture were made.

Elevated Temperature Effects

- Creep-ratcheting evaluations were made for the superheater and reheater steam-side heads, tubes, and salt-side shell (upper leg).
- Creep-fatigue interaction evaluations were made for the superheater and reheater tubesheet ligaments and head-tubesheet-shell juncture (upper leg).
- Creep-buckling evaluations were made for the superheater and reheater U-bend support region (upper leg).

OPERATION AND CONTROL

The controls for the steam generator subsystem are based on conventional boiler operating practice and provide the following functions required of any well-designed boiler control system:

(a) The superheater steam outlet pressure is controlled by adjustment of molten salt flow rate (analogous to fuel input in conventional practice).

(b) The supply of feedwater is controlled on average to match steam flow and also regulated to maintain a pre-determined steam drum water level.

(c) Final steam temperature is maintained within prescribed limits through attemperation with saturated steam.

(d) Flexibility of operation is provided by including operator automatic-manual transfer stations at appropriate points in the various control loops.

(e) Protection against equipment damage is provided by activation of functional operations which limit temperatures when an established criteria is reached.

A computer simulation was completed to examine the dynamic response of the system to selected normal and upset transients. The events which were analyzed are summarized in Table 5.

TABLE 5

Normal and Transient Events

100-80-100% Load Swing
100-60-100% Load Swing
100-30-100% Load Swing
Turbine Trip
One Recirculation Pump Tripped
Both Recirculaton Pumps Tripped
Feedwater Valve Failed Open
Feedwater Valve Failed Closed
Two Relief Valves Failed Open
One Relief Valve Failed Open
Superheater Salt Valve Closure
Hot Salt Pump Trip
Feedwater Pump Trip
Reheater Salt Valve Closure

The results of the analyses were used to assess the proposed control system, to refine control gain settings, and to provide corroboration of previously predicted steady-state plant performance. In no case were unacceptable thermal loads imposed on the components.

BALANCE OF SUBSYSTEM

The physical arrangement of the major components in the stand-alone (100 MWe) plant is shown in Figure 4. The heat exchangers and steam drum are incorporated into a typical turbine cycle feedwater heater bay that would be associated with a new plant design. The bay is an open space frame structure providing support for the equipment, piping, and platforms. The equipment has been arranged on four levels to accommodate cascade draining of the salt through the heat exchangers into the sump which is located below grade.

The most economical arrangement for the repowering (50 MWe) plant was found to be an outdoor, slab-mounted design with all components except the steam drum located on a concrete pad at grade level. The drum need be elevated only high enough to provide adequate net positive suction head for the recirculation pumps. The salt sump is located adjacent to the slab and below grade to facilitate draining the salt from the heat exchangers and piping. It is clear that the ability to physically lay out the components at grade level is dependent on the choice of forced circulation as the basic operating mode.

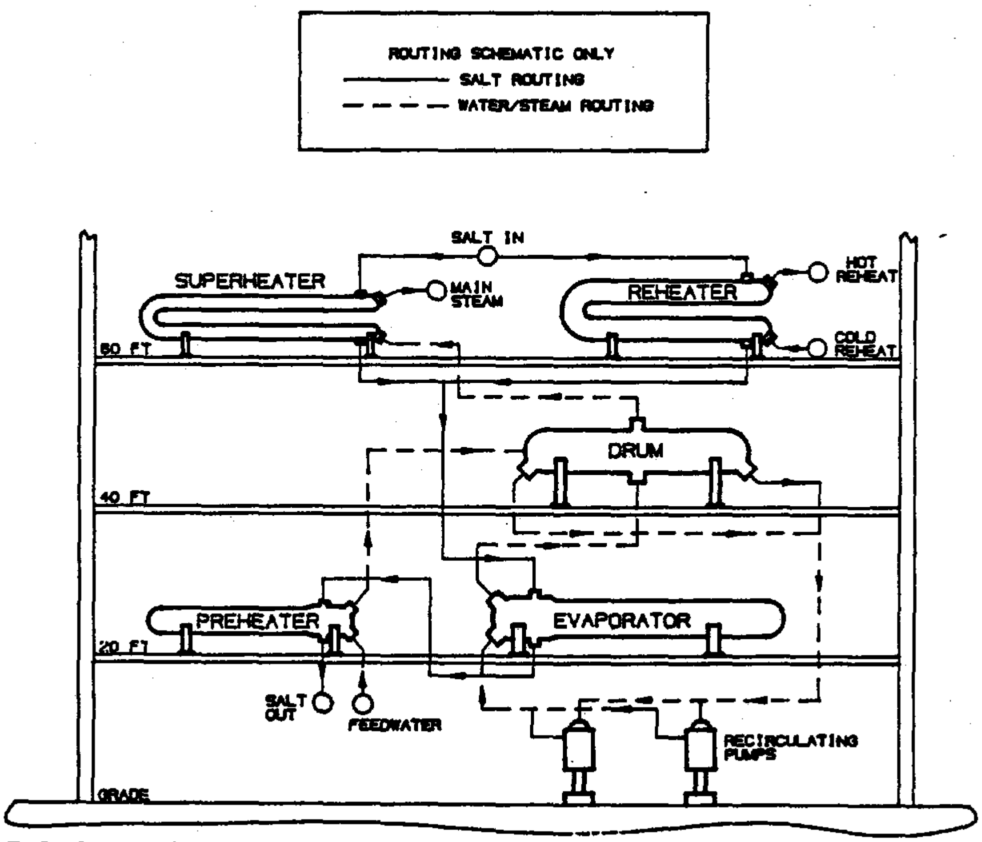


FIG. 4 LAYOUT OF STEAM GENERATOR COMPONENTS IN TURBINE BUILDING

FABRICATION AND FIELD ERECTION PLAN AND COST ESTIMATE

Plans, schedules, and cost estimates were developed for shop fabrication and field erection of the steam generator subsystem. Both a 100 MWe stand-alone plant and a 50 MWe repowered plant were considered.

The proposed construction methods are based on proven, qualified techniques and present no unusual risks or uncertainties. An integrated design, shop fabrication, and field erection schedule, applicable to both the 100 MWe and 50 MWe plants, is shown in Figure 5. Cost estimates are presented in Tables 6 and 7 and have been prepared using standards data, actual costs escalated from previous contracts, material vendor quotations, and catalog prices. Costs are expressed in current dollars (May, 1982).

DEVELOPMENT PLAN

A development program has been proposed with the principal objectives of (a) demonstrating to potential users, through subscale modeling (SRE), the design adequacy and operational capability of the steam generator subsystem, and (b) resolving, through SRE and laboratory testing, design and performance uncertainties associated with the full-scale system. The development goals are summarized in the following paragraphs.

Performance Tests - Performance tests have been proposed to verify the analytical predictions upon which the heat exchanger designs are based. Specifically, the accuracy of the salt-side heat transfer correlations would be assessed and the specific heat of the molten nitrate salt would be determined from appropriate flow and temperature distribution measurements and heat balances. The data would then be compared with the assumptions used in the design analyses.

DNB Tests - DNB tests have been proposed to verify that the circulation ratio selected is sufficiently high to maintain nucleate boiling throughout the evaporator. Based on our review of available data, it is believed that the circulation ratio has been chosen conservatively. Accepting that this will be confirmed, the design margin would then be re-assessed to determine if the circulation rate can be reduced. Reduction of the circulation rate would result in some corresponding reducton in heat transfer surface and savings in pump costs.

Corrosion Tests - Corrosion tests have been proposed to better define the salt-side corrosion allowances to be applied to the heat exchanger tubing. If the test data compare favorably with analytical assumptions, no design adjustments would be required. If the data deviate significantly from the analytical assumptions, it would be necessary to re-size the tube bundles of the affected components. Assuming that the original assumptions were conservative, re-sizing would involve a reduction of heat transfer surface.

TABLE 6

Cost Estimate for 100 MWe Steam Generator Subsystem

Steam Generator		
Engineering		$ 689,000
Shop Fabrication		
Preheater		1,393,000
Evaporator		1,504,000
Superheater		756,000
Reheater		786,000
Steam Drum		412,000
	Sub-Total	$ 5,540,000
Controls and Instrumentation		$ 415,000
Balance of Subsystem		
Piping		$ 693,000
Valves		577,000
Pumps		317,000
Heat Exchanger and Steam Drum Erection		510,000
Electrical Equipment		254,000
Building/Structure		395,000
Other (insulation, spring hangers, yard pipe supports, and sump)		255,000
	Sub-Total	$ 3,001,000
Construction and Procurement		$ 1,611,000
(field costs, engineering and procurement, and construction services and management)		
	Grand Total	$ 10,567,000

TABLE 7

Cost Estimate for 50 MWe Steam Generator Subsystem

Steam Generator		
Engineering		$ 689,000
Shop Fabrication		
Preheater		1,045,000
Evaporator		1,046,000
Superheater		502,000
Reheater		597,000
Steam Drum		280,000
	Sub-Total	$ 4,159,000
Controls and Instrumentation		$ 415,000
Balance of Subsystem		
Piping		$ 293,000
Valves		427,000
Pumps		293,000
Heat Exchanger and Steam Drum Erection		365,000
Electrical Equipment		242,000
Building/Structure		118,000
Other (insulation, spring hangers, yard pipe supports, and sump)		171,000
	Sub-Total	$ 1,909,000
Construction and Procurement		$ 1,123,000
(field costs, engineering and procurement, and construction services and management)		
	Grand Total	$ 7,606,000

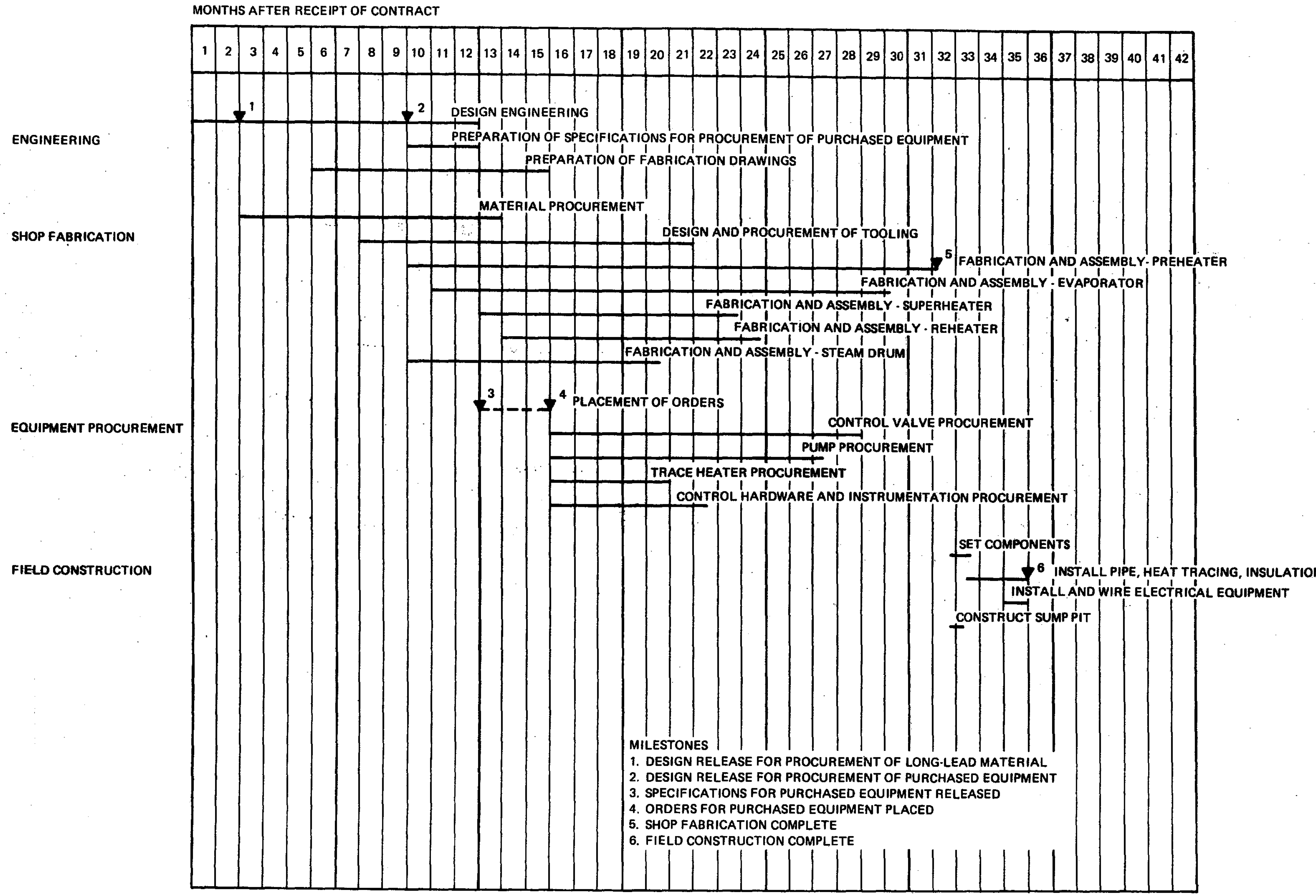


FIG. 5 DESIGN, FABRICATION AND FIELD ERECTION SCHEDULE

1.0 INTRODUCTION

1.1 Purpose of Study

This report describes the results of a study conducted by the Babcock & Wilcox Company, under contract to Sandia National Laboratories (SNLL), to develop steam generator subsystem and component designs for central receiver solar power applications using molten nitrate salt as the heat transfer medium. The principal objectives of the program were:

(a) to select an optimum steam system arrangement for both stand-alone and repowering applications

(b) to establish cost-effective heat exchanger designs based on conventional fabrication processes

(c) to prepare comprehensive subsystem and component specifications

(d) to develop shop fabrication and field erection plans, schedules, and cost estimates

(e) to prepare development plans intended to resolve design uncertainties and assure user confidence and acceptance

Steam generator subsystem (SGS) designs were developed for a 100 MWe stand-alone plant and for a 100 MWe fossil-fueled plant which has been 50 percent repowered by solar energy. The arrangement of the subsystem within the balance of plant is shown for each application in Figures 1-1 and 1-2.

1.2 Technical Approach

During the course of the study, subsystem and component specifications were developed; performance, structural, and control system analyses were completed; and construction plans and cost estimates were prepared. So that the work would proceed in an orderly manner, the effort was organized into the following nine tasks:

Task 1 - Review of SGS Definition and Interface Requirements

System interface requirements were reviewed and integrated in the design specifications. A component list was prepared, and applicable codes and standards were compiled.

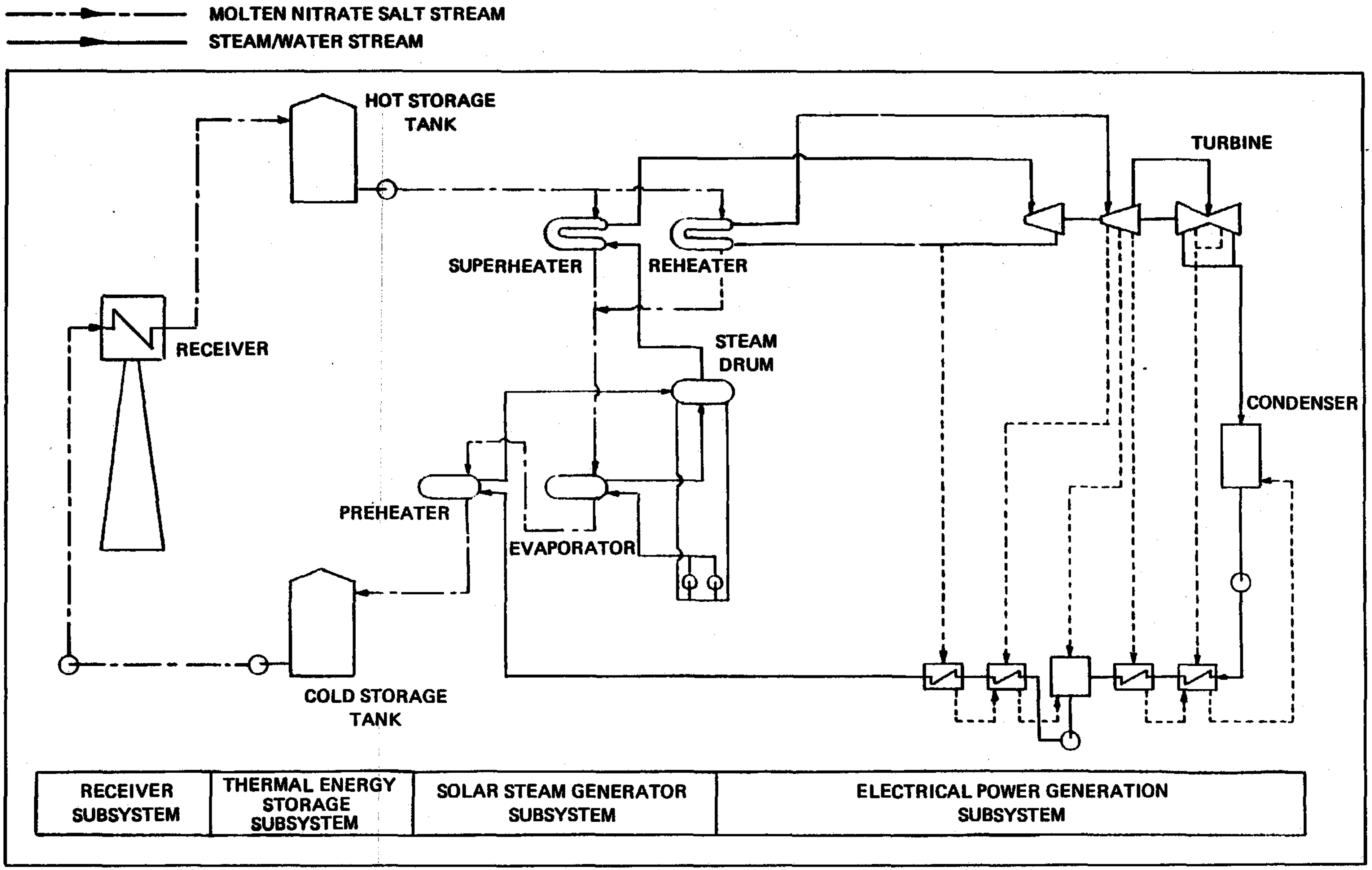


FIG. 1-1 SOLAR STAND-ALONE PLANT

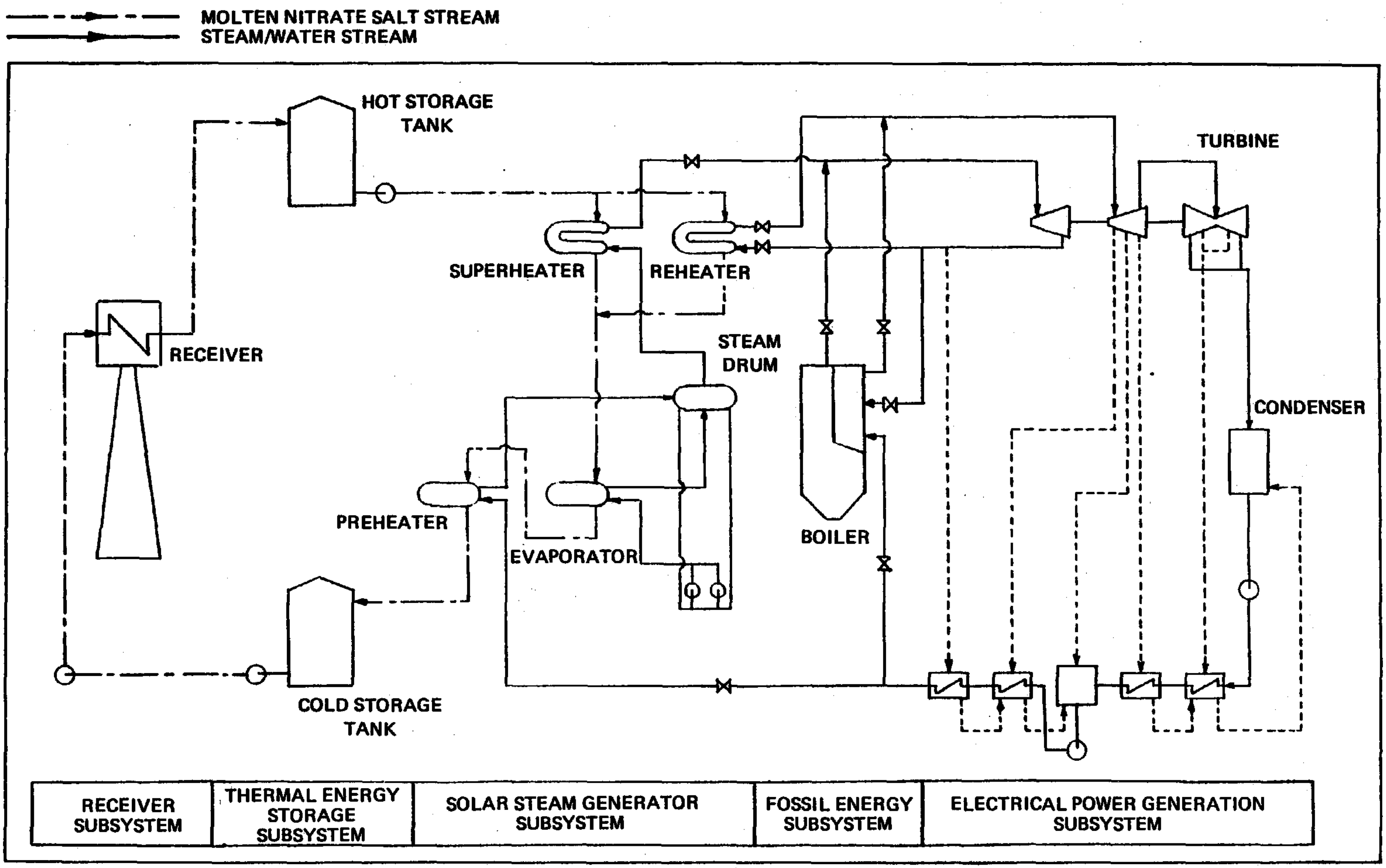


FIG. 1-2 SOLAR REPOWERED PLANT

Task 2 - Definition of SGS Requirements

The subsystem and component specifications document was prepared. Included in this document were requirements for design, a description of normal operating modes, definition of upset and emergency transients, and requirements for fabrication and maintenance. This document was revised periodically as the study proceeded.

Task 3 - SGS Concept Selection

A parametric evaluation of candidate steam system arrangements and heat exchanger designs was completed. Included in the evaluation criteria were:

(a) subsystem and component performance characteristics
(b) component structural integrity for high-temperature, cyclic solar service
(c) capital and operating cost
(d) reliability and serviceability
(e) development needs

Based on this evaluation, a forced recirculation system was selected for both stand-alone and repowering applications. Horizontal U-tube/straight shell heat exchanger configurations were chosen for the preheater and evaporator, and horizontal U-tube/U-shell configurations for the superheater and reheater.

Task 4 - SGS Design

A Design Analysis Plan was prepared, and detailed design of the subsystem and component concepts selected during Task 3 was completed in accordance with this plan. The design effort addressed:

(a) sizing of the heat transfer surface and thermal-hydraulic analysis of each heat exchanger component

(b) code analysis and supplementary thermal-mechanical evaluations of each heat exchanger component

(c) selection of material

(d) physical arrangement of the major subsystem components and associated piping layout

(e) design and characterization of the control system

(f) simulation of system response to selected upset transients

Drawings were prepared to illustrate important design features and to form the basis for fabrication plans and cost estimates.

Task 5 - SGS Cost and Fabrication Plan

Shop fabrication and field erection plans were prepared. Construction schedules and cost estimates were developed.

Task 6 - SGS Subsystem Research Experiment (SRE) and Development Plan

A development plan was prepared. The principal objectives of the recommended program would be (a) to demonstrate to potential users through subscale modeling the design adequacy and operational capability of the steam generator subsystem, and (b) to resolve through SRE and laboratory testing design and performance uncertainties associated with the full-scale system. The resolution of uncertainties would be directed to prudent and economic reduction of design margins.

Task 7 - Phase II Plan and Proposal

The development plans prepared during Task 6 were formalized in a Phase II proposal submitted for SNLL consideration.

Tasks 8 and 9 - Reports and Data and Program Management

Reports were prepared in accordance with SNLL requirements and to document significant accomplishments. The program was managed and controlled to assure satisfactory completion of all tasks in accordance with the program plan.

1.3 Project Organization

The Babcock & Wilcox Company was supported in the completion of this study by Martin Marietta Aerospace, Black & Veatch Consulting Engineers, and the Arizona Public Service Company. The specific responsibilities of each team member are summarized in Table 1-1.

TABLE 1-1

Project Organization

Babcock & Wilcox Company

- program management
- steam system and heat exchanger concept evaluation and selection
- design of major system components
- system simulation analysis
- preparation of shop fabrication plans and cost estimates
- definition of development plans

Martin Marietta Aerospace

- integration of molten salt technology
- preparation of specifications document
- control system design and analysis
- preparation of Phase II proposal

Black & Veatch Consulting Engineers

- physical arrangement of major system components
- piping layout
- preparation of field erection plans and cost estimates

Arizona Public Service Company

- determination and review of operation and maintenance requirements
- design review

2.0 STEAM GENERATOR CONCEPT SELECTION

A parametric evaluation of steam system arrangements and component designs considered most appropriate for solar power service was completed. Based on this evaluation, a forced recirculation system was selected for both stand-alone and repowering applications. Horizontal U-tube/straight shell heat exchanger configurations were chosen for the preheater and evaporator, and horizontal U-tube/U-shell configurations for the superheater and reheater. In the following paragraphs, the candidate designs are described, and the results of pertinent trade studies and other investigations are presented.

2.1 Steam System

The steam system arrangements considered the most promising candidates for solar power applications were the recirculation boiler, once-through (or Bensen) boiler, and a modification of the steam motive arrangement sometimes called the spill-over (or Sulzer) boiler. The principal selection criteria were overall cycle efficiency, total plant and component cost, operational requirements and control system complexity, and water chemistry and balance of plant material limitations. An overview of the selection process was provided by the Arizona Public Service Company (APS). The Southern California Edison, Pacific Gas and Electric, and Southwestern Public Service Companies were canvassed to supplement the APS review.

The essential advantages and disadvantages of the candidate steam systems are summarized in Table 2-1. Based on comparative evaluations, a recirculation boiler was selected as the preferred arrangement for both stand-alone and repowering applications. This type of system is used exclusively throughout the power industry where frequent startups and load swings must be accommodated. It is uniquely suited to the diurnal cyclic service required in solar power plants.

A brief description and assessment of each candidate steam system is provided in the following paragraphs:

Recirculation Steam System - The arrangement of the recirculation boiler system is shown in Figure 2-1. The characteristic feature of this system is a steam drum, where the water/steam mixture is separated. Separation of the phases assures that there is no "carryover" of water in the saturated steam delivered to the superheater, or "carryunder" of steam in the recirculated water directed to the downcomers.

	RECIRCULATING STEAM SYSTEM	ONCE-THROUGH STEAM SYSTEM	SULZER STEAM SYSTEM
CYCLE EFFICIENCY	SLIGHTLY LOWER THAN ONCE-THROUGH SYSTEM OWING TO PARASITIC POWER CONSUMED BY RECIRCULATION PUMPS AND ENERGY LOST IN BLOWDOWN PENALTY IS REDUCED IF NATURAL CIRCULATION IS EMPLOYED	TYPICALLY HIGHEST OF THE THREE SYSTEMS SOMEWHAT LIMITED BY PARASITIC POWER CONSUMED BY FEED PUMP (INHERENTLY HIGH PRESSURE LOSS SYSTEM)	ESTIMATED TO BE APPROXIMATELY TWO-PERCENT LOWER THAN OTHER SYSTEMS BECAUSE OF UNRECOVERABLE HEAT LOSSES IN LET-DOWN SYSTEM
OPERATION AND CONTROL	READILY ACCOMMODATES FREQUENT STARTUPS AND LOAD SWINGS TO FACILITATE DIURNAL CYCLING SIMPLE CONTROL SYSTEM FAMILIAR TO MOST USERS LARGE STEAM DRUM WATER INVENTORY TO SOFTEN IMPACT OF THERMAL TRANSIENTS	TIME-CONSUMING FEEDWATER CLEANUP REQUIRED TO ACHIEVE ACCEPTABLE WATER QUALITY PRIOR TO STARTUP MORE SENSITIVE CONTROL SYSTEM THAN REQUIRED FOR RECIRCULATING SYSTEM	OPERATIONAL CONSIDERATONS SIMILAR TO ONCE-THROUGH SYSTEM; HOWEVER, MORE COMPLEX CONTROL SYSTEM REQUIRED TO HANDLE IMPACT OF LOAD VARIATIONS ON LET-DOWN SYSTEM
WATER QUALITY REQUIREMENTS	LESS DEMANDING WATER QUALITY REQUIREMENTS THAN OTHER SYSTEMS WATER PURITY REQUIREMENTS GENERALLY CONSISTENT WITH CAPABILITY OF OLDER PLANTS WHICH MAY BE CANDIDATES FOR REPOWERING	HIGH-PURITY FEEDWATER REQUIRED; USUALLY ACHIEVED ONLY WITH FULL-FLOW CONDENSATE POLISHING AND HIGH-ALLOY FEEDWATER HEATERS	WATER QUALITY REQUIREMENTS SIMILAR TO ONCE-THROUGH SYSTEM (PARTICULARLY WITH HIGH EVAPORATOR EXIT QUALITY)
PLANT COST	SOMEWHAT HIGHER THAN OTHER SYSTEMS BECAUSE OF: . STEAM DRUM, DOWNCOMERS, AND RISERS . RECIRCULATING PUMP (IF FORCED CIRCULATION IS EMPLOYED) . GREATEST HEAT TRANSFER SURFACE IN EVAPORATOR (NECESSARY TO ACCOMMODATE RECIRCULATED BOILER WATER)	TYPICALLY LOWEST OF THE THREE SYSTEMS RESULTING FROM: . ABSENCE OF DRUM, DOWNCOMERS, AND RISERS . LEAST HEAT TRANSFER SURFACE IN EVAPORATOR COST SAVING LIMITED BY NEED FOR FULL-FLOW CONDENSATE POLISHING DEMINERALIZERS AND HIGH-ALLOY FEEDWATER HEATERS	SLIGHTLY HIGHER THAN ONCE-THROUGH SYSTEM OWING TO: . SMALL DRUM AND LET-DOWN SYSTEM . SOMEWHAT GREATER HEAT TRANSFER SURFACE IN EVAPORATOR

 TABLE 2-1 Comparison Of Steam System Characteristics

Thus, deposition and corrosion of heat transfer surfaces in the superheater is avoided, and adequate head is provided in the downcomers either for circulation (natural circulation mode) or to protect pumps from cavitation (forced circulation mode). The drum also provides a zone for feedwater to mix with recirculated water, assuring that chemical treatment additives are uniformly distributed before the mixed stream enters the heat absorbing zone of the evaporator.

The recirculation boiler is designed to preclude departure from nucleate boiling (DNB) throughout the evaporator. This can be accomplished by using high water recirculation rates. However, as the recirculation rate is increased, the thermal driving force is reduced, and the heat transfer surface area must be increased proportionally, thereby increasing capital costs. To partially overcome this drawback, ribbed tubes can be used in the evaporator surfaces. The swirling flow and centrifugal forces produced by the ribs keep the tube surface wetted and maintain nucleate boiling to a higher quality for a given pressure, heat flux, and mass velocity than with a smooth bore tube of the same general dimensions.

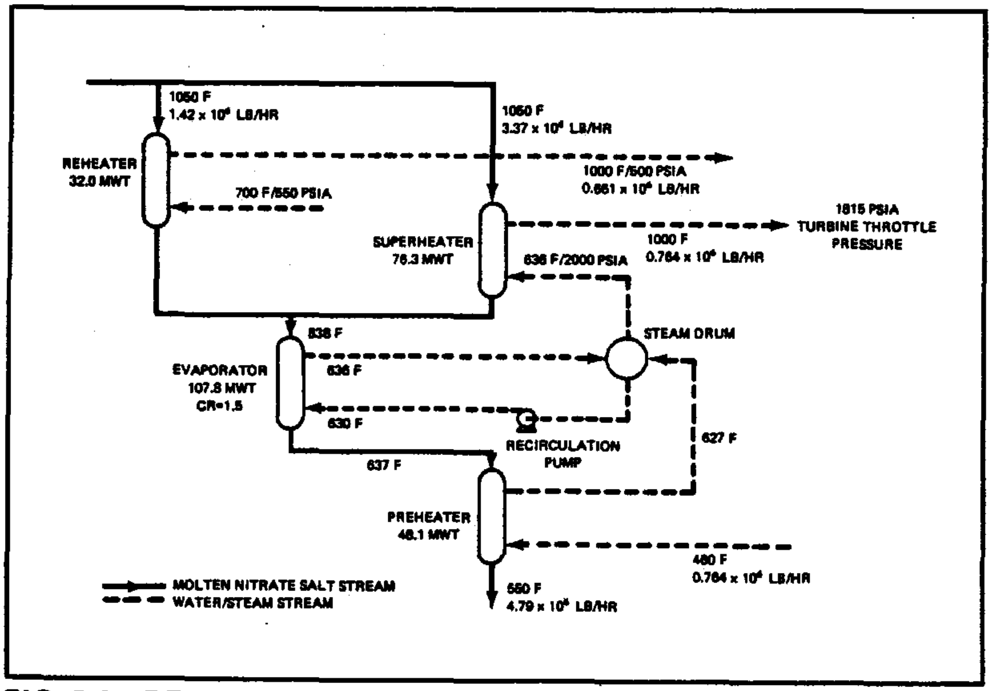


FIG. 2-1 RECIRCULATION STEAM SYSTEM

Because DNB is avoided in the evaporator, water quality requirements are not so demanding as for once-through systems. Water quality specifications can generally be maintained through down periods of short duration, and subsequent startup accomplished without time-consuming feedwater cleanup. For this reason, recirculation boilers are used exclusively throughout the power industry where frequent startups and load swings must be accommodated.

Once-Through Steam System - The arrangement of the once-through boiler system is shown in Figure 2-2. It was originally developed to reduce the capital costs associated with drums, downcomers, and risers required for recirculation boilers in conventional fossil power service. Because there is no recirculation of boiler water, the evaporator heat transfer surface area also can be reduced, and further capital cost savings achieved.

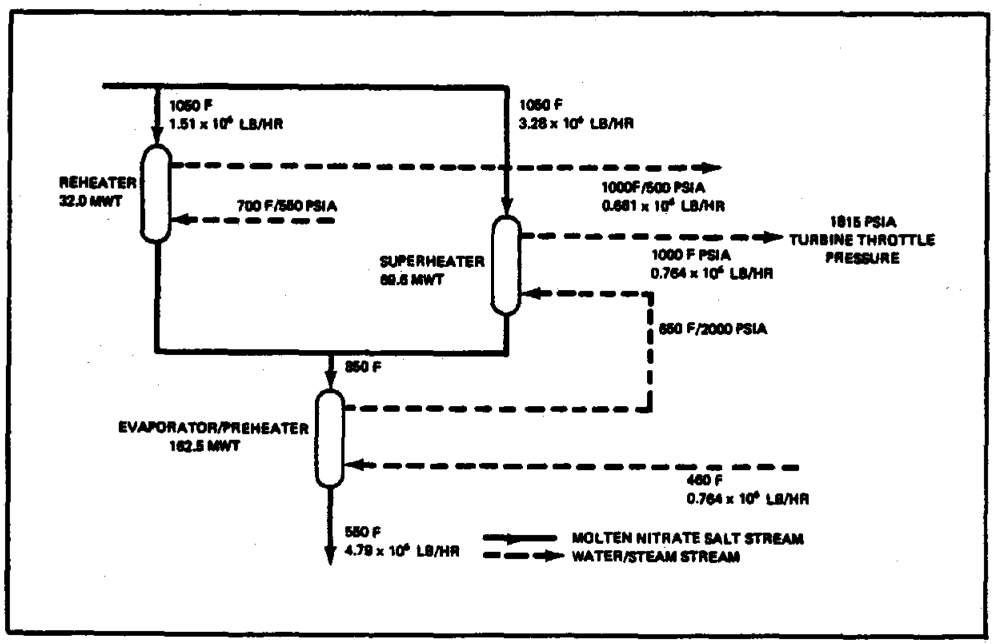


FIG. 2-2 ONCE-THROUGH STEAM SYSTEM

Because DNB occurs within the evaporator, feedwater of high purity must be provided to avoid deposition of contaminants and resultant corrosion of heat transfer surfaces. Achievement of the necessary water quality at candidate repowering sites would be prohibitively expensive as it would require substantial outlays for condensate polishing demineralizers and high-alloy feedwater heaters.

Should adequate water treatment equipment be in place, considerable time would be required to return the feedwater to once-through quality prior to each daily startup. This operational requirement is unattractive for both stand-alone and repowering applications, where the common objective is to maximize the time available for power production and minimize thermal energy storage requirements.

Sulzer Steam System - The arrangement of the Sulzer steam boiler system is shown in Figure 2-3. It is essentially a modified once-through system employing a small steam drum between the evaporator and superheater. The liquid effluent from the drum is usually let-down through recovery heat exchangers and returned to the system. Otherwise, the let-down effluent must be dumped with substantial heat loss.

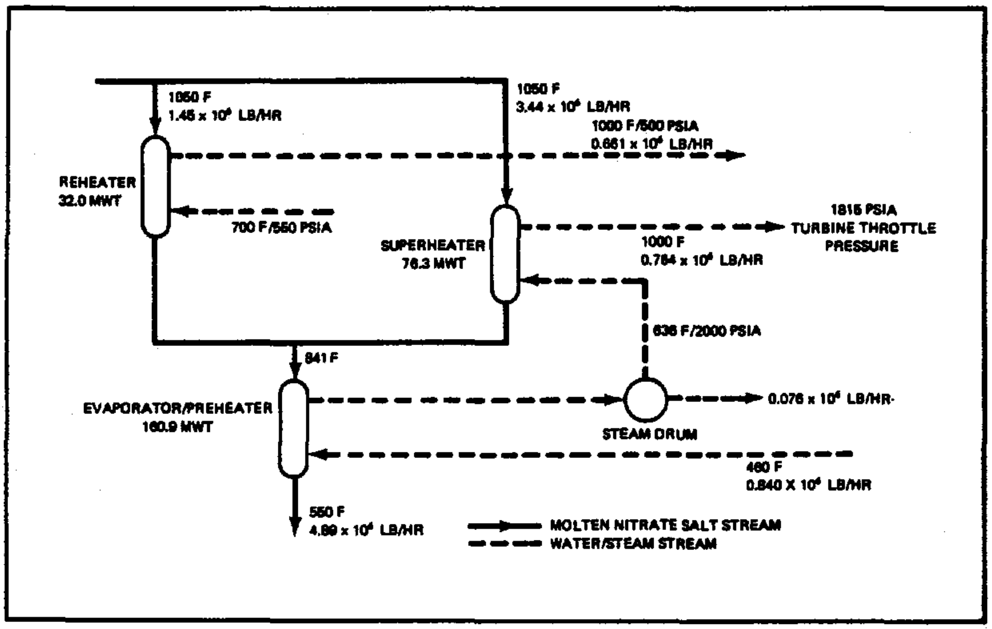


FIG. 2-3 SULZER STEAM SYSTEM

Depending on system operating parameters, such as pressure, mass velocity, and heat flux, DNB may or may not occur in the evaporator. However, to avoid DNB, exit steam quality must generally be low. It is clear that as the exit quality is decreased, and the amount of effluent correspondingly increases, greater demands are placed on the let-down system, and unrecoverable heat losses become prohibitively large. The resultant poor cycle efficiency leads to unacceptable collector and receiver subsystem cost increases.

2.2 Heat Exchangers

Numerous heat exchanger configurations, including U-tube, straight tube, helical coil tube, and serpentine tube bundles, were considered for application in the recirculation steam system. These candidate designs were compared on the basis of performance characteristics, structural integrity and capability for withstanding operational transients, component cost and fabricability, reliability, and maintainability. Although certain system considerations affecting the interface with collector/receiver/storage and electrical power generation subsystems make steam generator design for solar power service somewhat unique, fundamental principals developed through extensive design and operating experience in both fossil and nuclear power service apply to solar applications as well. These principals, considered essential in a steam generator design of high technical quality, are summarized in Table 2-2.

TABLE 2-2

Steam Generator Design Criteria

- Heating surfaces oriented to promote efficient heat transfer and hydraulic stability of the heating fluid and steam/water mixture
- Materials selected to provide adequate strength and corrosion/erosion resistance in the operating environment
- Uniform distribution of flow to all heating surfaces assured
- Sufficient flexibility provided between tubes, and between tubes and their supports, to preclude high stresses resulting from differential thermal expansion
- Excitation frequencies limited and tube supports arranged to prevent potential damage resulting from flow-induced and machinery-induced vibration
- Quality weld configurations and weld inspection standards provided to assure pressure boundary integrity
- Access provided for inspection and corrective maintenance
- The vessel capable of being fully drained and vented

The relative merits of the candidate heat exchanger concepts are summarized in Table 2-3. For the more promising design concepts, sizing calculations were made to provide a more quantitative assessment of the design variables most heavily influencing heat transfer surface requirements, tube bundle configuration, and overall vessel weight. These design variables are described in Table 2-4. Component fabrication costs are compared in Figure 2-4.

TABLE 2-4

Steam Generator Design Variables

- Tube outside diameter
- Tube length
- Tube pitch
- Tube-side pressure drop
- Shell-side velocity and pressure drop
- Shell-side flow direction (crossflow or longitudinal flow)
- Circulation ratio

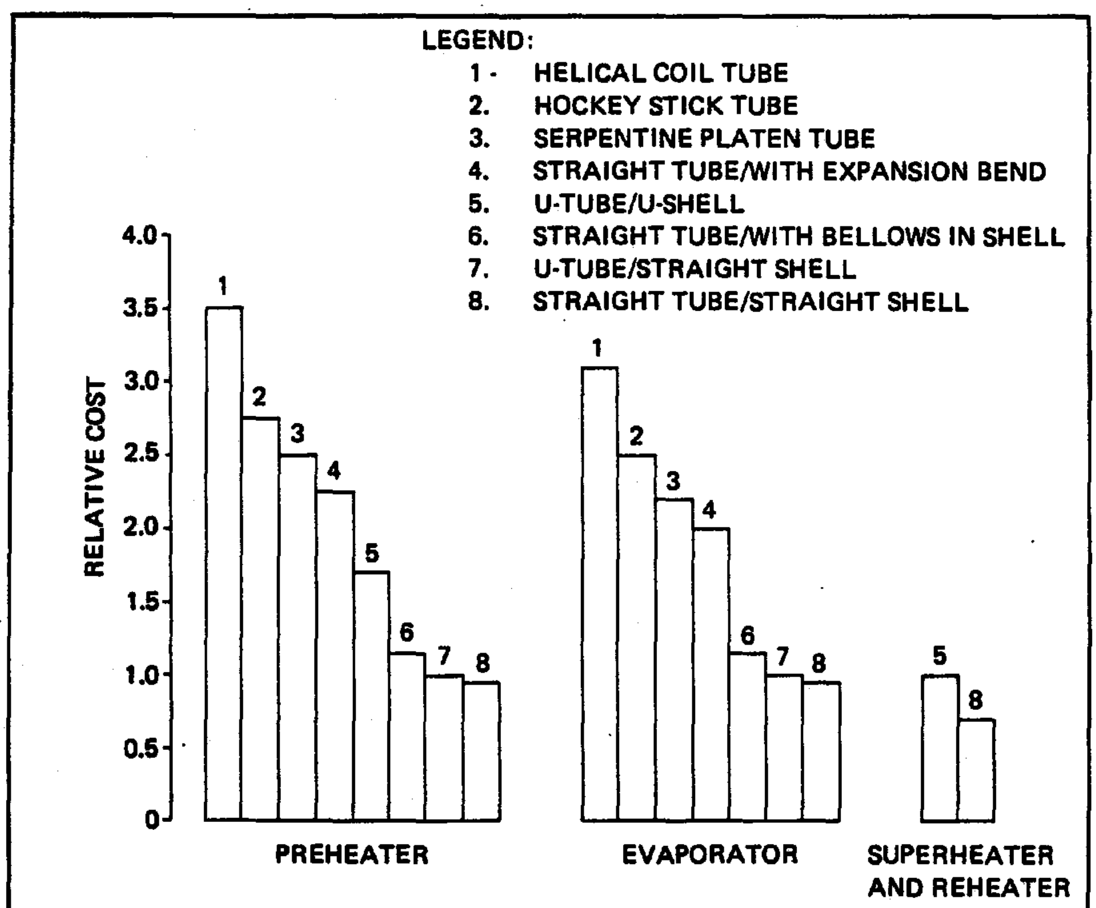
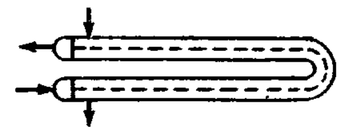
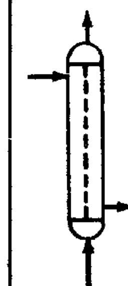
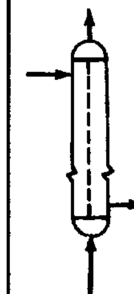
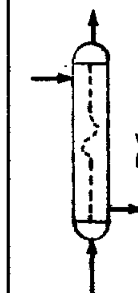
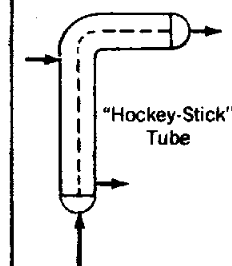
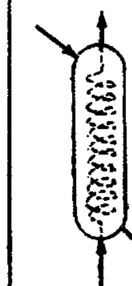
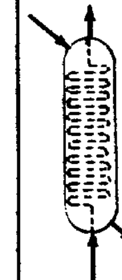
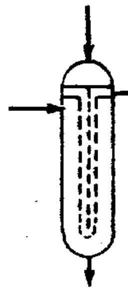


FIG. 2-4 COMPONENT FABRICATION COST COMPARISON

:changer uration:	 U-Tube/ Straight Shell	 U-Tube/U-Shell	 Straight Tube	 Straight Tube with Bellows	 Straight Tube with Short Expansion Bend	 "Hockey-Stick" Tube	 Helical Coil Tube	 Serpentine Platen Tube	 Bayonet Tube
tages:	• Efficient use of heat transfer surface • Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Low-cost fabrication • Vessel support at ground level - minimal seismic restraint required • Easy access for tubeside inspection and repair	• Efficient use of heat transfer surface • Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Vessel support at ground level - minimal seismic restraint required • Easy access for tubeside inspection and repair	• Efficient use of heat transfer surface • Low-cost fabrication • Easy access for tubeside inspection and repair	• Efficient use of heat transfer surface • Readily accomodates tube/shell differential expansion during normal and transient operation • Low-cost fabrication • Easy access for tubeside inspection and repair	• Efficient use of heat transfer surface • Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Easy access for tubeside inspection and repair	• Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Easy access for tubeside inspection and repair	• Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Tube bundle consists of a few long tubes - minimizes tubesheet size and/or number of pressure boundary penetrations • Easy access for tubeside inspection and repair	• Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation • Tube bundle consists of a few long tubes - minimizes tubesheet size and/or number of pressure boundary penetrations	• Readily accomodates tube/shell and tube/tube differential expansion during normal and transient operation
antages:	• Relatively large tubesheet • Large tubesheet thermal stresses where tubeside thermal gradients are large	• High-cost closure weld in shell bend region	• Large thermal stresses induced by tube/shell and tube/tube differential expansion during normal and transient operation • Tube/tube differential expansion may be aggravated by flow maldistribution	• Complex analytical and/or experimental qualification of bellows • Lateral support for bellows required • Periodic in-service inspection of bellows required • Large thermal stresses may be induced by tube/tube differential expansion during normal and transient operation • Tube/tube differential expansion may be aggravated by flow maldistribution	• High-cost tube bundle assembly necessitated by bent tubes • High-cost tube support arrangement in bend region	• High-cost closure weld in shell bend region • High-cost tube bundle assembly necessitated by bent tubes • High cost tube support arrangement in bend region • Large inactive heat transfer surface	• High-cost fabrication resulting from complex tube bundle assembly and tube support arrangement • High material weight-to-performance ratio	• High material weight-to-performance ratio • Reduced access for tubeside inspection and repair	• High-cost fabrication resulting from complex tube bundle assembly • High material weight-to-performance ratio • Inefficient use of heat transfer surface • Tubeside not drainable • Reduced access for tubeside inspection and repair

LE 2-3 COMPARISON OF STEAM GENERATOR DESIGN CHARACTERISTICS

Based on these evaluations, the concept chosen for the preheater and evaporator is a U-tube bundle housed in a single straight shell. While U-tubes were also selected for the superheater and reheater, the large steam-side terminal temperature differences impose unacceptable thermal stresses on a single tubesheet. Therefore, a U-shell assembly was selected to enclose the bundles. The general configuration of the component designs is shown in Figure 2-5

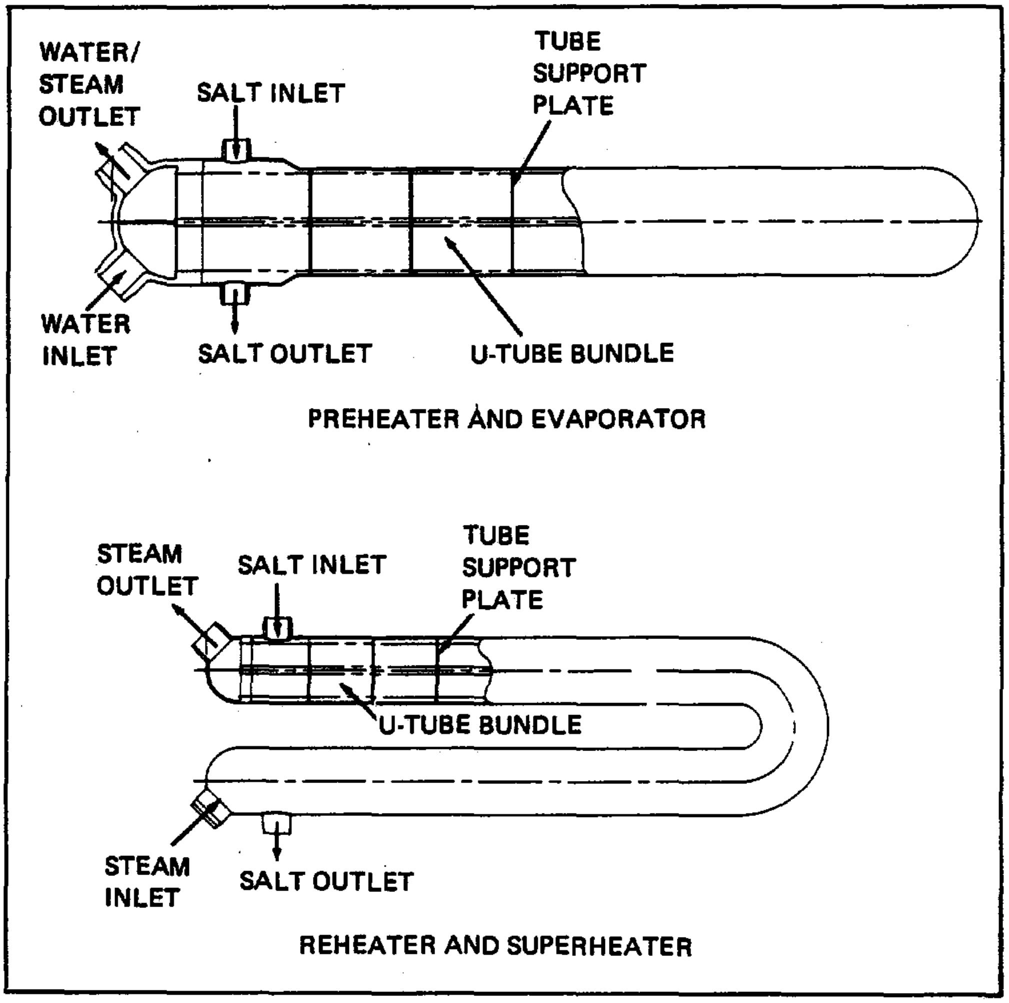


FIG. 2-5 STEAM GENERATOR COMPONENTS

The heat exchangers are oriented horizontally, with nozzles arranged to facilitate venting and draining. With components in the horizontal orientation, building height is reduced, with associated cost savings in construction and component support arrangements. For plant locations in the southwestern United States, where turbine buildings may not be required, the components may be easily supported on foundations at grade level.

