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FOSTER WHEELER SOLAR DEVELOPMENT CORPORATION

VOLUME 1

PHASE 1--FINAL REPORT

MOLTEN SALT STEAM GENERATOR SUBSYSTEM RESEARCH EXPERIMENT

Sandia Contract 20-9909B

Prepared for

Sandia National Laboratories
Livermore, California

September 1982

Sandia #82-8179
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September 30, 1982

Sandia National Laboratories
Purchasing Organization 8264
7011 East Avenue
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Attention: Jack P. Hubner
Sandia Contracting Representative

Gentlemen:

Subject: Molten Salt Steam Generator Subsystem Research Experiment
Transmittal of Final Report

Enclosed are eight copies and one reproducible master of the following:

"Volume 1, Phase 1--Final Report, Molten Salt Steam Generator Subsystem Research Experiment"

"Volume 2, Phase 1--Final Report (Appendices A through F), Molten Salt Steam Generator Subsystem Research Experiment"

The Final Report presents the results of the work performed during the period June 12, 1981 to April 14, 1982 by Foster Wheeler Solar Development Corporation under Sandia Contract 20-9909B.

Very truly yours,

Stephen J. Goidich

Stephen J. Goidich
Project Manager

SJG:jgr
Enclosures

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SECTION 1

EXECUTIVE SUMMARY

Section 1

EXECUTIVE SUMMARY

1.1 STUDY OBJECTIVES

The study conducted by the Foster Wheeler Solar Development Corporation (FWSDC) is Phase 1 of a two-phase project whose objectives are:

- Develop a reliable, cost-effective molten salt steam generating subsystem for solar thermal plants
- Minimize uncertainty in steam generator subsystem capital, operating, and maintenance costs
- Demonstrate the ability of molten salt to generate high-pressure, high-temperature steam.

The Phase 1 study involved the conceptual design of molten salt steam generating subsystems for a nominal 100-MWe net solar central receiving electric generating plant (100-MWe solar stand-alone) and a nominal 100-MWe net fossil-fueled electric power generating plant that is 50 percent repowered by a solar central receiver system (50-MWe hybrid). As part of Phase 1, a proposal was prepared for Phase 2, which will involve the design, construction, testing and evaluation of a Subsystem Research Experiment (SRE) of sufficient size to ensure successful operation of the full-size subsystem designed in Phase 1.

1.2 TECHNICAL APPROACH

To achieve the aforementioned objectives, the Phase 1 study was divided into the following tasks:

Task	Description
1	Review of SGS Definition and Interface Requirements
2	Definition of SGS Requirements
3	SGS Concept Selection
4	SGS Design
5	SGS Cost and Fabrication/Erection Plan
6	SGS SRE and Development Plan
7	Phase 2 Plan and Proposal
8	Project Management

In general, work on each task proceeded in sequential order, except for Task 8, which extended over the entire 10-month project schedule. Because of SNLL's request for submittal of the Phase 2 proposal 1 month ahead of schedule, we based the proposal solely on the 100-MWe solar stand-alone SGS. The 50-MWe hybrid SGS design was completed after the Phase 2 proposal was submitted.

1.3 PROJECT TEAM

The Phase 1 project team consisted of the following companies:

Prime Contractor:	Scope of Work:
Foster Wheeler Solar Development Corporation	Project coordination, concept selection, preliminary thermal design, dynamic modeling, structural analyses, modes of operation
Subcontractors:	
FW Energy Applications, Inc.	Thermal/hydraulic design and analysis, mechanical design
Foster Wheeler Energy Corporation	Fabrication requirements, heat exchanger cost estimate
Foster Wheeler Special Projects Engineering and Construction, Inc.	Maintenance requirements, SRE installation
Gibbs and Hill, Inc.	SGS auxiliary equipment and support structure design and cost estimate, SGS interface requirements, SRE layout
Utility Advisors:	
Arizona Public Service company	Review of subsystem-level requirements
Sierra Pacific Power Company	Review of subsystem-level requirements

1.4 SUMMARY OF RESULTS

1.4.1 Requirements and Specifications

The nominal interface requirements for a 100-MWe solar stand-alone SGS and a 50-MWe hybrid SGS identified by SNLL were refined and used as the basis for preparation of an SGS Requirements and Specification document. The document was written to be site-independent to the greatest extent possible. However, to quantify site-dependent design parameters, we selected Yerington, Nevada, as the location.