Brief descriptions of the candidate heat exchanger configurations are provided in the following paragraphs:

U-Tube Configurations - U-tube configurations may employ either straight shells or U-shells. Both arrangements offer compact tube bundles and counterflow heat transfer. The inherent tube flexibility readily accommodates tube/shell and tube/tube differential thermal expansion during normal and transient operation.

The straight shell configuration is a low-cost assembly which has been used reliably in many applications. It is not, however, a suitable arrangement if the tubeside inlet-to-outlet terminal temperature difference is sufficiently large to develop unacceptable thermal stresses in the tubesheet. If the tubeside terminal temperature difference is large, the U-shell configuration is preferred. This arrangement is structurally superior, but does present fabrication cost penalties associated with the complex shell closure weld in the bend region.

Straight -Tube Configurations - Straight tube arrangements also offer compact tube bundles and counterflow heat transfer. The cost of fabrication is typically low. However, when straight tubes are pinned between fixed tubesheets, thermal stresses produced by tube/shell and tube/tube differential thermal expansion are often unacceptable, particularly in high-temperature, cyclic service.

Use of an expansion bellows within the pressure boundary structure is one means of absorbing tube/shell thermal motion. However, a bellows assembly is considered undesirable because it (a) requires rigorous analytical and/or experimental qualification, (b) requires lateral support, and (c) demands periodic in-service inspection and possibly replacement. It fails to alleviate thermal stresses produced by tube/tube differential expansion, which may occur in superheaters, reheaters, or heat exchangers having a significant amount of surface in subcooled heat transfer. Expansion bends within straight tubes, such as sine-wave bends, may also be used to absorb thermal motion. However, the complexity of the tubing process and tube support arrangement serves to substantially increase fabrication cost.

Hockey Stick Tube Configuration - The hockey stick tube arrangement also offers a compact tube bundle and counterflow heat transfer. Adequate tube flexibility exists to accommodate tube/shell and tube/tube differential expansion. However, the complex tubing process and tube support arrangement in the bend region, as well as the complex shell closure welds in the bend region, introduce relatively high fabrication costs in comparison to U-tube assemblies.

Helical Coil Tube Configuration - The helical coil tube arrangement provides outstanding capability for accommodating tube/shell and tube/tube differential expansion produced by severe thermal gradients. Its major shortcomings are (a) the high fabrication cost associated with its complex tube bundle and tube support assembly, and (b) a high material weight-to-performance ratio for applications in which plant thermal ratings are small. It is considered a superior technical design, but not cost effective for solar power service.

Serpentine Tube Configuration - The serpentine tube configuration consists of a bundle of tubes arranged in platens similar to a fossil boiler economizer or superheater. It shares the major advantages and shortcomings of the helical coil tube arrangement. Like the helical coil tube configuration, it is considered unsuitable for solar power service.

Bayonet Tube Configuration - The bayonet tube arrangement offers excellent tube/shell and tube/tube differential thermal expansion capability. However, it also exhibits inefficient use of heat transfer surface, high cost fabrication resulting from a complex tube bundle assembly, and limited access for maintenance and repair.

2.3 Forced Versus Natural Circulation

The relative merits of forced versus natural circulation of flow between the steam drum and the evaporator were examined, and a forced circulation system was selected. The principal advantage of natural circulation operation is the elimination of recirculation pumps, with a corresponding reduction in equipment capital cost and operating and maintenance expense. However, to make natural circulation feasible, the available driving force (static head of downcomer fluid) must exceed the hydraulic resistance in the circuit. Thus, cost savings associated with elimination of the recirculation pumps is often offset by expenditures associated with (a) elevating the steam drum to increase downcomer head, and (b) enlarging riser-side piping and/or evaporator tubes to reduce pressure losses.

From the evaluation, it was found that a capital cost savings equal to approximately one percent of the installed subsystem cost favored the natural circulation scheme. The analysis was based on an arrangement in which the components were housed in a four level structure adjacent to the turbine building (see Figure 5-1). However, it was considered that for many applications, particularly those in the southwestern United States, it would be appropriate to place components on foundations at grade level, without need of a protective enclosure (see Figure 5-2). In this case, it would be desirable to arrange the components compactly, with the steam drum elevation minimized. If forced recirculation were employed for such applications, the drum elevation could be limited to no more than that required to provide adequate net positive suction head for the recirculation pumps (approximately 15 feet). The additional drum elevation necessary to assure sufficient driving head for natural circulation is shown in Figure 2-6.

It is apparent that for each potential solar power application, stand-alone or repowering, the choice of forced or natural circulation operation is dependent to a large extent on site-specific considerations. For this study, forced circulation was selected as the basic operating mode because of the important flexibility which it offers in the physical layout of the major subsystem components.

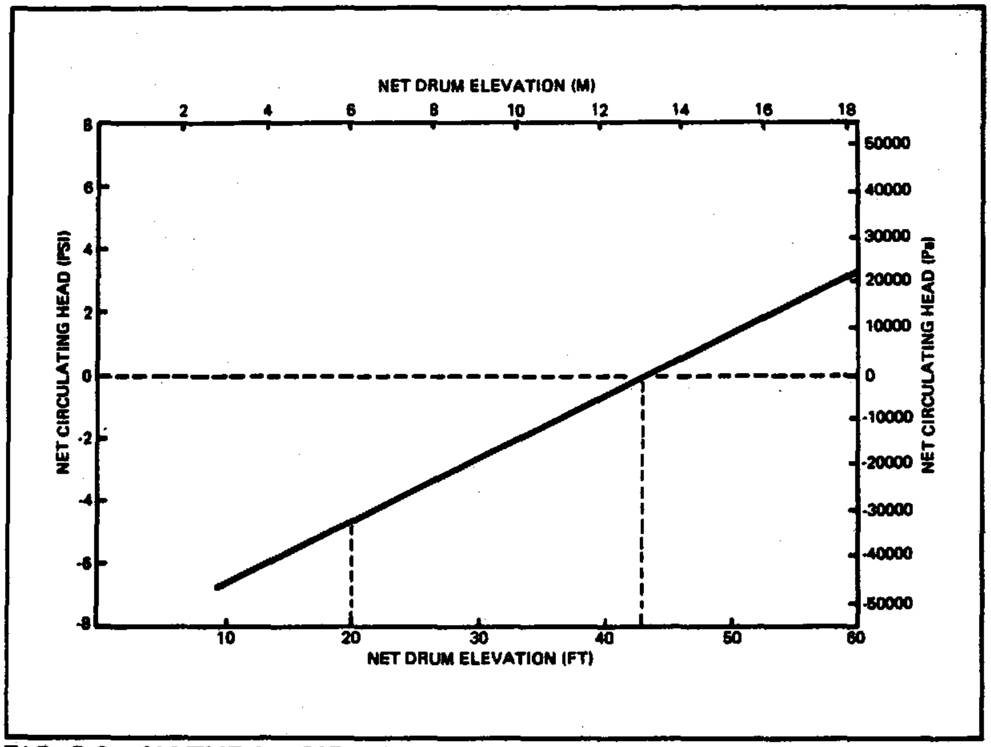


FIG. 2-6 NATURAL CIRCULATION HEAD AT 100% LOAD

3.0 STEAM GENERATOR DESIGN

The steam generator subsystem design employs a forced recirculation steam system arrangement using molten nitrate salt (60% $NaNO_3$, 40% KNO_3) as the primary heat transfer fluid. A simplified flow schematic is shown in Figure 3-1.

Hot salt is pumped to the steam generator from the high-temperature thermal storage tank. The stream is split and apportioned between the superheater and reheater. The two streams are mixed and delivered in turn to the evaporator and preheater. The cold salt is then returned to the low-temperature thermal storage tank.

Feedwater is preheated to near saturation temperature and delivered to the steam drum. In the drum, it is mixed with recirculated water and pumped to the evaporator, where a high-quality steam/water mixture is produced. This steam/water mixture is returned to the drum, and the steam and water phases are separated. The saturated steam is then superheated and delivered to the high-pressure turbine. A portion of the high-pressure turbine exhaust steam is reheated for expansion in the intermediate and low-pressure stages of the turbine.

The use of a separate preheater constructed of low-alloy material having relatively high thermal conductivity enables the overall heat transfer surface to be utilized most economically.

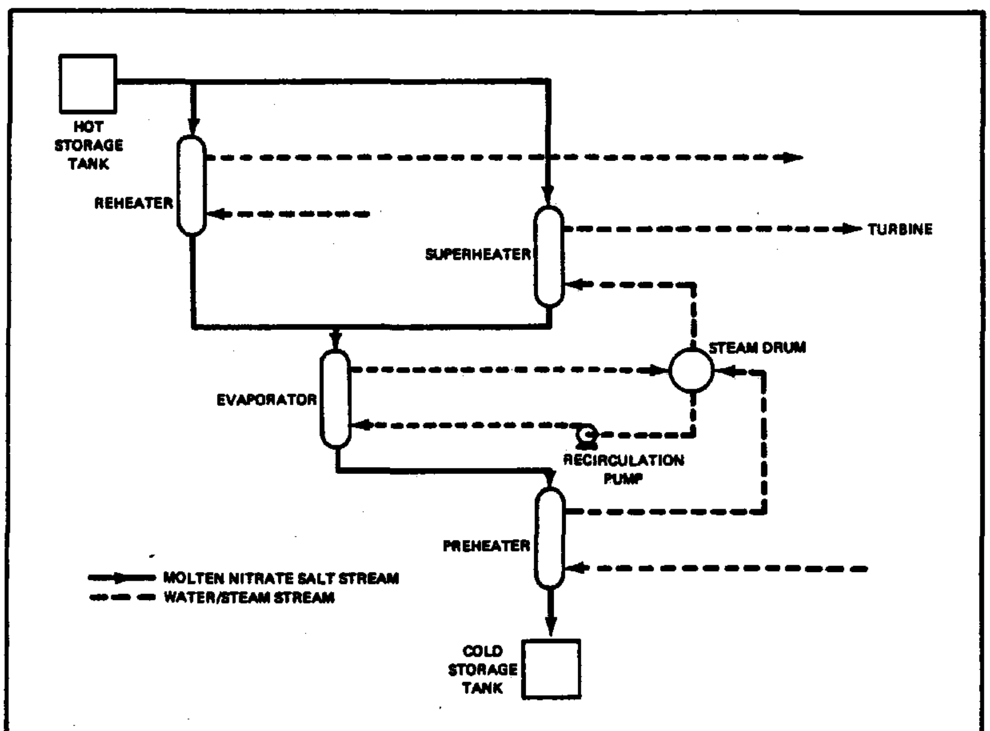


FIG. 3-1 RECIRCULATION STEAM CYCLE

3.1 Description Of Components

The preheater and evaporator are horizontally oriented U-tube/straight shell heat exchangers with the inlet and outlet legs of the U-tubes arranged in a common vertical plane. A central baffle is provided on the shell side to assure efficient counterflow heat transfer. A divider plate is used in the water-side plenum to separate the inlet and outlet streams.

The superheater and reheater are horizontally oriented U-tube/U-shell heat exchangers with the inlet and outlet legs also arranged in a common vertical plane. The U-shell configuration is employed because the tube side terminal temperature differences are large and would induce unacceptable thermal stresses in a single tubesheet.

The U-tube configurations offer compact tube bundles and tube flexibility to readily accommodate tube/shell and tube/tube differential thermal expansion during normal and transient operation.

The steam drum is a standard Babcock & Wilcox fossil steam drum design employing conventional cyclone steam separators and scrubbers.

Table 3-1 lists the principal geometric characteristics of the 100 MWe steam generator components.

Specific design features of the steam generator components are described in the following paragraphs:

Tube supports - Broached-hole support plates support the tubes along their straight lengths at spans determined to preclude potential damage from flow-induced or machinery-induced vibration. Similarly, "wiggle-bar" assemblies support the tubes in the bend region. These concepts are illustrated in Figures 3-2 and 3-3.

TABLE 3-1

Principal Geometric Characteristics

100 MWe Components

Component	Main Shell O.D. m(in)	Total Length m(ft)	Tube Dimensions O.D. X Wall mm(in)	No. Of Tubes	Avg. Active Tube Length m(ft)	Primary Surface Area $m^2(ft^2)$
Preheater	1.422 (56.0)	10.448 (34.28)	12.700 X 1.473 (.500 X .058)	2461	18.288 (60.0)	1803 (19410)
Evaporator	1.588 (62.5)	15.225 (49.95)	22.225 X 3.759 (.875 X .148)	1236	27.218 (89.3)	2349 (25280)
Superheater	0.610 (24.0)	7.373 (24.19)	12.700 X 1.651 (.500 X .065)	774	13.868 (45.5)	428 (4610)
Reheater	0.762 (30.0)	8.251 (27.07)	15.875 X .889 (.625 X .035)	870	15.027 (49.3)	652 (7020)

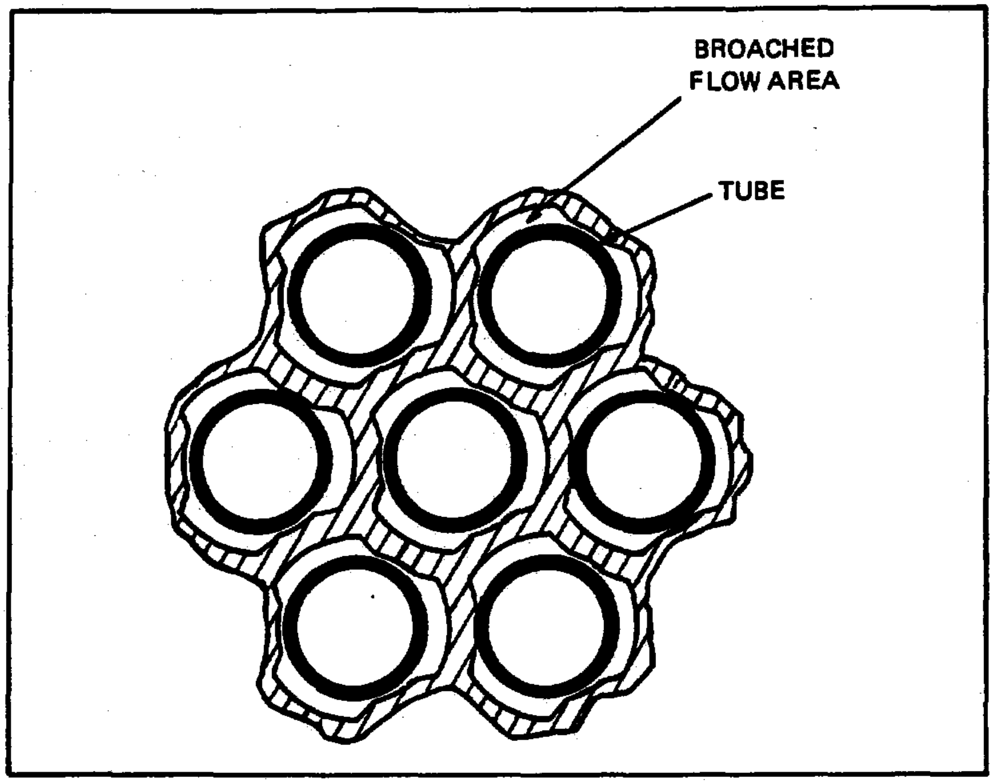


FIG. 3-2 BROACHED-HOLE TUBE SUPPORT PLATE

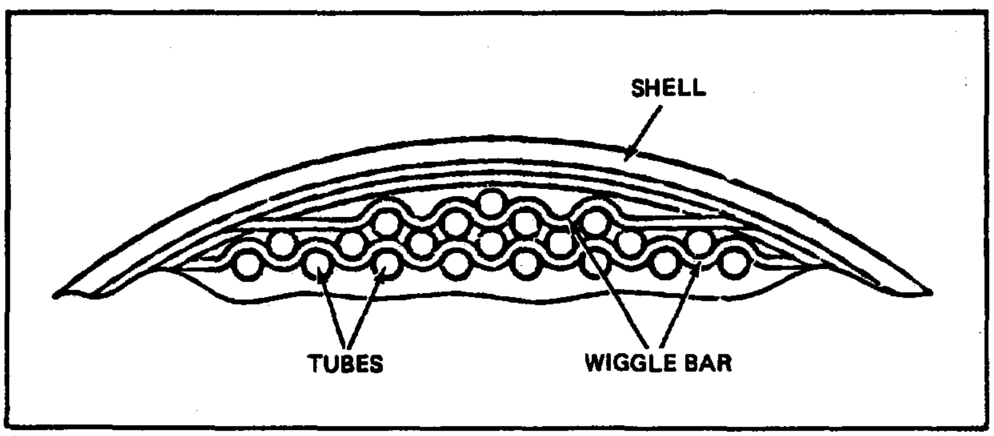


FIG. 3-3 WIGGLE BAR TUBE SUPPORTS

Salt Distribution - Distribution boxes at the salt inlet and outlet nozzles promote uniform flow distribution as the salt enters and discharges from the tube bundles. Flow baffles protect the tube bundles from excessive salt inlet impingement velocites. Figure 3-4 illustrates the salt distribution box concept.

Preferentially broached tube support plates promote salt mixing to enhance heat transfer and inhibit salt stratification (See Figure 3-5).

Tube/Tubesheet Weld - Tubes are terminated at tubesheets by expansion and flush-type seal welds as illustrated in Figure 3-6. This weld configuration has demonstrated excellent integrity in nuclear service.

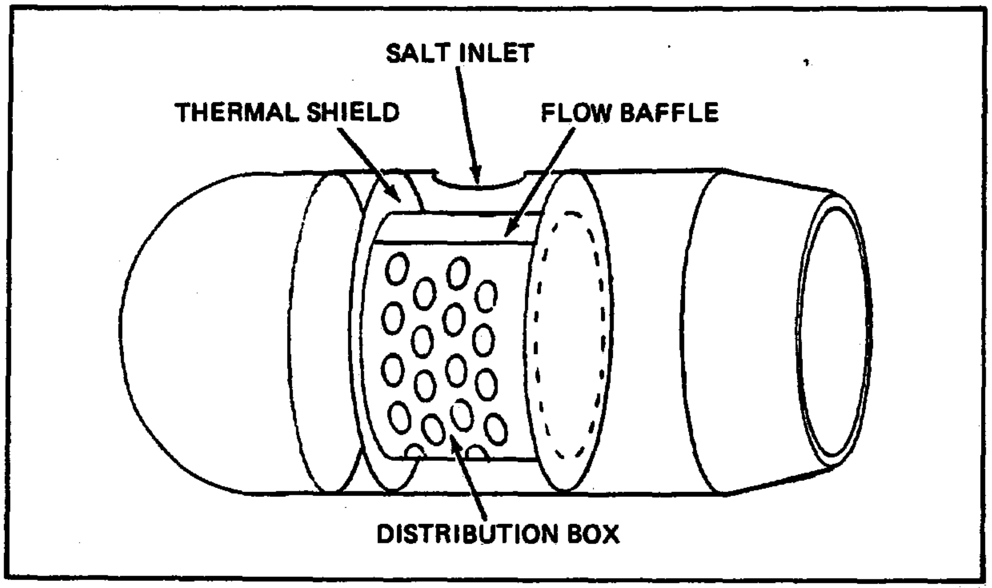


FIG. 3-4 SALT DISTRIBUTION BOX

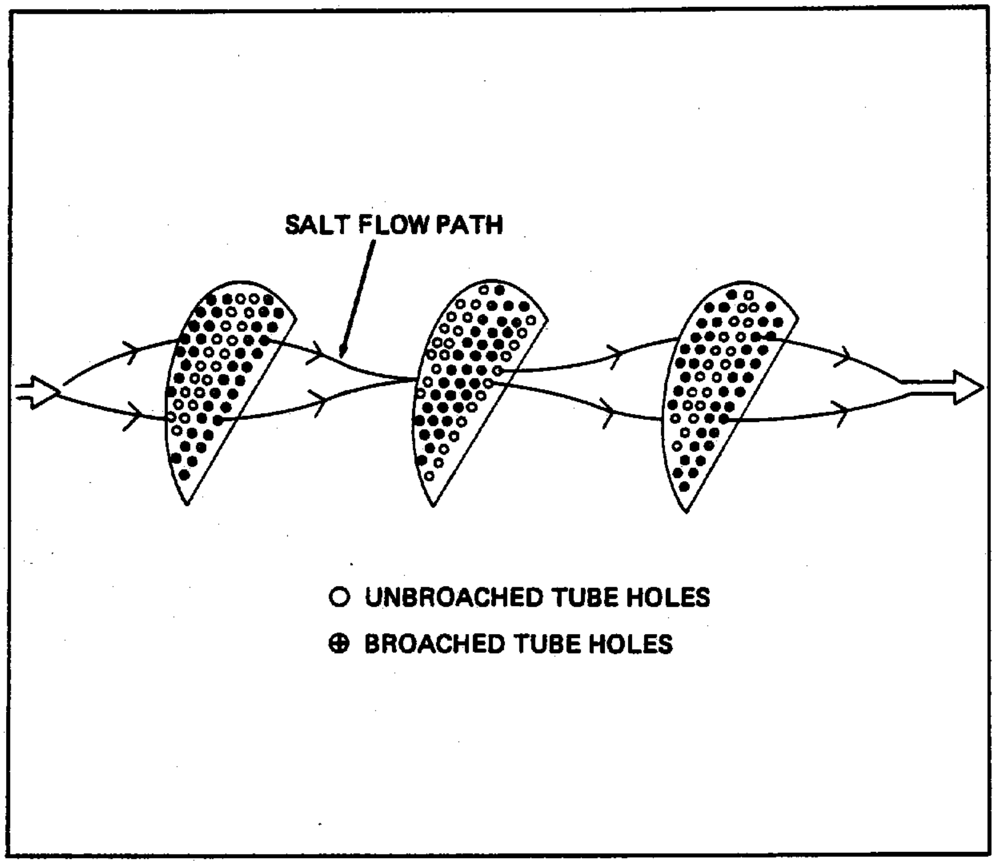


FIG. 3-5 HALF SUPPORT PLATE PREFERENTIAL BROACHING

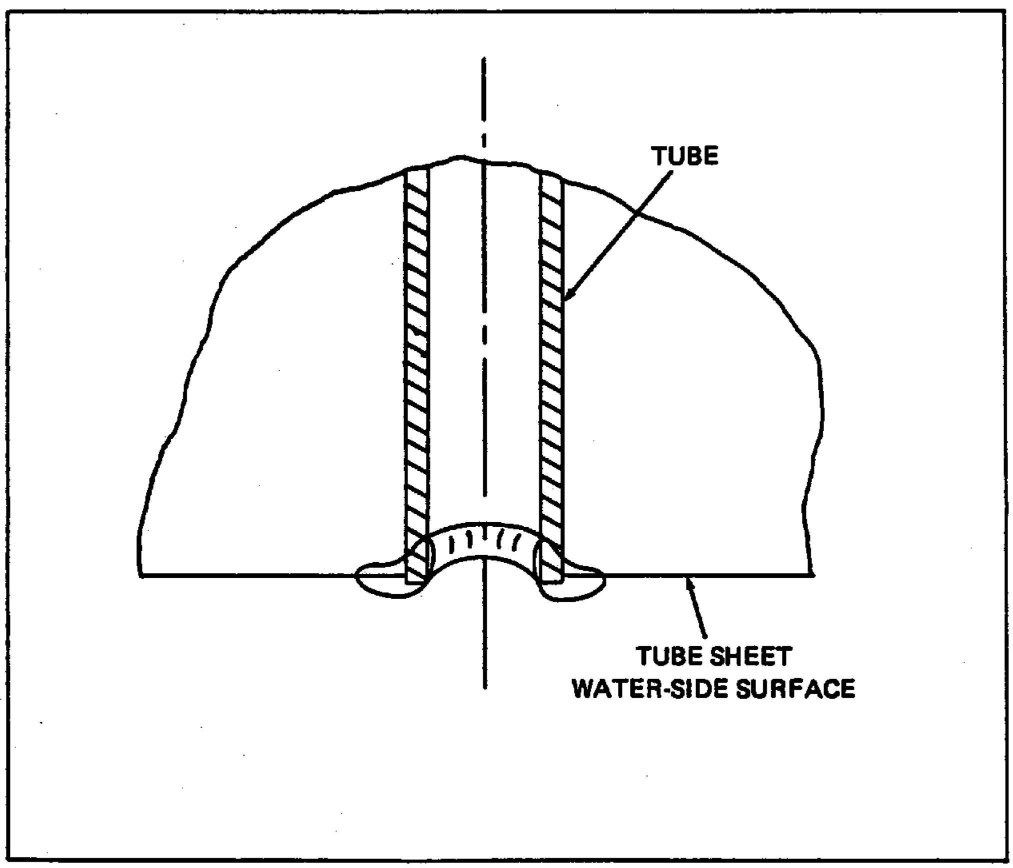


FIG. 3-6 TUBE-TO-TUBE SHEET FLUSH WELD

Ribbed Tubing - Thirty-two feet of multi-lead ribbed tubing is butt welded to smooth bore tubing in the evaporator. This construction, illustrated in Figure 3-7, is intended to inhibit the development of DNB in high heat flux, high quality regions of the tube bundle. The design considerations for using ribbed tubing are described in Section 3.3.1.

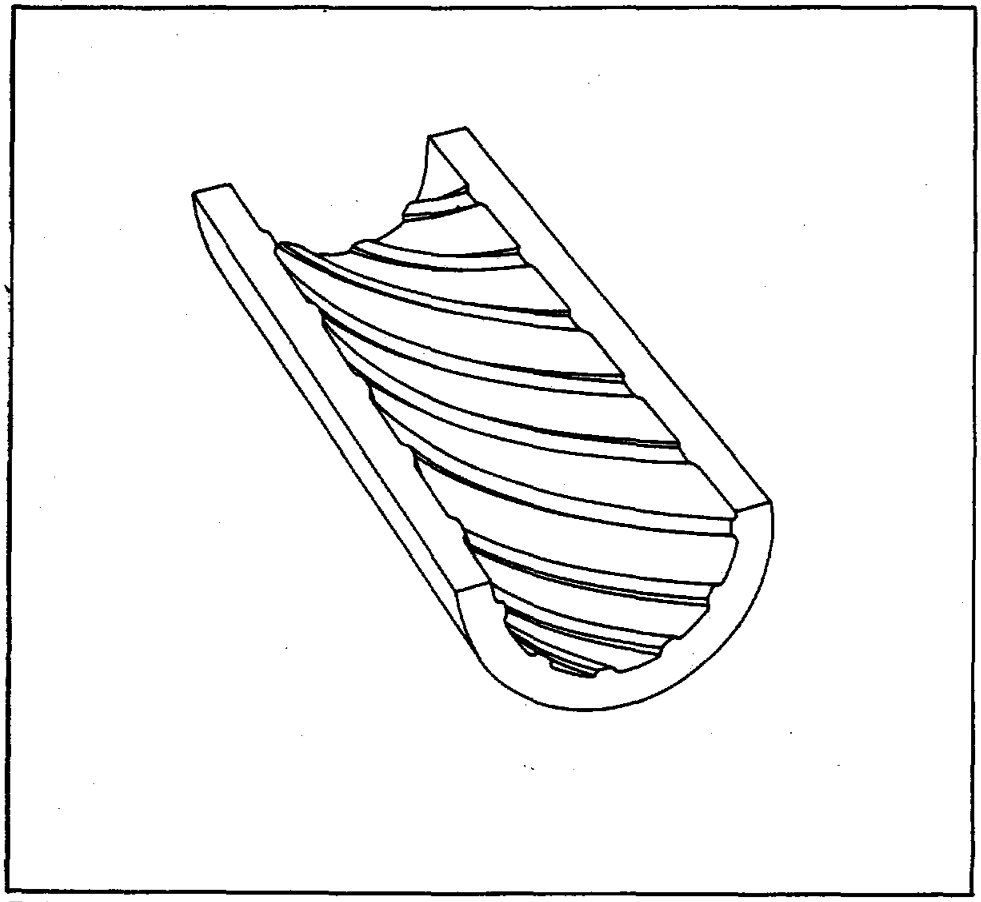


FIG. 3-7 CROSS-SECTION RIBBED TUBING

Thermal Shields - Thermal shields are attached to the central baffles on the shell side of the U-tube/straight shell heat exchangers. A stagnant salt region results between baffle and thermal shield, buffering thermal transients otherwise imposed on the baffle and precluding excessive baffle thermal stresses and/or deflections. Figure 3-8 illustrates the central baffle arrangement. The assembly is guided in and welded to the shells at the channels.

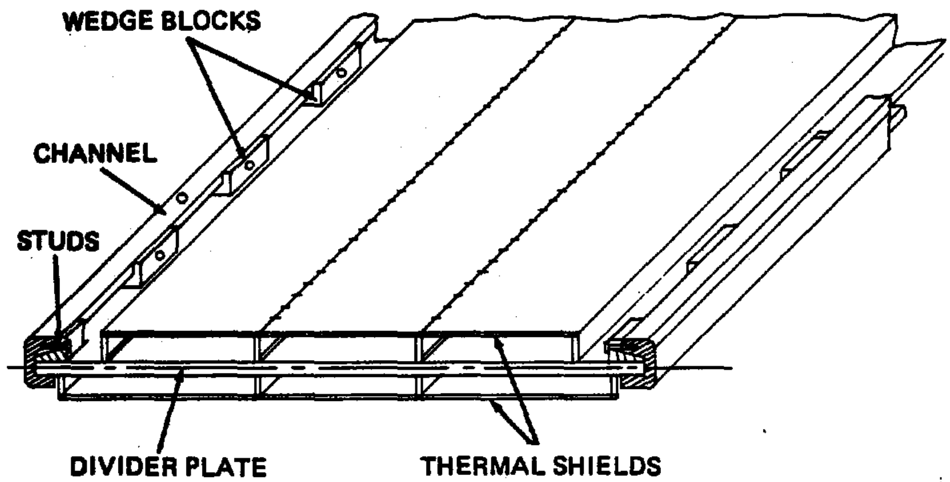


FIG. 3-8 CENTRAL BAFFLE ARRANGEMENT

Unbroached tube support plates, positioned near the salt face of the tubesheets, protect the tubesheet from excessive thermally induced stresses. A relatively stagnant salt region results between tubesheet and thermal shield which buffers thermal transients otherwise imposed directly on the tubesheet. Figure 3-4 shows the tubesheet thermal shield which is a part of the salt distribution box complex.

Inspection and Maintenance - Inlet water/steam nozzles are located to permit complete drainage of tubes. Water and shell side inspection openings provide internal access to maintain component reliability. A small notch is cut in the bottom of each tube support plate to provide an unobstructed flow path to the salt drain openings. The drain connections are located to facilitate gravitational draining of the salt. Figures 3-9, 3-10, and 3-11 illustrate the above design features.

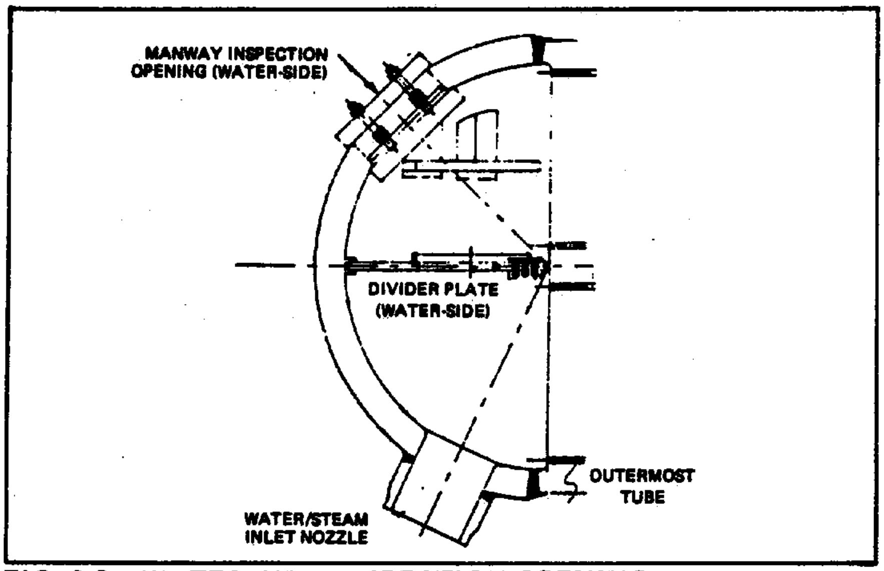


FIG. 3-9 WATER-SIDE INSPECTION OPENING

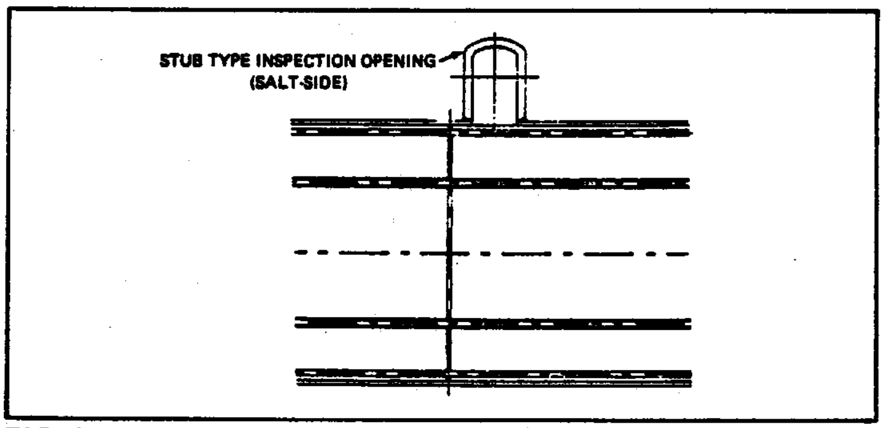


FIG. 3-10 SALT-SIDE INSPECTION OPENING

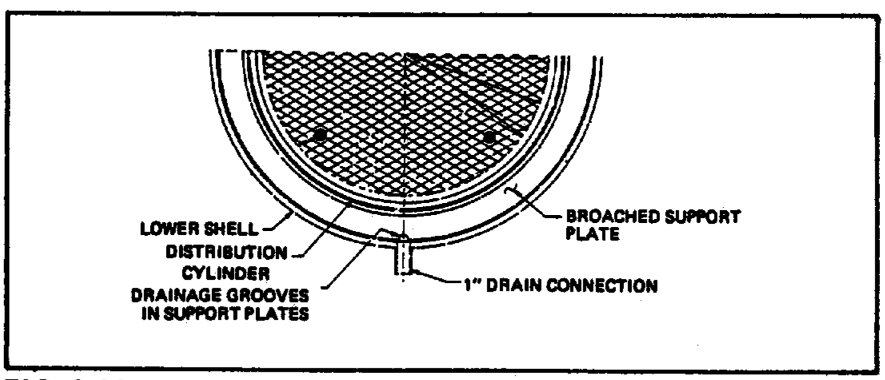


FIG. 3-11 SALT-SIDE DRAINAGE PROVISIONS

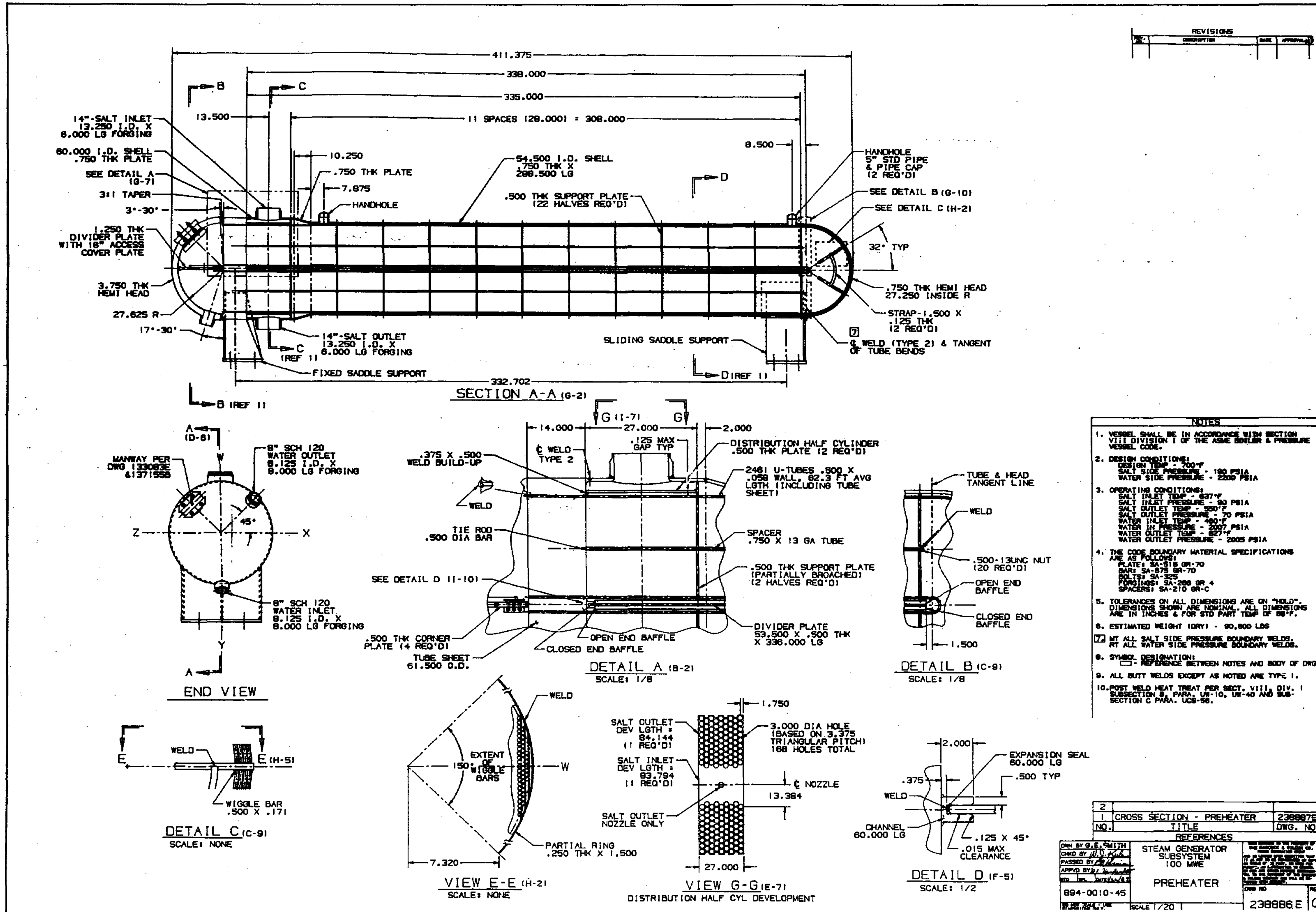
3.1.1 Preheater

The preheater is a horizontal U-tube/straight shell heat exchanger, as illustrated in Figures 3-12 and 3-13. Feedwater enters the lower leg of the tubes, flows upward around the bend, and discharges from the upper leg. Molten salt enters the shell side of the heat exchanger near the terminal of the upper, or outlet, leg and flows through the bundle to discharge near the terminal of the lower, or inlet, leg.

Functionally, the preheater design incorporates a water-side inlet/outlet divider plate to separate the water-side plenum. A salt-side central baffle is required to assure efficient counterflow heat transfer, and preferentially broached-hole tube support plates promote salt mixing to enhance heat transfer and inhibit salt stratification.

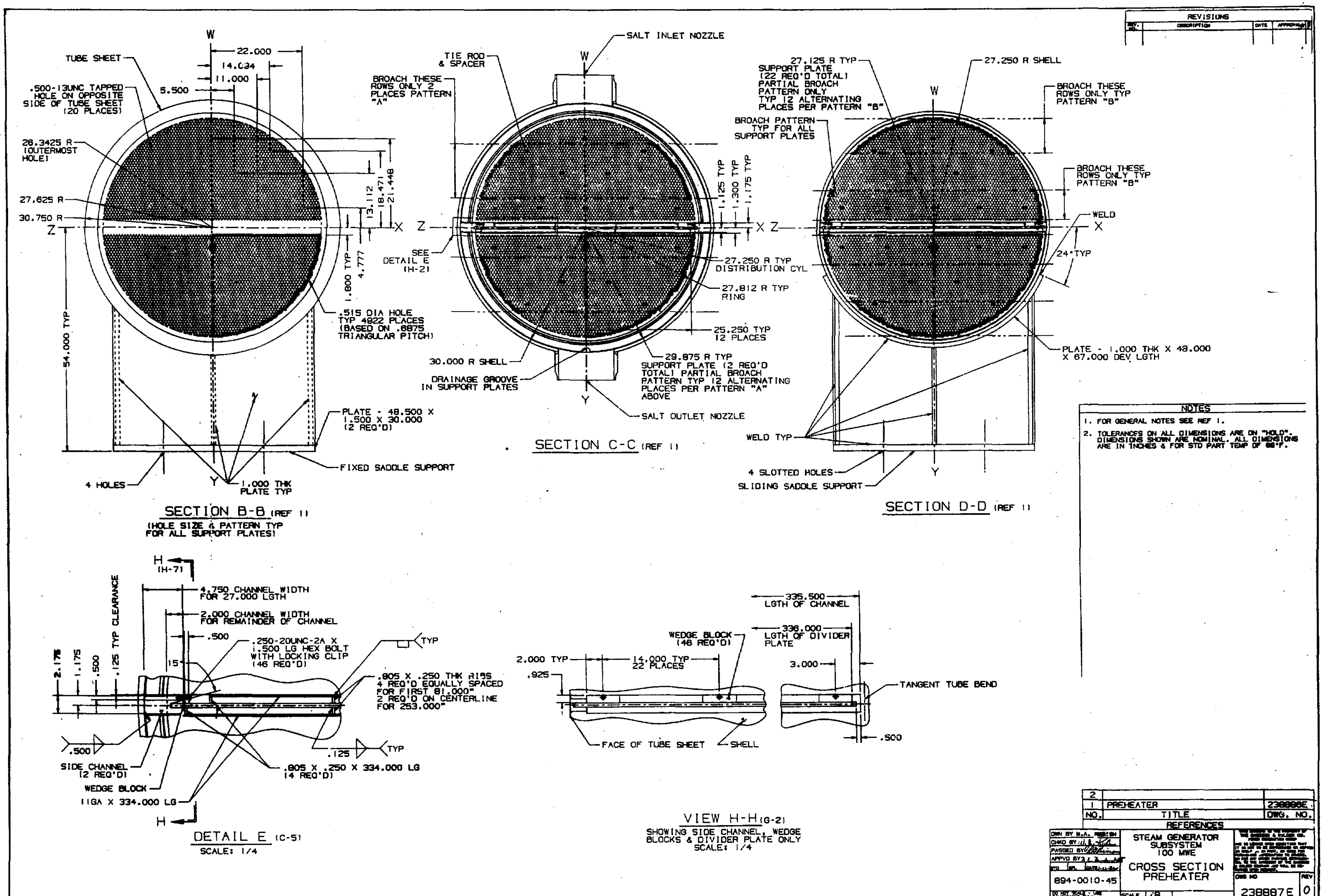
Structurally, thermal shields attached to the salt side central baffle prevent potentially excessive baffle deflections and/or stresses resulting from thermal transients, and close a potential flow bypass lane. A tubesheet thermal shield is not required since steady-state and transient thermal loadings do not produce excessive tubesheet stresses. Main vessel saddle supports have been sized considering deadweight, seismic, and thermal loads. A fixed saddle support is provided at the tubesheet, and a sliding saddle suport at the tube bend region assures unrestricted longitudinal expansion and contraction.

Provisions for inspection and maintenance include (a) a 16-inch access opening in the water-side divider plate to offer access to both inlet and outlet sides of the tubesheet through a single manway opening in the head, and (b) stub-type handholes in the salt side for inspecting the shell side of the vessel.



SECTION A-A (G-2)
END VIEW
DETAIL A (B-2)
SCALE: 1/8
DETAIL B (C-9)
SCALE: 1/8
DETAIL C (C-9)
SCALE: NONE
VIEW E-E (H-2)
SCALE: NONE
VIEW G-G (E-7)
DISTRIBUTION HALF CYL DEVELOPMENT
DETAIL D (F-5)
SCALE: 1/2
411.375
338.000
335.000
11 SPACES (28.000) = 308.000
332.702
14"-SALT INLET 13.250 I.D. X 8.000 LG FORGING
60.000 I.D. SHELL .750 THK PLATE
54.500 I.D. SHELL .750 THK X 298.500 LG
.500 THK SUPPORT PLATE (22 HALVES REQ'D)
HANDHOLE 5" STD PIPE & PIPE CAP (2 REQ'D)
1.250 THK DIVIDER PLATE WITH 16" ACCESS COVER PLATE
3.750 THK HEMI HEAD
.750 THK HEMI HEAD 27.250 INSIDE R
STRAP-1.500 X .125 THK (2 REQ'D)
14"-SALT OUTLET 13.250 I.D. X 6.000 LG FORGING
FIXED SADDLE SUPPORT
SLIDING SADDLE SUPPORT
MANWAY PER DWG 133083E & 137155B
8" SCH 120 WATER OUTLET 8.125 I.D. X 8.000 LG FORGING
8" SCH 120 WATER INLET 8.125 I.D. X 8.000 LG FORGING
DISTRIBUTION HALF CYLINDER .500 THK PLATE (2 REQ'D)
2461 U-TUBES .500 X .058 WALL, 62.3 FT AVG LGTH (INCLUDING TUBE SHEET)
DIVIDER PLATE 53.500 X .500 THK X 336.000 LG
TUBE SHEET 61.500 O.D.
EXPANSION SEAL 60.000 LG
CHANNEL 60.000 LG
NOTES
1. VESSEL SHALL BE IN ACCORDANCE WITH SECTION VIII DIVISION I OF THE ASME BOILER & PRESSURE VESSEL CODE.
2. DESIGN CONDITIONS: DESIGN TEMP - 700°F SALT SIDE PRESSURE - 190 PSIA WATER SIDE PRESSURE - 2200 PSIA
3. OPERATING CONDITIONS: SALT INLET TEMP - 637°F SALT INLET PRESSURE - 90 PSIA SALT OUTLET TEMP - 550°F SALT OUTLET PRESSURE - 70 PSIA WATER INLET TEMP - 460°F WATER IN PRESSURE - 2007 PSIA WATER OUTLET TEMP - 627°F WATER OUTLET PRESSURE - 2005 PSIA
4. THE CODE BOUNDARY MATERIAL SPECIFICATIONS ARE AS FOLLOWS: PLATE: SA-516 GR-70 BAR: SA-675 GR-70 BOLTS: SA-325 FORGINGS: SA-266 GR 4 SPACERS: SA-210 GR-C
5. TOLERANCES ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
6. ESTIMATED WEIGHT (DRY) - 90,800 LBS
7. MT ALL SALT SIDE PRESSURE BOUNDARY WELDS. RT ALL WATER SIDE PRESSURE BOUNDARY WELDS.
8. SYMBOL DESIGNATION: - REFERENCE BETWEEN NOTES AND BODY OF DWG.
9. ALL BUTT WELDS EXCEPT AS NOTED ARE TYPE 1.
10. POST WELD HEAT TREAT PER SECT. VIII, DIV. 1 SUBSECTION B, PARA. UW-10, UW-40 AND SUB-SECTION C PARA. UCS-56.
1 CROSS SECTION - PREHEATER 238887E
NO. TITLE DWG. NO.
REFERENCES
DWN BY G.E. SMITH
STEAM GENERATOR SUBSYSTEM 100 MWE
PREHEATER
894-0010-45
SCALE 1/20
238886 E 0

FIG. 3-12



SECTION B-B (REF 1)
(HOLE SIZE & PATTERN TYP FOR ALL SUPPORT PLATES)
TUBE SHEET
.500-13UNC TAPPED HOLE ON OPPOSITE SIDE OF TUBE SHEET (20 PLACES)
28.3425 R (OUTERMOST HOLE)
27.625 R
30.750 R
.515 DIA HOLE TYP 4922 PLACES (BASED ON .6875 TRIANGULAR PITCH)
PLATE - 48.500 X 1.500 X 30.000 (2 REQ'D)
FIXED SADDLE SUPPORT
4 HOLES
1.000 THK PLATE TYP
54.000 TYP
SECTION C-C (REF 1)
SALT INLET NOZZLE
TIE ROD & SPACER
BROACH THESE ROWS ONLY 2 PLACES PATTERN "A"
SEE DETAIL E (H-2)
27.250 R TYP DISTRIBUTION CYL
27.812 R TYP RING
25.250 TYP 12 PLACES
29.875 R TYP SUPPORT PLATE (2 REQ'D TOTAL) PARTIAL BROACH PATTERN TYP 12 ALTERNATING PLACES PER PATTERN "A" ABOVE
30.000 R SHELL
DRAINAGE GROOVE IN SUPPORT PLATES
SALT OUTLET NOZZLE
SECTION D-D (REF 1)
27.125 R TYP SUPPORT PLATE (22 REQ'D TOTAL) PARTIAL BROACH PATTERN ONLY TYP 12 ALTERNATING PLACES PER PATTERN "B"
27.250 R SHELL
BROACH PATTERN TYP FOR ALL SUPPORT PLATES
BROACH THESE ROWS ONLY TYP PATTERN "B"
WELD
24° TYP
PLATE - 1.000 THK X 48.000 X 67.000 DEV LGTH
WELD TYP
4 SLOTTED HOLES
SLIDING SADDLE SUPPORT
DETAIL E (C-5)
SCALE: 1/4
4.750 CHANNEL WIDTH FOR 27.000 LGTH
2.000 CHANNEL WIDTH FOR REMAINDER OF CHANNEL
.250-20UNC-2A X 1.500 LG HEX BOLT WITH LOCKING CLIP (46 REQ'D)
.805 X .250 THK RIBS 4 REQ'D EQUALLY SPACED FOR FIRST 81.000" 2 REQ'D ON CENTERLINE FOR 253.000"
.805 X .250 X 334.000 LG (4 REQ'D)
SIDE CHANNEL (2 REQ'D)
WEDGE BLOCK
11GA X 334.000 LG
.125 TYP CLEARANCE
VIEW H-H (G-2)
SHOWING SIDE CHANNEL, WEDGE BLOCKS & DIVIDER PLATE ONLY
SCALE: 1/4
WEDGE BLOCK (46 REQ'D)
14.000 TYP 22 PLACES
335.500 LGTH OF CHANNEL
336.000 LGTH OF DIVIDER PLATE
TANGENT TUBE BEND
FACE OF TUBE SHEET
SHELL
REVISIONS
NOTES
1. FOR GENERAL NOTES SEE REF 1.
2. TOLERANCES ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
1 PREHEATER 238888E
NO. TITLE DWG. NO.
REFERENCES
STEAM GENERATOR SUBSYSTEM 100 MWE
CROSS SECTION PREHEATER
894-0010-45
SCALE 1/8
238887E

FIG. 3-13

3.1.2 Evaporator

The evaporator is also a horizontal U-tube/straight shell heat exchanger and is illustrated in Figures 3-14 and 3-15. The water and salt-side flow schemes are similar to those described for the preheater.

The functional design features of the evaporator are similar to those described for the preheater. Additionally, multi-lead ribbed tubing is used for the last 32 feet of the upper or outlet leg. The boundary layer turbulence and centrifugal forces produced by the ribs maintain nucleate boiling to a higher quality for a given pressure, heat flux, and mass flow rate than a smooth bore tube of the same general dimensions. A discussion of the considerations and trade-offs for application of ribbed tubing to preclude DNB is presented in Section 3.3.1.

Structurally, thermal shields are attached to the salt-side central baffle to serve the same purpose as described for the preheater. An unbroached tube support plate, located near the salt-side face of the tubesheet, shields the tubesheet from excessive thermally induced stresses. The salt-side head is ellipsoidal to provide accessability to its circumferential butt weld (final closure weld) which must be radiographed during fabrication. A fixed saddle support with stiffeners is located at the mid-span of the vessel to provide dead weight and seismic load support and to prevent the shell from bowing upward during transient and steady state operating conditions. It is important to prevent the shell from bowing to maintain the inherent advantages of the U-tube configurations; namely tube/tube and shell/tube flexibility.

Provisions for inspection and maintenance include (a) a 16-inch access opening in the water side divider plate to access both inlet and outlet sides of the tubesheet through a single manway opening in the head, and (b) master-type handholes in the salt-side ellipsoidal head for shell-side inspection.

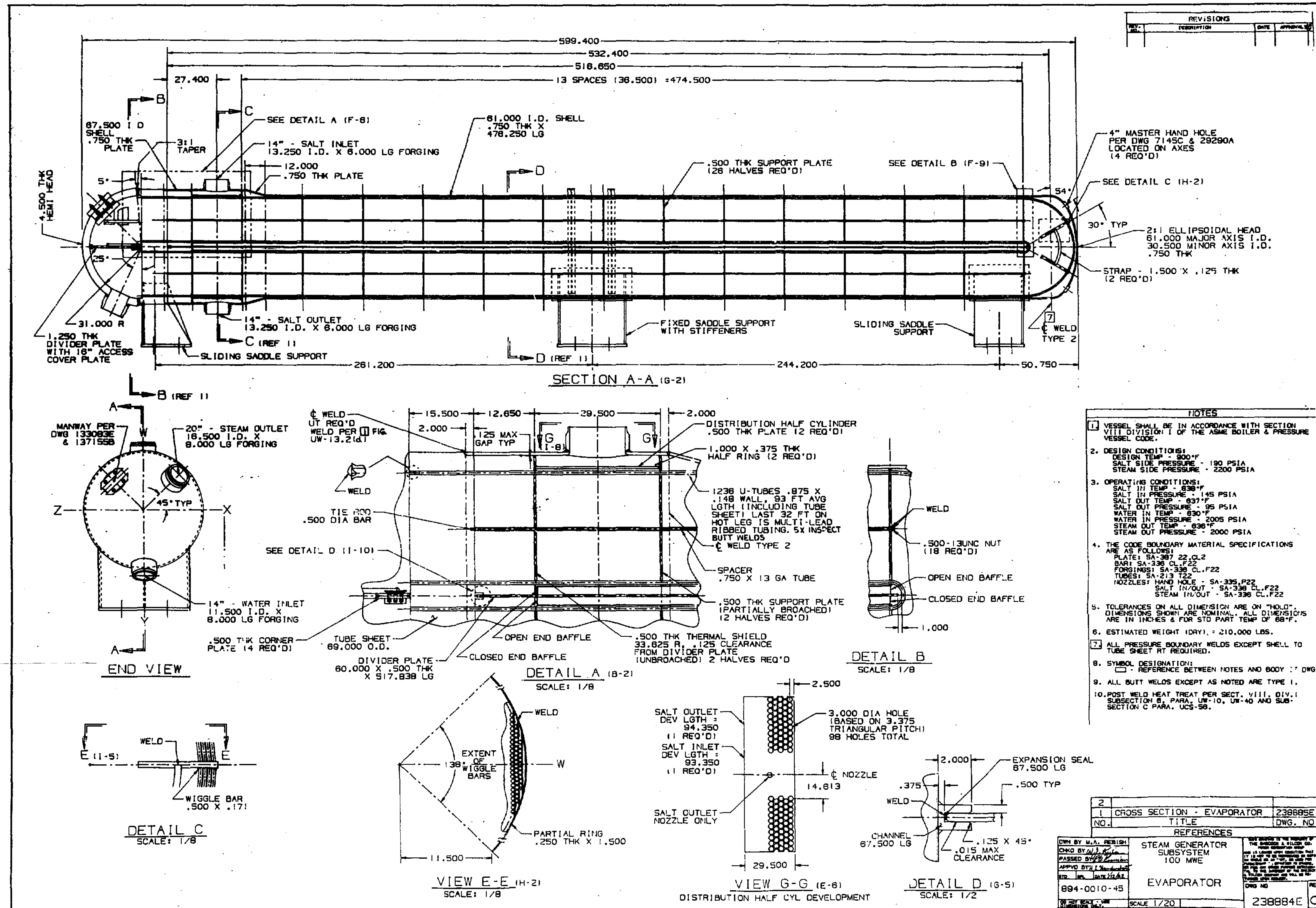
3.1.3 Superheater

The superheater is a horizontal U-tube/U-shell heat exchanger as illustratred in Figures 3-16 and 3-17, with inlet and outlet legs arranged in a common vertical plane. Saturated steam enters the lower leg of the tubes, flows upward around the bend, and discharges from the upper leg. Molten salt enters the shell-side near the terminal of the upper, or outlet, leg and flows through the bundle to discharge near the terminal of the lower, or inlet, leg.

Functionally, the superheater design requires neither steam-side nor salt-side central baffles since the U-shell provides the necessary flow boundaries to promote efficient counterflow heat transfer. Tube support plates are broached, but not preferentially broached, since the salt flow velocity is high and turbulent enough to promote sufficient salt mixing, even at part load conditions.

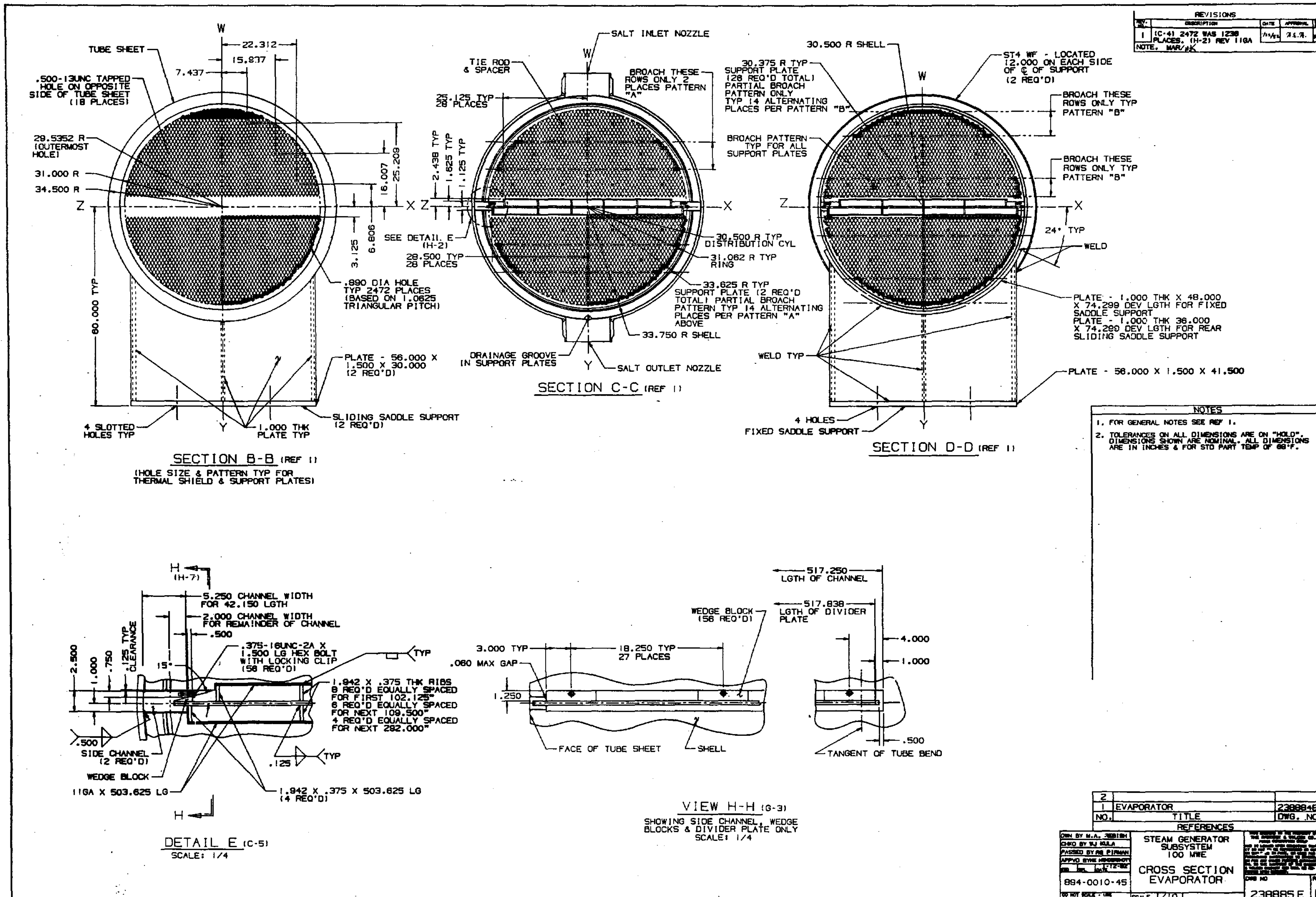
Structurally, the tube bundle diameter is small enough to permit standard seamless pipe and fittings to be economically used for pressure shell parts. Unbroached tube support plates located near the salt-side surfaces of the tubesheets shield the tubesheets from potentially excessive thermal loadings. Lower leg saddle supports are fixed at the tubesheet and sliding at the U-bend to accommodate dead weight, seismic, and thermal loads. Upper leg U-bolt supports are attached to constant load hangers to maintain flexibility in the U-bend region. Lateral ties on the shell of the upper leg provide support attachments for out-of-plane loadings.

Stub-type handholes provide internal access to steam and salt-sides for inspection and maintenance.



SECTION A-A (G-2)
599.400
532.400
518.650
27.400
13 SPACES (36.500) =474.500
87.500 I D SHELL .750 THK PLATE
3:1 TAPER
SEE DETAIL A (F-8)
14" - SALT INLET 13.250 I.D. X 6.000 LG FORGING
12.000 .750 THK PLATE
61.000 I.D. SHELL .750 THK X 478.250 LG
.500 THK SUPPORT PLATE (26 HALVES REQ'D)
SEE DETAIL B (F-9)
4" MASTER HAND HOLE PER DWG 7145C & 29290A LOCATED ON AXES (4 REQ'D)
SEE DETAIL C (H-2)
2:1 ELLIPSOIDAL HEAD 61.000 MAJOR AXIS I.D. 30.500 MINOR AXIS I.D. .750 THK
STRAP - 1.500 X .125 THK (2 REQ'D)
4.500 THK HEMI HEAD
31.000 R
1.250 THK DIVIDER PLATE WITH 16" ACCESS COVER PLATE
14" - SALT OUTLET 13.250 I.D. X 6.000 LG FORGING
SLIDING SADDLE SUPPORT
FIXED SADDLE SUPPORT WITH STIFFENERS
SLIDING SADDLE SUPPORT
℄ WELD TYPE 2
281.200
244.200
50.750
END VIEW
MANWAY PER DWG 133083E & 137155B
20" - STEAM OUTLET 16.500 I.D. X 8.000 LG FORGING
45° TYP
14" - WATER INLET 11.500 I.D. X 8.000 LG FORGING
.500 THK CORNER PLATE (4 REQ'D)
DETAIL A (B-2) SCALE: 1/8
℄ WELD UT REQ'D WELD PER FIG. UW-13.2(d)
TIE ROD .500 DIA BAR
SEE DETAIL D (I-10)
TUBE SHEET 69.000 O.D.
DIVIDER PLATE 60.000 X .500 THK X 517.838 LG
OPEN END BAFFLE
CLOSED END BAFFLE
.125 MAX GAP TYP
DISTRIBUTION HALF CYLINDER .500 THK PLATE (2 REQ'D)
1.000 X .375 THK HALF RING (2 REQ'D)
1236 U-TUBES .875 X .148 WALL, 93 FT AVG LGTH (INCLUDING TUBE SHEET) LAST 32 FT ON HOT LEG IS MULTI-LEAD RIBBED TUBING. 5X INSPECT BUTT WELDS
℄ WELD TYPE 2
SPACER .750 X 13 GA TUBE
.500 THK SUPPORT PLATE (PARTIALLY BROACHED) (2 HALVES REQ'D)
.500 THK THERMAL SHIELD 33.625 R, .125 CLEARANCE FROM DIVIDER PLATE (UNBROACHED) 2 HALVES REQ'D
DETAIL B SCALE: 1/8
.500-13UNC NUT (18 REQ'D)
OPEN END BAFFLE
CLOSED END BAFFLE
DETAIL C SCALE: 1/8
WIGGLE BAR .500 X .171
VIEW E-E (H-2) SCALE: 1/8
EXTENT OF WIGGLE BARS
PARTIAL RING .250 THK X 1.500
VIEW G-G (E-6) DISTRIBUTION HALF CYL DEVELOPMENT
SALT OUTLET DEV LGTH = 94.350 (1 REQ'D)
SALT INLET DEV LGTH = 93.350 (1 REQ'D)
SALT OUTLET NOZZLE ONLY
3.000 DIA HOLE (BASED ON 3.375 TRIANGULAR PITCH) 98 HOLES TOTAL
℄ NOZZLE
DETAIL D (G-5) SCALE: 1/2
EXPANSION SEAL 67.500 LG
.500 TYP
CHANNEL 67.500 LG
.125 X 45°
.015 MAX CLEARANCE
NOTES
1. VESSEL SHALL BE IN ACCORDANCE WITH SECTION VIII DIVISION I OF THE ASME BOILER & PRESSURE VESSEL CODE.
2. DESIGN CONDITIONS:
DESIGN TEMP - 900°F
SALT SIDE PRESSURE - 190 PSIA
STEAM SIDE PRESSURE - 2200 PSIA
3. OPERATING CONDITIONS:
SALT IN TEMP - 838°F
SALT IN PRESSURE - 145 PSIA
SALT OUT TEMP - 637°F
SALT OUT PRESSURE - 95 PSIA
WATER IN TEMP - 630°F
WATER IN PRESSURE - 2005 PSIA
STEAM OUT TEMP - 636°F
STEAM OUT PRESSURE - 2000 PSIA
4. THE CODE BOUNDARY MATERIAL SPECIFICATIONS ARE AS FOLLOWS:
PLATE: SA-387 22,CL2
BAR: SA-336 CL.F22
FORGINGS: SA-336 CL.F22
TUBES: SA-213 T22
NOZZLES: HAND HOLE - SA-335,P22
SALT IN/OUT - SA-336 CL.F22
STEAM IN/OUT - SA-336 CL.F22
5. TOLERANCES ON ALL DIMENSION ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
6. ESTIMATED WEIGHT (DRY) = 210,000 LBS.
7. ALL PRESSURE BOUNDARY WELDS EXCEPT SHELL TO TUBE SHEET RT REQUIRED.
8. SYMBOL DESIGNATION: - REFERENCE BETWEEN NOTES AND BODY OF DWG.
9. ALL BUTT WELDS EXCEPT AS NOTED ARE TYPE 1.
10. POST WELD HEAT TREAT PER SECT. VIII, DIV.I SUBSECTION B, PARA. UW-10, UW-40 AND SUBSECTION C PARA. UCS-56.
REVISIONS
1 CROSS SECTION - EVAPORATOR 238885E
NO. TITLE DWG. NO.
REFERENCES
STEAM GENERATOR SUBSYSTEM 100 MWE
EVAPORATOR
894-0010-45
SCALE 1/20
238884E 0

FIG. 3-14



REVISIONS
1 (C-4) 2472 WAS 1238 PLACES. (H-2) REV 11GA NOTE. MAR/#K
TUBE SHEET
22.312
15.837
7.437
.500-13UNC TAPPED HOLE ON OPPOSITE SIDE OF TUBE SHEET (18 PLACES)
29.5352 R (OUTERMOST HOLE)
31.000 R
34.500 R
16.007
25.209
3.125
6.806
60.000 TYP
.890 DIA HOLE TYP 2472 PLACES (BASED ON 1.0625 TRIANGULAR PITCH)
PLATE - 56.000 X 1.500 X 30.000 (2 REQ'D)
SLIDING SADDLE SUPPORT (2 REQ'D)
4 SLOTTED HOLES TYP
1.000 THK PLATE TYP
SECTION B-B (REF 1)
(HOLE SIZE & PATTERN TYP FOR THERMAL SHIELD & SUPPORT PLATES)
SALT INLET NOZZLE
TIE ROD & SPACER
BROACH THESE ROWS ONLY 2 PLACES PATTERN "A"
25.125 TYP 28 PLACES
2.438 TYP
1.625 TYP
1.125 TYP
SEE DETAIL E (H-2)
28.500 TYP 28 PLACES
30.500 R TYP DISTRIBUTION CYL
31.062 R TYP RING
33.625 R TYP SUPPORT PLATE (2 REQ'D TOTAL) PARTIAL BROACH PATTERN TYP 14 ALTERNATING PLACES PER PATTERN "A" ABOVE
33.750 R SHELL
DRAINAGE GROOVE IN SUPPORT PLATES
SALT OUTLET NOZZLE
SECTION C-C (REF 1)
30.500 R SHELL
30.375 R TYP SUPPORT PLATE (28 REQ'D TOTAL) PARTIAL BROACH PATTERN ONLY TYP 14 ALTERNATING PLACES PER PATTERN "B"
BROACH PATTERN TYP FOR ALL SUPPORT PLATES
ST4 WF - LOCATED 12.000 ON EACH SIDE OF ℄ OF SUPPORT (2 REQ'D)
BROACH THESE ROWS ONLY TYP PATTERN "B"
BROACH THESE ROWS ONLY TYP PATTERN "B"
24° TYP
WELD
PLATE - 1.000 THK X 48.000 X 74.299 DEV LGTH FOR FIXED SADDLE SUPPORT
PLATE - 1.000 THK X 36.000 X 74.299 DEV LGTH FOR REAR SLIDING SADDLE SUPPORT
WELD TYP
PLATE - 56.000 X 1.500 X 41.500
4 HOLES
FIXED SADDLE SUPPORT
SECTION D-D (REF 1)
NOTES
1. FOR GENERAL NOTES SEE REF 1.
2. TOLERANCES ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
H (H-7)
5.250 CHANNEL WIDTH FOR 42.150 LGTH
2.000 CHANNEL WIDTH FOR REMAINDER OF CHANNEL
.500
.375-16UNC-2A X 1.500 LG HEX BOLT WITH LOCKING CLIP (56 REQ'D)
TYP
1.942 X .375 THK RIBS 8 REQ'D EQUALLY SPACED FOR FIRST 102.125" 6 REQ'D EQUALLY SPACED FOR NEXT 109.500" 4 REQ'D EQUALLY SPACED FOR NEXT 282.000"
2.500
1.000
.750
.125 TYP CLEARANCE
15°
.500
SIDE CHANNEL (2 REQ'D)
WEDGE BLOCK
11GA X 503.625 LG
.125
1.942 X .375 X 503.625 LG (4 REQ'D)
DETAIL E (C-5)
SCALE: 1/4
3.000 TYP
18.250 TYP 27 PLACES
.060 MAX GAP
1.250
WEDGE BLOCK (56 REQ'D)
FACE OF TUBE SHEET
SHELL
517.250 LGTH OF CHANNEL
517.838 LGTH OF DIVIDER PLATE
4.000
1.000
.500
TANGENT OF TUBE BEND
VIEW H-H (G-3)
SHOWING SIDE CHANNEL, WEDGE BLOCKS & DIVIDER PLATE ONLY
SCALE: 1/4
2
1 EVAPORATOR 238884E
NO. TITLE DWG. NO.
REFERENCES
STEAM GENERATOR SUBSYSTEM 100 MWE
CROSS SECTION EVAPORATOR
894-0010-45
SCALE 1/10
238885E 1

FIG. 3-15

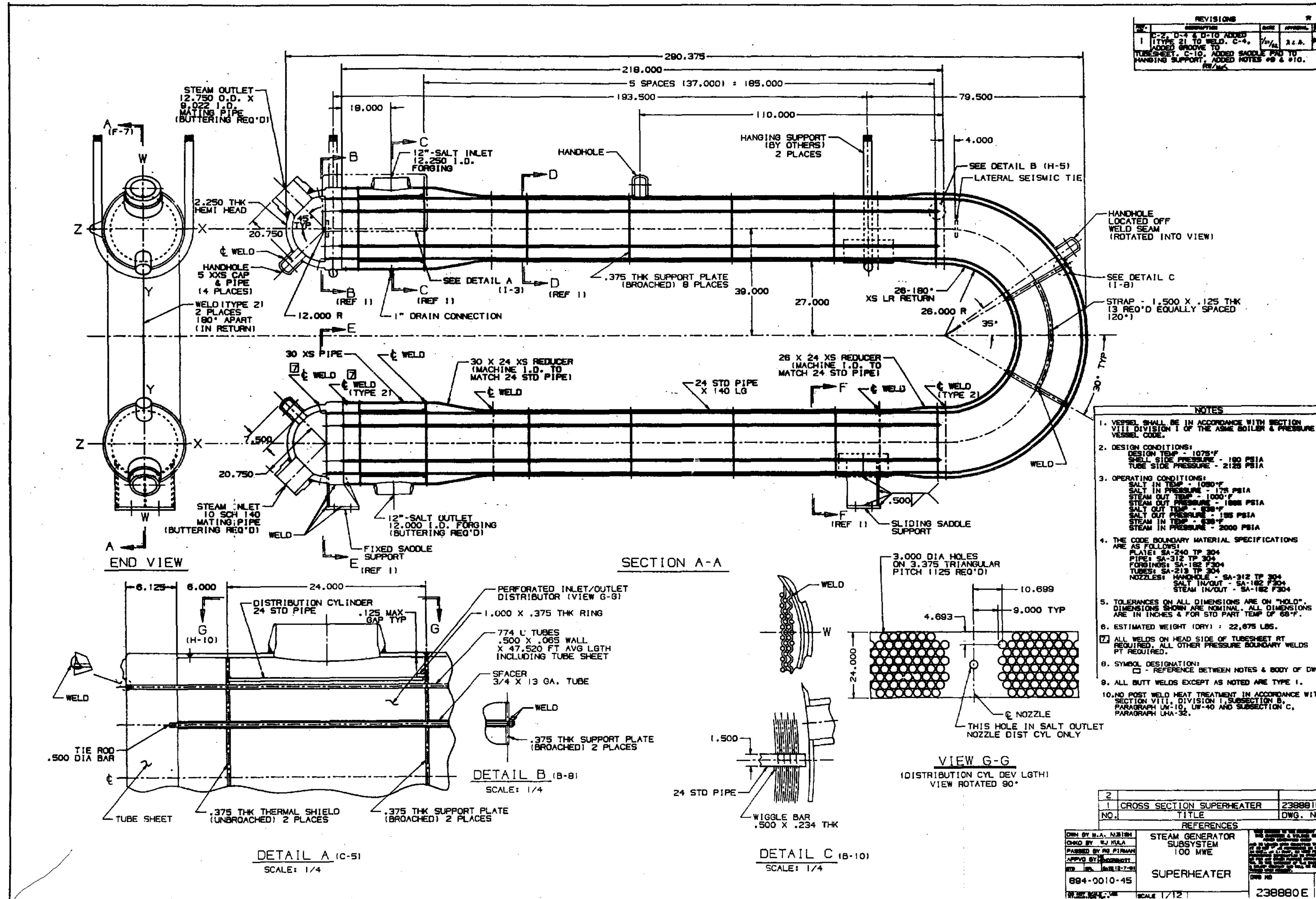
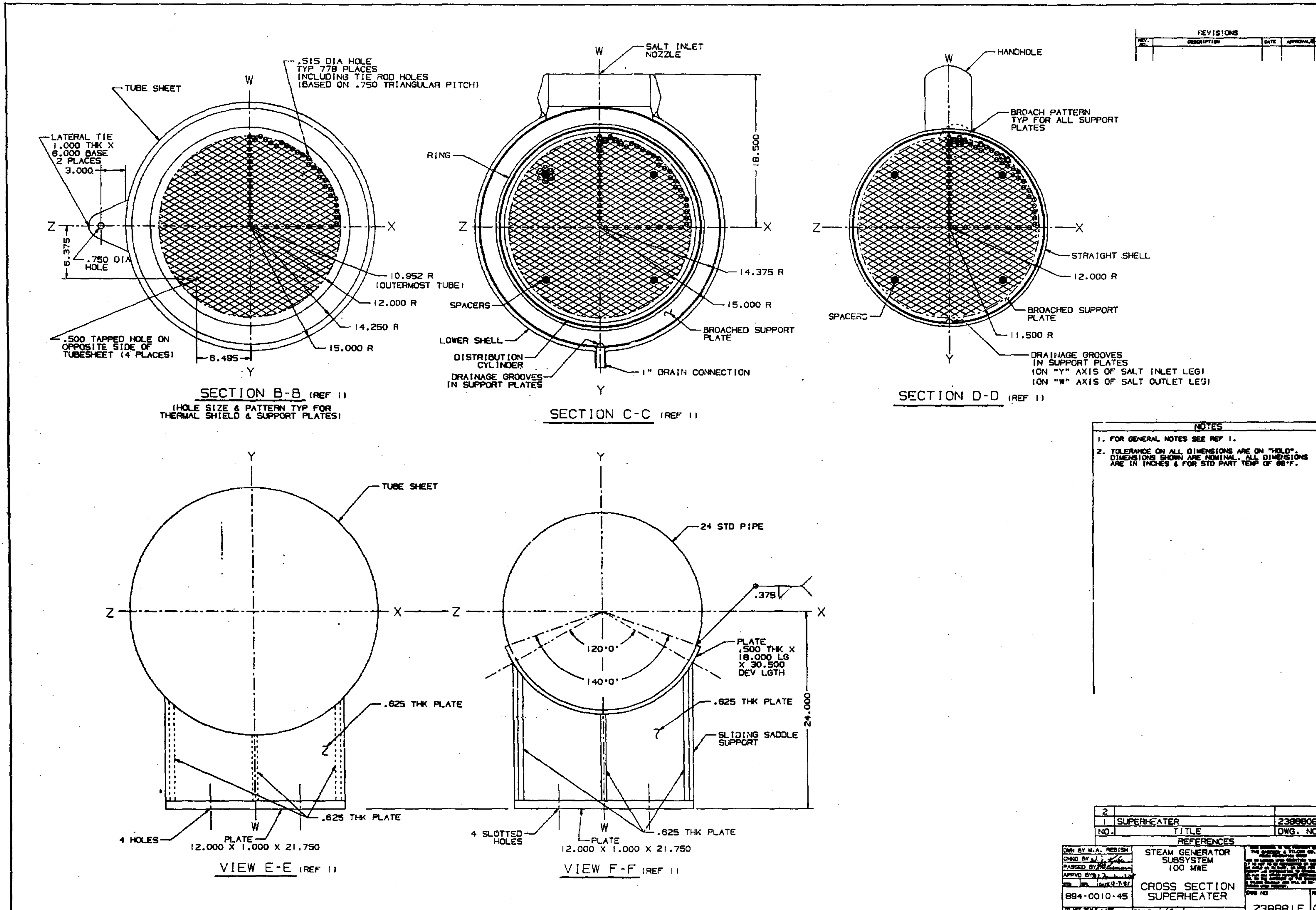

REVISIONS
C-2, D-4 & D-10 ADDED (TYPE 2) TO WELD. C-4, ADDED GROOVE TO TUBESHEET. C-10, ADDED SADDLE PAD TO HANGING SUPPORT, ADDED NOTES #9 & #10.
290.375
218.000
5 SPACES (37.000) = 185.000
193.500
79.500
110.000
18.000
4.000
STEAM OUTLET 12.750 O.D. X 9.022 I.D. MATING PIPE (BUTTERING REQ'D)
12"-SALT INLET 12.250 I.D. FORGING
HANDHOLE
HANGING SUPPORT (BY OTHERS) 2 PLACES
SEE DETAIL B (H-5)
LATERAL SEISMIC TIE
2.250 THK HEMI HEAD
20.750
℄ WELD
HANDHOLE 5 XXS CAP & PIPE (4 PLACES)
WELD (TYPE 2) 2 PLACES 180° APART (IN RETURN)
12.000 R
SEE DETAIL A (I-3)
1" DRAIN CONNECTION
.375 THK SUPPORT PLATE (BROACHED) 8 PLACES
39.000
27.000
26-180° XS LR RETURN
26.000 R
35°
HANDHOLE LOCATED OFF WELD SEAM (ROTATED INTO VIEW)
SEE DETAIL C (I-8)
STRAP - 1.500 X .125 THK (3 REQ'D EQUALLY SPACED 120°)
30° TYP
30 XS PIPE
℄ WELD
℄ WELD (TYPE 2)
30 X 24 XS REDUCER (MACHINE I.D. TO MATCH 24 STD PIPE)
24 STD PIPE X 140 LG
26 X 24 XS REDUCER (MACHINE I.D. TO MATCH 24 STD PIPE)
℄ WELD (TYPE 2)
7.500
20.750
STEAM INLET 10 SCH 140 MATING PIPE (BUTTERING REQ'D)
WELD
12"-SALT OUTLET 12.000 I.D. FORGING (BUTTERING REQ'D)
FIXED SADDLE SUPPORT
SLIDING SADDLE SUPPORT
.500
WELD
END VIEW
SECTION A-A
NOTES
1. VESSEL SHALL BE IN ACCORDANCE WITH SECTION VIII DIVISION I OF THE ASME BOILER & PRESSURE VESSEL CODE.
2. DESIGN CONDITIONS: DESIGN TEMP - 1075°F SHELL SIDE PRESSURE - 190 PSIA TUBE SIDE PRESSURE - 2125 PSIA
3. OPERATING CONDITIONS: SALT IN TEMP - 1050°F SALT IN PRESSURE - 175 PSIA STEAM OUT TEMP - 1000°F STEAM OUT PRESSURE - 1885 PSIA SALT OUT TEMP - 838°F SALT OUT PRESSURE - 155 PSIA STEAM IN TEMP - 636°F STEAM IN PRESSURE - 2000 PSIA
4. THE CODE BOUNDARY MATERIAL SPECIFICATIONS ARE AS FOLLOWS: PLATE: SA-240 TP 304 PIPE: SA-312 TP 304 FORGINGS: SA-182 F304 TUBES: SA-213 TP 304 NOZZLES: HANDHOLE - SA-312 TP 304 SALT IN/OUT - SA-182 F304 STEAM IN/OUT - SA-182 F304
5. TOLERANCES ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
6. ESTIMATED WEIGHT (DRY) : 22,675 LBS.
7. ALL WELDS ON HEAD SIDE OF TUBESHEET RT REQUIRED. ALL OTHER PRESSURE BOUNDARY WELDS PT REQUIRED.
8. SYMBOL DESIGNATION: □ - REFERENCE BETWEEN NOTES & BODY OF DWG
9. ALL BUTT WELDS EXCEPT AS NOTED ARE TYPE 1.
10. NO POST WELD HEAT TREATMENT IN ACCORDANCE WITH SECTION VIII, DIVISION I, SUBSECTION B, PARAGRAPH UW-10, UW-40 AND SUBSECTION C, PARAGRAPH UHA-32.
6.125
6.000
24.000
DISTRIBUTION CYLINDER 24 STD PIPE
.125 MAX GAP TYP
PERFORATED INLET/OUTLET DISTRIBUTOR (VIEW G-G)
1.000 X .375 THK RING
774 U TUBES .500 X .065 WALL X 47.520 FT AVG LGTH INCLUDING TUBE SHEET
SPACER 3/4 X 13 GA. TUBE
TIE ROD .500 DIA BAR
TUBE SHEET
.375 THK THERMAL SHIELD (UNBROACHED) 2 PLACES
.375 THK SUPPORT PLATE (BROACHED) 2 PLACES
DETAIL A (C-5)
SCALE: 1/4
.375 THK SUPPORT PLATE (BROACHED) 2 PLACES
DETAIL B (B-8)
SCALE: 1/4
1.500
24 STD PIPE
WIGGLE BAR .500 X .234 THK
DETAIL C (B-10)
SCALE: 1/4
3.000 DIA HOLES ON 3.375 TRIANGULAR PITCH (125 REQ'D)
10.699
9.000 TYP
4.693
24.000
℄ NOZZLE
THIS HOLE IN SALT OUTLET NOZZLE DIST CYL ONLY
VIEW G-G
(DISTRIBUTION CYL DEV LGTH) VIEW ROTATED 90°
1 CROSS SECTION SUPERHEATER 238881E
NO. TITLE DWG. NO.
REFERENCES
STEAM GENERATOR SUBSYSTEM 100 MWE
SUPERHEATER
894-0010-45
SCALE 1/12
238880E

FIG. 3-16



REVISIONS
.515 DIA HOLE
TYP 778 PLACES
INCLUDING TIE ROD HOLES
(BASED ON .750 TRIANGULAR PITCH)
TUBE SHEET
LATERAL TIE
1.000 THK X
6.000 BASE
2 PLACES
3.000
6.375
.750 DIA
HOLE
10.952 R
(OUTERMOST TUBE)
12.000 R
14.250 R
15.000 R
.500 TAPPED HOLE ON
OPPOSITE SIDE OF
TUBESHEET (4 PLACES)
6.495
SECTION B-B (REF 1)
(HOLE SIZE & PATTERN TYP FOR
THERMAL SHIELD & SUPPORT PLATES)
SALT INLET
NOZZLE
RING
18.500
14.375 R
15.000 R
SPACERS
BROACHED SUPPORT
PLATE
LOWER SHELL
DISTRIBUTION
CYLINDER
DRAINAGE GROOVES
IN SUPPORT PLATES
1" DRAIN CONNECTION
SECTION C-C (REF 1)
HANDHOLE
BROACH PATTERN
TYP FOR ALL SUPPORT
PLATES
STRAIGHT SHELL
12.000 R
BROACHED SUPPORT
PLATE
SPACERS
11.500 R
DRAINAGE GROOVES
IN SUPPORT PLATES
(ON "Y" AXIS OF SALT INLET LEG)
(ON "W" AXIS OF SALT OUTLET LEG)
SECTION D-D (REF 1)
NOTES
1. FOR GENERAL NOTES SEE REF 1.
2. TOLERANCE ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
TUBE SHEET
.625 THK PLATE
.625 THK PLATE
4 HOLES
PLATE
12.000 X 1.000 X 21.750
VIEW E-E (REF 1)
24 STD PIPE
.375
120°0'
140°0'
PLATE
.500 THK X
18.000 LG
X 30.500
DEV LGTH
.625 THK PLATE
SLIDING SADDLE
SUPPORT
24.000
4 SLOTTED
HOLES
PLATE
12.000 X 1.000 X 21.750
.625 THK PLATE
VIEW F-F (REF 1)
2
1 SUPERHEATER 238880E
NO. TITLE DWG. NO.
REFERENCES
DWN BY M.A. REBISH
CHKD BY
PASSED BY
APPVD BY
DATE 12-7-81
894-0010-45
DO NOT SCALE - USE DIMENSIONS ONLY.
STEAM GENERATOR
SUBSYSTEM
100 MWE
CROSS SECTION
SUPERHEATER
SCALE 1/4
DWG NO
238881E
REV
0

FIG. 3-17

3.1.4 Reheater

The reheater is also a horizontal U-tube/U-shell heat exchanger, as illustrated in Figures 3-18 and 3-19. The water and salt-side flow schemes are similar to those described for the superheater.

The reheater meets the same functional design requirements as those described for the superheater with the following exception: preferentially broached tube support plates are necessary to assure salt mixing and enhance heat transfer.

The reheater meets the same structural design requirements as those described for the superheater.

Steam and salt-side inspection openings are similar to those utilized in the superheater.

3.1.5 Steam Drum

The steam drum is a standard Babcock & Wilcox fossil steam drum design employing conventional cyclone steam separators and scrubbers, as illustrated in Figures 3-20 and 3-21. The high quality steam/water mixture from the evaporator is delivered to the drum where the phases are separated. The saturated steam is delivered to the superheater, while the saturated water is mixed with incoming preheated feedwater and returned via the downcomer to the evaporator.