The stand-alone and hybrid SGS's were designed to generate main and reheat steam for a nominal 100-MWe net steam turbine-generator. For the purpose of defining SGS performance requirements over the operating load range, the Sierra Pacific Power Company Fort Churchill Unit 1 turbine-generator was selected. The turbine "valves wide open" (VWO) steam flow requirements that each SGS design must satisfy are as follows:

	100-MWe Solar Stand-Alone, kg/s (lb/h)	50-MWe Hybrid kg/s (lb/h)
Superheated steam [541°C (1005°F) and 13.48 MPa gage (1955 lb/in²g)]	96.1 (762,900)	48.1 (381,400)
Reheat steam [541°C (1005°F) and 2.86 MPa gage (415 lb/in²g)]	83.2 (660,300)	41.6 (330,150)

The heat source for the SGS is hot, molten salt stored in a thermal storage tank at 566°C (1050°F). To compensate for heat losses in the salt piping between thermal storage and the SGS, and to provide a design margin, an inlet salt temperature of 563°C (1045°F) was selected. A salt exit temperature of 293°C (560°F) was selected to provide a reasonable pinch-point temperature difference [7.3°C (13.1°F)].

1.4.2 Concept Selection

We identified 29 selection criteria for evaluation of candidate SGS concepts. Combinations of the following parameters were considered:

Surface Arrangements:

- Straight Tube
- Hockey Stick
- Helical Coil
- Bayonet Tube
- Involute Tube (serpentine)
- U-Tube (common tubesheet)
- U-Tube (U-shaped shell)
- U-Tube (involute)

Circulation Methods:

- Benson Once-Through
- Sulzer Once-Through
- Natural Circulation
- Forced recirculation

Orientation:

- Horizontal
- Vertical

Qualitative and preliminary quantitative evaluation of the candidate configurations resulted in selection of a four-component (preheater, evaporator, superheater, and reheater) straight-tube, vertical, natural-circulation arrangement as the concept that best satisfies the selection criteria.

Based on a review of available molten salt corrosion data, the following materials were selected for component fabrication:

Component	Material
Preheater	Carbon Steel
Evaporator	1-1/4%Cr-1/2%Mo
Superheater	Type 304 Stainless Steel
Reheater	Type 304 Stainless Steel

1.4.3 SGS Design

The physical arrangement of the 100-MWe solar stand-alone SGS and 50-MWe hybrid SGS are essentially the same except for the variations in size resulting from the difference in thermal rating. The equipment arrangement for the 100-MWe solar stand-alone SGS is illustrated in Figure 1.1. The process flow diagrams for the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS are shown in Figures 1.2 and 1.3.

Hot molten salt entering the system at 563°C (1045°F) flows in parallel through the superheater and reheater, combines, and passes in series through the evaporator and preheater; cold salt leaves the preheater at approximately 293°C (560°F). All heat exchangers are oriented vertically with all heated steam/water flowing upward. The preheater, superheater, and reheater are counterflow; the evaporator is parallel flow to improve natural circulation. An integral vertical steam drum is mounted atop the evaporator. A drum water recirculation pump is provided to maintain the feedwater at a temperature above the salt freezing point [221°C (430°F)] during start-up and part-load operation.