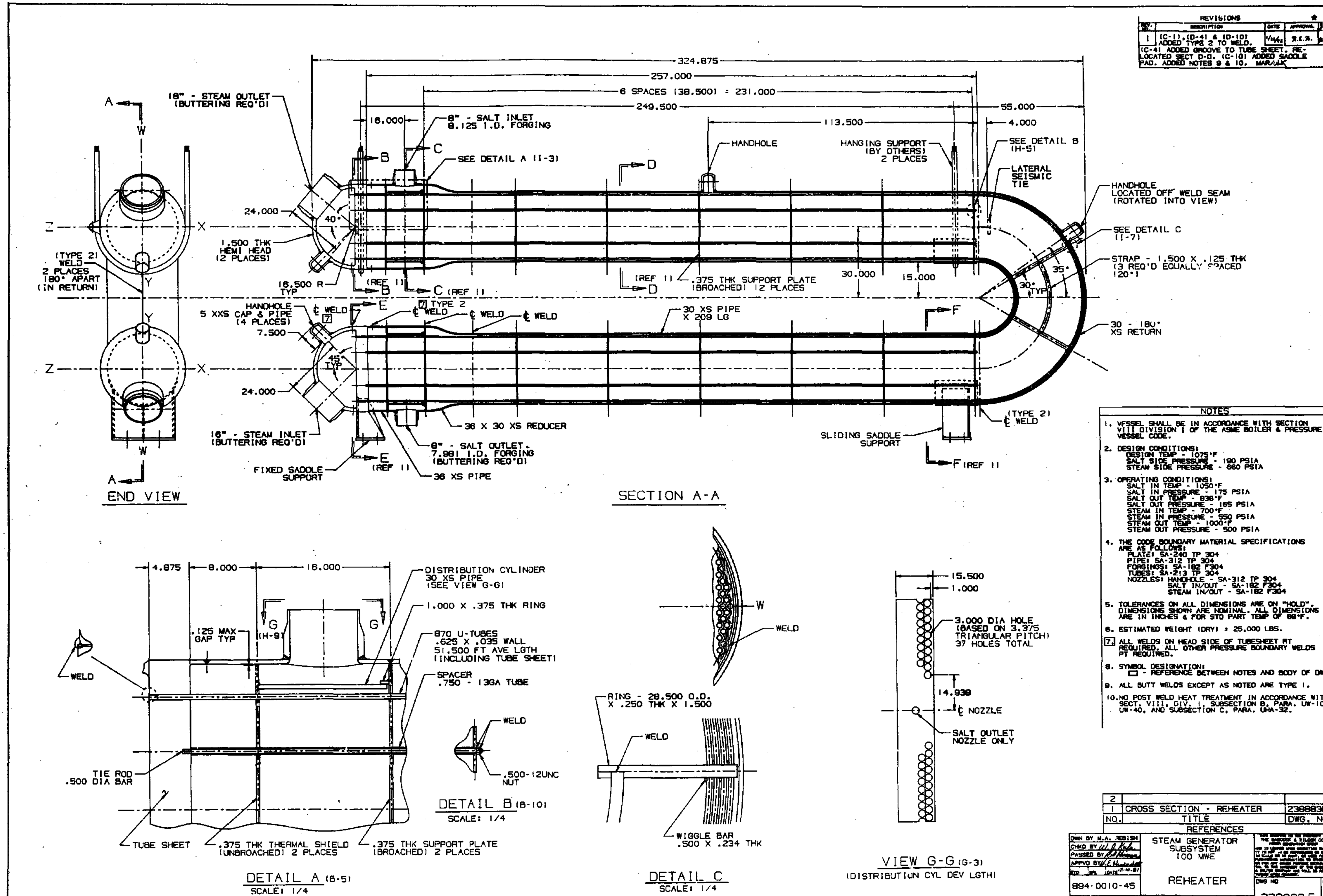

REVISIONS
1 (C-1), (D-4) & (D-10) ADDED TYPE 2 TO WELD. (C-4) ADDED GROOVE TO TUBE SHEET. RELOCATED SECT D-D. (C-10) ADDED SADDLE PAD. ADDED NOTES 9 & 10.
324.875
257.000
6 SPACES (38.500) = 231.000
249.500
55.000
4.000
113.500
16.000
18" - STEAM OUTLET (BUTTERING REQ'D)
8" - SALT INLET 8.125 I.D. FORGING
SEE DETAIL A (I-3)
HANDHOLE
HANGING SUPPORT (BY OTHERS) 2 PLACES
SEE DETAIL B (H-5)
LATERAL SEISMIC TIE
HANDHOLE LOCATED OFF WELD SEAM (ROTATED INTO VIEW)
SEE DETAIL C (I-7)
STRAP - 1.500 X .125 THK (3 REQ'D EQUALLY SPACED (20°)
24.000
40°
1.500 THK HEMI HEAD (2 PLACES)
16.500 R TYP
(TYPE 2) WELD 2 PLACES 180° APART (IN RETURN)
(REF 1)
C (REF 1)
(REF 1)
.375 THK SUPPORT PLATE (BROACHED) 12 PLACES
30.000
15.000
35°
30° TYP
HANDHOLE 5 XXS CAP & PIPE (4 PLACES)
℄ WELD
TYPE 2 ℄ WELD
℄ WELD
℄ WELD
30 XS PIPE X 209 LG
30 - 180° XS RETURN
7.500
45° TYP
24.000
18" - STEAM INLET (BUTTERING REQ'D)
36 X 30 XS REDUCER
8" - SALT OUTLET, 7.981 I.D. FORGING (BUTTERING REQ'D)
FIXED SADDLE SUPPORT
36 XS PIPE
SLIDING SADDLE SUPPORT
(TYPE 2) ℄ WELD
F (REF 1)
END VIEW
SECTION A-A
4.875
8.000
16.000
.125 MAX GAP TYP
(H-9)
DISTRIBUTION CYLINDER 30 XS PIPE (SEE VIEW G-G)
1.000 X .375 THK RING
870 U-TUBES .625 X .035 WALL 51.500 FT AVE LGTH (INCLUDING TUBE SHEET)
SPACER .750 - 13GA TUBE
WELD
TIE ROD .500 DIA BAR
TUBE SHEET
.375 THK THERMAL SHIELD (UNBROACHED) 2 PLACES
.375 THK SUPPORT PLATE (BROACHED) 2 PLACES
DETAIL A (B-5)
SCALE: 1/4
WELD
.500-12UNC NUT
DETAIL B (B-10)
SCALE: 1/4
WELD
RING - 28.500 O.D. X .250 THK X 1.500
WELD
WIGGLE BAR .500 X .234 THK
DETAIL C
SCALE: 1/4
15.500
1.000
3.000 DIA HOLE (BASED ON 3.375 TRIANGULAR PITCH) 37 HOLES TOTAL
14.938
℄ NOZZLE
SALT OUTLET NOZZLE ONLY
VIEW G-G (G-3)
(DISTRIBUTION CYL DEV LGTH)
NOTES
1. VESSEL SHALL BE IN ACCORDANCE WITH SECTION VIII DIVISION I OF THE ASME BOILER & PRESSURE VESSEL CODE.
2. DESIGN CONDITIONS:
DESIGN TEMP - 1075°F
SALT SIDE PRESSURE - 190 PSIA
STEAM SIDE PRESSURE - 660 PSIA
3. OPERATING CONDITIONS:
SALT IN TEMP - 1050°F
SALT IN PRESSURE - 175 PSIA
SALT OUT TEMP - 838°F
SALT OUT PRESSURE - 165 PSIA
STEAM IN TEMP - 700°F
STEAM IN PRESSURE - 550 PSIA
STEAM OUT TEMP - 1000°F
STEAM OUT PRESSURE - 500 PSIA
4. THE CODE BOUNDARY MATERIAL SPECIFICATIONS ARE AS FOLLOWS:
PLATE: SA-240 TP 304
PIPE: SA-312 TP 304
FORGINGS: SA-182 F304
TUBES: SA-213 TP 304
NOZZLES: HANDHOLE - SA-312 TP 304
SALT IN/OUT - SA-182 F304
STEAM IN/OUT - SA-182 F304
5. TOLERANCES ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
6. ESTIMATED WEIGHT (DRY) = 25,000 LBS.
7. ALL WELDS ON HEAD SIDE OF TUBESHEET RT REQUIRED. ALL OTHER PRESSURE BOUNDARY WELDS PT REQUIRED.
8. SYMBOL DESIGNATION:
☐ - REFERENCE BETWEEN NOTES AND BODY OF DWG
9. ALL BUTT WELDS EXCEPT AS NOTED ARE TYPE 1.
10. NO POST WELD HEAT TREATMENT IN ACCORDANCE WITH SECT. VIII, DIV. 1, SUBSECTION B, PARA. UW-10, UW-40, AND SUBSECTION C, PARA. UHA-32.
2
1 CROSS SECTION - REHEATER 238883E
NO. TITLE DWG. NO.
REFERENCES
DWN BY M.A. REBISH
CHKD BY
PASSED BY
APPVD BY
DATE 12-10-81
STEAM GENERATOR SUBSYSTEM 100 MWE
REHEATER
894-0010-45
SCALE 1/14
DWG NO 238882 E
REV 1

FIG. 3-18

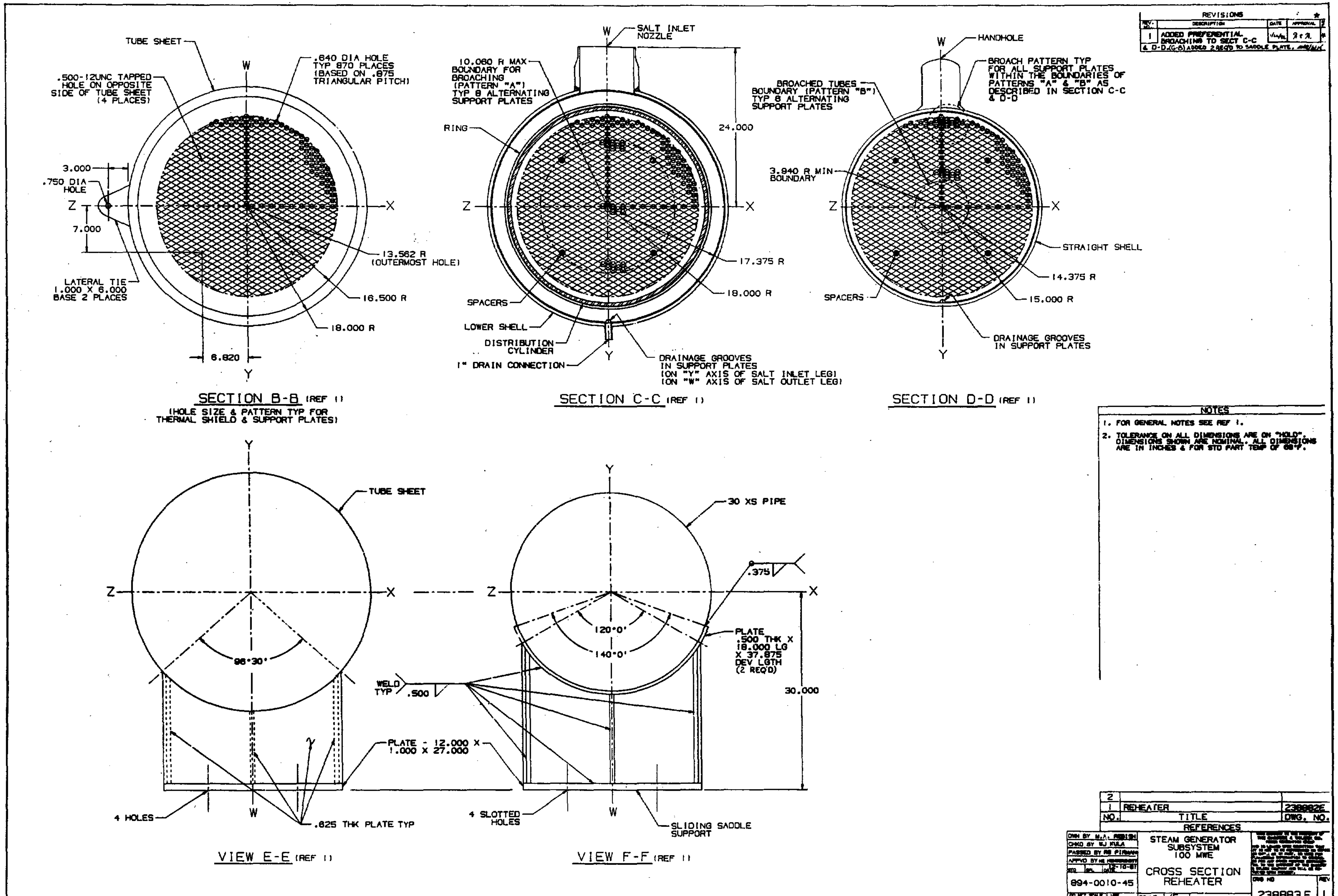

REVISIONS
1 ADDED PREFERENTIAL BROACHING TO SECT C-C & D-D (C-8) ADDED 2 REQ'D TO SADDLE PLATE
TUBE SHEET
W
.500-12UNC TAPPED HOLE ON OPPOSITE SIDE OF TUBE SHEET (4 PLACES)
.640 DIA HOLE TYP 870 PLACES (BASED ON .875 TRIANGULAR PITCH)
3.000
.750 DIA HOLE
Z
X
7.000
13.562 R (OUTERMOST HOLE)
LATERAL TIE 1.000 X 6.000 BASE 2 PLACES
16.500 R
18.000 R
6.820
Y
SECTION B-B (REF 1)
(HOLE SIZE & PATTERN TYP FOR THERMAL SHIELD & SUPPORT PLATES)
SALT INLET NOZZLE
10.060 R MAX BOUNDARY FOR BROACHING (PATTERN "A") TYP 8 ALTERNATING SUPPORT PLATES
RING
24.000
17.375 R
18.000 R
SPACERS
LOWER SHELL
DISTRIBUTION CYLINDER
1" DRAIN CONNECTION
DRAINAGE GROOVES IN SUPPORT PLATES (ON "Y" AXIS OF SALT INLET LEG) (ON "W" AXIS OF SALT OUTLET LEG)
SECTION C-C (REF 1)
HANDHOLE
BROACHED TUBES BOUNDARY (PATTERN "B") TYP 8 ALTERNATING SUPPORT PLATES
BROACH PATTERN TYP FOR ALL SUPPORT PLATES WITHIN THE BOUNDARIES OF PATTERNS "A" & "B" AS DESCRIBED IN SECTION C-C & D-D
3.940 R MIN BOUNDARY
STRAIGHT SHELL
14.375 R
15.000 R
DRAINAGE GROOVES IN SUPPORT PLATES
SECTION D-D (REF 1)
NOTES
1. FOR GENERAL NOTES SEE REF 1.
2. TOLERANCE ON ALL DIMENSIONS ARE ON "HOLD". DIMENSIONS SHOWN ARE NOMINAL. ALL DIMENSIONS ARE IN INCHES & FOR STD PART TEMP OF 68°F.
TUBE SHEET
96°30'
4 HOLES
PLATE - 12.000 X 1.000 X 27.000
.625 THK PLATE TYP
VIEW E-E (REF 1)
30 XS PIPE
.375
120°0'
140°0'
PLATE .500 THK X 18.000 LG X 37.875 DEV LGTH (2 REQ'D)
WELD TYP .500
30.000
4 SLOTTED HOLES
SLIDING SADDLE SUPPORT
VIEW F-F (REF 1)
1 REHEATER 238882E
NO. TITLE DWG. NO.
REFERENCES
STEAM GENERATOR SUBSYSTEM 100 MWE
CROSS SECTION REHEATER
894-0010-45
SCALE 1/5
238883 E 1

FIG. 3-19

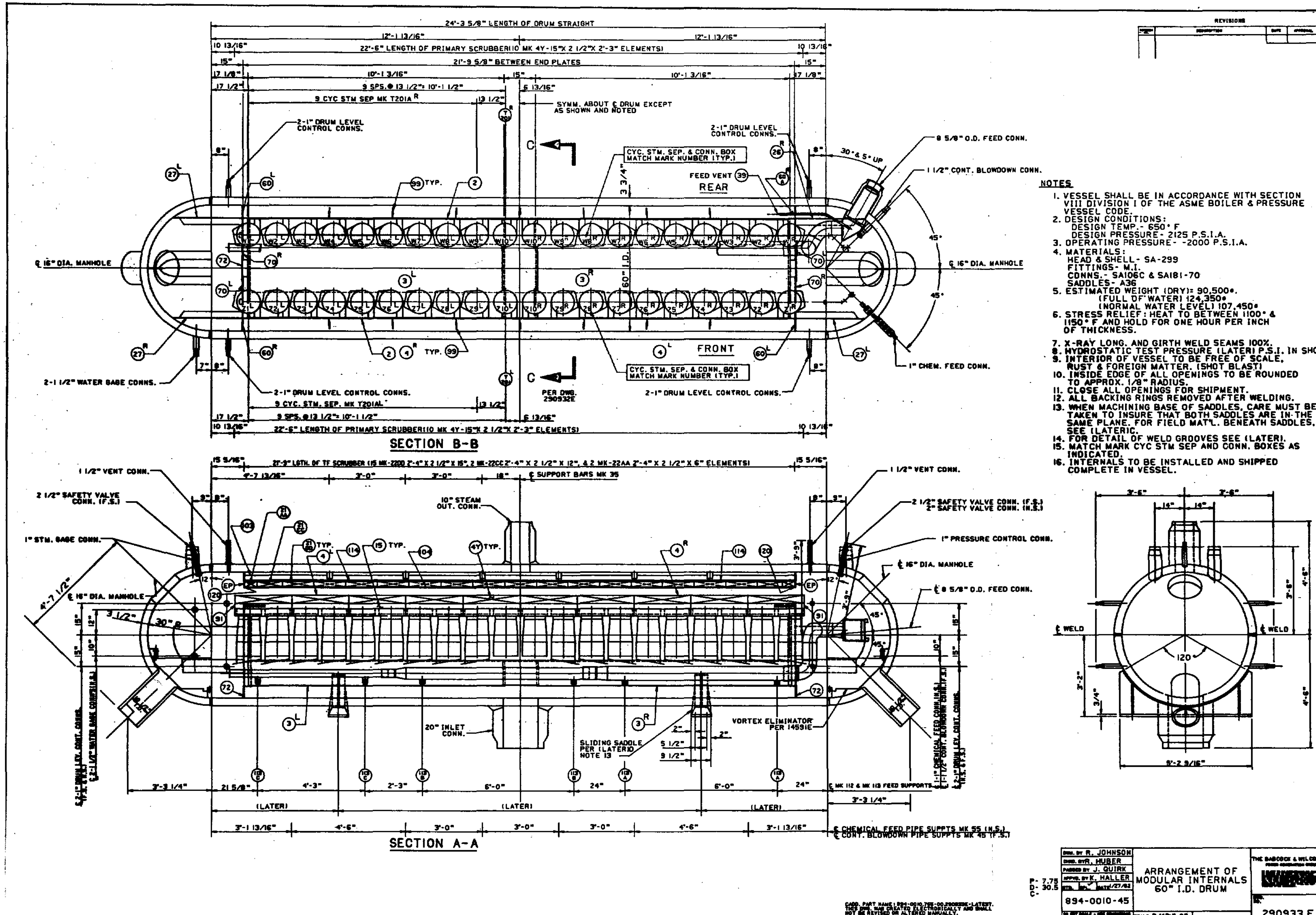


FIG. 3-20 100 MEGAWATT STEAM DRUM, LONGITUDINAL SECTION

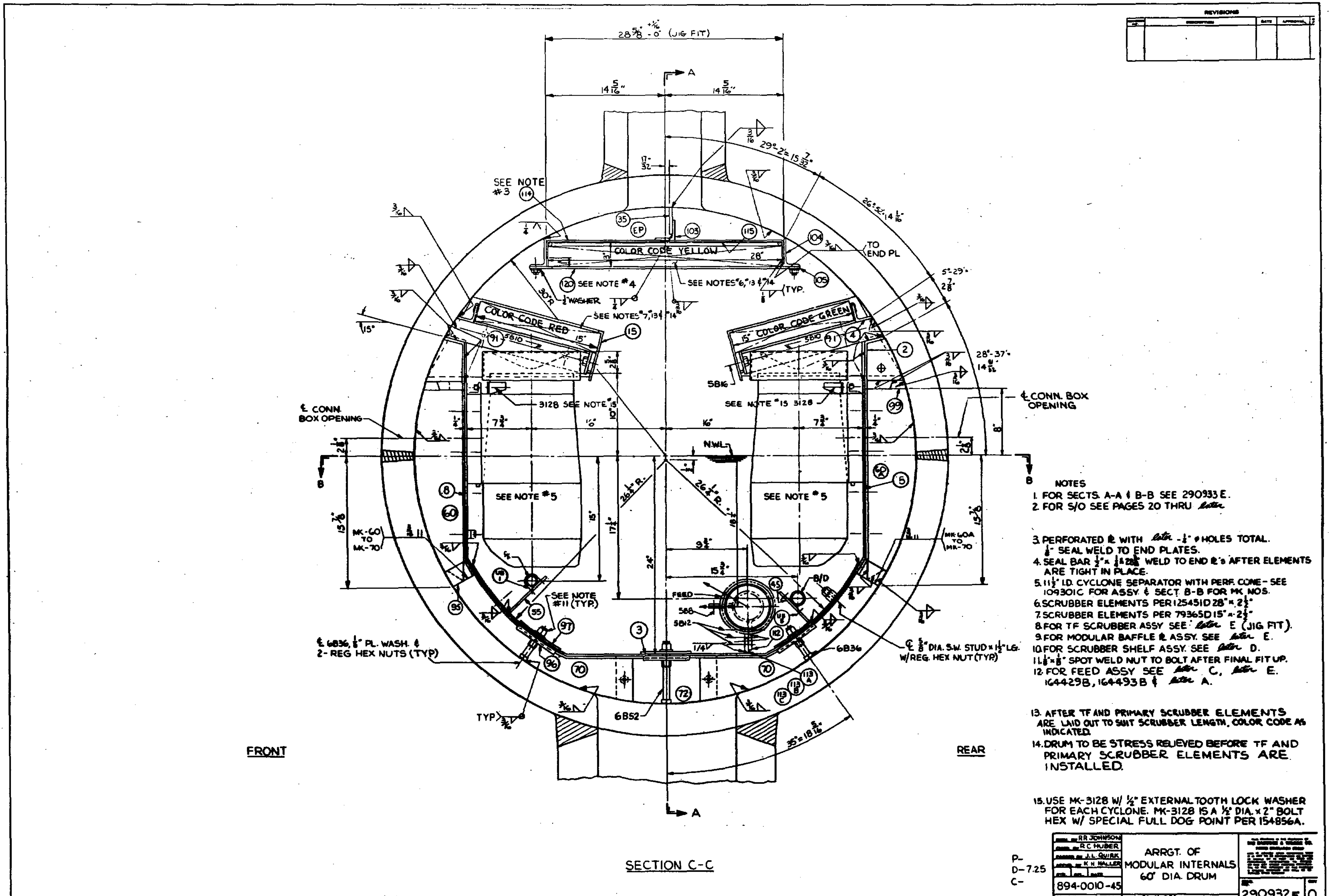


FIG. 3-21 100 MEGAWATT STEAM DRUM, CROSS SECTION

3.2 Material

Materials were selected to provide adequate strength and corrosion resistance in the operating environments with the additional objective to minimize base material costs.

3.2.1 Material Selection

Carbon steel was selected for the low temperature service of the preheater and steam drum. Carbon steel provides favorable strength and acceptable corrosion resistance at temperatures below the maximum operating temperature of 336° C (637°F).

The evaporator operates in a higher temperature environment (normally as high as 448° C (838°F)); consequently the increased mechanical strength and corrosion resistance of 2 1/4 Cr-1 Mo was considered necessary.

The superheater and reheater operate in temperature environments up to 566°C (1050°F). Consequently, type 304 stainless steel was selected to provide satisfactory strength and corrosion resistance in the high temperature environments. Incoloy 800 is also an excellent material for high temperature applications. However, economic penalties imposed by the use of high cost material in this application are not offset by advantages accruing from its superior strength and corrosion resistance.

The mechanical strength of the component alloys are compared in Figure 3-22. The corrosion rate data upon which corrosion allowances were based are described in Section 3.2.2. The available data pertaining to the corrosion resistance of the selected alloys, particularly the low-chromium alloys, in molten nitrate salt is very limited and often exhibits wide scatter. Thus, it will be prudent to reassess the material choices based on the results of on-going and future test programs. The need for further corrosion testing is discussed in Section 7.4. The results of sensitivity studies made to assess the potential impact of material changes on component design are presented in Section 3.2.3.

3.2.2. Corrosion Allowances

Table 3-2 summarizes the corrosion allowances used in sizing the heat transfer surface for the steam generator components.

TABLE 3-2

Material Corrosion Allowances

Component	Material	Maximum Operating Temperaures		Corrosion Rates	
		Salt-Side °C(°F)	Water-Side °C(°F)	Salt-Side MMPY(MPY)	Water-Side MMPY(MPY)
Preheater	Crbn Stl	336(637)	330(627)	.008(.300)	.009(.360)
Evaporator	2 1/4 CR-1 MO	448(838)	336(636)	.030(1.200)	.013(.530)
Superheater	304 Stnless Steel	566(1050)	538(1000)	.005(.200)	.003(.130)
Reheater	304 Stnless Steel	566(1050)	538(1000)	.005(.200)	.003(.130)

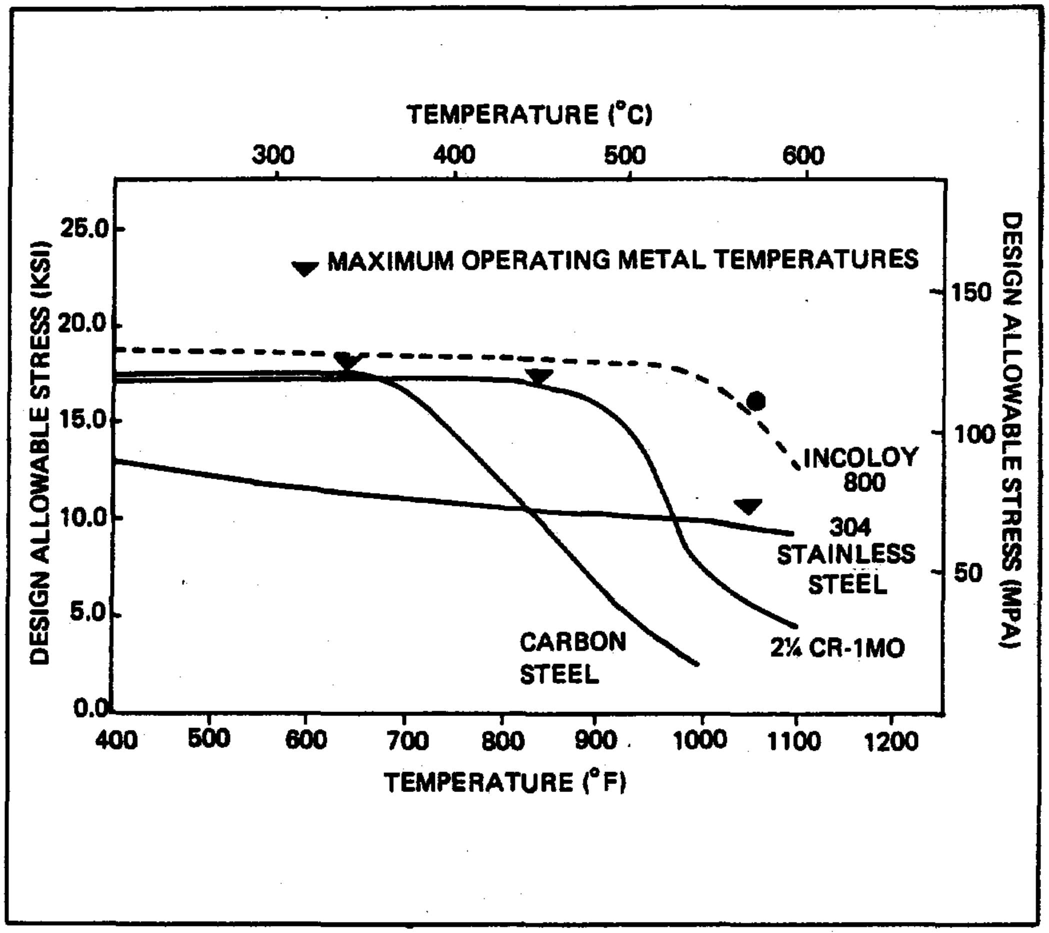


FIG. 3-22 DESIGN ALLOWABLES VERSUS TEMPERATURE

Limited data are available for salt-side corrosion rates; however, data presented in Reference 1 was used to establish salt-side corrosion rates for the carbon steel component, and static immersion test data per Reference 2 was used to establish salt-side corrosion rates for the 2 1/4 Cr-1 Mo and 304 stainless steel components. Significant data are available for water/steam corrosion rates and these results are presented in Figures 3-23 through 3-25.

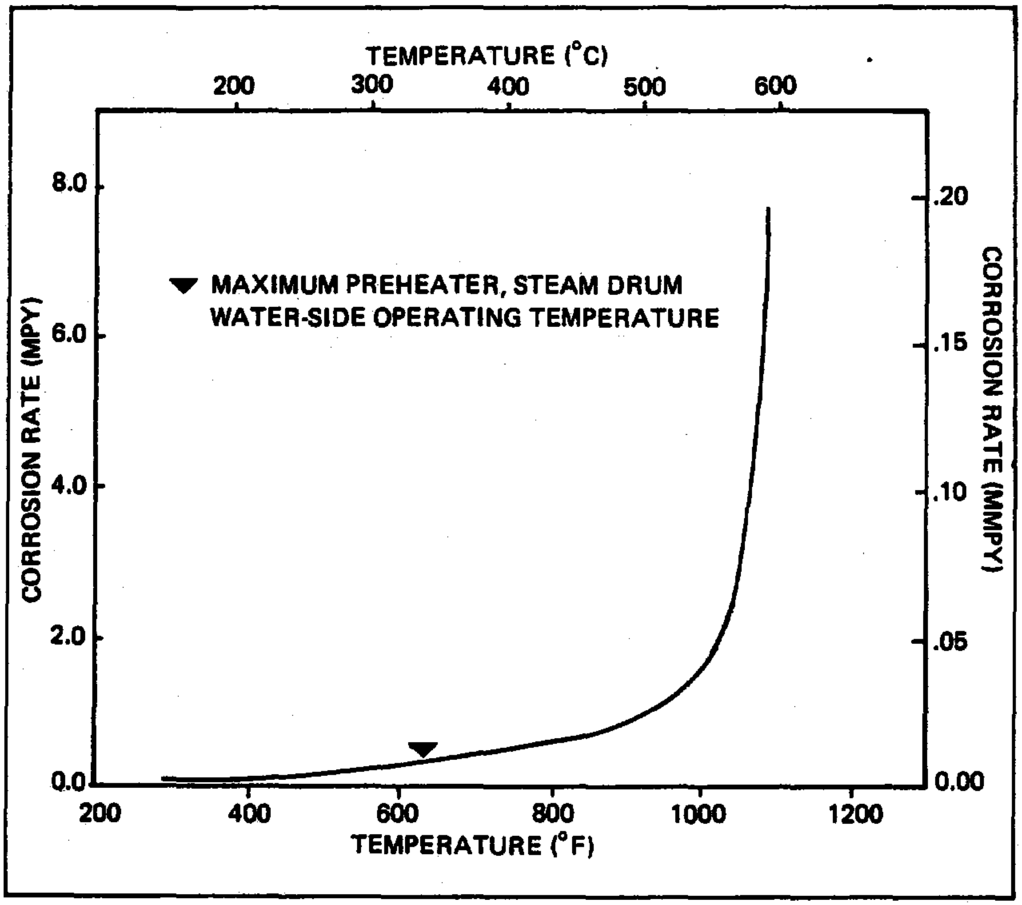


FIG. 3-23 WATER-SIDE CORROSION RATE FOR CARBON STEEL

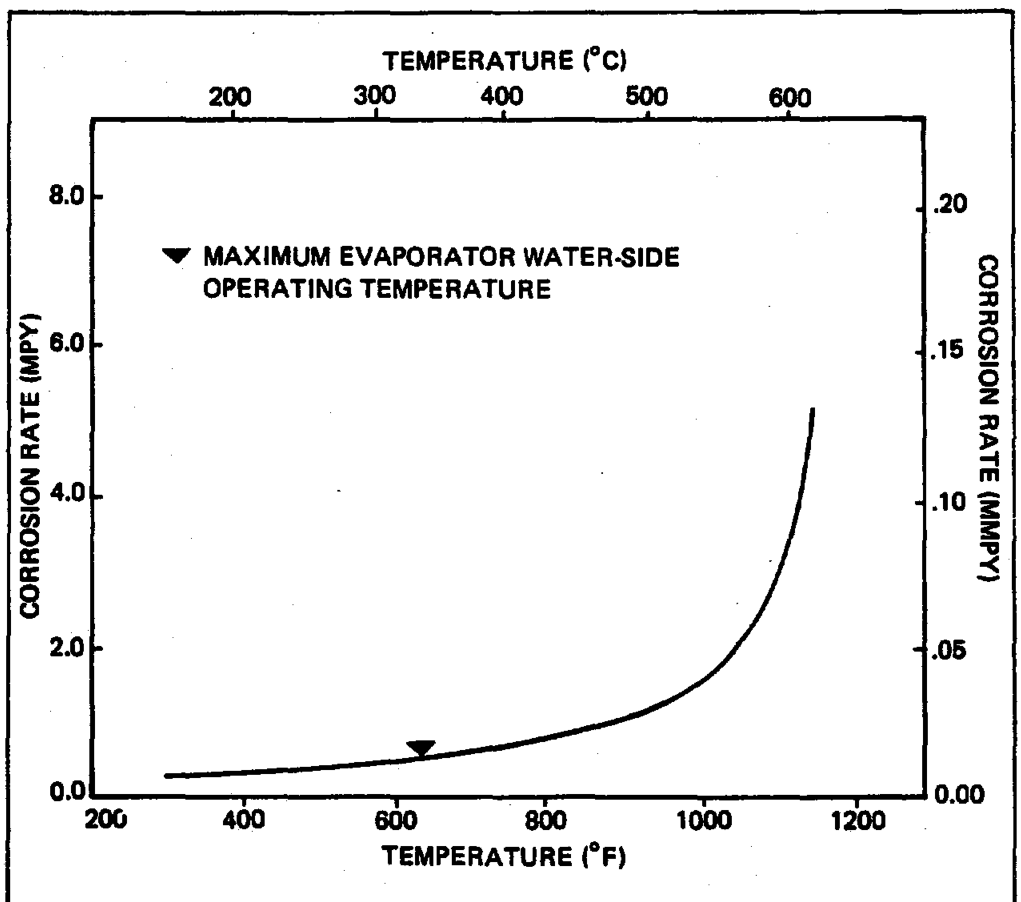


FIG. 3-24 WATER-SIDE CORROSION RATE FOR 2¼ CR-IMO

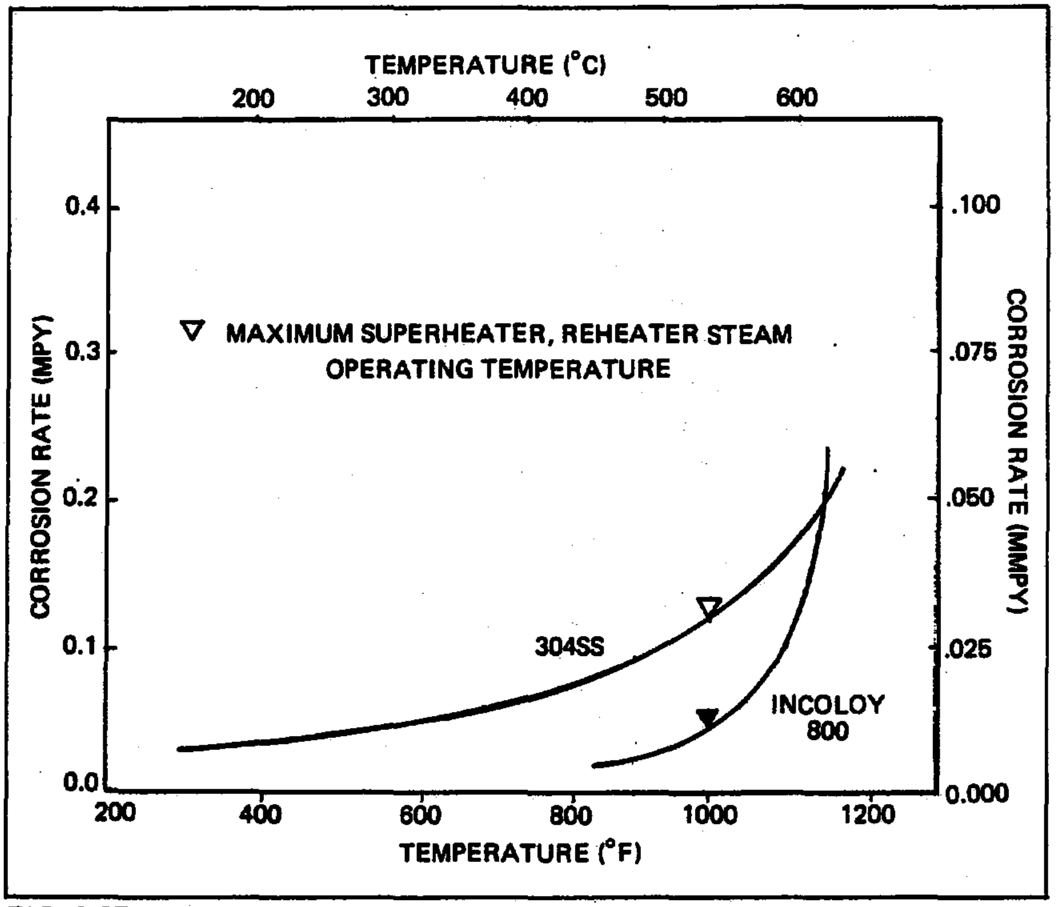


FIG. 3-25 STEAM-SIDE CORROSION RATE FOR 304 STAINLESS STEEL AND INCOLOY 800

3.2.3 Material Sensitivity Studies

Two sensitivity studies were made to assess the potential impact of material changes on the component designs. Such changes would be required if on-going and future test programs demonstrate that either the selected materials are not suitable for the intended service or that the assigned salt-side corrosion allowances are not sufficient for the operating temperatures.

In the first study, 5 Cr-1/2 Mo, 9 Cr-1 Mo, and 13 Cr were considered as potential substitutes for the 2 1/4 Cr - 1 Mo tubes of the evaporator. As the chromium content of the alloy is increased, resistance to general corrosion is increased and thermal conductivity is decreased. Thus, the overall heat transfer coefficient is reduced, resulting in a need to increase the heat transfer surface if performance is to be maintained. The results of the analysis are shown in Figure 3-26.

In the second study, the effect of increasing the salt-side corrosion allowance of the type 304 stainless steel reheater and superheater tubes from 0.0002 inch/year (0.005 mm/year) to 0.0005 inch/year (0.013 mm/year) and 0.001 inch/year (0.025 mm/year) was investigated. As the corrosion allowance, and thus the tube wall thickness, is increased, the overall heat transfer coefficient is reduced. The heat transfer surface must then be increased if performance is to be maintained. The results of the analysis are shown in Figure 3-27.

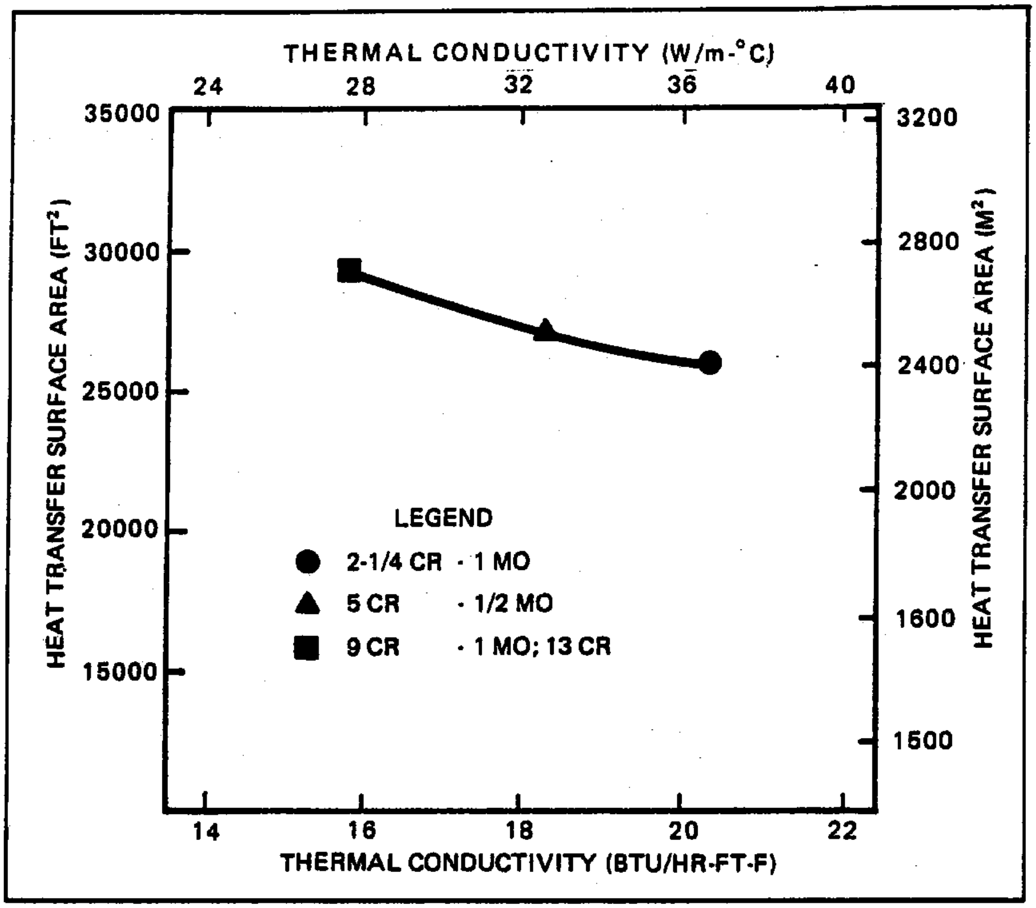


FIG. 3-26 EFFECT OF TUBE MATERIAL ON EVAPORATOR HEAT TRANSFER SURFACE

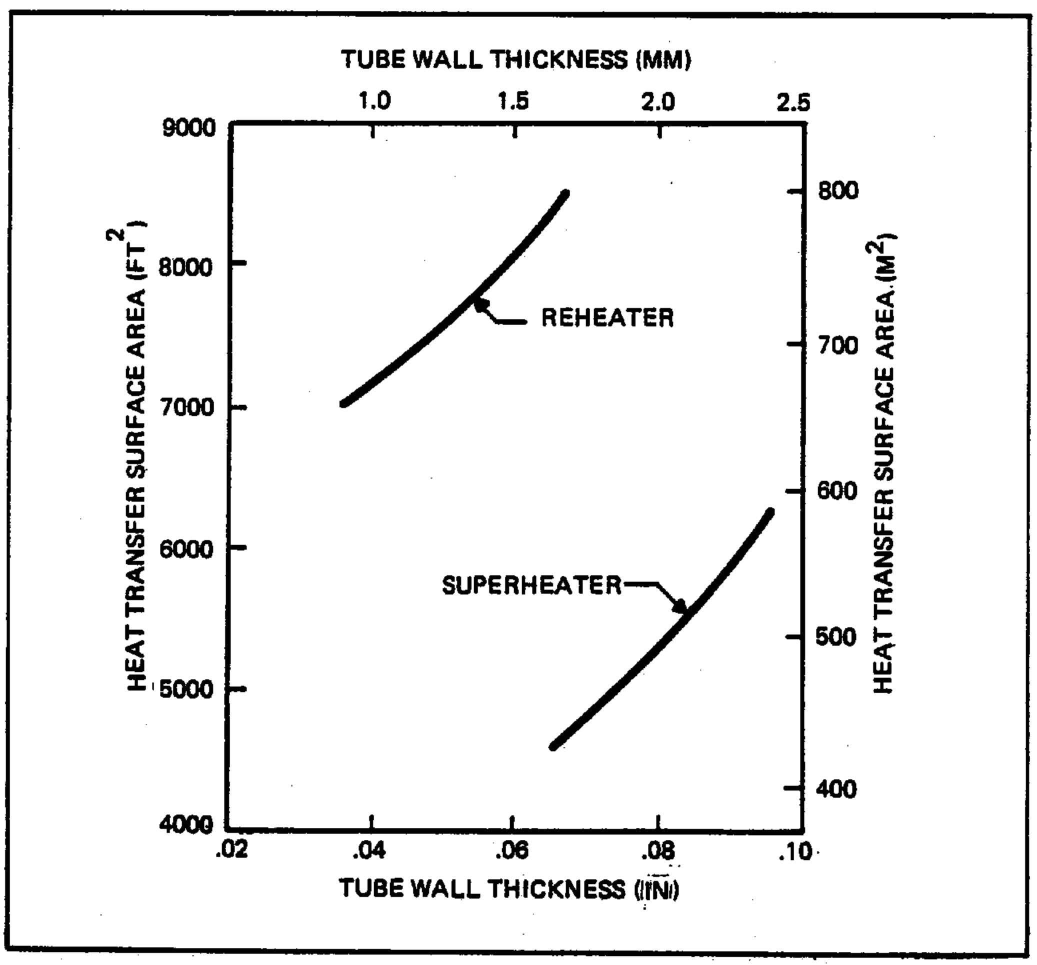


FIG. 3-27 EFFECT OF TUBE WALL THICKNESS ON SUPERHEATER AND REHEATER HEAT TRANSFER SURFACE

3.3 Thermal-Hydraulic Design

The steam generator performance characteristics are summarized in Table 3-3. These characteristics represent subsystem boundary conditions upon which system performance analyses and component sizing and design calculations were based.

The principal design analyses were accomplished using the Babcock & Wilcox VAGEN computer code (Reference 3). This code is a generalized analytical tool suitable for completing steam generator sizing and/or performance calculations. As a sizing tool, required heat transfer surface and number and length of tubes is determined as a function of specified fluid flow rates and temperatures, tubeside pressure drop limitations, and tube material and configuration. Conversely, as a performance tool, steam generator outlet conditions, such as steam temperature, pressure, and quality, are determined when the tube bundle geometry and fluid inlet conditions are fixed. Numerous fluid property and heat transfer subroutines are available, enabling the code to be used with equal efficiency for subcooled, superheated, and two-phase steam/water mixtures. Temperature variable physical properties of molten nitrate salts were added to the program library for this contract.

A process flow schematic is shown in Figure 3-28. Individual heat exchanger performance characteristics are described in Table 3-4 and the predicted fluid temperature profiles are shown in Figure 3-29.

The remainder of this section is devoted to a presentation of the important thermal-hydraulic features of the selected steam generator component designs. Specifically:

1. Design to preclude departure from nucleate boiling (DNB)

2. Tube-side flow distribution

3. Shell-side flow distribution

4. Hydraulic stability

5. Flow-induced vibration

6. Salt-side pressure relief

TABLE 3-3

Steam Generator Subsystem Performance Characteristics

	Stand-Alone (100MWe)	Repowering (50Mwe)
Thermal Rating - MWt	264.2	132.2
Salt Inlet Temp - °C (°F)	566(1050)	566(1050)
Salt Outlet Temp - °C (°F)	288(550)	288(550)
Salt Flow Rate - kg/sec (lb/hrx10^{-6})	603(4.79)	301(2.39)
Steam Throttle Temp - °C(°F)	538(1000)	538(1000)
Steam Throttle Pres - MPa(psia)	12.5(1815)	12.5(1815)
Main Steam Flow Rate - kg/sec (lb/hr x 10^{-6})	96.3(.764)	48.2(.382)
Cold Reheat Stm Temp - °C(°F)	371(700)	371(700)
Hot Reheat Stm Temp - °C(°F)	538(1000)	538(1000)
Hot Reheat Stm Pres - MPa(psia)	3.4(500)	3.4(500)
Reheat Steam Flow Rate - kg/sec (lb/hr x10^{-6})	83.4(.661)	41.6(.330)
Saturation Pressure - MPa(psia)	13.8(2000)	13.8(2000)
Feedwater Temp - °C (°F)	238(460)	238(460)

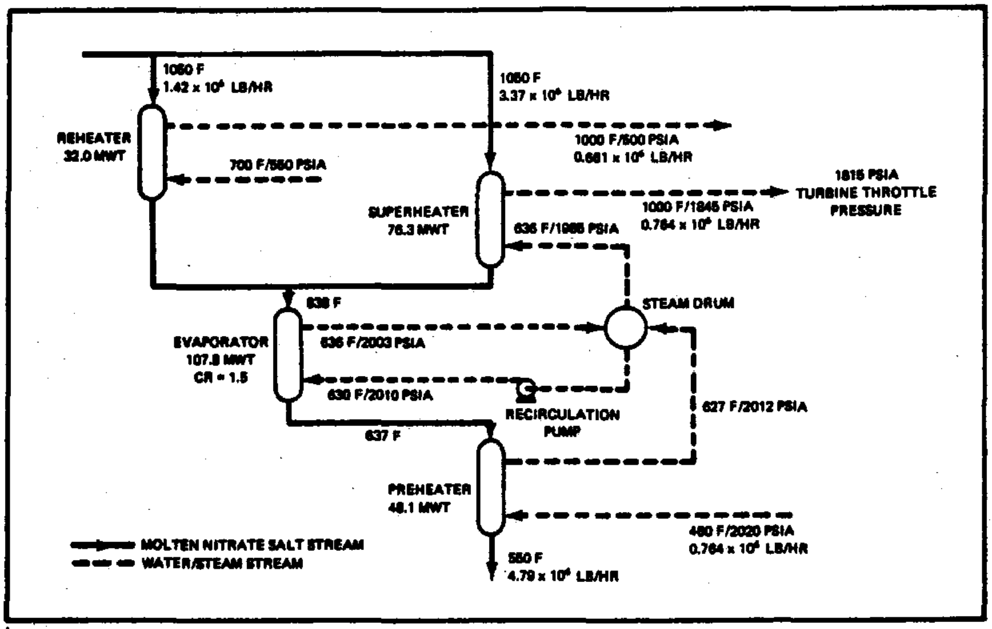


FIG. 3-28 RECIRCULATION STEAM SYSTEM - PROCESS FLOW SCHEMATIC

COMPONENT	PRIMARY M^2 (FT^2)	SURFACE AREA (% OF TOTAL)	HEAT LOAD (MWT)	(% OF TOTAL)
PREHEATER	1803 (19410)	34.5	48.1	18.2
EVAPORATOR	2349 (25280)	45.0	107.8	40.8
SUPERHEATER	428 (4610)	8.0	76.3	28.9
REHEATER	652 (7020)	12.5	32.0	12.1

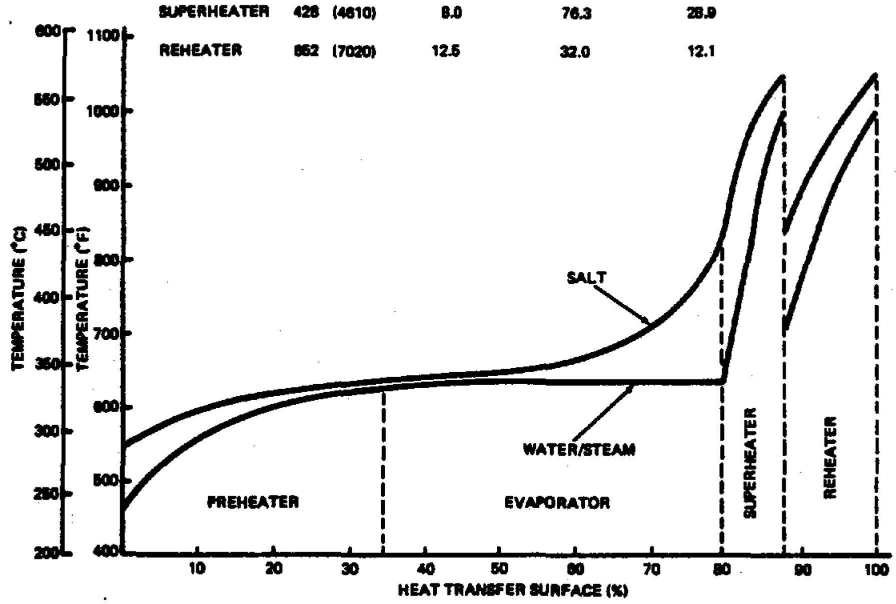


FIG. 3-29 FLUID TEMPERATURE PROFILES

TABLE 3-4

Heat Exchanger Performance Characteristics

Component	Salt Velocity M/sec (ft/sec)	Mean Salt Side H.T.C.* $W/M^2 {}^oC$ (BTU/hr ft^2F)	Mean Water Side H.T.C.** $W/M^2 {}^oC$ (BTU/hr ft^2F)	Mean Tube Wall H.T.C.* $W/M^2 {}^oC$ (BTU/hr ft^2F)	Mean Overall H.T.C.* $W/M^2 {}^oC$ (BTU/hr ft^2F)	Surface Design Margin (%)
Preheater	0.9(3.0)	3134(552)	7949(1400)	27142(4780)	1845(325)	15
Evaporator	1.1(3.5)	3730(657)	22145(3900)	7893(1390)	1987(350)	15
Superheater	1.6(5.2)	7495(1320)	9426(1660)	11527(2030)	2703(476)	15
Reheater	0.5(1.6)	2669(470)	2669(470)	2350(4140)	1136(200)	15

*Heat transfer coefficient based on tube O.D.

** Heat transfer coefficient based on tube I.D.

3.3.1 Design To Preclude DNB

In conventional steam generating equipment, experience has shown that departure from nucleate boiling (DNB) must be prevented to avoid failure of tubes from overheating in an extremely short time, and corrosion of tubes in zones where overheating is not a consideration. In solar steam generators, where heat input is limited by the temperature of the heating fluid, overheating of tubes is not of concern. However, under-deposit corrosion where DNB occurs in the presence of porous water-side deposits is a major design and operational consideration. This corrosion can occur either rapidly or over a long period depending on the level of heat flux and the boiler water purity.

In traditional steam generator design practice, the circulation ratio (or mass flow rate through the evaporator) is fixed high enough to maintain nucleate boiling in all circuits. However, high circulation rates lead to increased heat transfer surface, increased pumping power requirements, and reduced cycle efficiency. Thus, ribbed tubes are often used in the evaporator surfaces. This tube construction produces a swirling flow and centrifugal forces that keep the tube surface wetted and maintain nucleate boiling to a higher quality for a given pressure, heat flux, and mass flow rate than with a smooth-bore tube of the same general dimensions.

The factors controlling the onset of DNB have been determined for the range of practical interest through extensive research in boiling heat transfer. These factors are indicated in Table 3-5, including the effect each has on the onset of DNB.

TABLE 3-5

Factors Influencing

Onset Of DNB

Factors Influencing DNB	Likelihood of DNB Decreased by:
Pressure	Decreasing Pressure
Heat Flux	Decreasing Heat Flux
Mass Flow Rate	Increasing Mass Flow Rate
Quality	Decreasing Quality
Tube Surface Condition (Ribbed, Smooth)	Ribbed Tube Surfaces
Tube Orientation (Vertical, Horizontal, Inclined)	Vertical Tube Orientation
Tube Diameter	Decreasing Tube Diameter

To prevent DNB, the evaporator design employs multi-lead ribbed tubing for the last 32 feet of the upper, or steam/water outlet, leg. This represents the zone of highest heat flux and quality, and is consequently, most susceptible to DNB. Smooth-bore tubes are used in the remaining low heat flux sections of the components. Figure 3-30 illustrates steam quality and heat flux as a function of average tube length and indicates the smooth-bore to ribbed tube juncture. Figure 3-31 qualitatively illustrates the effect of quality on required mass velocity to avoid DNB. This figure clearly shows the significantly higher "safe" steam qualities for a given mass velocity using ribbed tubing.

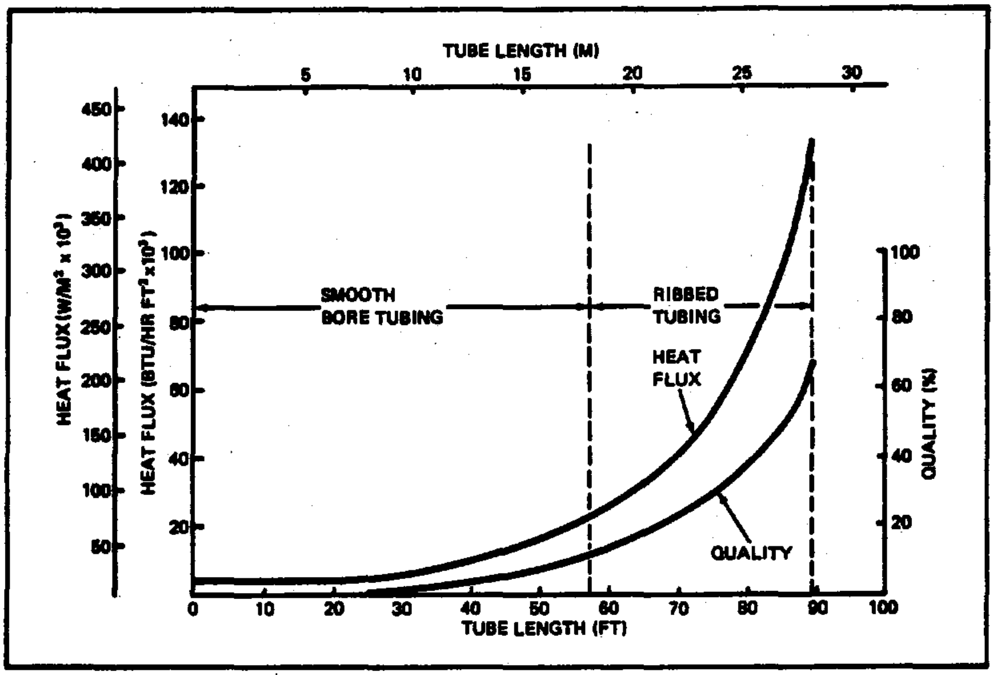


FIG. 3-30 STEAM QUALITY AND HEAT FLUX FOR AVERAGE EVAPORATOR TUBE

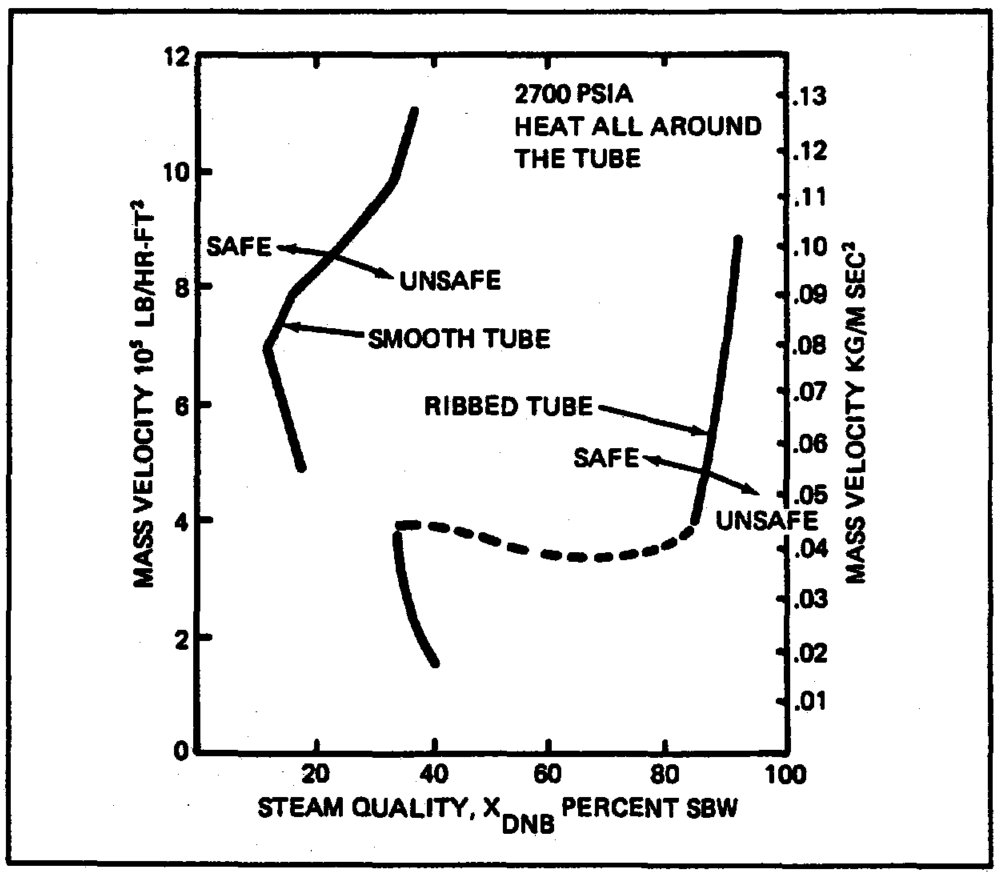


FIG. 3-31 EFFECT OF QUALITY ON REQUIRED MASS VELOCITY TO AVOID DNB

Table 3-6 lists the advantages and disadvantages of using the ribbed tubing. Because specific data applying to small diameter ribbed tubes in horizontal orientations is not available, the most applicable and published Babcock & Wilcox proprietary data was reviewed, prudent design margins added, and a conservative circulation ratio (CR) selected. The chosen CR is 1.5.

TABLE 3-6

Advantages And Disadvantages Of Ribbed Tubes

Advantages:

- heat transfer surface requirements minimized resulting from lower CR
- reduced recirculating pump power requirements

Disadvantages:

- increased base material tubing costs of approximately 30% over smooth-bore tubes
- increased hydraulic resistance by about 15% over smooth-bore tubes

The horizontal orientation of the evaporator mandates the use of ribbed tubing to assure absence of DNB under all operating conditions and assure component reliability. The incremental cost of ribbed tubing is offset by the advantages accruing from its use.

3.3.2 Tube-Side Flow Distribution

The U-tube heat exchanger configurations provide tube flexibility to accommodate tube/shell and tube/tube differential thermal expansion during normal and transient operation. Because U-tubes are employed in the bundles of all heat exchanger components, variations in tube length exist between long and short bend radii tubes (or innermost and outermost tubes). This variation in length leads to a variation in hydraulic resistance which produces a certain amount of flow mal-distribution. The maximum calculated difference in flow rates ranged from 3.8 percent in the evaporator to 7.1 percent in the reheater. Resulting from this flow mal-distribution, a maximum tube/tube temperature difference of 12 ^{o}C (22^{o}F) between long and short bend radii tubes was calculated (reheater). The maximum calculated tube/tube differential expansion was less than 1/16 of an inch, which the U-tube flexibility readily accommodates.

3.3.3. Shell-Side Flow Distribution

Since the thermal conductivity of the molten salt is relatively low, the overall resistance to heat transfer is dominated by the shell-side heat transfer resistance. Consequently, salt velocity is maximized to promote efficient heat transfer within salt-side pressure drop constraints and limits necessary to preclude tube vibration.

Salt inlet flow distribution baffles control tube impingement velocities and limit excitation frequencies. Preferentially broached tube support plates promote crossflow and serve to enhance heat transfer. The preferentially broached plates also serve to inhibit salt stratification, although the possibility of such stratification is considered remote. Preferentially broached plates are employed in all heat exchangers except the superheater. Superheater salt velocities are high and flow turbulent enough (even at part load conditions) to preclude the necessity and additional expense for preferentially broached plates.

3.3.4 Hydraulic Stability

Water, steam/water mixture, or steam is heated in up flow to promote hydraulic stability. Conversely, molten salt is cooled in down-flow. That is, fluid bouyant forces are in the correct direction.

Heat exchangers that experience phase change, such as the evaporator, are candidates for static instabilities. If instabilities exist, the evaporator may (a) fail to meet basic plant heat load requirements, (b) reduce effective life due to excessive thermal cycling from an oscillatory condition, or (c) disrupt normal plant operation. The following discussion shows that the evaporator is stable for various operating conditions or demands.

The source of potential instability is on the water-side where pressure drops are sensitive to flow rate changes as a result of density (phase) changes. A flow is subject to static instability if, when flow conditions change by a small step from original steady-state conditions, the flow oscillates between new steady-state conditions. Figure 3-32 represents a theoretical unstable system. Points A, B, and C represent three potential operating points for a given pump head, or supply. Figure 3-33 represents the actual calculated molten salt evaporator design supply and demand curves. Since the evaporator demand curve is constantly increasing with flow rate, there is only one operating point that can exist for a given flow rate (intersection of supply and demand curves). Consequently, the evaporator is a stable system.

3.3.5 Flow-Induced Vibration

Flow-induced vibration due to shell-side cross flow and/or very high parallel flow can potentially cause tube failures. The design objective, considering flow-induced vibration, is to maintain excitation frequencies below tube natural frequencies.

Flow-induced vibration is prevented by (a) limiting shell-side velocities, (b) providing adequate tube support, and (c) controlling salt inlet impingement velocities. With respect to the first point, excessive conservatism is undesirable since efficient heat transfer (and thus component performance) is highly dependent on high shell-side salt velocity. With repsect to the second point, tube support plates are spaced at appropriate spans to limit tube natural frequencies. And finally, with respect to the third point, salt inlet distribution baffles control impingement velocities.

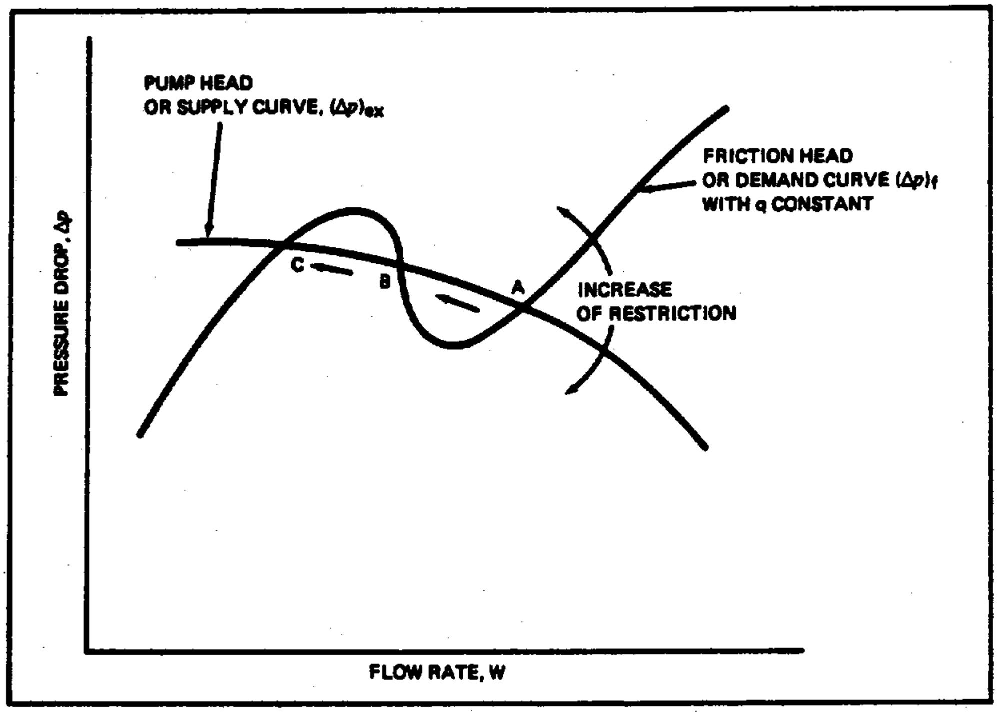


FIG. 3-32 SUPPLY AND DEMAND OF PRESSURE HEADS

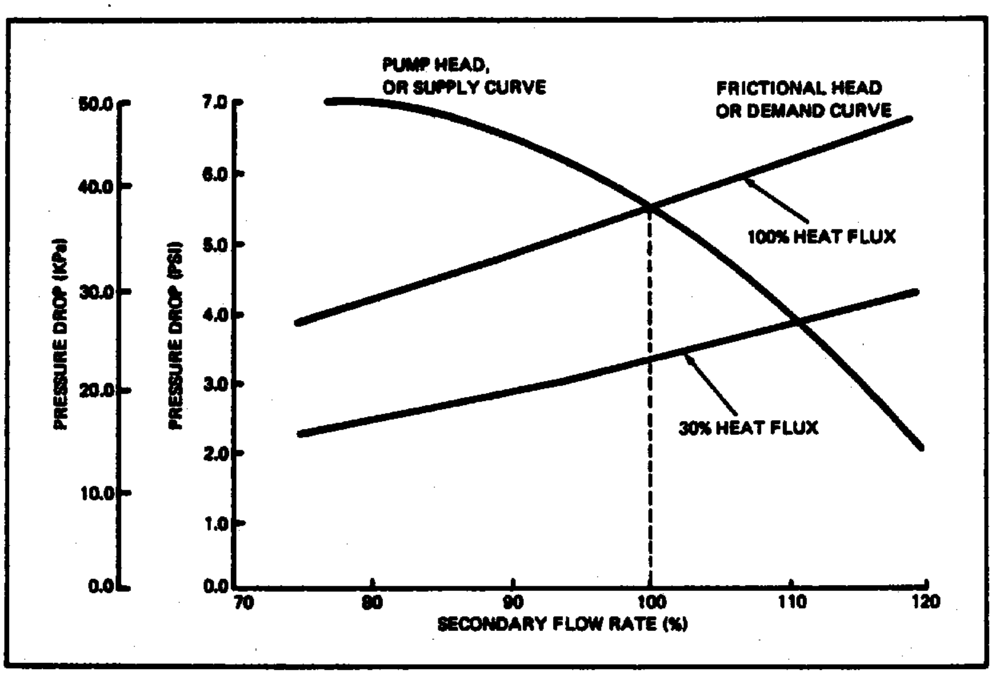


FIG. 3-33 EVAPORATOR STATIC STABILITY FOR 30% AND 100% HEAT FLUX (AVERAGE TUBE)

The types of flow-induced vibration mechanisms most commonly associated with the heat exchanger design configurations include cross-flow vortex shedding and fluidelastic whirling. The analytical criteria used include those established by Y.N. Chen (Reference 4) for vortex shedding and J. J.Connors, Jr. (Reference 5) for fluidelastic whirling. The heat exchangers were designed to preclude tube failues due to flow induced vibration by meeting the criteria specified in the above references.

3.3.6 Salt-Side Pressure Relief

Salt-side pressure relief is required to prevent over pressurization in the event of a heat exchanger tube leak. Three basic types of pressure relief devices were examined:

1. Relief valves - automatically open in proportion to the amount of pressure

2. Safety valves - automatically "pop" to full open position

3. Rupture disks - thin metal diaphram held between flanges "bursts" open; diaphram must be replaced

Rupture disks were selected for this application. They avoid the unique possibility of salt freezing in a valve, which would prevent the valve from properly reseating. The locations and sizes of required salt-side rupture disks are identified in Figure 3-34.

The rupture disks were sized and located to preclude heat exchanger shell stresses above yield due to over pressurization resulting from a single tube guillotine type failure. Safety margin is provided however, for the following numbers of guillotine type tube failures before yield stresses in the shell are exceeded:

	Number of Guillotine Type Tube Failures
Preheater	6
Evaporator	7
Superheater	4
Reheater	17

All rupture disks are exhausted back to the salt drain sump.

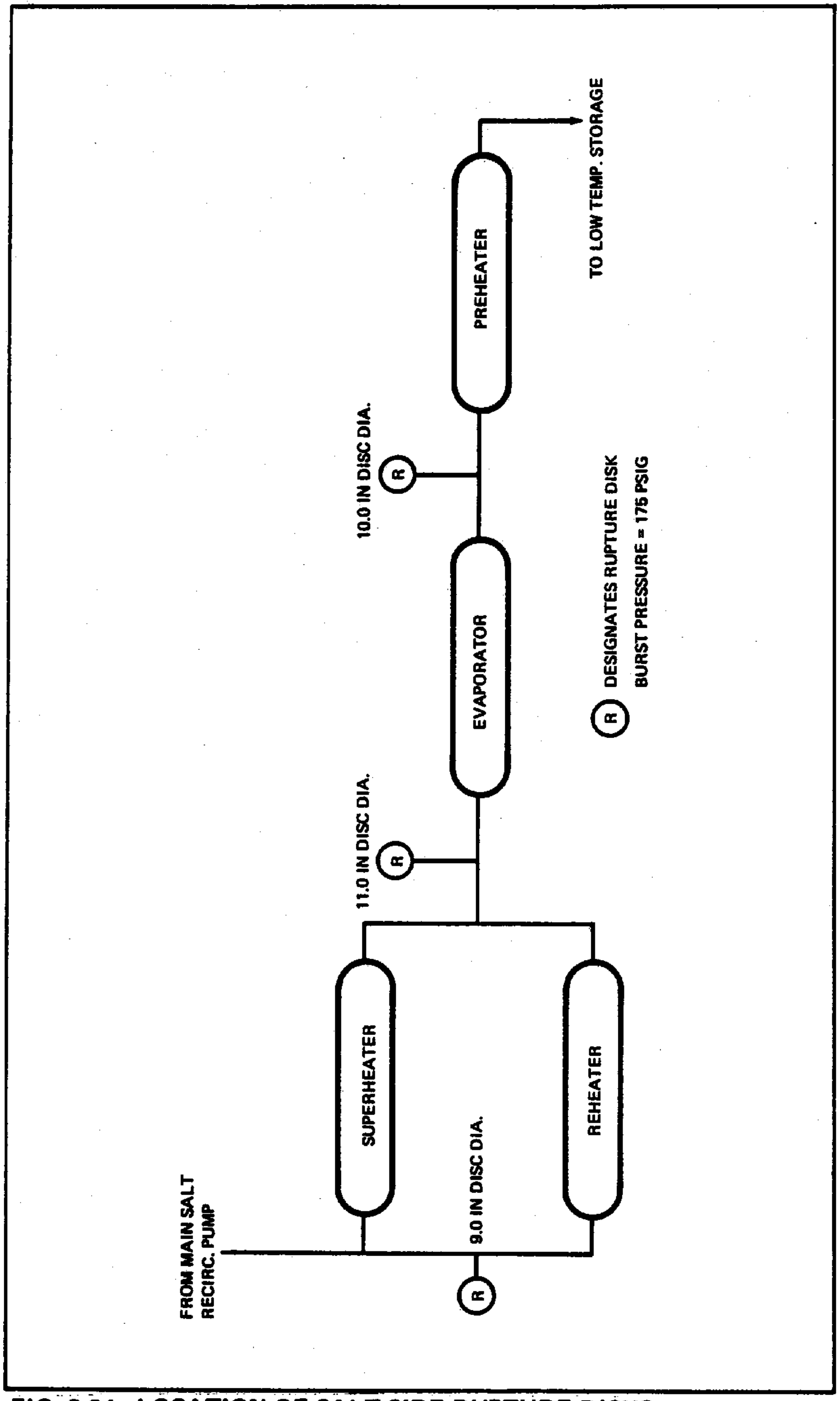


FIG. 3-34 LOCATION OF SALT SIDE RUPTURE DISKS

3.4 Structural Design

All components were structurally designed in accordance with ASME Section VIII, Division 1 and the considerations defined in the Design Analysis Plan (Reference 6). Criteria for non-code thermal/mechanical analyses were developed to supplement the code calculations using ASME Section VIII, Division 2 as a guideline and these are shown in Table 3-7. Mechanical stress, thermal stress, and elevated temperature analyses were performed to meet code requirements and to assure structural adequacy at critical locations in the vessels. Appendix B presents the pressure boundary code calculations and Appendix C presents a summary of the stress analysis results.

TABLE 3-7

Supplementary Design Criteria

Tubesheet:

Condition	Pm	Pm+Pb	Pm+Pb+Q	Pm+Pb+Q+F
Design Pressure	Sy	1.35 Sy	-	-
Operating Pressure	S	Sy	-	-
Operating Pressure plus Temperature	-	-	2 Sy	$\Sigma u \leqq 1.0$

Head And Shell Junctures At Tubesheet:

Condition	P_L	P_L+Q	P_L+Q+F
Design Pressure	Sy	-	-
Operating Pressure	Sy	-	-
Operating Pressure plus Temperature	-	2 Sy	$\Sigma u \leqq 1.0$

Shell Stresses At Saddle Supports:

Type Stress	Allowable*
Dead Load Longitudinal Bending + Pressure	S
Dead Load Circumferential Bending Across Thickness	1.25S
Dead Load Tangential Shear Stress In Wall	0.80S

* Allowable for additional operating seismic load increased by 1/3 per Reference 10.

P_m - General primary membrane stress

$P_m + P_b$ - Primary membrane + primary bending stress

P_L - Local primary membrane stress

$P_m + P_b + Q$ - Primary + secondary stress

$P_m + P_b + Q + F$ - Primary + secondary + peak stress

3.4.1 Mechanical Stress

Mechanical stress analyses were performed for design conditions and full load steady state operating conditions, and evaluations were made considering the relative severity of the upset and emergency event transients. Table 3-8 lists the design temperatures and pressures for the steam generator subsystem components.

TABLE 3-8

Component Design Temperatures and Pressures

Component	Design Temp. °C(°F)	Salt-Side Design Pressure MPa(psia)	Water-Side Design Pressure MPa(psia)
Preheater	371(700)	1.31(190)	15.17(2200)
Evaporator	482(900)	1.31(190)	15.17(2200)
Superheater	580(1075)	1.31(190)	14.66(2125)
Reheater	580(1075)	1.31(190)	4.55(660)
Steam Drum	343(650)	N/A	14.66(2125)

Specific mechanical design analyses included the following:

1. Pressure boundary code calculations were made for all components, and wall thicknesses were established to meet minimum code requirements. Appendix B summarizes the results of the code calculations and lists actual wall thicknesses.

2. Pressure discontinuity analyses per the methods of Reference 7 were made to supplement TEMA (Reference 8) calculations for tubesheet design to assure acceptable tubesheet ligament stress levels and shell-tubesheet-head discontinuity stress levels. Figure 3-35 shows the six element model used for the interaction analysis. Tubesheet peak stress intensities and juncture principal stresses and stress intensities for internal pressure and/or thermal loadings were calculated in accordance with the method of Reference 7.

3. Dead weight and seismic loads were considered to design the main vessel saddle supports. Hand calculations per the methods of Reference 9 indicate the shell longitudinal and circumferential bending and tangential shear stresses meet the criteria in Table 3-7.

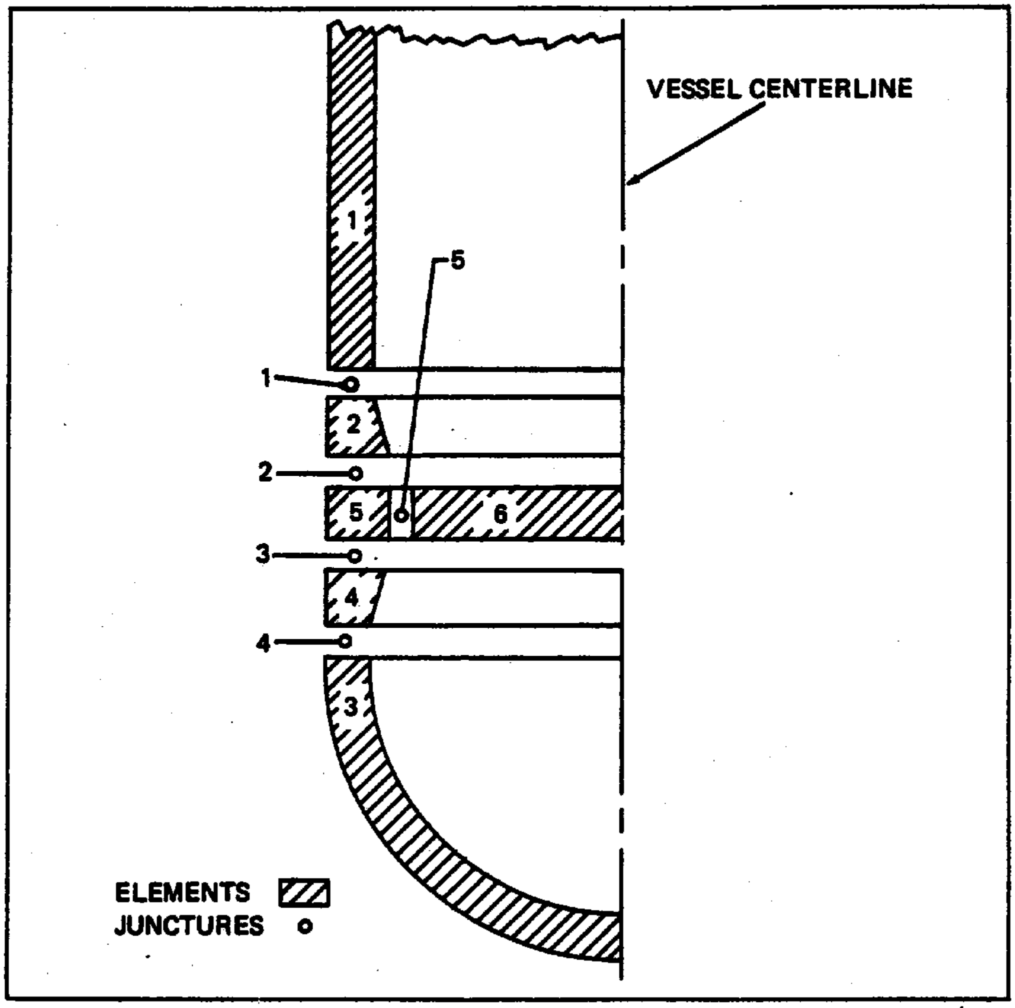


FIG. 3-35 SHELL/TUBE SHEET/HEAD DISCONTINUITY MODEL

4. Piping loads have not been established; however, a qualitative evaluation of allowable piping loads was done per the methods of Reference 11 to establish allowable moments and forces as a percentage of yield to assure shell-side adequacy without additional reinforcement. The results are tabulated in Table 3-9.

TABLE 3-9

Allowable Piping Loads

Pipe Terminal	Allowable Load In % Of Yield Loads Of Moment My And Force Py	
	Allowable M	Allowable P
Superheater Salt Inlet	0.14 My	0.20 Py
Reheater Salt Inlet	0.23 My	0.14 Py
Evaporator Salt Inlet	0.28 My	0.19 Py
Preheater Salt Inlet	0.35 My	0.26 Py
Superheater Steam Outlet	1.00 My	1.00 Py
Reheater Steam Outlet	1.00 My	1.00 Py
Evaporator Steam Outlet	1.00 My	1.00 Py
Preheater Steam Outlet	1.00 My	1.00 Py

3.4.2 Thermal Stress

Thermally induced stresses were analyzed for worst case radial and longitudinal gradients, discontinuity, and fatigue. The critical temperature plus pressure loading condition analyzed was full load steady-state operating conditions. Table 3-10 summarizes the locations analyzed including the type of analysis performed.

Analytical methods and results are summarized in the following paragraphs:

- A plane stress quadrilateral type finite element analysis of the preheater tubesheet was performed to analyze the tubesheet for secondary stresses resulting from divider lane temperature gradients. The results from this model, combined with primary membrane plus bending stress results from the methods of Reference 7 for isothermal operating pressure conditions, show that primary plus secondary stress intensities meet allowable limits. Figure 3-36 llustrates the finite element model used for the analysis.

TABLE 3-10

Thermal Mechanical Analysis Summary

Component	Radial Gradient	Longitudinal Gradient	Discontinuity	Fatigue
Preheater	. Tubesheet Divider Lane Plane Stress Quadralateral Type F.E. Analysis		. Head/Tubesheet/Shell Juncture Axisymmetric Six-Element Model	. Tubesheet Ligaments
Evaporator		. Salt-Side Shell Plate/Shell F.E. Analysis	. Head/Tubesheet/Shell Juncture Axisymmetric Six-Element Model	. Shell/Tubesheet Juncture @ Weld Hand Calculated Usage Factor
		. Short Bend-Radius Tube Beam Type F.E. Analysis		
		. Salt-Side Transition Three-Element Interaction Analysis		
Superheater			. Head/Tubesheet/Shell Juncture Axisymmetric Six-Element Model	
Reheater			. Head/Tubesheet/Shell Juncture Axisymmetric Six-Element Model	

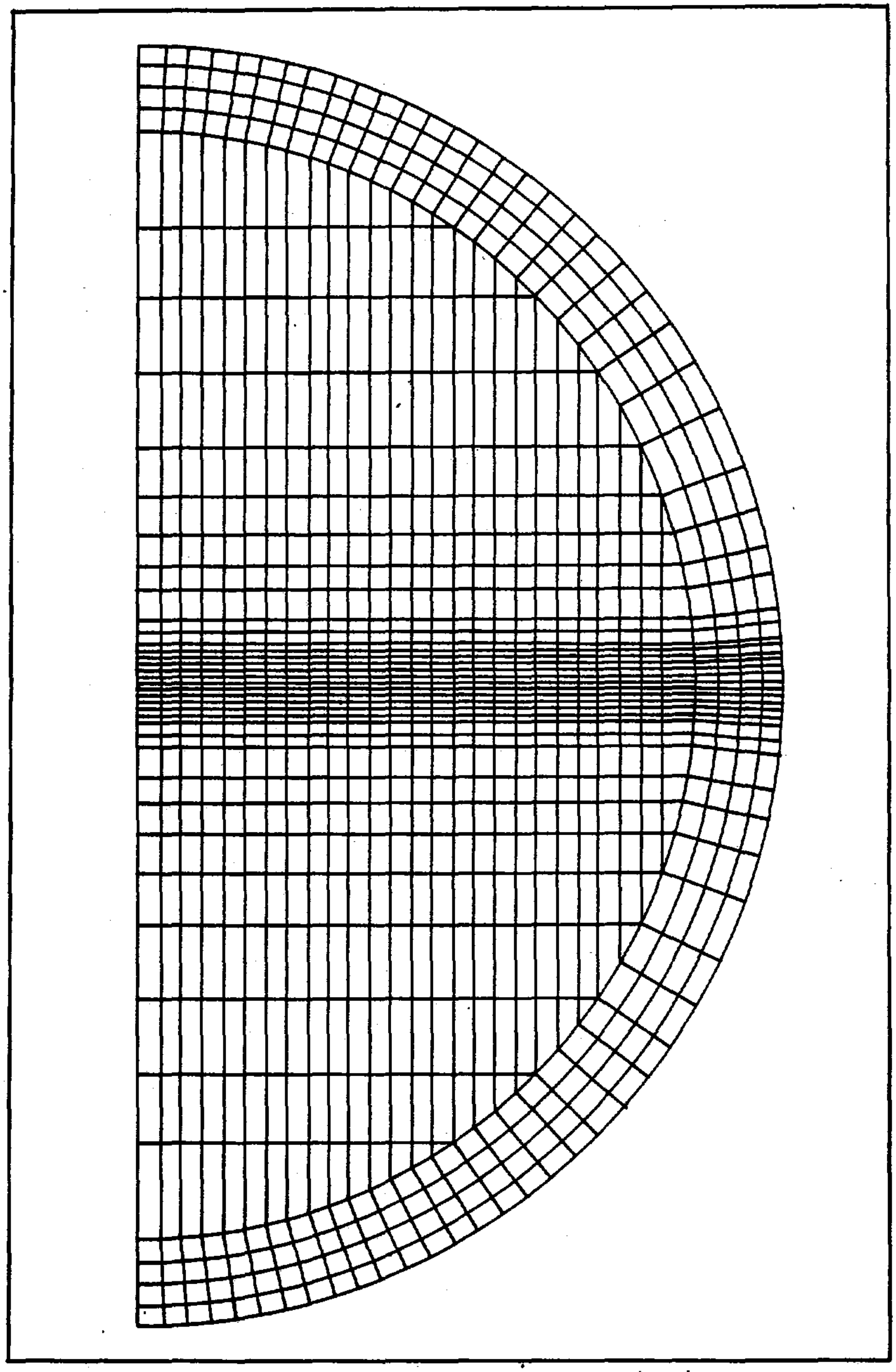


FIG. 3-36 FINITE ELEMENT PLANAR MODEL FOR PREHEATER TUBE SHEET

- A plate/shell finite element analysis of the evaporator shell was performed to determine the free body response of the shell to the imposed longitudinal thermal gradient. The axisymmetric shell model is fixed at the tubesheet and allowed to slide at the tube bend region. Figure 3-37 llustrates the finite element model and the free body response. The results indicate that the shell must be externally restrained to prevent the shell from interfering with tube bundle expansion and contraction. Excessive shell deflections would tend to restrict the U-tube bundle from moving independently and degradate the inherent advantages of the U-tube configuration.

- A beam type finite element analysis of the short bend radius tube in the evaporator was also analyzed for imposed longitudinal thermal gradients. The short bend radius tube was analyzed since the most rigid tube results in maximum tube bending stresses. The tube model illustrated in Figure 3-38 is fixed at the tubesheet and unrestricted at the U-bend to accommodate longitudinal expansion. Radial displacements at support plate locations were hand calculated and input at the corresponding support plate location. The purpose of the analysis was to determine maximum tube displacements and stresses. The results indicate that the magnitude of tube displacements and rotations will not cause the tubes to lock-up in the support plates. Figure 3-39 indicates that the maximum available angle of rotation is 0.6°; the actual rotation is less than 0.6°. The stress analysis indicates that the maximum stress intensity is less than yield and occurs at the tube support nearest the tube bend in the lower leg.

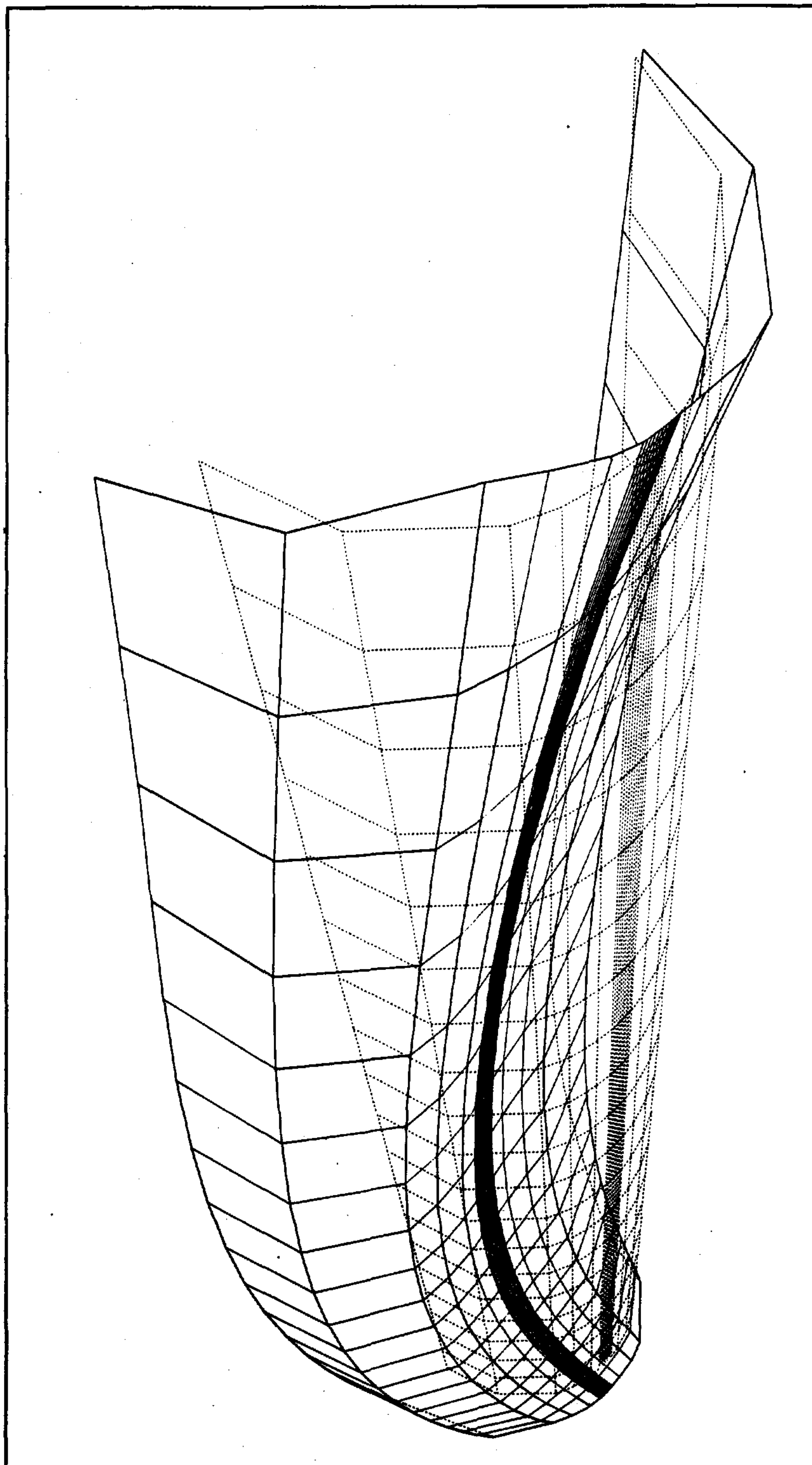


FIG. 3-37 EVAPORATOR-SHELL/PLATE AXISYMMETRIC MODEL

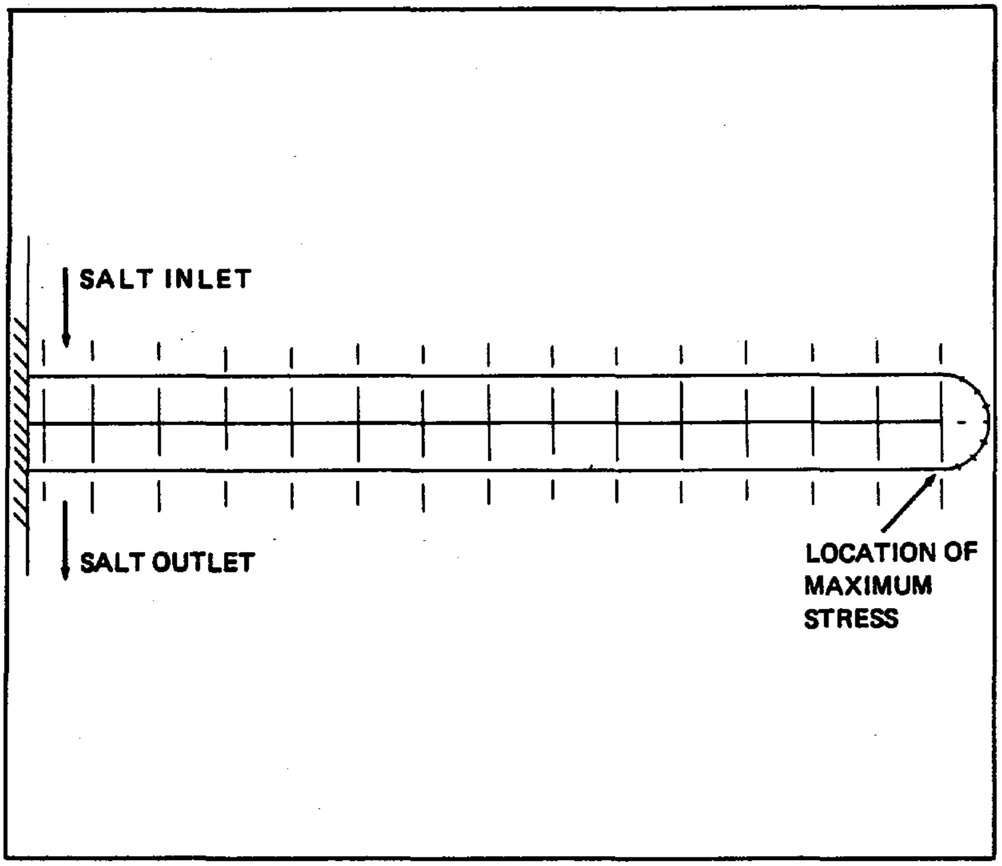


FIG. 3-38 EVAPORATOR - SHORT BEND RADIUS TUBE MODEL

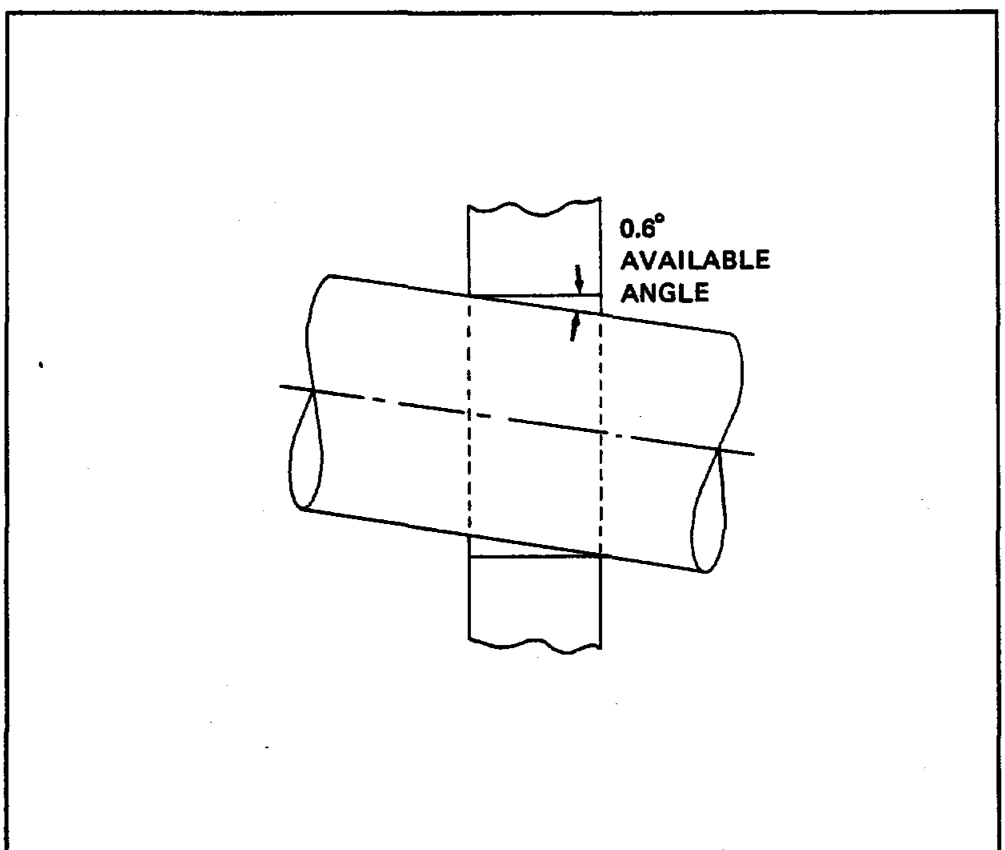


FIG. 3-39 MAXIMUM TUBE DEFORMATION

- The salt inlet shell-to-cone and cone-to-main shell junctures of the evaporator were analyzed for longitudinal thermal gradient, discontinuity stress. The model for the discontinuity analysis is shown in Figure 3-40. The analysis was performed per the methods of Reference 12, based on the shell-plate discontinuity methods given in Article 4-7 of Reference 13. Radial deflection and end slope boundary conditions at the cone-to-shell junctions were hand calculated based on the longitudinal temperature gradient. The results of the discontinuity analysis indicate that the primary plus secondary stress intensities meet allowable limits.

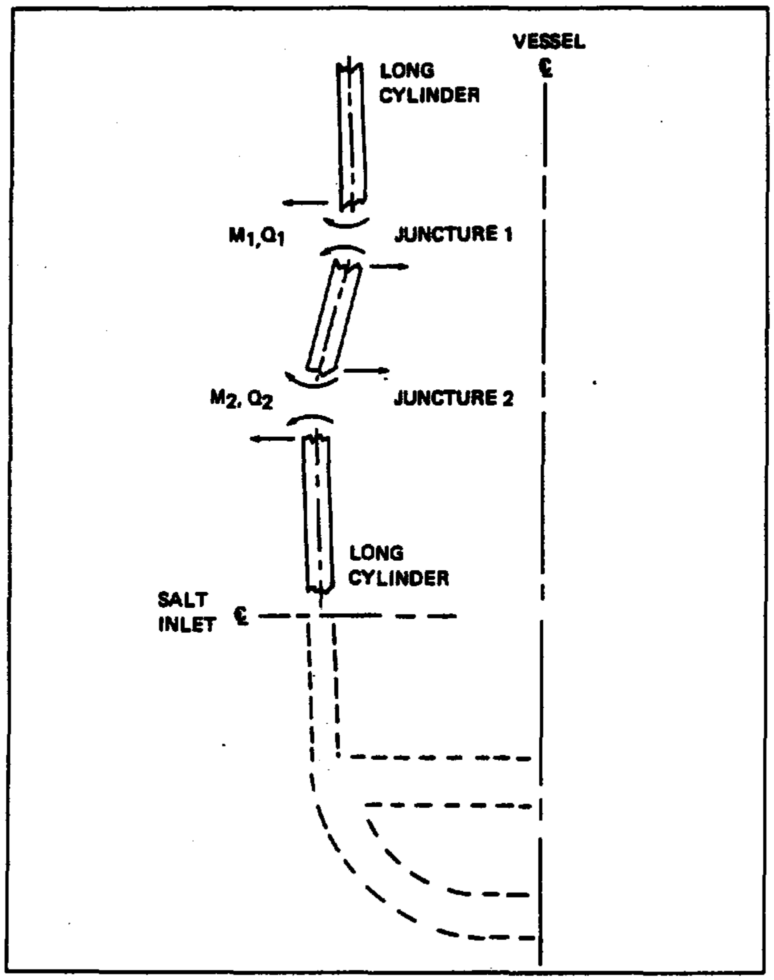


FIG. 3-40 EVAPORATOR - CONICAL SHELL DISCONTINUITY MODEL

- The shell/tubesheet/head junctures of all heat exchangers were analyzed using the methods of Reference 7 (see Figure 3-35), for full load steady-state operating conditions. Discontinuity stress levels were shown to meet the non-code criteria in Table 3-7.

- Cyclic loading conditions were evaluated for a design life of 10,000 cycles (approximately one start-up/shut-down per day for 365 days per year for 30 years). Fatigue analyses for the preheater tubesheet ligaments, including through-thickness and divider lane temperature gradients and pressure stresses, indicate a safe fatigue life of 12,000 cycles. Fatigue analyses of the evaporator shell/tubesheet juncture also indicate a safe fatigue life; greater than the design life of 10,000 cycles.

- Upset and emergency event transients were evaluated and judged less severe than the full load steady-state operating condition which was analyzed.

3.4.3. Elevated Temperature Effects

Supplementary rules for elevated temperature design are presented in Appendix D as developed from ASME Code Case N-47 and Sandia Report for Interim Design Standards, Reference 14. Both superheater and reheater were analyzed for creep-ratcheting, creep-fatigue and creep-buckling effects for the supplementary rules. The results are summarized in Appendix C.