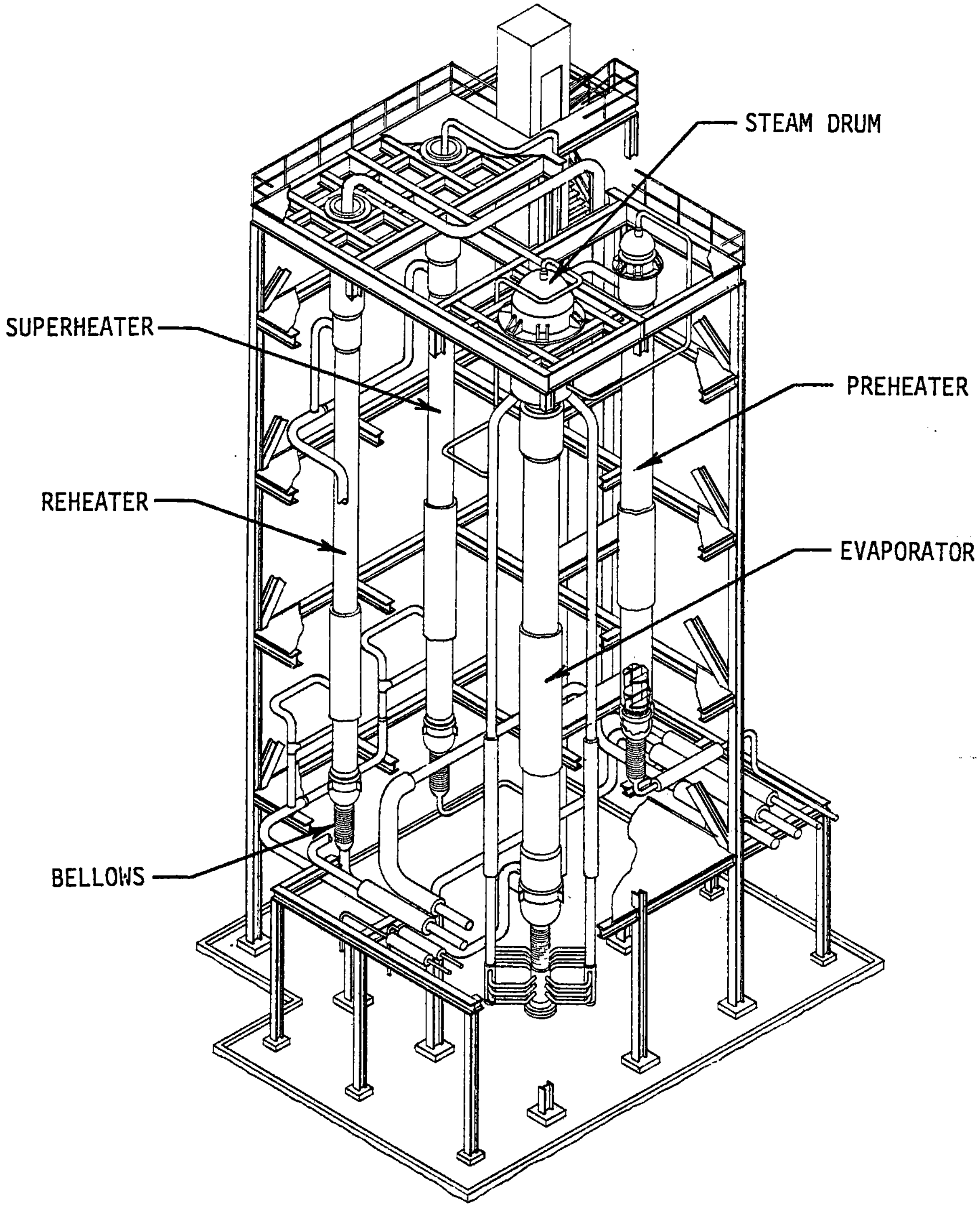


Figure 1.1 SGS Isometric View--100-MWe Solar Stand-Alone SGS