Creep-Ratcheting

In accordance with the supplementary rules, Para. 3251.2 (a), the temperatures above which the start of creep is significant is 510°C (950°F) for type 304 stainless steel. The locations analyzed for creep-ratcheting in the superheater and reheater were the steam-side head, tubes, and salt-side shell in the upper, or outlet, leg. The salt-side shells are thin and have (a) small radial temperature gradients during normal event transients, (b) no radial temperature gradients during full load steady-state operation, and (c) relatively low pressure stresses. Consequently, the primary plus secondary stress parameters fall into the non-creep regime, E, of Figure 3252.2.2 in Appendix D. The tubes were evaluated using tube metal temperature radial gradients established from the Babcock & Wilcox VAGEN code for the worst case full load steady-state operating conditions. The steam-side heads were evaluated using radial gradients conservatively estimated for a hot standby start-up to full load condition where the duration was assumed to be 3 hours. The results indicate that all locations meet the strain criteria established in Appendix D.

Creep-Fatigue Interaction

The creep-fatigue interaction evaluation of the superheater and reheater was performed for the normal event transient (standby to full load operating condition) considering creep hold times and fatigue cycling. This transient was chosen since it is the most significant for both hold time at temperature ($T=10^5$ hours) and number of cycles ($N=10^4$). The critical locations were judged to be the tubesheet ligaments and the head/tubesheet/shell junctures of the steam outlets. The analysis was performed per the methods of Reference 7 and the results indicate that the interaction factors meet allowable limits.

Creep-Buckling

The critical locations for creep-buckling effects were judged to be at the U-bend support for the upper leg of the superheater and reheater where cross-sectional bending is the greatest. To provide safe margins, both time-independent load factors and time-dependent load factors were used in accordance with the supplementary rules in Appendix D. Compressive cross-sectional bending was calculated and compared to the critical values determined per the methods of Reference 15, and the results indicate a considerable margin.

3.5 50 MWe Steam Generator Components

The design characteristics of the steam generator components for a 100 MWe stand-alone plant were described in Section 3.1. The design features of the components for a 50 MWe repowered plant are similar to the 100 MWe components.

The principal geometric characteristics of the 50 MWe steam generator components are presented in Table 3-11. The design configurations of the heat exchangers are shown in Figures 3-41 through 3-44.

TABLE 3-11

Principal Geometric Characteristics

50 MWe Components

Component	Main Shell O.D. m(in)	Total Length m(ft)	Tube Dimensions O.D. X Wall mm(in)	No. Of Tubes	Avg. Active Tube Length m(ft)	Primary Surface Area $m^2(ft^2)$
Preheater	1.257 (49.5)	9.836 (32.27)	12.700 X 1.473 (.500 X .058)	1710	17.404 (57.1)	1187 (12780)
Evaporator	1.308 (51.5)	15.057 (49.40)	22.225 X 3.759 (.875 X .148)	775	27.157 (89.1)	1470 (15820)
Superheater	.457 (18.0)	7.705 (25.28)	12.700 X 1.651 (.500 X .065)	396	14.417 (47.3)	228 (2452)
Reheater	.560 (22.0)	8.019 (26.3)	15.875 X .889 (.625 X .035)	442	15.301 (50.2)	337 (3630)

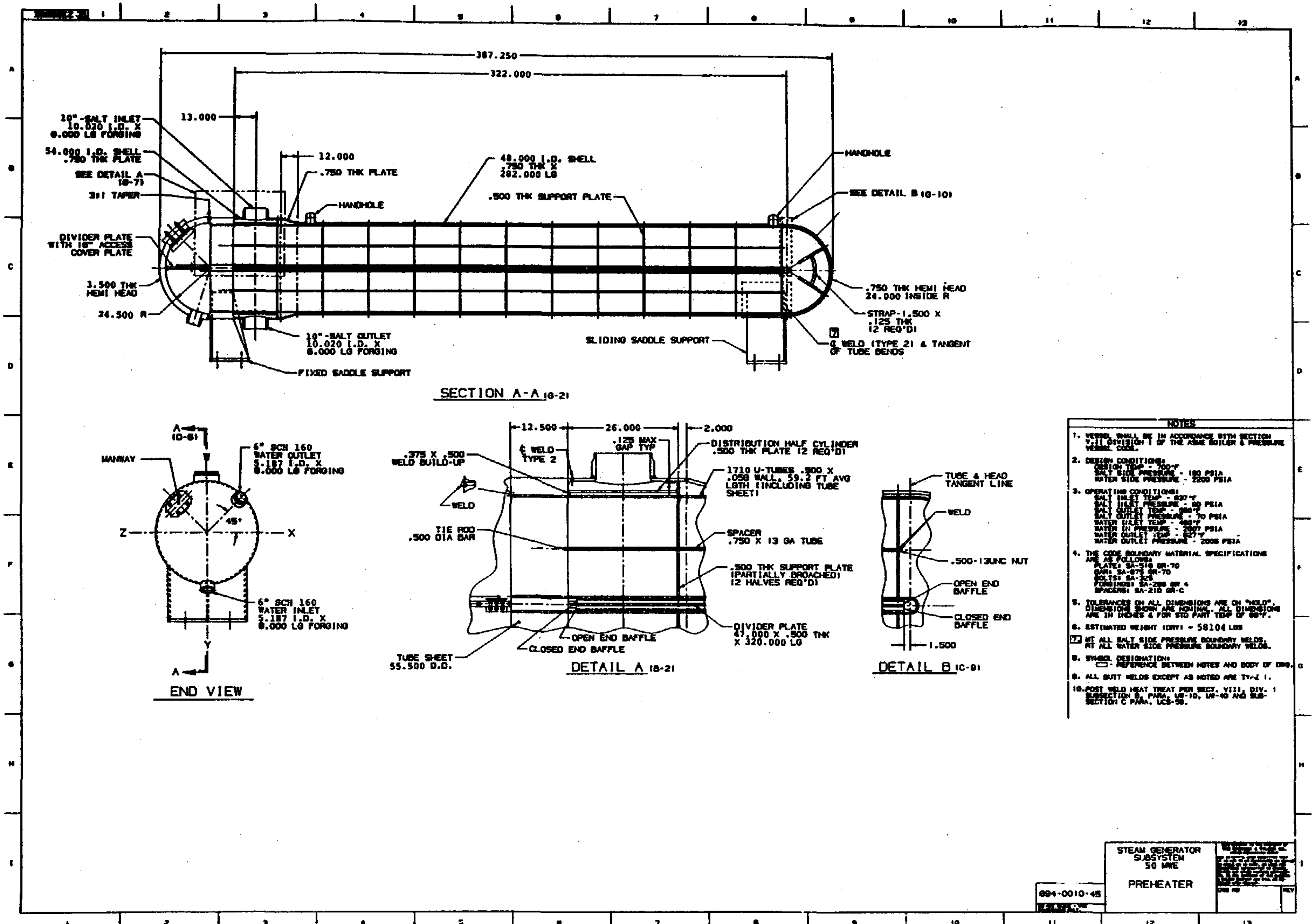


FIG. 3-41 50 MWe PREHEATER

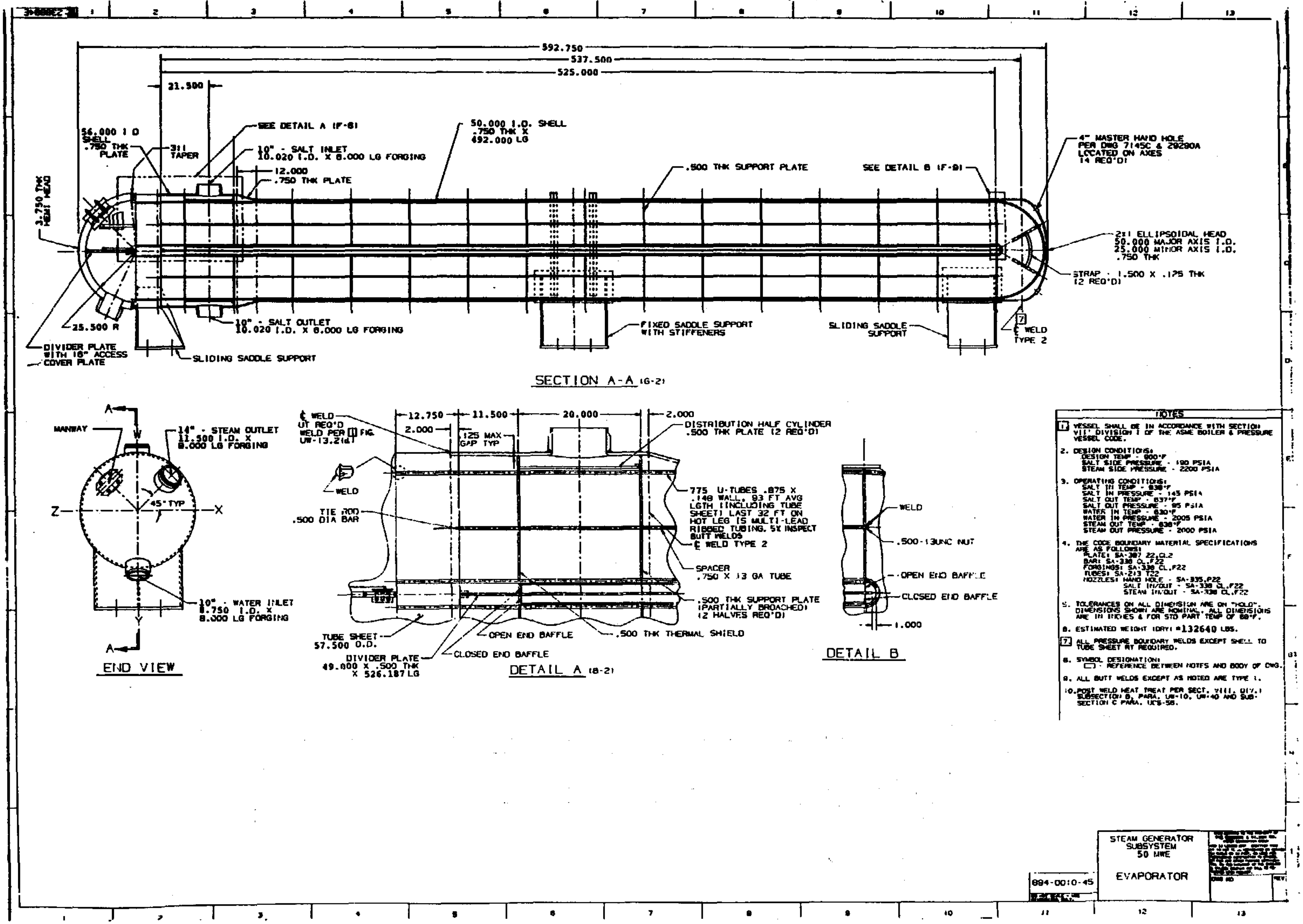


FIG. 3-42 50 MWe EVAPORATOR

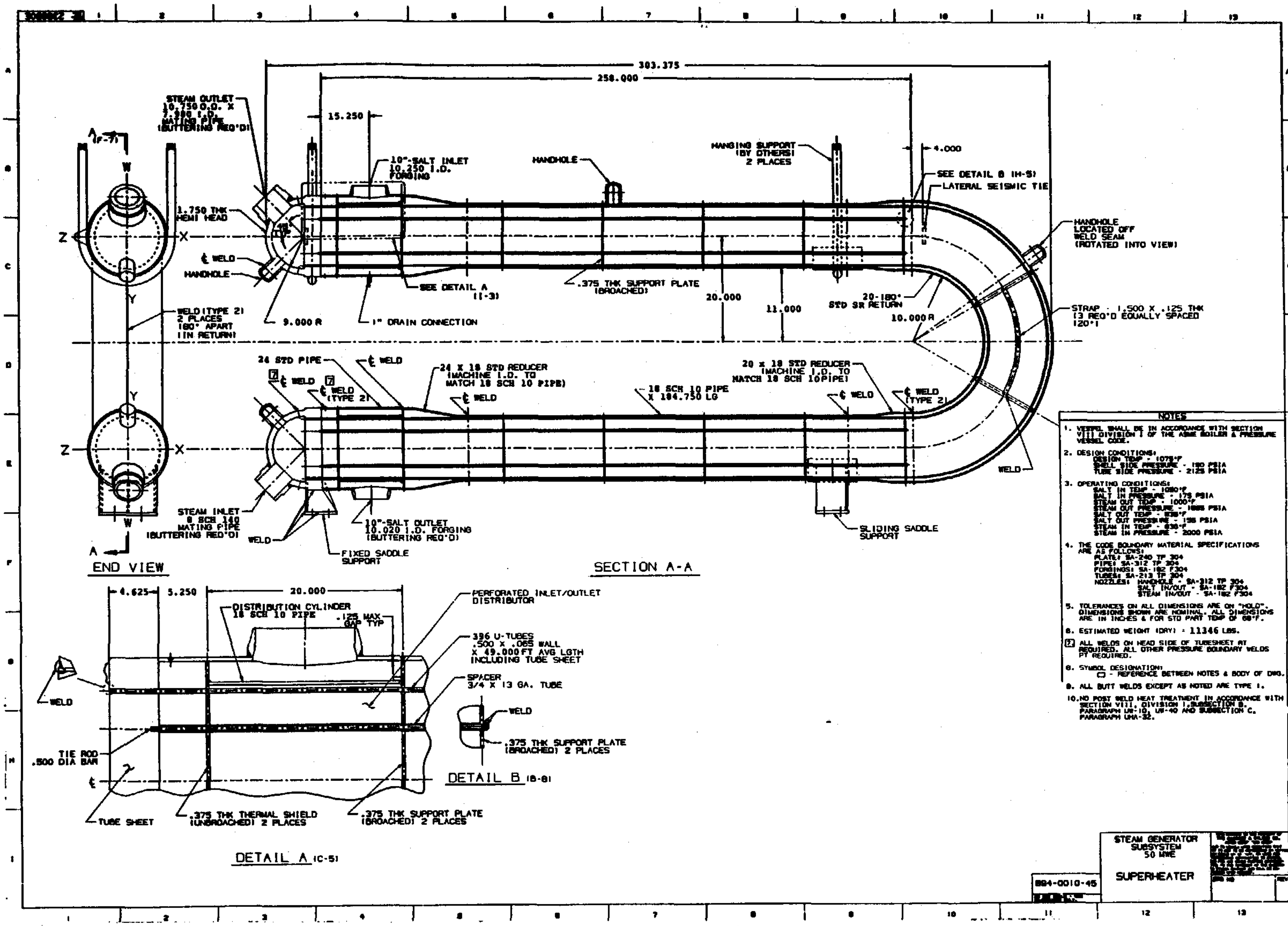


FIG. 3-43 50 MWe SUPERHEATER

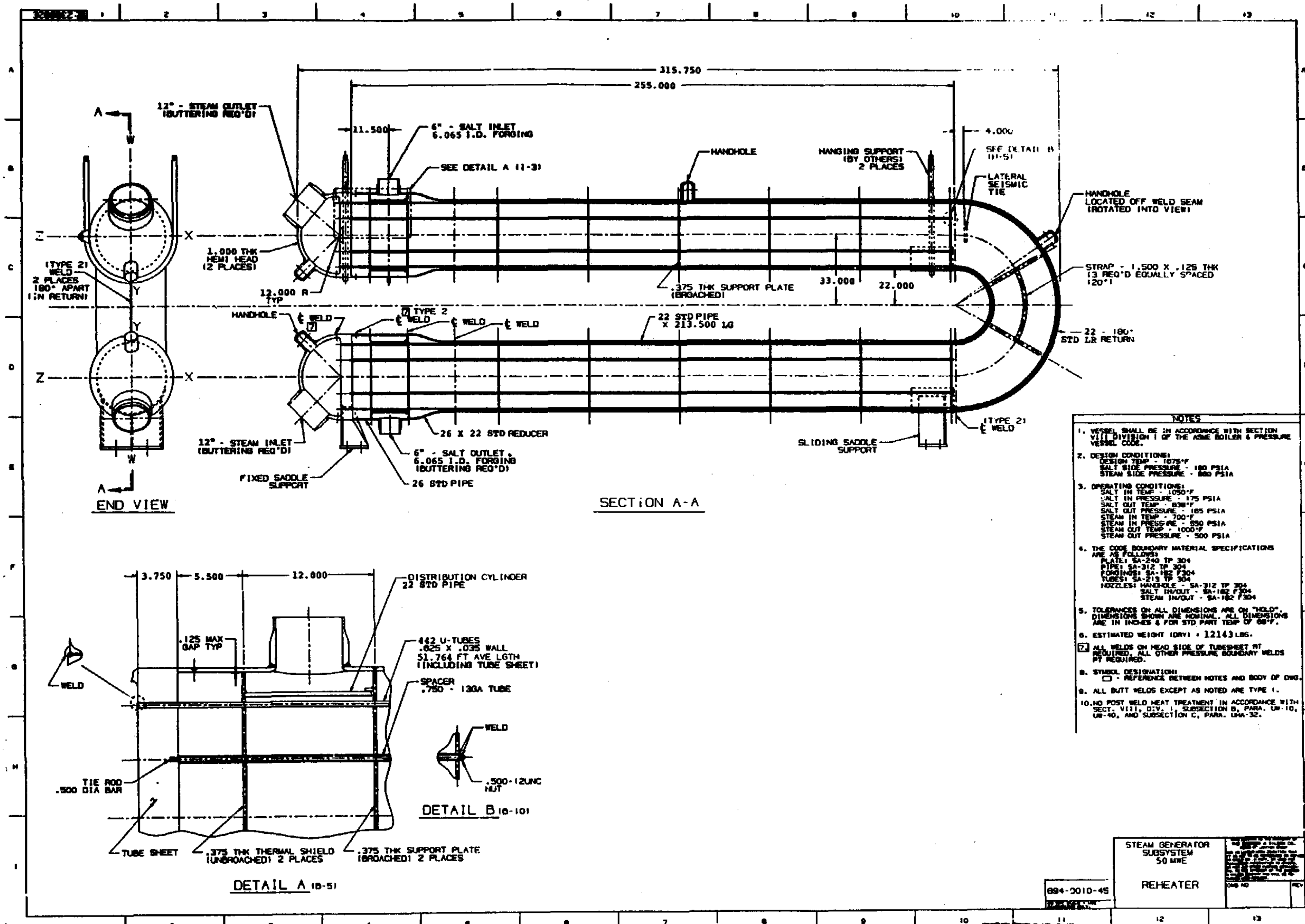


FIG. 3-44 50 MWe REHEATER

3.6 Inspection and Maintenance

In the design of the steam generator components, provisions have been made for inspection and maintenance. The emphasis has been placed on developing component designs which can be serviced by existing utility personnel using standard fossil boiler maintenance practice.

Important inspection and maintenance considerations for the heat exchanger components are described in the following paragraphs:

In-service Inspection - Manways and handholes provide for visual inspection of components in accordance with code requirements. Access is provided for in-service inspection of tubing (for example, by eddy current, ultrasonic or leak check examination techniques) to insure component reliability.

Tube Plugging - Manways or handholes provide access to the water/steam side of tubesheets for plugging defective heat exchanger tubes. Plugged tubes must be pierced to allow for salt drainage, and the design and installation of plugs must meet all code pressure boundary requirements.

Layup - When the steam generator components are shut down for short periods, adequate corrosion protection can be provided by blanketing the heat transfer surfaces with inert gas. For longer down periods, the components must be filled with treated water in addition to the inert cover gas.

Chemical Cleaning - Deposition on the water/steam side of steam generator components is generally produced by transport of corrosion products from the condensate and preboiler system. This deposit build-up may contribute to corrosion of heat transfer surfaces and may eventually degrade the thermal-hydraulic performance of the equipment. To remove the deposits, chemical cleaning solvents can be flushed through the components.

4.0 OPERATION AND CONTROL

A steam-electric generating system deriving thermal energy from a solar system with significant thermal storage is inherently a base load plant. Load maneuvering requirements are essentially limited to diurnal startup and shutdown. The control of such a system is therefore relatively straight-forward.

The control system for the steam generator is based on conventional boiler operating practice and provides the following functions required of any well-designed boiler control system:

(a) The superheater steam outlet pressure is controlled by adjustment of molten salt flow rate (analogous to fuel input in conventional practice).

(b) The supply of feedwater is controlled on average to match steam flow and also regulated to maintain a pre-determined steam drum water level.

(c) Final steam temperature is maintained within prescribed limits through attemperation.

(d) Flexibility of operation is provided by including operator automatic-manual transfer stations at appropriate points in the various control loops.

(e) Protection against equipment damage is provided by activation of functional operations which limit temperatures when an established criteria is reached.

The control system design, process instrumentation, and the results of the system simulation analysis are described in the following paragraphs. In subsequent paragraphs, part-load performance characteristics are discussed and startup and shutdown procedures are outlined.

4.1 Control System Design and System Simulation Analysis

A computer simulation of the steam generator subsystem was made to examine the dynamic thermal response of each steam generator component within the system as a whole. This work was implemented by developing and applying a model capable of predicting the response of the system to normal and upset transients. The detailed results of the analysis were used to determine if any limiting thermal loads occur in any components during upset transients, to both assess the proposed control system and to refine the analytically determined control gain settings, and to provide corroboration of the previously predicted steady-state plant performance.

4.1.1 System Simulation Analysis

The components included in the system simulation model are shown in Figure 4-1. To clarify the operation of the system, the hot and cold thermal storage tanks are shown in this figure. However, these tanks are not specifically included in the model. A general control scheme is implicit in the schematic. The individual and interactive indices used for control are discussed later.

The simulation model was implemented using a hybrid computer consisting of two parallel EAI 680 analog computers controlled by a CDC 1700 through a Digital-to-Analog Converter (DAC). The spatial analog solutions were sampled and returned to the CDC 1700 through an Analog-to-Digital Converter (ADC). An AP-120B array processor assisted the CDC 1700 by handling all floating point operations. The computing hardware and the relationship between components is shown in Figure 4-2.

The evaporator and preheater were modeled by modifying Continuous-Space-Discrete-Time (CSDT) models previously developed by Babcock & Wilcox for use in PWR nuclear system simulation. These components were both modeled on one of the EAI 680 analog computers. The superheater and reheater models were newly developed using a CSDT model. Because single-phase fluids exist on both the primary and secondary sides of these components, simplified models using average parameters were deemed adequate and were implemented on the other EAI 680 computer.

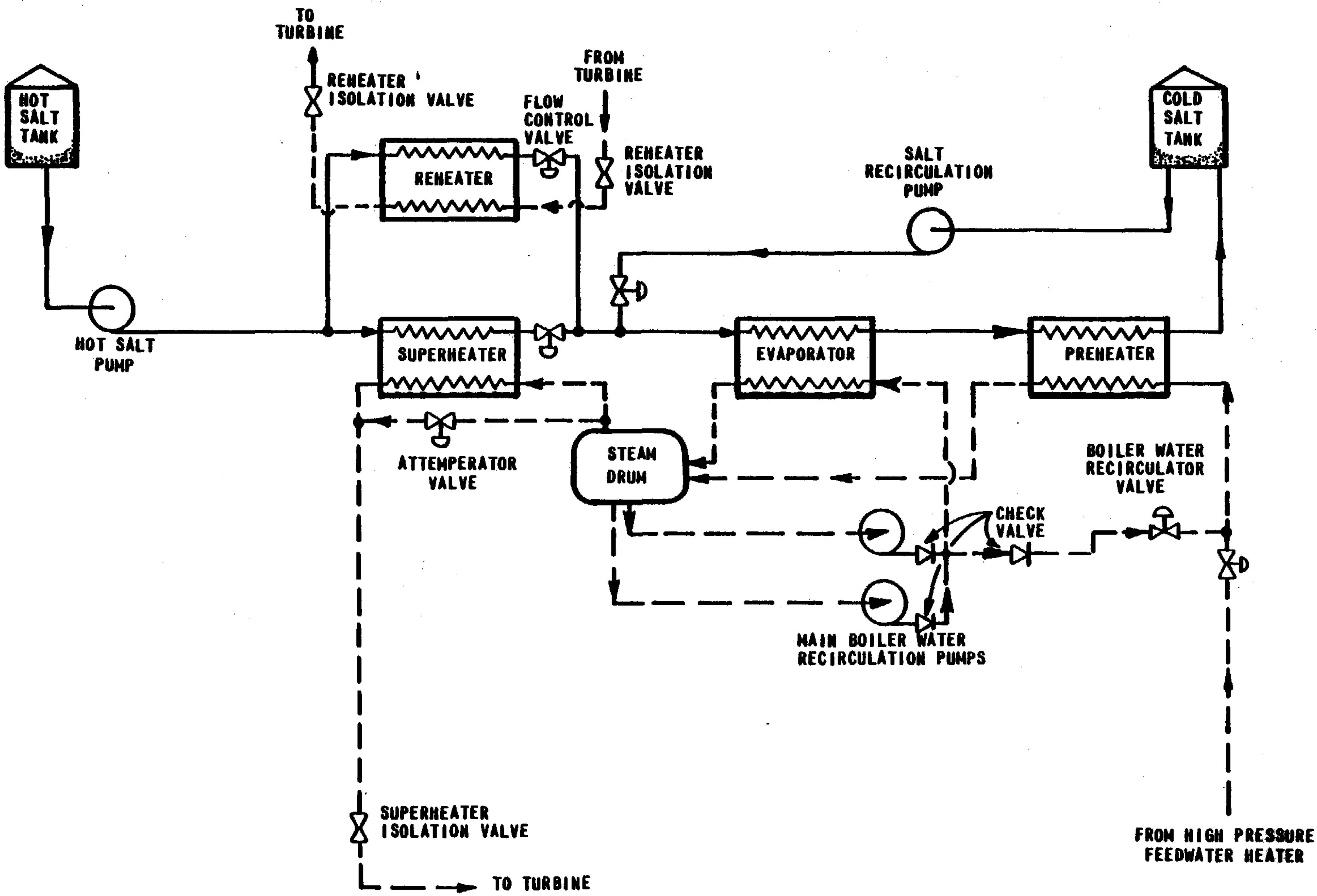


FIG. 4-1 SYSTEM SIMULATION MODEL

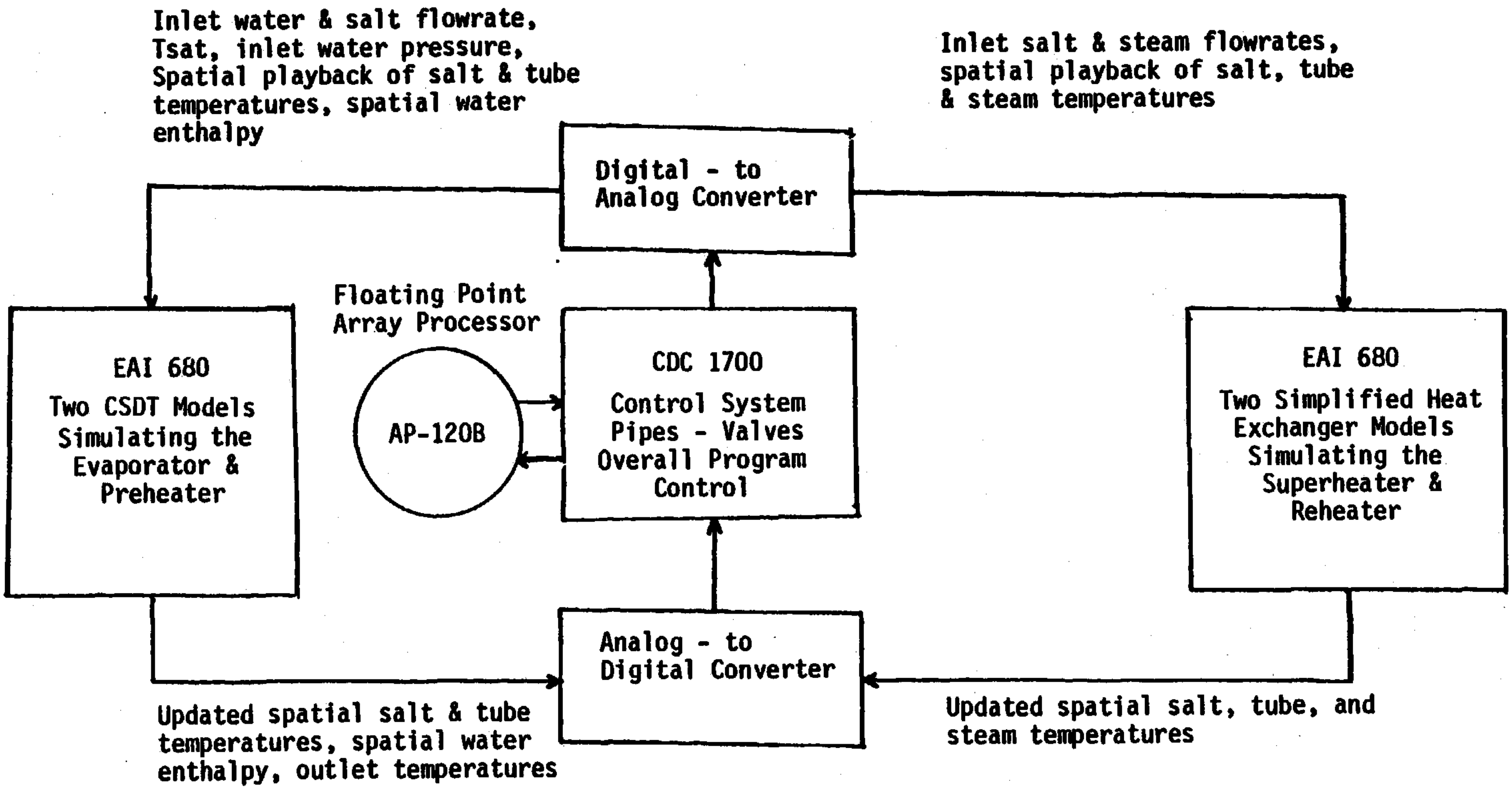


FIG. 4-2 HYBRID MODEL COMPUTING HARDWARE

The piping, valves, steam drum, and the controls were simulated digitally on the CDC 1700. The AP-120B, because of its high speed capability in vector mathematics, was used to obtain the pressure and velocity fields for the subsystem hydraulic network.

Further details of the component modeling, including analytical assumptions, are provided in Appendix E.

The proposed control system was modeled, but not fully implemented, in the initial steady-state and upset transient simulations. The control scheme is generally described in the following paragraphs, while its detailed design and characterization is presented in Section 4.1.2. Functional arrangement diagrams are shown in Figures 5-3 and 5-4.

The total salt flow through the system is controlled in two parallel paths through the superheater and reheater (see Figure 4-1). Both the superheater and reheater salt flow are indexed to a feedforward load signal (steam flow). The major salt flow path (approximately 70 percent of total flow) is through the superheater and is trimmed by an error signal developed between the measured turbine throttle pressure and the operator-established set point. Similarly, the salt flow through the reheater is trimmed by an error signal between the reheater outlet steam temperture and its set point.

Feedwater flow is controlled by a signal developed by comparing feed flow to a feedforward index of steam flow which has been modified, if necessary, by a signal which is proportional to a drum water level signal. This control loop configuration is identical to that used on conventional fossil-fired utility boilers, commonly referred to in the industry as a 3-element feedwater controller.

The total salt flow is defined at any load by thermal balance as long as the salt terminal conditions are controlled to meet system specificatons. This, coupled with the surface margin provided in the superheater, results in the superheater exit steam temperature exceeding 1000°F (538°C) over the control range. A controlled saturated steam by-pass around the superheater, an attemperator, is provided to limit the final turbine steam temperature to 1000°F (538°C). An adaptive control, varying the proportional gain in the temperature feedback loop, is used to control the saturated bypass flow.

It should be noted that, should excessive fouling or excessive tube pluggage or both result in "negative surface margin" in the superheater, there is no convenient control strategy to correct the situation. Such a strategy is theoretically feasible, but would result in storage system thermal degradation, and is therefore considered not cost-effective and is not employed.

The final two control loops are essentially protective in nature. One loop is designed to limit the temperature of salt entering the evaporator to below the maximum temperature with which the evaporator materials of construction are compatible. The other loop limits the temperature of water entering the preheater to a minimum temperature established to assure salt solidification does not occur locally in the vicinity of the preheater salt exit region. Both these control loops are currently depicted and modeled as modulating controls using temperature feedback loops with the proportional gain varied by a load signal. Since the temperature of the salt entering the evaporator does not exceed the allowable (and in fact varies very little) through the load range, and since a very limited area of the preheater primary side tube wall temperature is in the liquidus-solidus range at 30 percent load, it is probable that simple on-off controls can be safely employed for these functions.

The details of the steady-state and upset transient simulations are presented in Appendix E. The basic steady-state results very closely match the results of the VAGEN program which was used to establish the design configuration and size of each of the components and to analyze performance throughout the operating load range. This match lends strong credibility to results of both the hybrid model simulation and the VAGEN analysis.

The upset transients investigated, by their nature, required the control system to be partially negated, since a properly designed control system generally mitigates system upsets. This procedure results in an overly conservative imposition of boundary conditions in some cases. For instance, the feedforward load signal in the reheater was cancelled by setting the proportional gain to zero. Thus, the results of the turbine trip were very similar to the results of the superheater valve closure without trip; yet, neither transient imposed unacceptable thermal loadings on the components.

A properly arranged control scheme would have essentially eliminated these thermal upsets either by throttling all salt flow proportionately in the case of the trip, or by tripping the unit on indication of inadvertent single salt valve closure with similar results. Similar remarks apply to most of the transients examined.

In general, performing the simulation analysis without a valid control system resulted in contrived and conservative thermal loadings the results of which did not invalidate the basic component designs. In several instances, the temperature of salt entering the evaporator was predicted to exceed that used for full-life design, but the short duration of the transient (about 100 seconds) made the event acceptable. Again, implementation of a properly arranged control system would eliminate almost all the more severe transient thermal loadings.

4.1.2 Control System Characterization

The general control scheme was described in Section 4.1.1. The simulated results of the load maneuvering transients are presented in Appendix F. Also included in Appendix F are the detailed basis for choosing between the several control options available and for the determination of initial control gain settings for function and stability over the load range. The control system interface requirements are defined, and the hardware and software options to meet these interface requirements are discussed and recommendations made.

After initial tuning, the combined subsystem thermal-hydraulic model and control model were maneuvered through several load swings. Figures 4-3 through 4-6 are examples of the results of these analyses. Shown in the figures are water and steam flow rates, steam drum and throttle pressures, salt flow rates, and the various subsystem temperatures for the swing of 100 to 30 to 100 percent of full load at 10 percent per minute. Examination of these figures shows that the feedwater flow tracks steam flow very closely after a brief period of over-feed and shorter period of under-feed, while all controlled temperatures and pressures are held to within very narrow and acceptable limits.

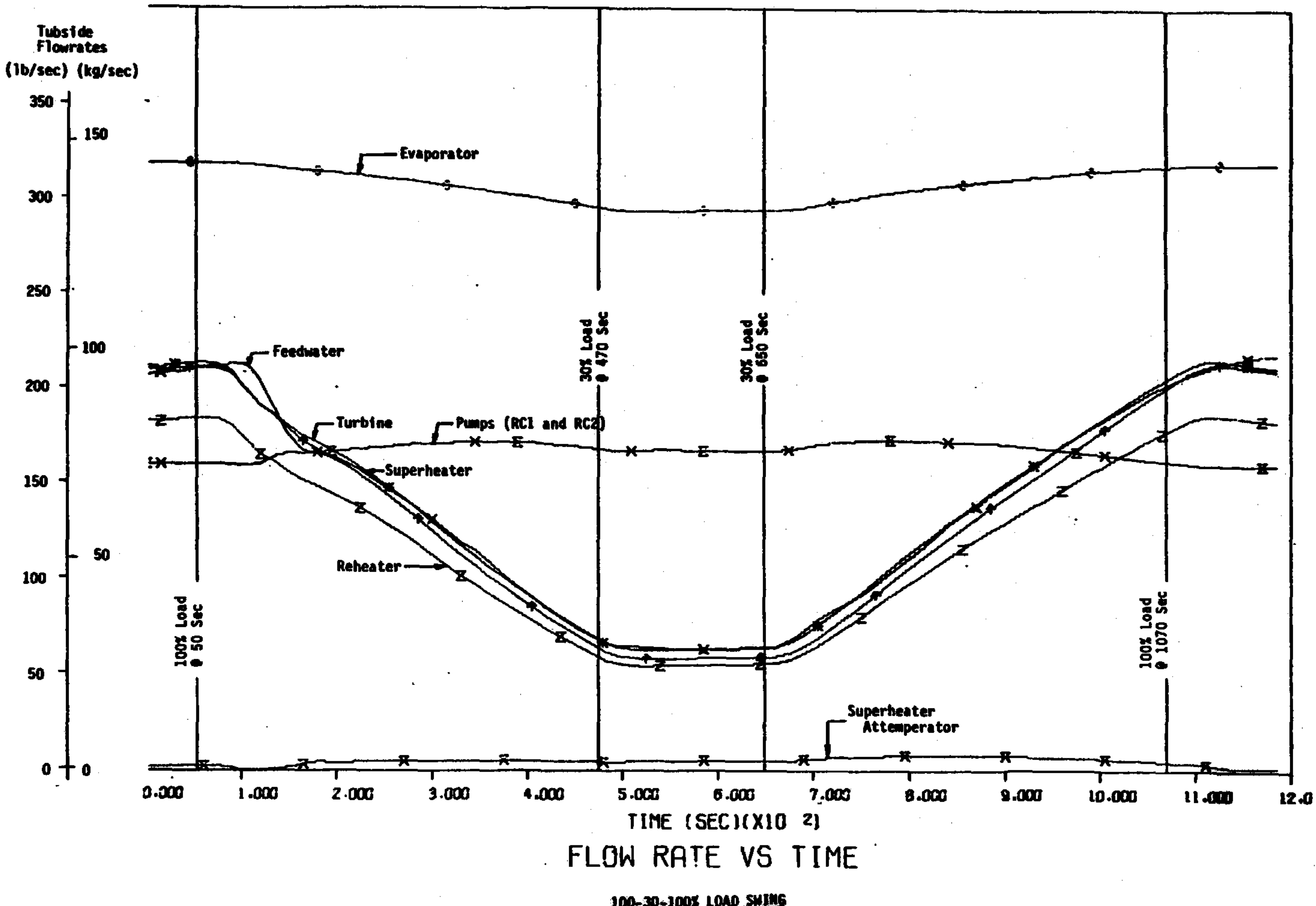


FIG. 4-3 100-30-100% LOAD SWING WATER AND STEAM FLOW

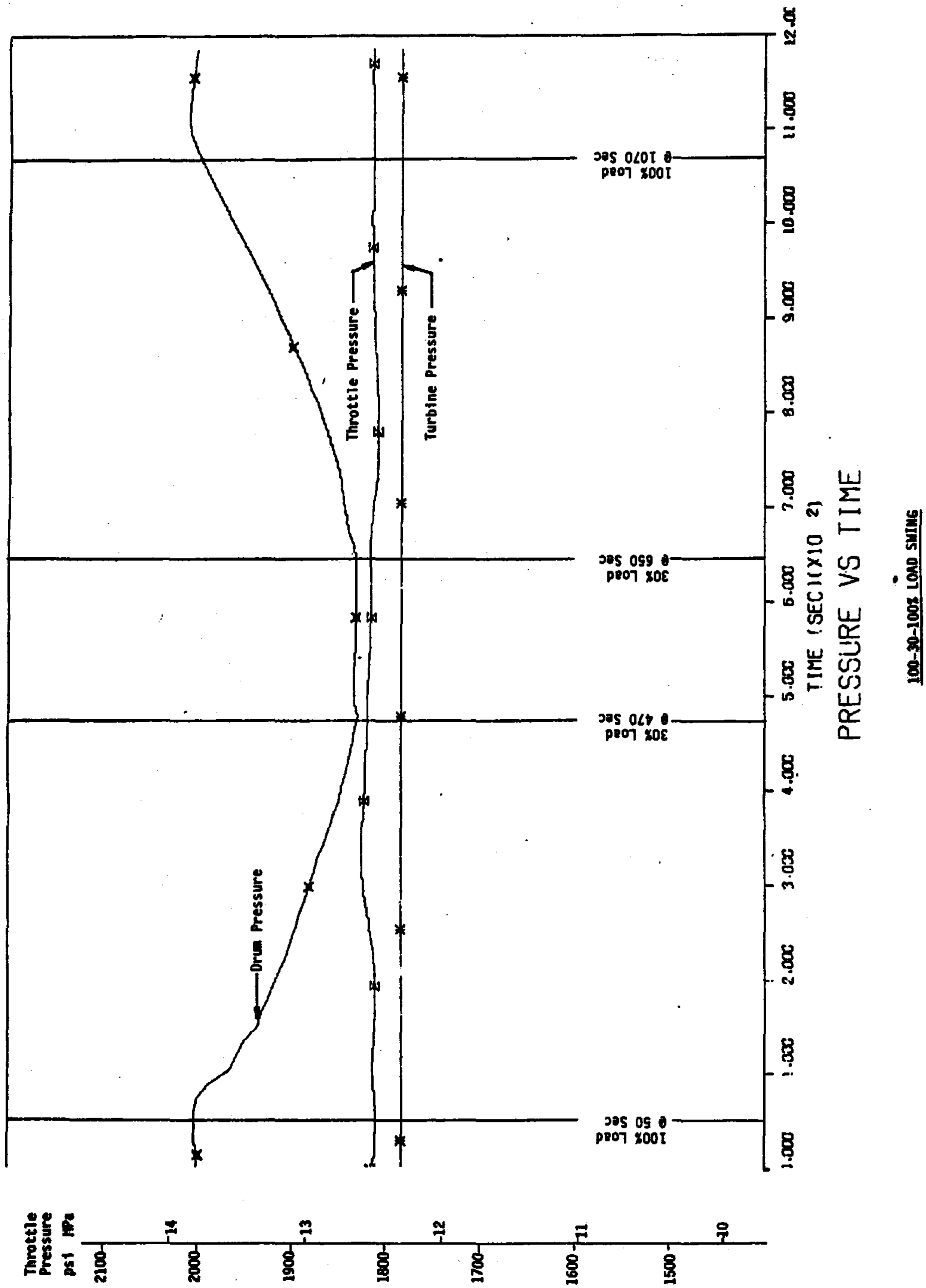


FIG. 4-4 100-30-100% LOAD SWING DRUM AND THROTTLE PRESSURE

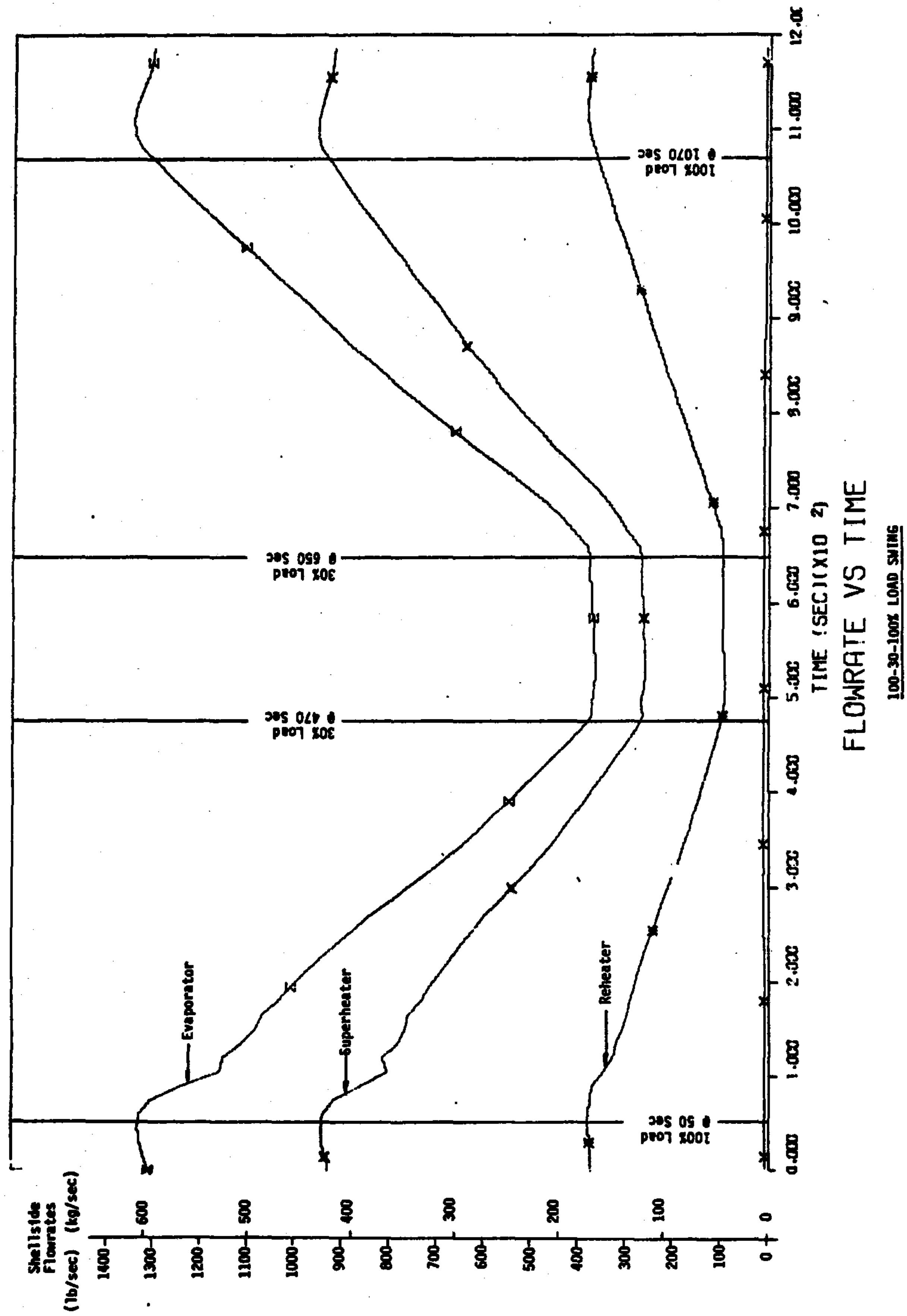


FIG. 4-5 100-30-100% LOAD SWING SALT FLOW

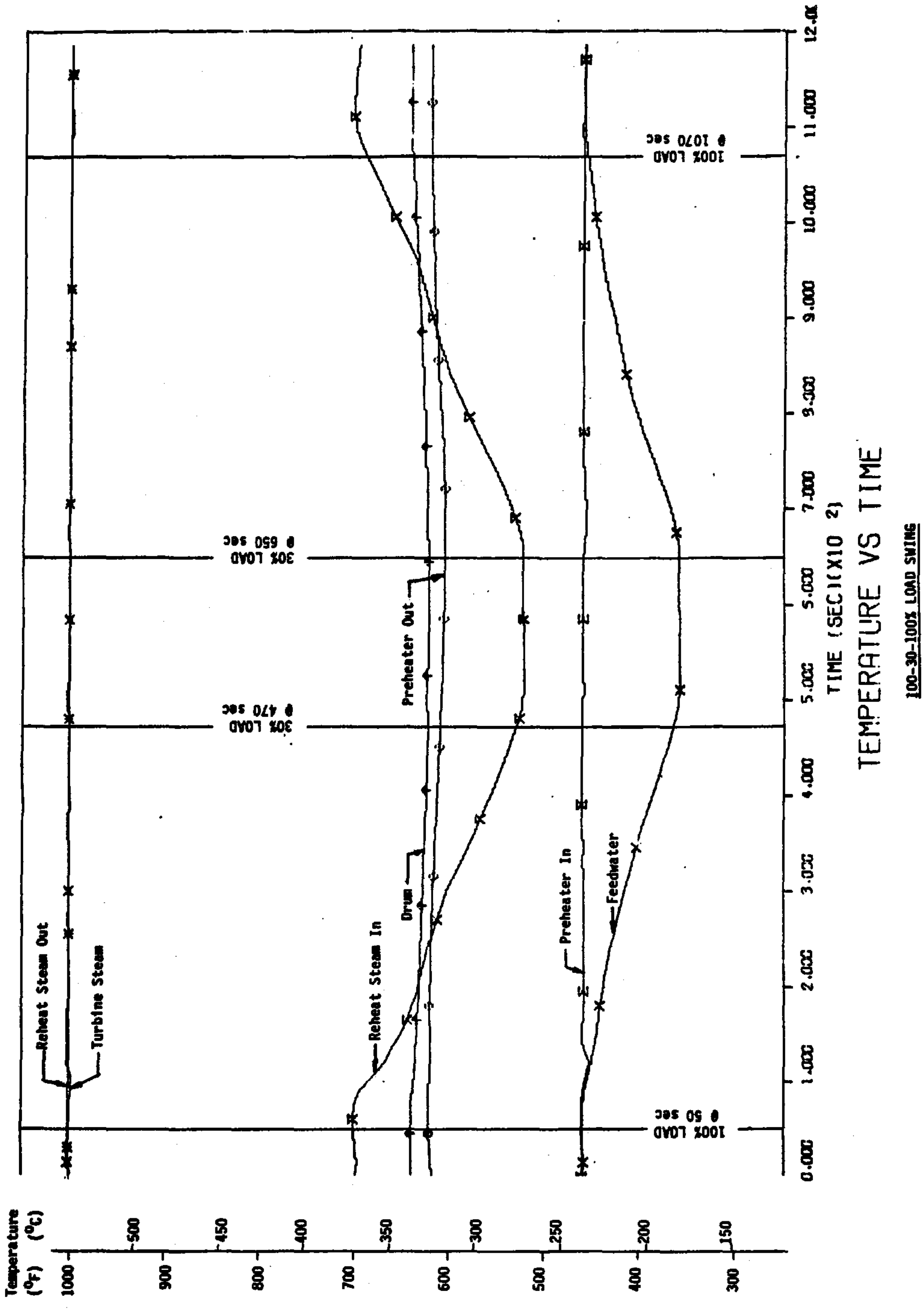


FIG. 4-6 100-30-100% LOAD SWING SUBSYSTEM TEMPERATURES

The hardware reviewed included the Bailey Network 90 and the Taylor 3103 systems. The Bailey system is a distributed control system; that is, the control modules (electrically alterable read only memory microprocessors) are located physically near the process to be controlled. They are linked to the operator console in the control room by a "data highway". because the control algorithms are self-contained, the integrity of single loops is maintained if the data highway goes down. The Taylor system is based on a centrally-located mini-computer. The computer stores and implements all control algorithms; thus, the integrity of the data highway, or communication link, is crucial. The mini-computer can provide sequential control which is ideal for startup operations.

It was concluded that the advantages of the less expensive computer-based system are outweighed by its reduced operational reliability, and the distributed control system is preferred.

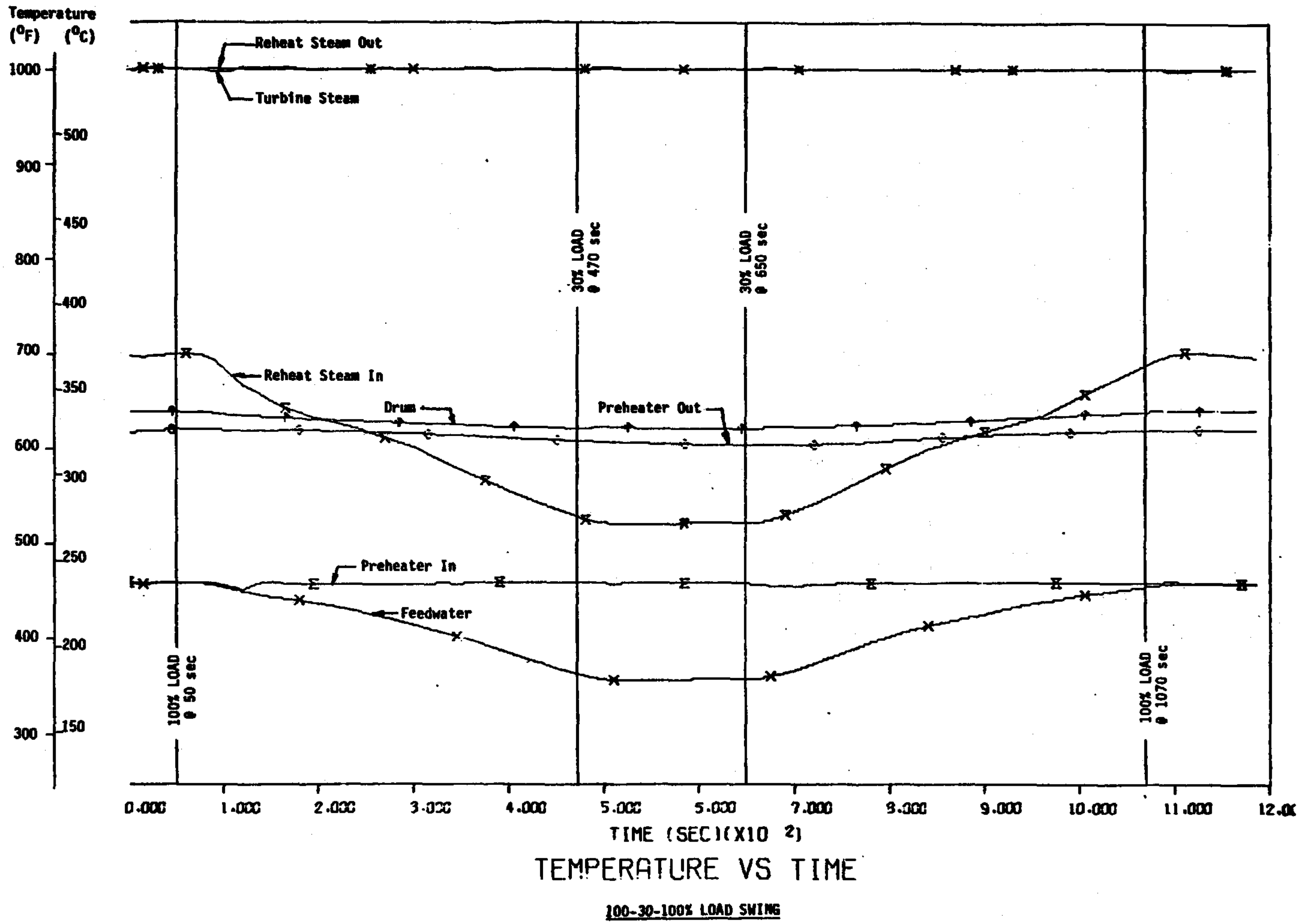


FIG. 4-6 100-30-100% LOAD SWING SUBSYSTEM TEMPERATURES

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It was concluded that the advantages of the less expensive computer-based system are outweighed by its reduced operational reliability, and the distributed control system is preferred.

4.2 Part-Load Performance

In order to perform a part load performance analysis, an operating scenario must be defined. Several options, each with certain advantages, are available. In one option, salt flow is held proportional to the required heat transport load and, thus, cold salt temperature remains constant over the load range. Another option is to hold turbine throttle pressure constant over the load range and provide the salt flow required to accomplish this. In this case, the cold salt temperature varies such that the combination of salt flow and temperature drop provides the requisite thermal energy for the steam system. The amount of deviation from the nominal cold salt temperature of 550°F is a function of both the full load steam line and superheater pressure drop and the surface margin provided in the evaporator and preheater.

Since the concept of repowering requires that low load solar superheater outlet pressure match the full load pressure of the fossil-fired unit, it was deemed appropriate to analyze operation over the load range holding the throttle pressure constant. For a solar stand-alone plant, either of the options described, or a combination of both, can be utilized.

Saturation pressure, boiler water circulation ratio, and fluid temperatures at strategic points throughout the system are shown in Figures 4-7, 4-8, and 4-9. It is apparent from the thermal profiles that the bulk salt outlet temperature falls to approximately 510 °F (266 °C) at 30 percent load. At this low bulk outlet temperature, it can be expected that local salt temperatures at the tube wall will approach the freezing point of the salt. For this reason, at low loads, it will be necessary to bypass a portion of the recirculated boiler water to warm the feedwater and limit the bulk salt outlet temperature to a safe minimum.

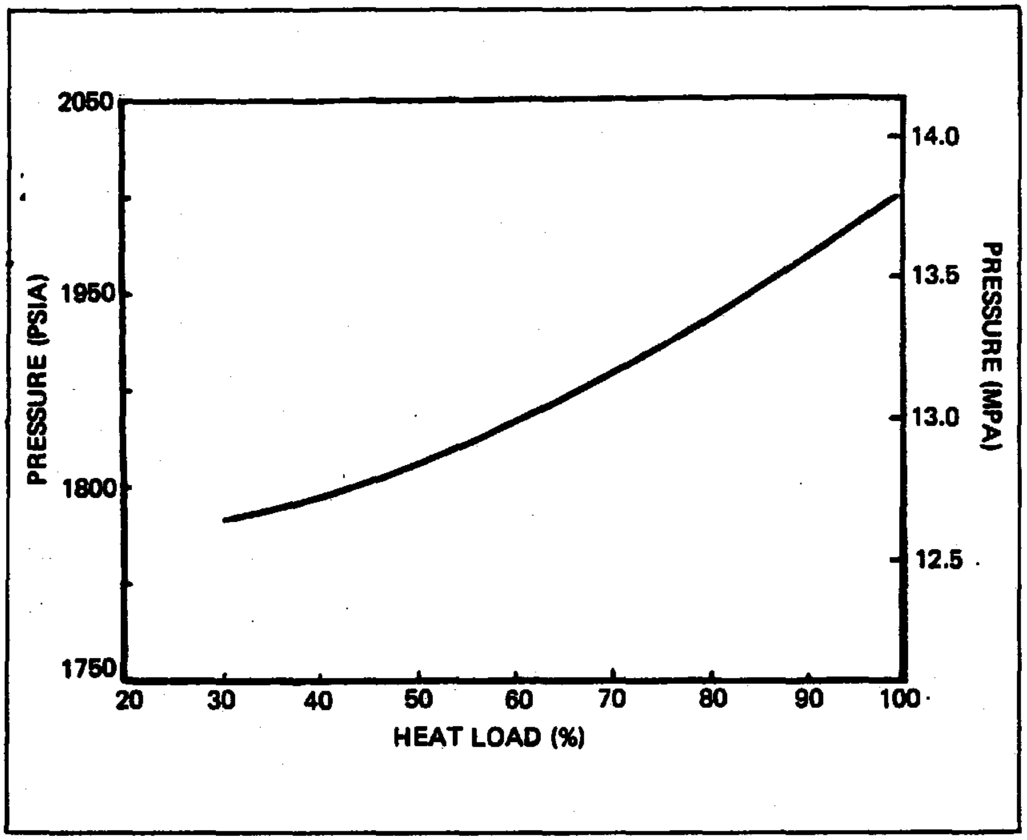


FIG. 4-7 SATURATION PRESSURE (PART LOAD OPERATION)

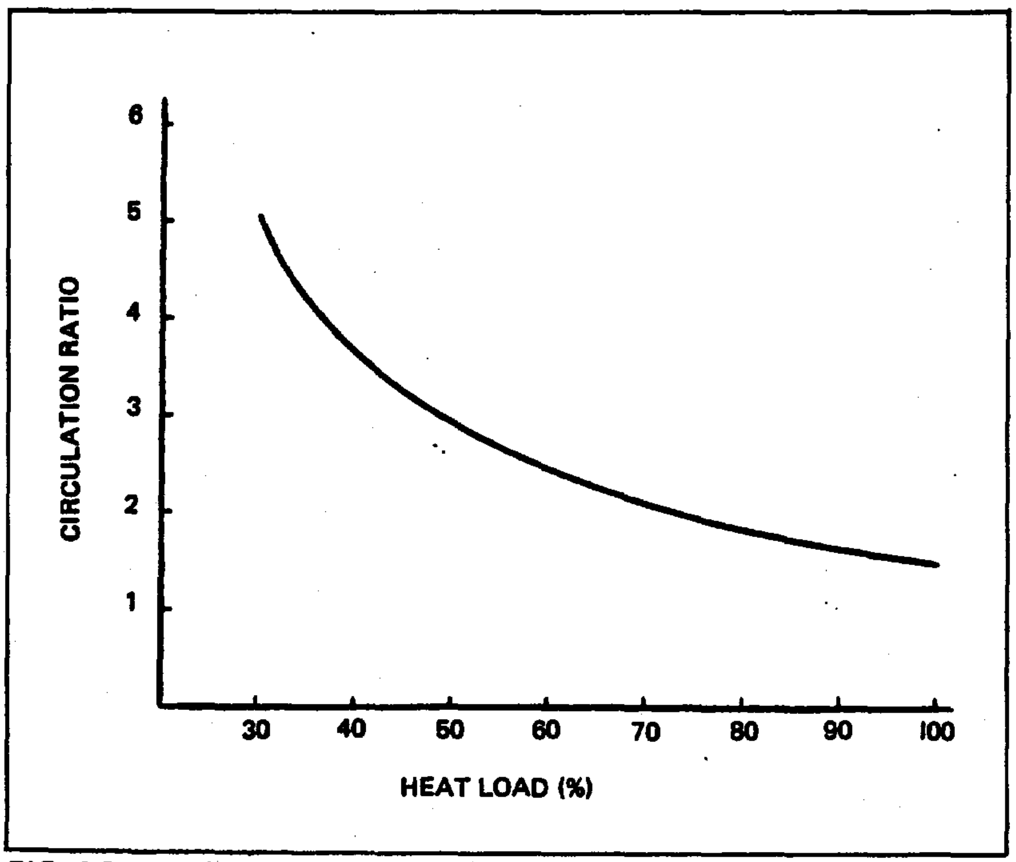


FIG. 4-8 CIRCULATION RATIO (PART LOAD OPERATION)

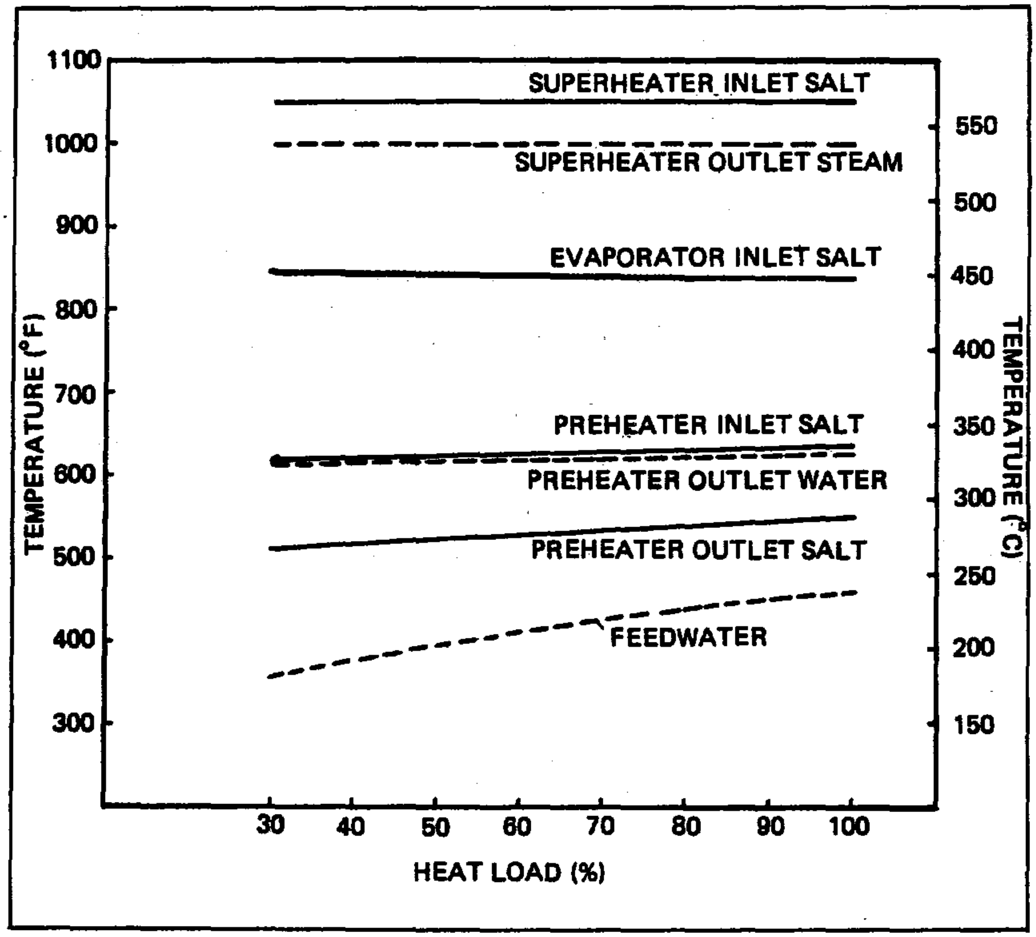


FIG. 4-9 THERMAL PROFILES (PART LOAD OPERATION)

4.3 Startup and Shutdown Procedures

Key steps in the procedures for cold startup, diurnal startup, and diurnal shutdown of the steam generator subsystem in a stand-alone solar plant are presented in Tables 4-1, 4-2 and 4-3. The startup and shutdown procedures for repowered plants are similar to those tabulated. However, for repowered plants, the following exceptions are of interest:

(a) Steam for cold startup is available from the existing plant, eliminating the need for a special auxiliary boiler to service the solar system.

(b) In bringing the solar reheater on line, special care is necessary to assure that the fossil-fired reheater is not starved of flow. It will probably be necessary to place a control valve in the reheat steam line to assure that flow is properly apportioned between the solar and fossil power systems.

TABLE 4-1

Startup from Cold, Dry Condition

1. Align all valves for startup - fill system with water of specified purity - start boiler recirculation pump.

2. Open steam line from auxiliary boiler - sparge steam into drum water - control steam flow to achieve pre-determined heatup rate - activate trace heaters on component shells and on salt piping.

3. After vigorous venting, close vents and permit steam generator pressure and temperature to ramp at a controlled pre-determined rate.

4. When the separately controlled reheater has reached 254 ^{o}C (489 ^{o}F) and the other components and piping have reached a minimum 260^{o}C (500^{o}F), introduce salt at 288 ^{o}C (550 ^{o}F) - recirculate salt through the components back to the cold salt storage tank at a pre-determined minimum rate.

5. Start to bleed hot salt into recirculated cold salt and ramp hot salt flow fraction of total flow - control rate of pressure rise by adjusting steam flow to the condenser - start to warm steam lines.

6. When steam temperature exceeds saturation temperature by 56 ^{o}C (100 ^{o}F), admit steam to turbine - close steam bypass to condenser lines - continue to ramp hot salt flow fraction during turbine run-up - after generator is synchronized, ramp total salt flow (while continuing to increase hot salt flow fraction) until 30 percent load is reached - if not already, bring hot salt flow fraction to unity.

7. Transfer controls from manual to automatic.

TABLE 4-2

Diurnal Startup

Assumptions:	Steam generator subsystem is at 288 °C (550 °F) with cold salt being circulated at minimum rate; and turbine throttle block temperature is between 288 °C (550 °F) and 538 °C (1000 °F)
Constraints:	Steam for initial turbine roll is a minimum of 56 °C (100 °F) of superheat; and, to the maximum extent possible, the steam temperature matches the turbine throttle block temperature (only possible if turbine temperature exceeds 316 °C (600 °F)

1. If turbine temperature exceeds 343 °C (650 °F), bleed hot salt into cold salt stream and ramp total salt temperature - hold steam generator pressure at 7.2 MPa (1040 psia) by control of steam flow to condenser - continue to ramp salt temperature at constant salt flow - when superheater outlet steam temperature matches turbine throttle block metal temperature, proceed to Step 3.

2. If turbine temperature is below 343 °C (650 °F), open turbine bypass to condenser and reduce steam generator pressure so that saturation temperature is 56 °C (100 °F) below that of the turbine metal temperature, but no lower than 260 °C (500 °F) - ramp salt temperature at constant salt flow - when superheater outlet steam temperature reaches saturation temperature plus 56°C (100 °F), proceed to Step 3.

3. If necessary, heat steamlines - open turbine throttle - continue to ramp salt temperature during turbine run-up.

4. After synchronization, ramp salt flow and salt temperature until 30 percent load is reached and hot salt flow only is achieved.

5. Transfer controls from manual to automatic.

TABLE 4-3

Diurnal Shutdown

1. Reduce load to 30 percent on automatic control-transfer controls to manual operating mode.

2. Reduce salt flow (and concomitant EPGS load) to minimum flow - trip turbine and salt pump.

3. Continue to circulate steam generator water until system equilibrates thermally - activate trace heaters on cold salt piping.

4. Monitor system temperatures and, if required, start salt pump to trickle flow cold salt through steam generator subsystem - bleed in hot salt if required.

5.0 BALANCE OF SUBSYSTEM

The major components in the Steam Generator Subsystem are the preheater, steam drum, evaporator, superheater, and reheater. However, proper operation of these components depends on the design of ancillary components and structures which are referred to here as balance of subsystem. This section begins with a discussion of the physical arrangement of subsystem components; that discussion is followed by descriptions of the piping and electrical distribution systems as well as of structural supports and foundations. Where appropriate, separate descriptions are provided for the 100 MW and 50 MW subsystem designs.

5.1 Physical Arrangement

The physical arrangements of the major components in the 100 MW and 50 MW Steam Generator Subsystems are shown in Figures 5-1 and 5-2, respectively. In both cases, consideration is given to effective utilization of available space and minimizing piping distances consistent with piping flexibility requirements. The 100 MW arrangement is designed as a new, stand-alone facility. The 50 MW arrangement is designed as a repowering unit, or supplementary steam source for an existing electrical power generation facility.

5.1.1 100 MW Subsystem Design

The 100 MW unit is designed as a new power generation facility not associated with an existing plant. The arrangement shown in Figure 5-1 is designed to allow the steam generator components to be incorporated into a typical turbine cycle feedwater heater bay that would be associated with a new plant design. The figure indicates the location of typical feedwater heaters relative to the major Steam Generator Subsystem components.

The bay is an open space frame structure providing structural support for the equipment, piping, and platforms. The equipment has been arranged on four levels to accommodate cascade draining of the salt through the heat exchangers into the sump which is located below grade. The steam drum is located about 12 m (40 ft) above the recirculation pumps to provide the NPSH required by the pumps.

5.1.2 50 MW Subsystem Design

The 50 MW unit is designed to be added to an existing power station. For this application, the most economical arrangement is an outdoor, slab-mounted design with all equipment except the steam drum located on a concrete pad. As indicated in Figure 5-2, the drum is elevated to provide adequate net positive suction head for the recirculation pumps; this arrangement prevents cavitation in the pumps. All other equipment including the recirculation pumps, preheater, evaporator, superheater, and reheater are mounted on the single, grade-level slab. The salt sump is located adjacent to the slab and below grade to facilitate draining the salt from subsystem components and piping.

5.2 Piping System

The functional arrangement of the salt piping associated with the Steam Generator Subsystem is shown in Figure 5-3. The functional arrangement of the feedwater and steam piping is shown in Figure 5-4. Interfaces with the piping in the Energy Storage and Electric Power Generation Subsystem exist at the edge of the equipment bay for the 100 MW design and at the edge of the grade level slab for the 50 MW design.

5.2.1 Functional Arrangement

As shown in Figure 5-3, hot salt is supplied to the Steam Generator Subsystem from the hot salt transfer pump. The flow of hot salt splits, with a larger flow rate of salt being supplied to the superheater than to the reheater. The two salt flow streams leaving the superheater and reheater merge before entering the evaporator. After leaving the evaporator, the salt flows through the preheater and to the cold salt storage tank. During start-up, cold salt is mixed with hot salt upstream of the superheater and reheater; this prevents thermal shock of the two heat exchangers. During shutdown and subsequent draining of salt from subsystem components, cold salt is introduced upstream of the evaporator; by mixing this cold salt with the higher temperature salt leaving the superheater and reheater, thermal shock of the evaporator and preheater is avoided.

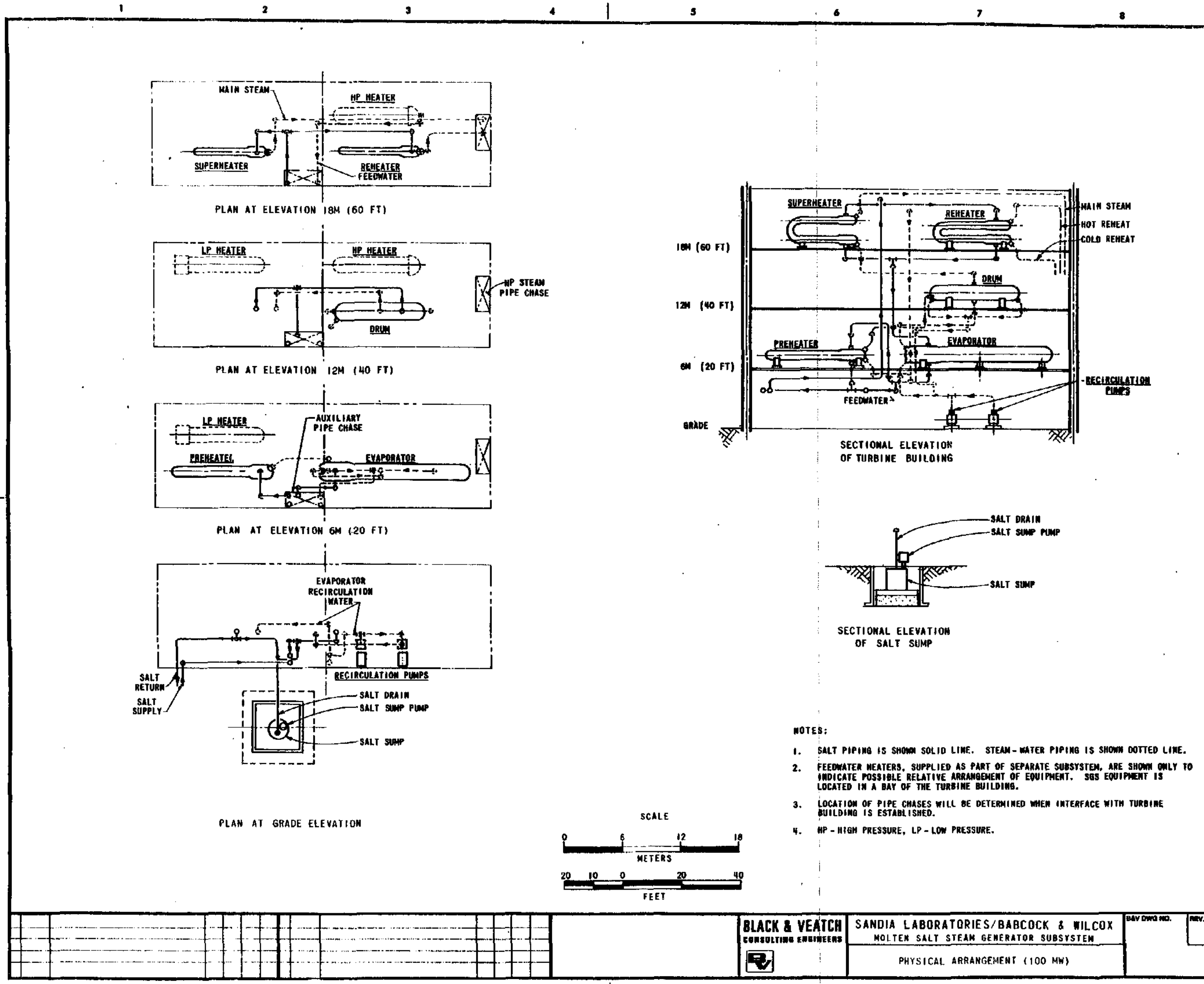


FIG. 5-1 PHYSICAL ARRANGEMENT (100 MW)

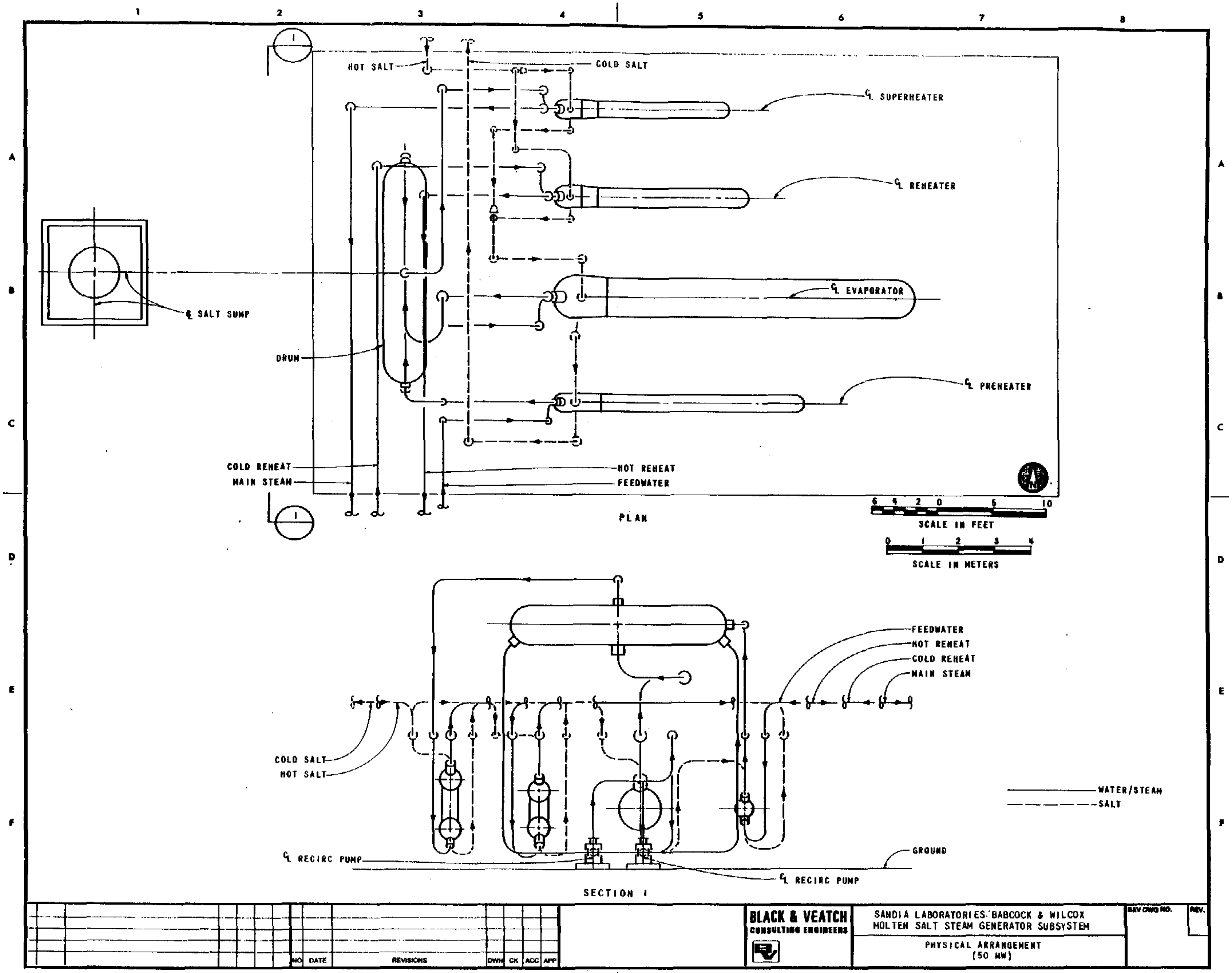


FIG. 5-2 PHYSICAL ARRANGEMENT (50 MW)

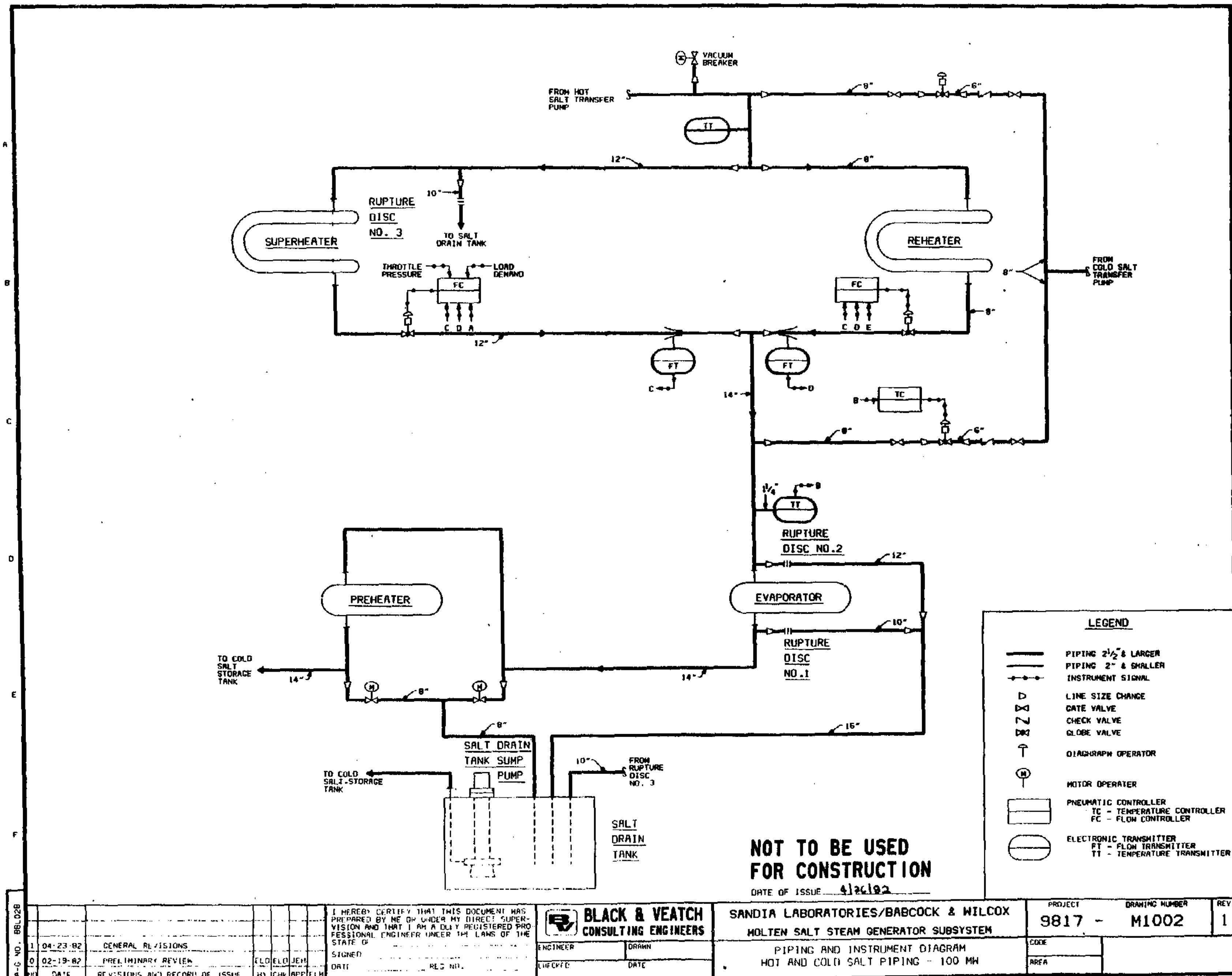


FIG. 5-3 PIPING AND INSTRUMENT DIAGRAM
HOT AND COLD SALT PIPING

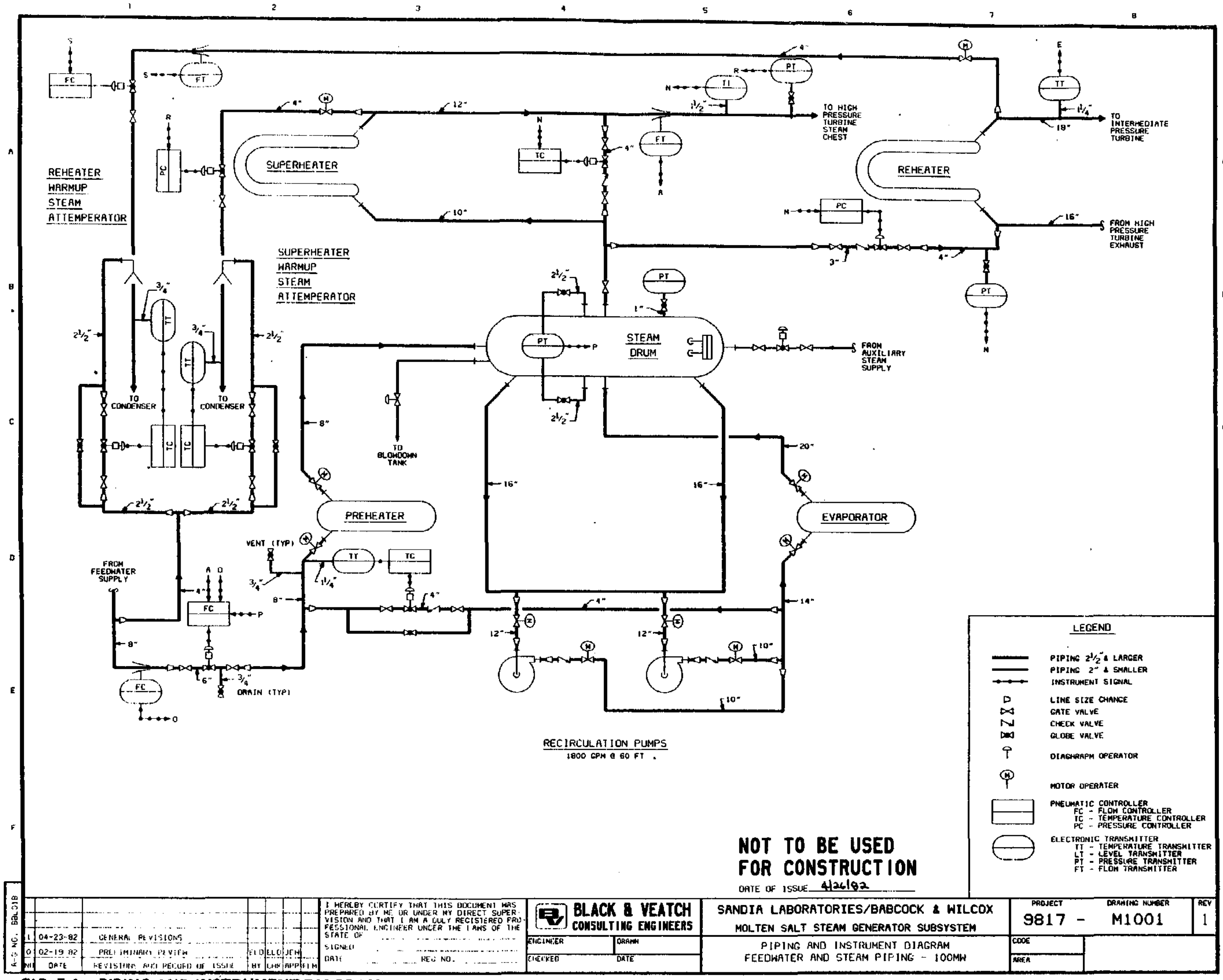


**FIG. 5-4 PIPING AND INSTRUMENT DIAGRAM
FEEDWATER AND STEAM PIPING**

In addition three rupture discs are provided to protect subsystem components from abnormal, high pressure conditions. Salt drained from the system and salt flowing through a rupture disc is routed to the sump. Salt in the sump is pumped to the cold salt storage tank.

Feedwater is supplied to the preheater as shown in Figure 5-4. After leaving the preheater, the feedwater enters the steam drum where it is combined with a mixture of saturated liquid and saturated vapor from the evaporator. Saturated liquid leaves the bottom of the steam drum through two parallel lines and enters a header, from which two half-capacity recirculation pumps take suction. These pumps provide sufficient head to overcome friction losses in the recirculation loop; pump discharge flow enters the evaporator. The mixture of saturated liquid and saturated vapor is introduced to the bottom of the steam drum. In the drum, saturated steam is separated from the saturated liquid. This steam is then routed to the superheater. Leaving the superheater, the steam is supplied to the high pressure turbine throttle valves. A by-pass line around the superheater provides steam temperature control by mixing saturated steam from the steam drum with steam from the superheater outlet. In a separate set of interfaces with the Electric Power Generation Subsystem, steam from the high pressure turbine exhaust flows through the reheater and is introduced to the intermediate pressure turbine inlet. During start-up, steam from an auxiliary steam supply is used to raise the temperature and pressure in the steam drum. In addition, steam from the drum is used to warm the superheater and reheater. In this warm up process, but before acceptably high steam temperatures are reached, superheater and reheater outlet steam is dumped to the condenser; feedwater is supplied to attemperators in both lines to desuperheat these steam flows. A by-pass line is provided around the recirculation pumps to protect them against overheating during low flow operation. Finally, a blowdown line permits removal of excess solids which accumulate in the bottom of the steam drum.

Routing of the major piping is indicated in Figures 5-1 and 5-2. These routing schemes were selected to minimize piping costs and accommodate the anticipated thermal stresses of the piping systems. Routing of the salt piping was also selected to allow the piping to be conveniently drained.

5.2.2 Pipe Selection

The pipe material, sizes, and wall thicknesses selected for all major piping in the Steam Generator Subsystem are listed in Table 5-1. Pipe materials were selected based on the design temperatures and the fluids carried by the various piping sections. Stainless steel pipe is used for salt piping with design temperatures above 538°C (1000°F), whereas alloy steel was selected for steam piping with the comparable design temperatures. Alloy steel piping is also utilized for salt and steam piping with design temperatures between 371°C (700°F) and 538°C (1000°F). For design temperatures below 371°C (700°F), carbon steel pipe was selected for salt, steam, and feedwater.

Pipe sizes were selected based on velocity and pressure loss considerations.

Specification of pipe wall thicknesses is based on the design pressures and temperature of the various piping sections. The thicknesses listed in Table 5-1 were selected such that the piping stresses due to the internal pressures do not exceed safe working levels and include an appropriate corrosion allowance. Wall thicknesses were selected in accordance with the American National Standards Institute code for Pressure Piping, B31.1, and the American Society of Mechanical Engineers Boiler and Pressure Vessel Code.

5.2.3 Valves

The materials and pressure classes specified for all major valves in the system are listed in Table 5-1. Valve material was selected for compatability with the pipe material and the design pressures and temperatures of the various piping sections. Valve pressure classes were selected in accordance with the American National Standards Institute code for Steel Butt-Welding End Valves B16.34.

TABLE 5-1 PIPELINE LISTING

PIPING SECTION	PRESSURE AND TEMPERATURE OPERATING	PRESSURE AND TEMPERATURE DESIGN	PIPE SIZE	PIPE MATERIAL	VALVES	INSULATION THICKNESS
Feedwater piping to preheater	238 C (460 F) 13.86 MPa (2010 psia)	243 C (470 F) 15.27 MPa (2215 psia)	100mm (4"), 150mm (6"), 200mm (8") Sch 120	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	89mm (3.5")
Feedwater piping from preheater to steam drum	331 C (627 F) 13.79 (2000 psia)	338 C (640 F) 15.27 MPa (2215 psia)	200mm (8") Sch 120	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	100mm (4")
Feedwater piping from steam drum to recirc pump	331 C (630 F) 13.79 MPa (2000 psia)	338 C (640 F) 14.75 MPa (2140 psia)	300mm (12"), 400mm (16") Sch 140	Carbon steel ASTM A106 Gr B.	Carbon steel CL 1500	140mm (5.5")
Feedwater piping from recirc pump to evaporator	331 C (630 F) 13.82 MPa (2005 psia)	338 C (640 F) 15.17 MPa (2200 psia)	100mm (4"), 250mm (10"), 350mm (14") Sch 140	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	4" Pipe - 89mm (3.5") 10" Pipe - 100mm (4") 14" Pipe - 140mm (5.5")
Steam piping from evaporator to steam drum	331 C (636 F) 13.79 MPa (2000 psia)	338 C (650 F) 15.17 MPa (2200 psia)	500mm (20") Sch 160	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	140mm (5.5")
Steam piping from steam drum to superheater	331 C (636 F) 13.79 MPa (2000 psia)	338 C (650 F) 14.75 MPa (2140 psia)	100mm (4"), 250mm (10") Sch 140	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	4" Pipe - 89mm (3.5") 10" Pipe - 100mm (4")
Steam piping from superheater to turbine	538 C (1000 F) 12.86 MPa (1865 psia)	546 C (1015 F) 14.75 MPa (2140 psia)	300mm (12") 1.631" Min Wall	Alloy steel ASTM A335 Gr P22	Alloy steel CL 2500 Special	150mm (6")
Salt piping from salt pumps to boiler area	566 C (1050 F)	579 C (1075 F) 1.41 MPa (205 psia)	350mm (14") Sch 10S	Stainless steel ASTM A312 TP 304	Stainless steel CL 300	150mm (6")

TABLE 5-1 PIPELINE LISTING (cont.)

PIPING SECTION	PRESSURE AND TEMPERATURE OPERATING	DESIGN	PIPE SIZE	PIPE MATERIAL	VALVES	INSULATION THICKNESS
Salt piping from boiler area to superheater	566 C (1050 F)	579 C (1075 F) 1.41 MPa (205 psia)	300mm (12") Sch 20S	Stainless steel ASTM A312 TP 304	Stainless steel CL 300	150mm (6")
Salt piping from boiler area to reheater	566 C (1050 F)	579 C (1075 F) 1.41 MPa (205 psia)	200mm (8") Sch 20S	Stainless steel ASTM A312 TP 304	Stainless steel CL 300	127mm (5")
Salt piping from superheater and reheater to evaporator	448 C (838 F) 1.07 MPa (155 psia)	482 C (900 F) 1.41 MPa (205 psia)	200mm (8"), 300mm (12"), 350mm (14") Std Wt	Alloy steel ASTM A335 Gr P22	Alloy steel CL 600	8" Pipe - 127mm (5") 12" Pipe - 150mm (6") 14" Pipe - 150mm (6")
Salt piping from evaporator to preheater	336 C (637 F) 1 MPa (145 psia)	371 C (700 F) 1.41 MPa (205 psia)	350mm (14") Std Wt	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	140mm (5.5")
Salt piping from preheater to cold salt storage	288 C (550 F)	316 C (600 F) 1.41 MPa (205 psia)	350mm (14") Std Wt	Carbon steel ASTM A106 Gr B	Carbon steel CL 1500	140mm (5.5")
Salt piping from rupture discs and main piping to drain tank	- -	482 C (900 F) 1.41 MPa (205 psia)	200mm (8") 250mm (10") 300mm (12") 400mm (16")	Alloy steel ASTM A335 Gr P22	Alloy steel CL 600	8" Pipe - 127mm (5") 10" Pipe - 150mm (6") 12" Pipe - 150mm (6") 16" Pipe - 150mm (6")

Salt valves in piping with design temperatures of 538°C (1000°F) and above are stainless steel. Salt valves in piping with design temperatures between 427°C (800°F) and 538°C (1000°F) are alloy steel. Salt valves in piping with design temperatures below 427°C (800°F) are carbon steel. Steam valves in pipe with design temperatures above 371°C (700°F) are made of alloy steel. Valves in steam and feedwater pipe with design temperatures lower than 371°C (700°F) are made of carbon steel.

5.2.4 Recirculation Pumps

The recirculation pumps are horizontal, wet motor pumps designed specifically for boiler circulating water service. The pumps circulate water from the steam drum through the evaporator and back to the drum. Two half-capacity pumps are used; each has a total developed head sized to overcome the piping and evaporator friction losses plus a suitable design margin. Each pump has a capacity of 409 m^3/hr (1800 gpm) for the 100 MW design and 204.5 m^3/hr (900 gpm) for the 50 MW design. Pump head is 18.3 m (60 ft) for both size subsystems.

5.2.5 Heat Tracing

Electrical heat tracing in the steam generator subsystem is designed to prevent salt solidification or water freezing in the heat exchangers, valves, and process piping. Heat tracing cable is flexible, self-limiting, parallel circuit type with a monitor conductor for surfaces of low (65°C, 149°F maximum) operating temperature. For surfaces of high operating temperatures (up to 572°C, 1061°F), heat tracing cable is mineral insulated with an Inconel sheath. Controls are provided to assure a minimum molten salt temperature of 277°C (530°F) when the plant is shut down; the trace heaters are de-energized during plant operation. Alarms are included to indicate system malfunction.

5.2.6 Insulation and Lagging

The thicknesses of the calcium silicate insulation which were selected for the various piping sections are listed in Table 5-1. These thicknesses were selected based on the operating temperatures of the pipe and represent a trade-off between initial capital costs of the insulation and energy loss costs. The insulation is protected and secured in place by aluminum lagging.

5.3 Electrical Distribution System

The Electrical Distribution System consists of cable, raceway, and all devices necessary to provide electrical power to required steam generator subsystem loads. The system equipment includes motor control center breakers and starters, dry type distribution transformers, cable, and raceway. Electrical power to the recirculation pumps, distribution transformers, and heat tracing equipment is 480 volts, three phase, 60 Hz. Control and instrumentation power is supplied at 120 volts, single phase, 60 Hz.

Power and control cable is 600 volt insulated, flame retardant, with a normal maximum operating temperature of 90°C (194°F). The minimum size for power cable is 12 AWG and for control cable is 14 AWG. Instrumentation cable is 300 volt insulated with a normal maximum operating temperature of 75°C (167°F) and a minimum size of 16 AWG.

Raceway consists of rigid galvanized steel conduit, cable tray, and wiring devices as required.

The Electrical Distribution System interfaces with the existing plant 480 volt system at the motor control center level, as shown in Figure 5-5.

Motor starters for the recirculation pumps and motor operated valves are required. Feeder breakers for the heat tracing and dry type transformers are also required. Significant electrical loads are listed in Table 5-2.

TABLE 5-2

STEAM GENERATOR SUBSYSTEM AUXILIARY POWER REQUIREMENTS

COMPONENT	100 MW Plant (kW)	50 MW Plant (kW)
Recirculation Pump A	20.8	11.1
Recirculation Pump B	20.8	11.1
Heat Tracing	140	105
Dry Type Transformers	10	7.5
TOTAL	191.6	134.7

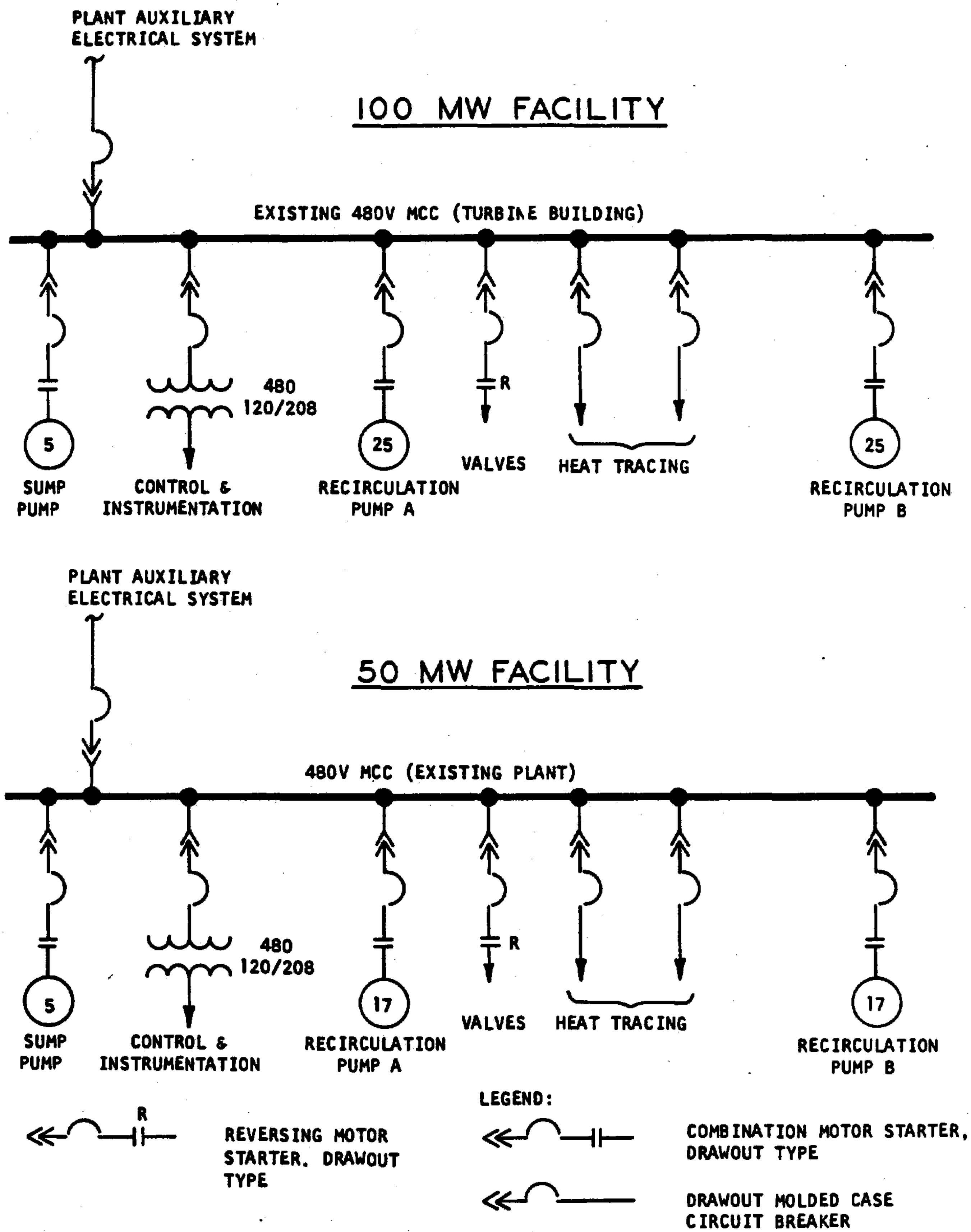


FIG. 5-5 NORMAL AC POWER SUPPLY ONE-LINE DIAGRAM

5.4 Structural Supports and Foundations

The design of the Steam Generator Subsystem included foundations and support structures for the following items:

(a) preheater

(b) evaporator

(c) superheater

(d) reheater

(e) steam drum

(f) salt and water/steam piping

Structural supports for salt and water/steam piping are designed according to standard industry practice. Expansion loops are provided as required to allow for thermal expansion. Where appropriate, piping is hung from structural support steel.

All equipment and piping supports are designed for gravity, wind, seismic, and thermal loads, as appropriate. Structural steel is ASTM Grade A36 and/or Grade A42, and satisfies the requirements of AISC. Concrete has a minimum design strength of 281 Kg/cm^2 (4000 psi) and steel reinforcement is Grade 60, and meets the requirements of ACI-318,77.

5.4.1 100 MW Subsystem Design

For the 100 MWe design, the preheater, evaporator, superheater, reheater, and steam drum are supported on a four-level open structural steel braced frame as shown in Figure 5-1. A reinforced concrete slab provides structural support for the steel frame structure, and forms the floor of the bottom level. Support saddles for the heat exchangers and steam drum are located directly on steel support beams. Lubrite pads are provided at required saddle support locations to allow for thermal expansion of the components; hanger rods for the superheater and reheater are supported from floor steel above the heat exchangers. Grating is provided in open areas of each level, and a stairway and a caged ladder are provided at opposite ends of the structure.

5.4.2 50 MW Subsystem Design

For the 50 MWe design, the preheater, evaporator, superheater, reheater, and steam drum are supported on a common reinforced concrete slab as shown in Figure 5-2. As shown in Figure 5-6, a structural steel braced frame, located at one end of the slab, supports the steam drum at the approprite elevation and serves as a fixed support for subsystem piping; a caged ladder provides access to the drum level. The superheater and evaporator are supported on saddles, which in turn are supported on the concrete slab. Lubrite pads are provided to allow for thermal expansion of the heat exchangers. The lower legs of the superheater and reheater are supported in a smiliar manner to the preheater and evaporator; however, the upper legs of the superheater and reheater are supported by steel hanger rods, which in turn are supported by small structural steel frames.

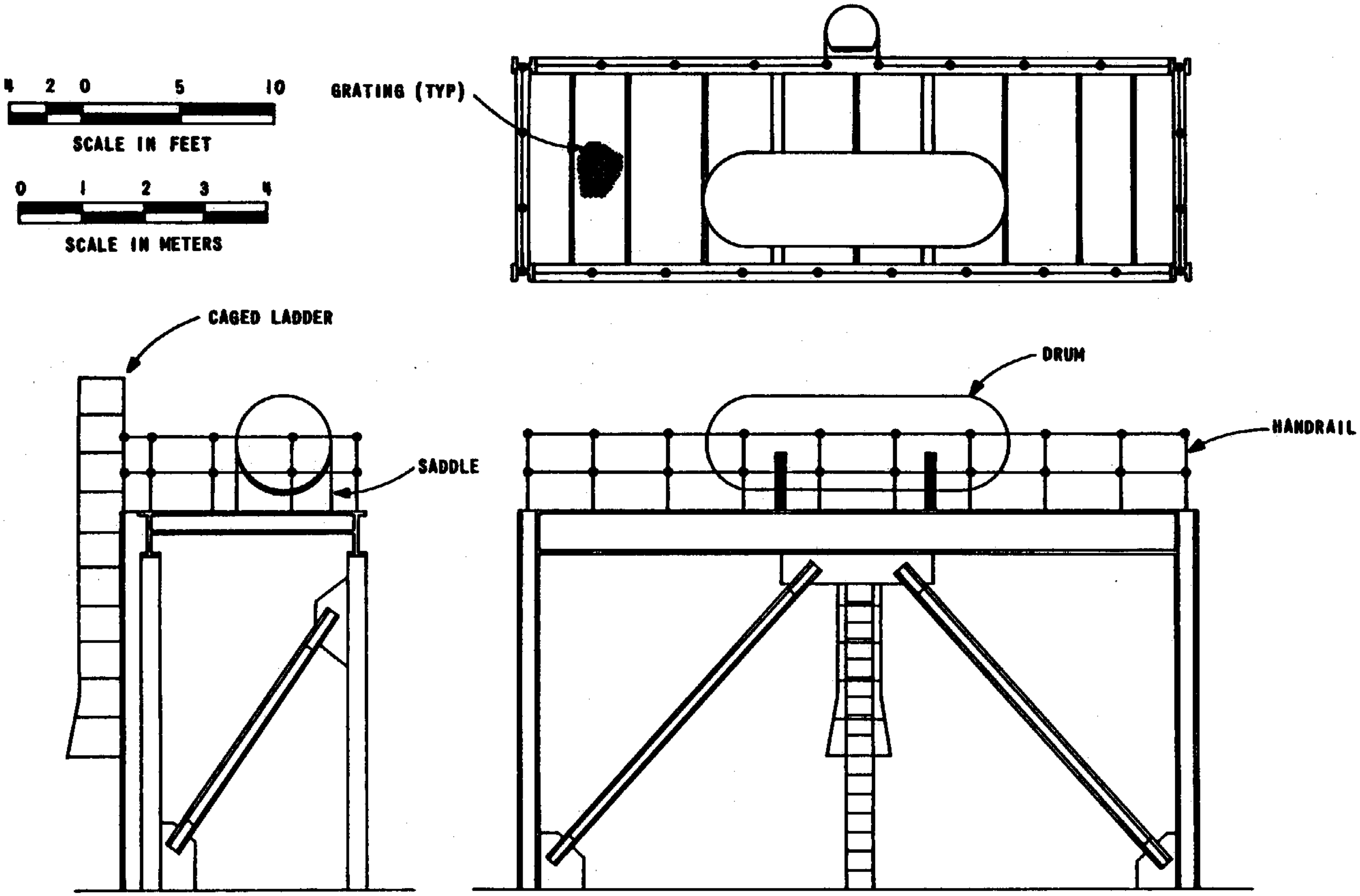


FIG. 5-6 50 MWe STEAM DRUM SUPPORT STEEL

6.0 FABRICATION AND FIELD ERECTION PLAN AND COST ESTIMATE

Plans, schedules, and cost estimates were developed for shop fabrication and field erection of the steam generator subsystem. Both a 100 MWe stand-alone plant and a 50 MWe repowered plant were considered.

Detailed process outlines were prepared to assure that schedules and cost estimates were properly established. The proposed fabrication methods are based on proven, qualified techniques and present no unusual risks or uncertainties. Cost estimates were developed using standards data, actual costs escalated from previous contracts, material vendor quotations, and catalog prices.

6.1 Shop Fabrication Plan

The manufacturing activities to be completed in fabricating the principal components of the steam generator subsystem (preheater, evaporator, superheater, reheater, and steam drum) are briefly described in the following paragraphs:

1. Detailed fabrication drawings will be prepared and reviewed to assure that design requirements are satisfied and that the components can be fabricated in a cost-effective manner.

2. Material ordering information will be prepared and reviewed.

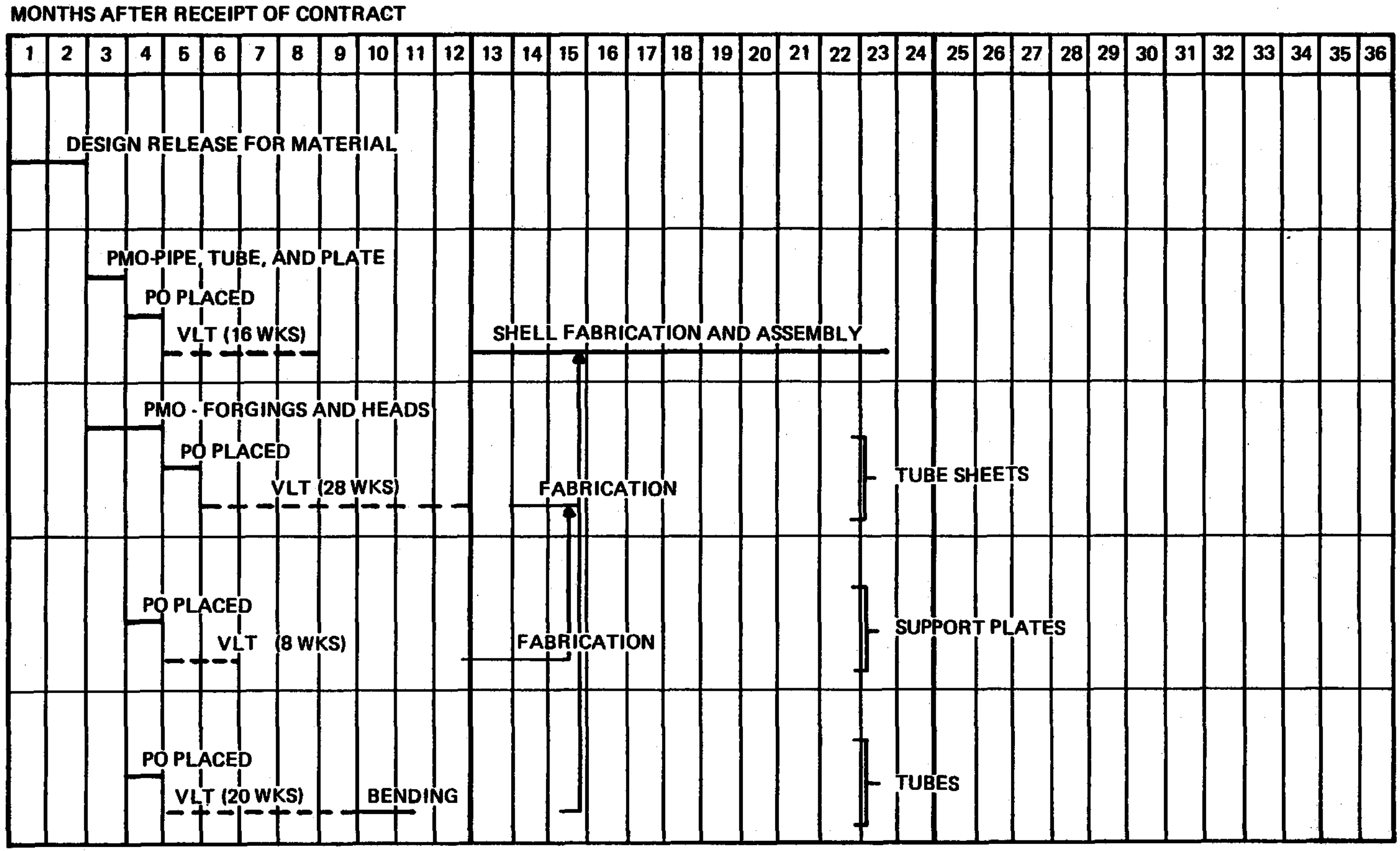
3. Detailed fabrication processing will be prepared. This processing will provide the manufacturing shop with the instructions required for fabrication, inspection, and testing of the components.

4. The detailed processing will be reviewed, and the manufacturing operations will be spanned and scheduled for completion. Fabrication activities will be monitored and expedited as necessary throughout the project.

5. Fabrication, inspection, and testing will be completed in accordance with the detailed processing instructions, design drawings, and approved procedures. All operations will be documented as they are completed.

6. The detailed processing will be reviewed to assure proper completion and documentation of all operations before the components are code stamped and shipped.

Fabrication schedules for each of the components are shown in Figures 6-1 through 6-4.



PMO - PROCUREMENT MATERIAL ORDER
PO - PURCHASE ORDER
VLT - VENDOR LEAD TIME

FIG. 6-1 FABRICATION SCHEDULE FOR SUPERHEATER AND REHEATER (100 MWE PLANT)

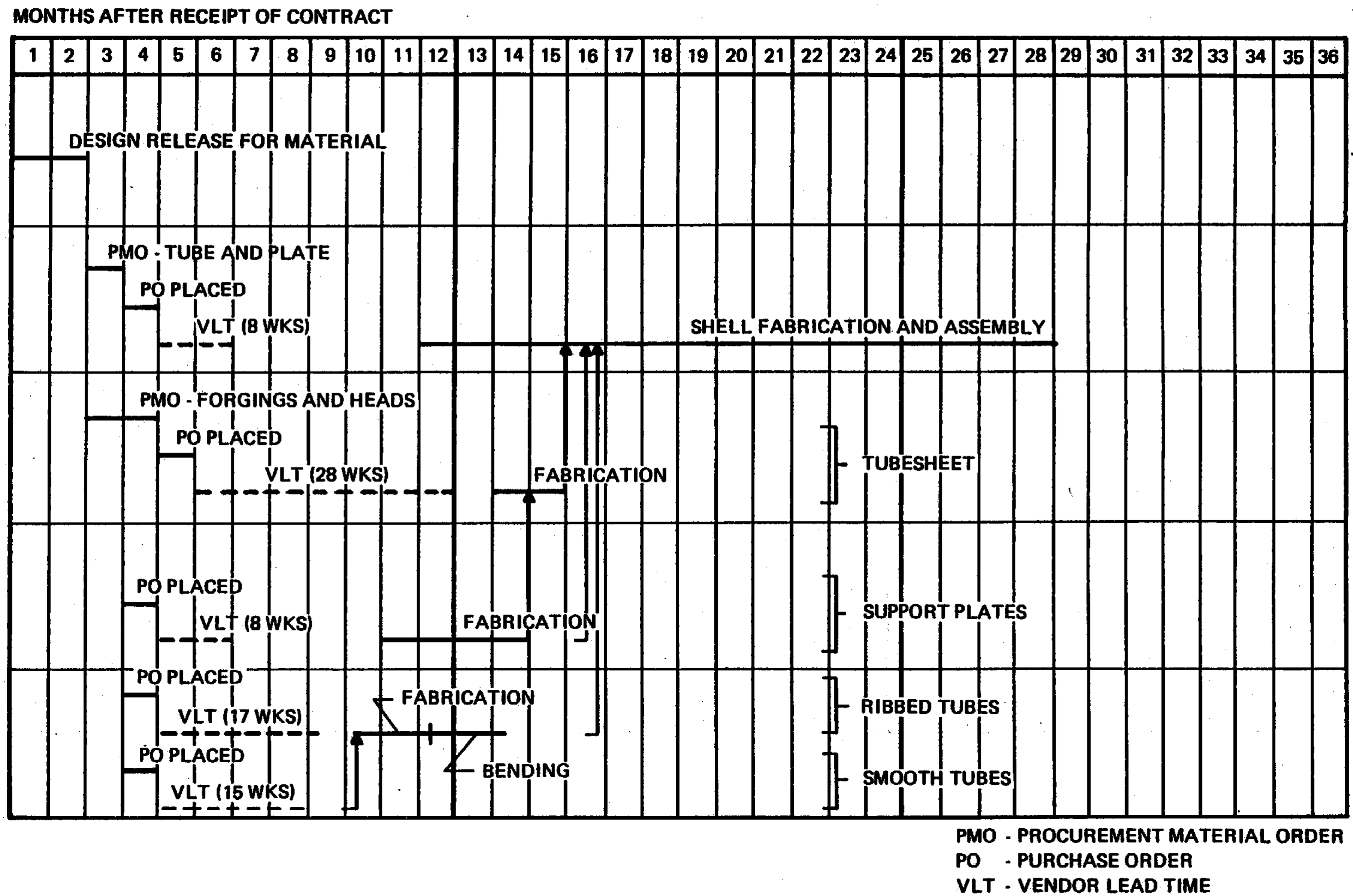
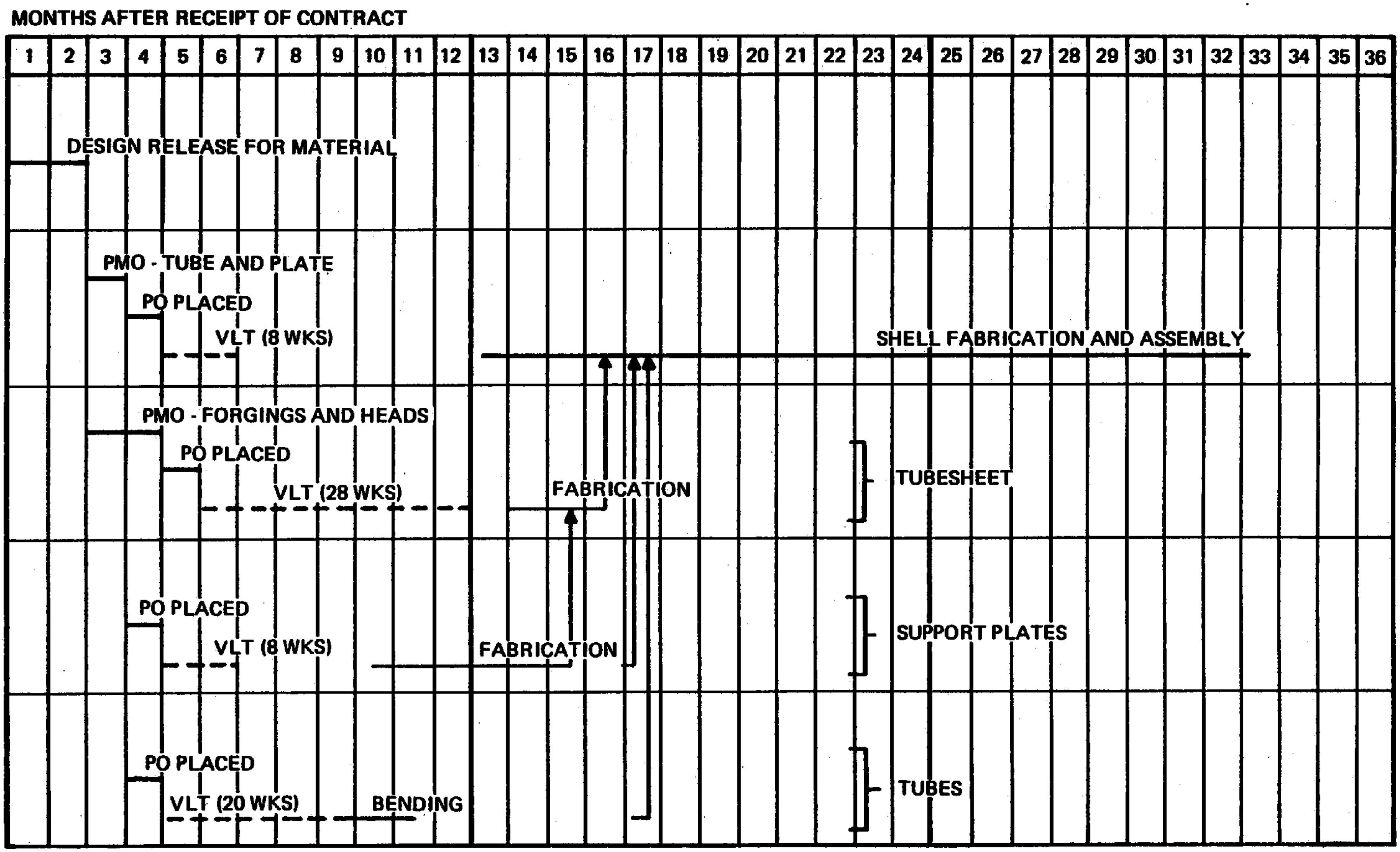
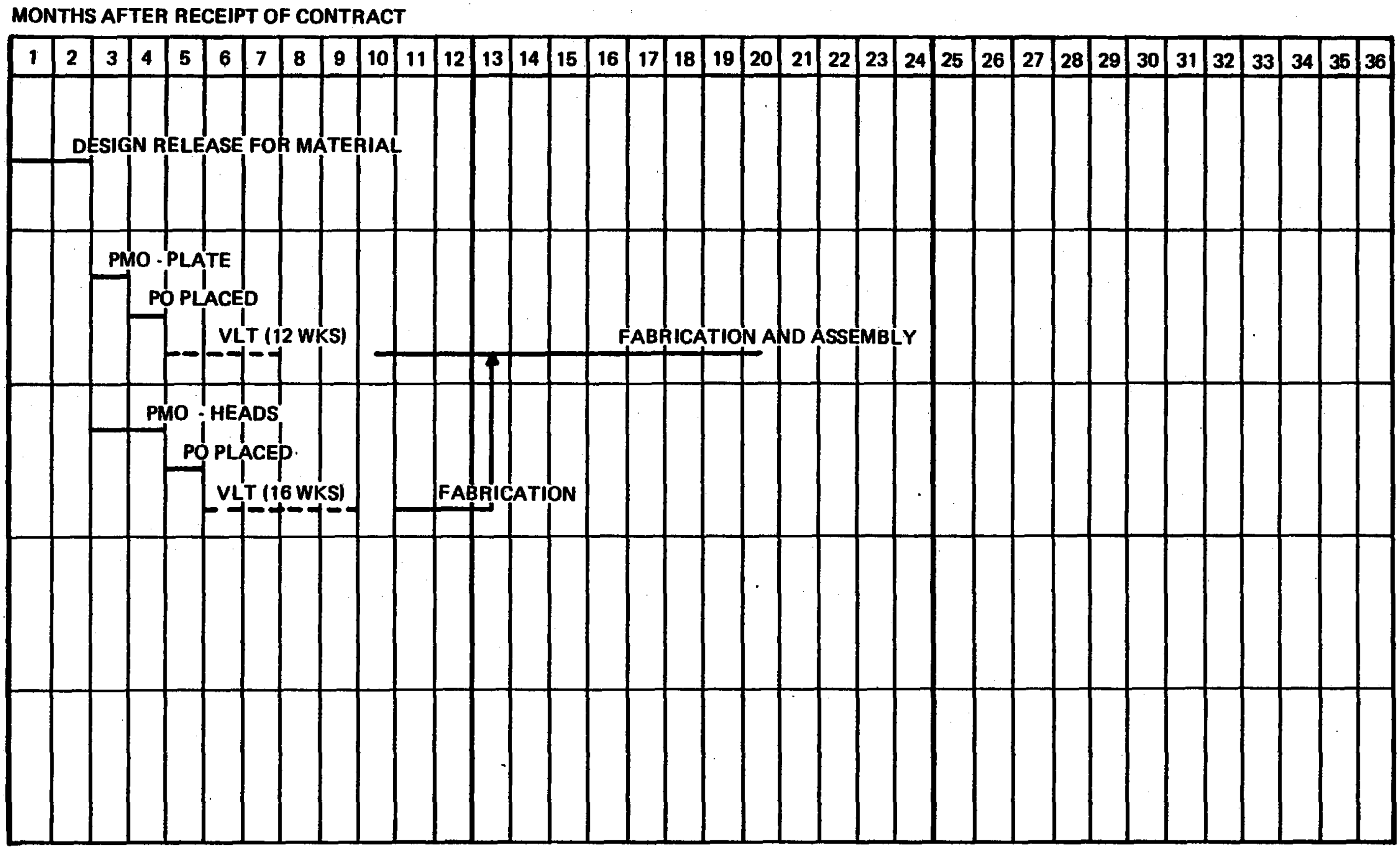


FIG. 6-2 FABRICATION SCHEDULE FOR EVAPORATOR
(100 MWE PLANT)



PMO - PROCUREMENT MATERIAL ORDER
PO - PURCHASE ORDER
VLT - VENDOR LEAD TIME

FIG. 6-3 FABRICATION SCHEDULE FOR PREHEATER (100 MWE PLANT)



PMO - PROCUREMENT MATERIAL ORDER
PO - PURCHASE ORDER
VLT - VENDOR LEAD TIME

FIG. 6-4 FABRICATION SCHEDULE FOR STEAM DRUM (100 MWE PLANT)

6.2 Field Erection Plan

The 100 MWe steam generator subsystem is part of a new stand-alone solar power plant. As such, field erection of this subsystem would be integrated with erection of the rest of the plant. Because most of the components are located in a bay of the turbine building, field erection of the steam generator subsystem begins after the turbine building structure is available. During the first month of field construction, the work consists mainly of setting the equipment in the turbine building and beginning the installation of piping and hangers. The second month involves completion of piping and hangers and the erection of the salt drain line on overhead supports outside the building. The drain line is routed to the sump pit. The last month of field construction is devoted to installing heat tracing and insulation, and making final piping and cable terminations.

The 50 MWe Steam Generator Subsystem is part of a retrofit plant or an addition of solar steam generation capability to an existing power plant. As such, field erection of subsystem components can be considered independently, except for the final physical interconnection of subsystem piping and controls with the existing plant. Field erection of the 50 MWe subystem begins with pouring the foundation and erecting steel for the steam drum. After the erection of the steam drum support structure, the mechanical equipment is set and supported before beginning piping and wiring. Heat tracing, insulating, and terminations are the final elements of field construction before system checkout and startup.

6.3 Integrated Schedule

The integrated design, shop fabrication, and field erection schedule, applicable to both the 100 MWe and 50 MWe plants, is shown in Figure 6-5. Important events are identified by milestones.

6.4 Cost Estimates

Cost estimates for design, shop fabrication, and field erection of the steam generator subsystem for the 100 MWe stand-alone plant and the 50 MWe repowered plant are presented in Tables 6-1 and 6-2. Important assumptions upon which the estimates are based are defined below:

(a) Costs are expressed in current dollars (May, 1982).

(b) The steam generator subsystem for the 100 MWe plant is part of a new stand-alone solar plant located near Tucson, Arizona. The steam generator subsystem for the 50 MWe plant is a repowering addition to the Arizona Public Service Company's Saguaro plant.

(c) The cost estimate excludes owner's costs such as land, licenses and permits, taxes, and cost of money.

(d) Building costs for the 100 MWe plant are for average cost of floor space in a turbine building. The building is assumed to be open construction (no walls). For the 50 MWe plant, structure costs include site preparation, foundation, support structure for the drum, and steel for saddles and hanger supports.

(e) For the 100 MWe plant, feedwater and steam piping is costed to the pipe chases, and salt piping to the edge of the turbine building. For the 50 MWe plant, all piping is costed to the edge of the steam generator subsystem foundation. Drain lines are costed to the sump.

(f) Electrical equipment includes heat tracing, cable, conduit, grounding ties, and instrumentation and control circuits. The motor control center and grounding grid are assumed to be part of the main electric system (100 MWe plant) or existing plant (50 MWe plant). Costs for convenience receptacles, lighting, and communication equipment are not included in the estimate.

(g) Sump costs include tank, concrete, insulating brick, excavation, backfill, and handrail.

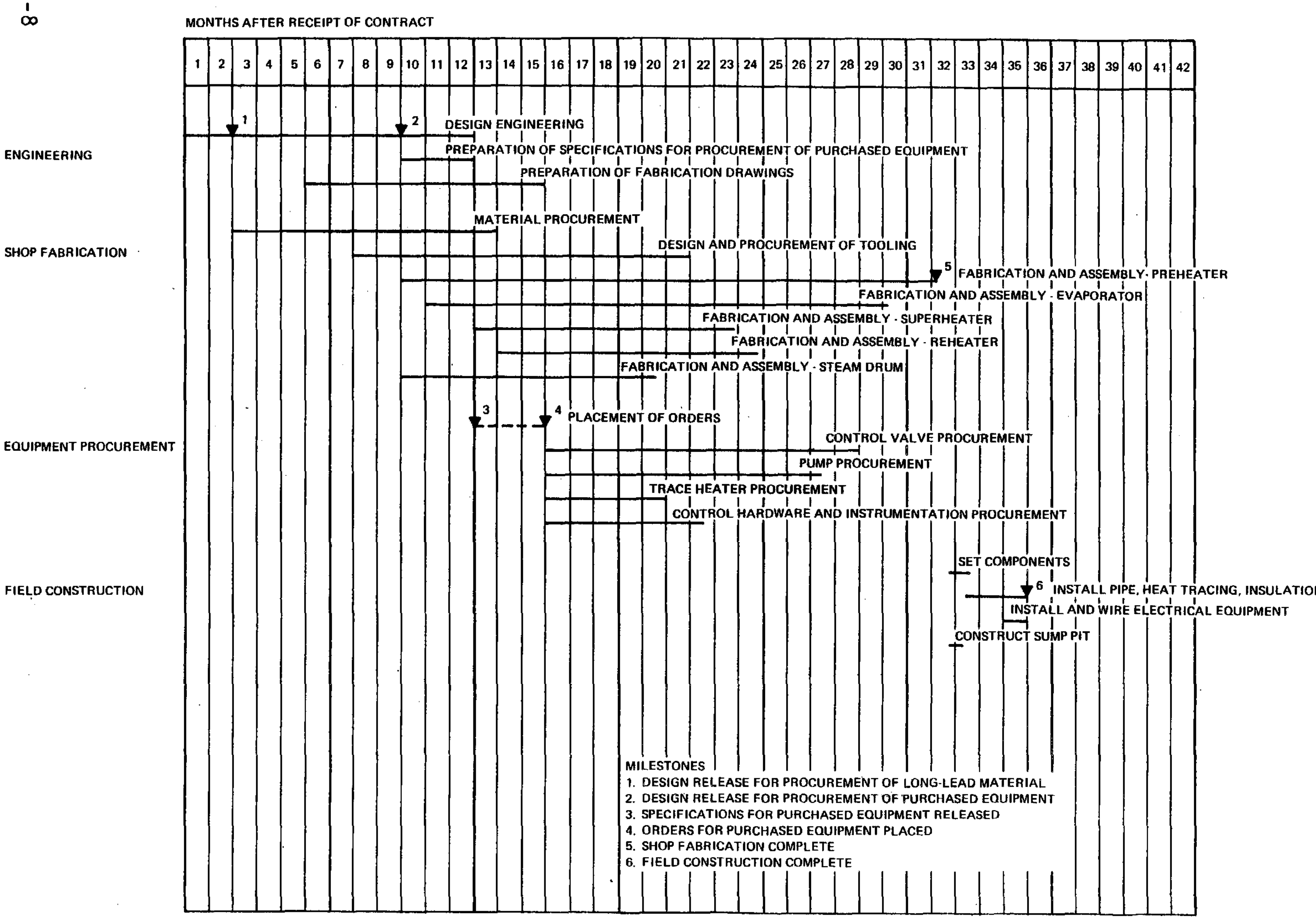


FIG. 6-5 DESIGN, FABRICATION, AND FIELD ERECTION SCHEDULE

TABLE 6-1

Cost Estimate for 100 MWe Steam Generator Subsystem

Steam Generator		
Engineering		$ 689,000
Shop Fabrication		
Preheater		1,393,000
Evaporator		1,504,000
Superheater		756,000
Reheater		786,000
Steam Drum		412,000
	Sub-Total	$ 5,540,000
Controls and Instrumentation		$ 415,000
Balance of Subsystem		
Piping		$ 693,000
Valves		577,000
Pumps		317,000
Heat Exchanger and Steam Drum Erection		510,000
Electrical Equipment		254,000
Building/Structure		395,000
Other (insulation, spring hangers, yard pipe supports, and sump)		255,000
	Sub-Total	$ 3,001,000
Construction and Procurement		$ 1,611,000
(field costs, engineering and procurement, and construction services and management)		
	Grand Total	$ 10,567,000

TABLE 6-2

Cost Estimate for 50 MWe Steam Generator Subsystem

Steam Generator		
Engineering		$ 689,000
Shop Fabrication		
Preheater		1,045,000
Evaporator		1,046,000
Superheater		502,000
Reheater		597,000
Steam Drum		280,000
	Sub-Total	$ 4,159,000
Controls and Instrumentation		$ 415,000
Balance of Subsystem		
Piping		$ 293,000
Valves		427,000
Pumps		293,000
Heat Exchanger and Steam Drum Erection		365,000
Electrical Equipment		242,000
Building/Structure		118,000
Other (insulation, spring hangers, yard pipe supports, and sump)		171,000
	Sub-Total	$ 1,909,000
Construction and Procurement		$ 1,123,000
(field costs, engineering and procurement, and construction services and management)		
	Grand Total	$ 7,606,000

7.0 DEVELOPMENT PLAN

7.1 Objectives

The principal objectives of the proposed development program are (a) to demonstrate to potential users through subscale modeling (SRE), the design adequacy and operational capability of the steam generator subsystem, and (b) to resolve, through SRE and laboratory testing, design and performance uncertainties associated with the full-scale system.

Experimental data derived from the proposed development program would be examined and compared with analytical results and assumptions used in the Phase I study (and upon which the commercial subsystem and component designs are based). If the data differ significantly from that anticipated, it would be necessary to modify the Phase I designs.

The development goals are summarized in the following paragraphs:

Performance Tests - The primary purpose of the performance tests would be to verify the analytical predictions upon which the heat exchanger designs are based. Specifically, the accuracy of the salt-side heat transfer correlations would be assessed and the specific heat of the molten nitrate salt would be determined from appropriate flow and temperature distribution measurements and heat balances. The data would then be compared with the assumptions used in the design analyses accomplished with the Babcock & Wilcox VAGEN computer code.

DNB Tests - the principal objective of the DNB tests would be to verify that the circulation ratio selected is sufficiently high to maintain nucleate boiling throughout the evaporator. Based on our review of available data, we believe that the circulation ratio has been chosen conservatively. Accepting that this will be confirmed, the design margin would be re-assessed to determine if the circulation rate can be reduced. Reduction of the circulation rate would result in some corresponding reducton in heat transfer surface and savings in pump costs.

Corrosion Tests - The corrosion tests would be intended to better define the salt-side corrosion allowances to be applied to the heat exchanger tubing. If the test data compare favorably with analytical assumptions, no design adjustments would be required. If the data deviate significantly from the analytical assumptions, it would be necessary to re-size the tube bundles of the affected components. Assuming that the original assumptions were conservative, re-sizing would involve a reduction of heat transfer surface.

The details of the proposed subsystem research experiment (SRE) and supplementary laboratory test programs were described in the Phase II Plan and Proposal (Reference 16), portions of which are provided in Appendix G. It is the purpose of the following paragraphs to define the broad test objectives and the general test methods.

7.2 PERFORMANCE TESTS

The proposed performance tests are directed to verification and/or improved characterization of the salt-side heat transfer correlation and the specific heat of the molten nitrate salts.

7.2.1 Salt-Side Heat Transfer

For the sizing and design analyses completed during the Phase I study, a modified version of the Dittus-Boelter correlation was chosen to model shell-side molten salt heat transfer. This correlation, normally written

$$Nu = 0.023\ K\ (Re)^{0.8}\ (Pr)^{1/3}$$

Where K is a function of pitch-to-diameter ratio

is long established for use with water and steam. Because the Prandtl number for molten nitrate salt is similar to that for water, application of the correlation to molten salt heat transfer was considered a logical extension of the technology. However, because its application with molten salts is not supported by well-defined test data, caution must be exercised.

Because of the poor thermal conductivity of the nitrate salts, the overall heat transfer resistance is generally dominated by the salt-side heat transfer resistance. Therefore, it is particularly important that the salt-side heat transfer resistance be known accurately, so that the tube bundle surface requirements can be prudently and economically established.

It is recommended that the validity of the selected correlation be confirmed. This can best be accomplished by performing heat balances on subscale heat exchanger models (SRE), and comparing the measured performance with that predicted using the VAGEN code. To assure that the performance data is representative, the important thermal-hydraulic characteristics must be properly modeled. Thus, parameters such as average tube length, tube diameter and wall thickness, and tube spacing must be duplicated in the models, so that the thermal profile along the tube (and thus the heat flux distribution) and the tube-side and shell-side mass flow rate and velocity are suitably represented. The number of tubes in the bundle is then proportional to the thermal ratings of the SRE and commercial components.

7.2.2 Specific Heat of Molten Salt

A temperature-variable property for the specific heat of molten nitrate salt, as specified by Sandia National Laboratories, was used for the Phase I analyses. Sandia has very recently revised this data in accordance with the results of new tests (see Appendix A). A cursory review of this revised data suggests that a favorable impact on overall heat transfer surface requirements can be expected.

Because proper sizing of the heat transfer surface is dependent upon accurate knowledge of the specific heat, it is important that any uncertainties in the correct value of this property be resolved. It is therefore recommended that average values of specific heat be deduced from heat balances performed on the subscale models described in Section 7.2.1.

7.3 DNB Tests

In traditional steam generator design practice, the circulaton ratio (or mass flow rate through the evaporator) is fixed high enough to maintain nucleate boiling in all circuits. However, high circulation rates lead to greater heat transfer surface requirements, as well as to increased pumping power requirements and reduced cycle efficiency. To partially overcome these drawbacks, ribbed tubes are often used in the evaporator surfaces. This tube construction produces a swirling flow and centrifugal forces that keep the tube surface wetted and maintain nucleate boiling to a higher quality for a given pressure, heat flux, and mass velocity than with a smooth-bore tube of the same general dimensions.

In the commercial evaporator design, ribbed tubes have been used for the final 32 feet (constituting the steam/water outlet end) of the evaporator surface. Smooth tubes are used in the remaining low heat flux sections of the component. To preclude DNB (departure from nucleate boiling), a circulation ratio of 1.5 was established.

A sample of typical DNB data is shown in Figure 7-1. The limiting mass velocity is described as a function of steam quality for various values of heat flux. The limiting curves uniquely depend on steam pressure, tube diameter and angle of inclination, and rib helix angle. Because specific data applying to small-diameter ribbed tubes in horizontal orientations is not available, the most applicable published and Babcock & Wilcox proprietary data has been reviewed, prudent design margins have been added, and a circulation ratio believed to be conservative has been selected. It is recommended that well-controlled DNB tests be completed for the applicable design configuration to confirm the choice of circulation ratio and to determine if design margins can be safely reduced.

Special problems in detecting the location of DNB can occur. With vertical tubes, DNB normally occurs at high steam quality where the amount of liquid available for tube wetting is small. In horizontal tubes, the same phenomenon at high quality is encountered, but DNB may also occur at low qualities where the liquid fraction is high but where the steam velocity is low. This behavior is shown in Figure 7-1 where a peak in the limit curve shows possible DNB at low quality. The mechanism of low quality DNB is stratification of the liquid and vapor phases. It is apparent that the DNB test should address both the high-quality and low-quality flow regimes.

For the DNB tests, a single-tube section installed in parallel with the evaporator in the SRE steam system has been proposed. The heat source would then be molten salt rather than the electrically heated elements used in conventional DNB tests. The resulting heat flux profile would be more similar to the evaporator profile than a constant heat flux electrically heated test section. The single-tube test apparatus and test matrix have been described in the Phase II Proposal (Reference 16). The facility is shown schematically in Figure 7-2.

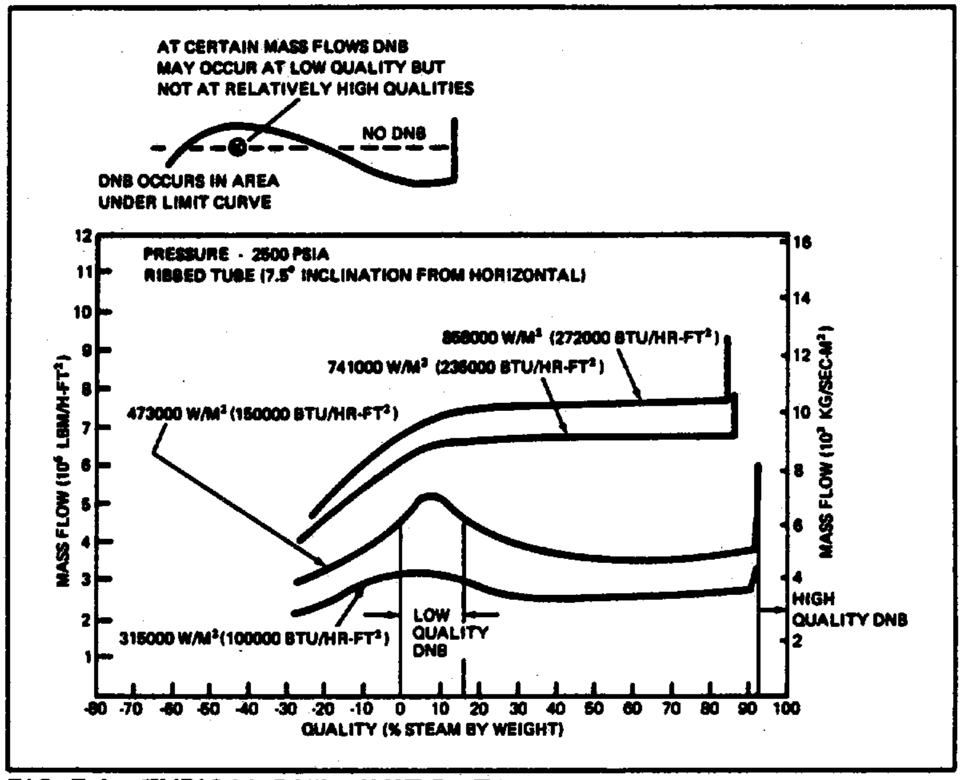


FIG. 7-1 TYPICAL DNB LIMIT DATA

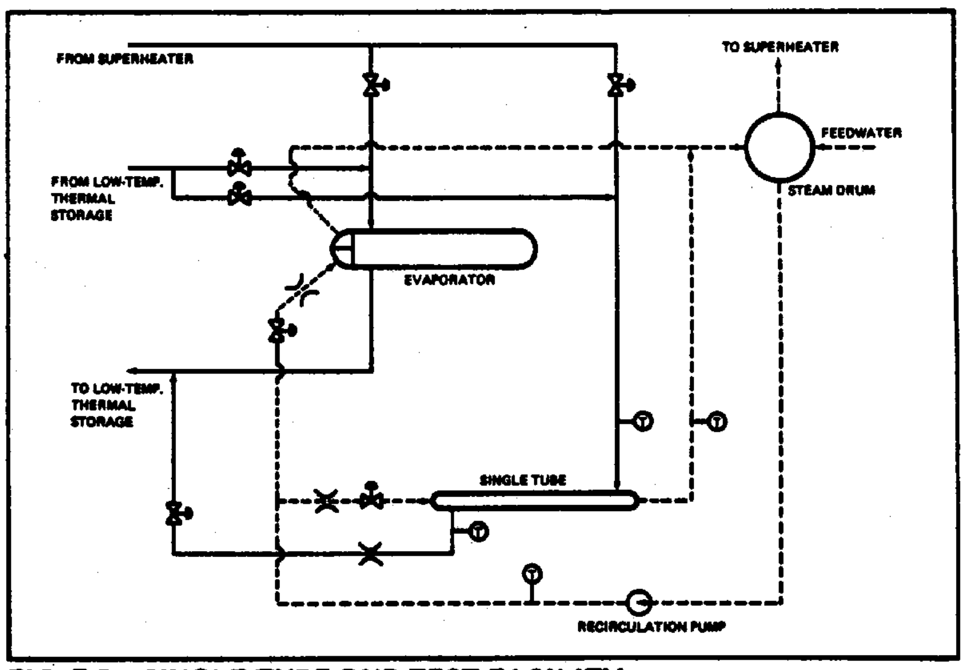


FIG. 7-2 SINGLE TUBE DNB TEST FACILITY

7.4 CORROSION TESTS

Suitable allowances must be added to the required thickness of pressure boundaries to account for corrosion under anticipated operating conditions. Particular care should be exercised in applying corrosion allowances to heat exchanger tubing, so that the margin is large enough to preclude any potential breach of the pressure containment, yet not so excessive as to impose unnecessary penalties in heat transfer surface.

For the Phase I study, the salt-side corrosion data developed during earlier programs (for example, see Reference 18), were used to establish corrosion allowances. In that work, isothermal corrosion rates were measured using weight-gain methods. However, because the data are greatly affected by exfoliation of corrosion products, the results are often sporadic. Consequently, it is difficult to extract from this data the information necessary to establish corrosion allowances with high confidence.

It is recommended that the corrosion test work be extended using electrochemical test methods. The test apparatus and techniques have been described in References 16 and 19. Using the same electrochemical methods, isothermal corrosion rates, as well as corrosion rates under cyclic thermal conditions, can be determined. This data, in conjunction with data derived from longer-term material test programs, can be used to establish corrosion allowances for the 30-year plant life of the steam generator components. The proposed matrix for the electrochemical tests is shown in Table 7-1.

TABLE 7-1

Corrosion Test Matrix

Isothermal Tests (50-day duration)

Material	Temperature, °C (°F)
Carbon Steel	288 (550) 371 (700)
2 1/4 Cr - 1 Mo	288 (550) 454 (850)
304 Stainless Steel	288 (550) 454 (850) 566 (1050)

Cyclic Thermal Tests (50-day duration, one cycle daily)

Material	Temp (1)	⟷	Temp (2), °C (°F)
Carbon Steel	288 (550)	⟷	371 (700)
2 1/4 Cr- 1 Mo	288 (550)	⟷	454 (850)
304 Stainless Steel	288 (550)	⟷	566 (1050)

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