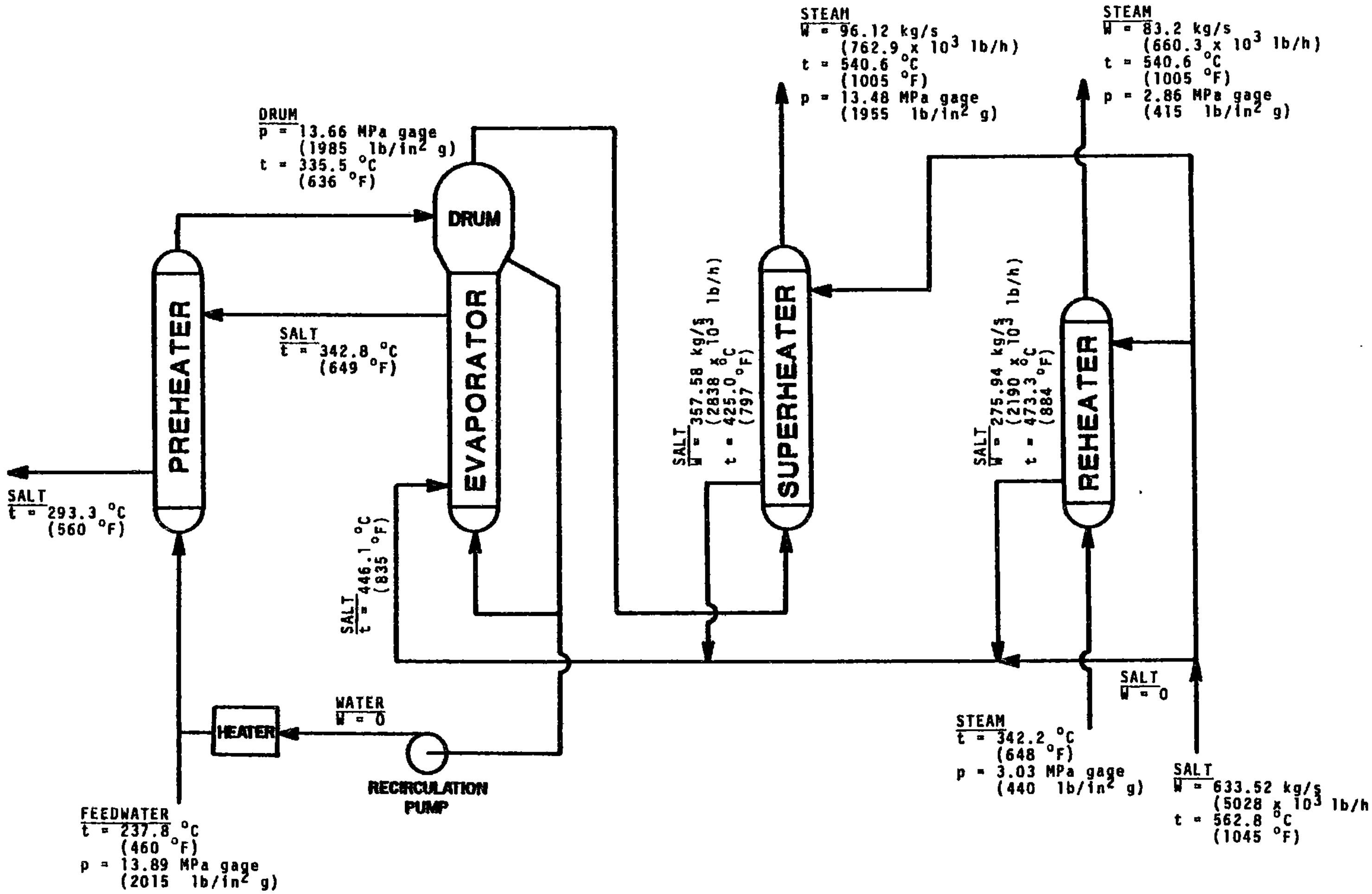


Figure 1.2 Process Flow Diagram--100-MWe Solar Stand-Alone SGS

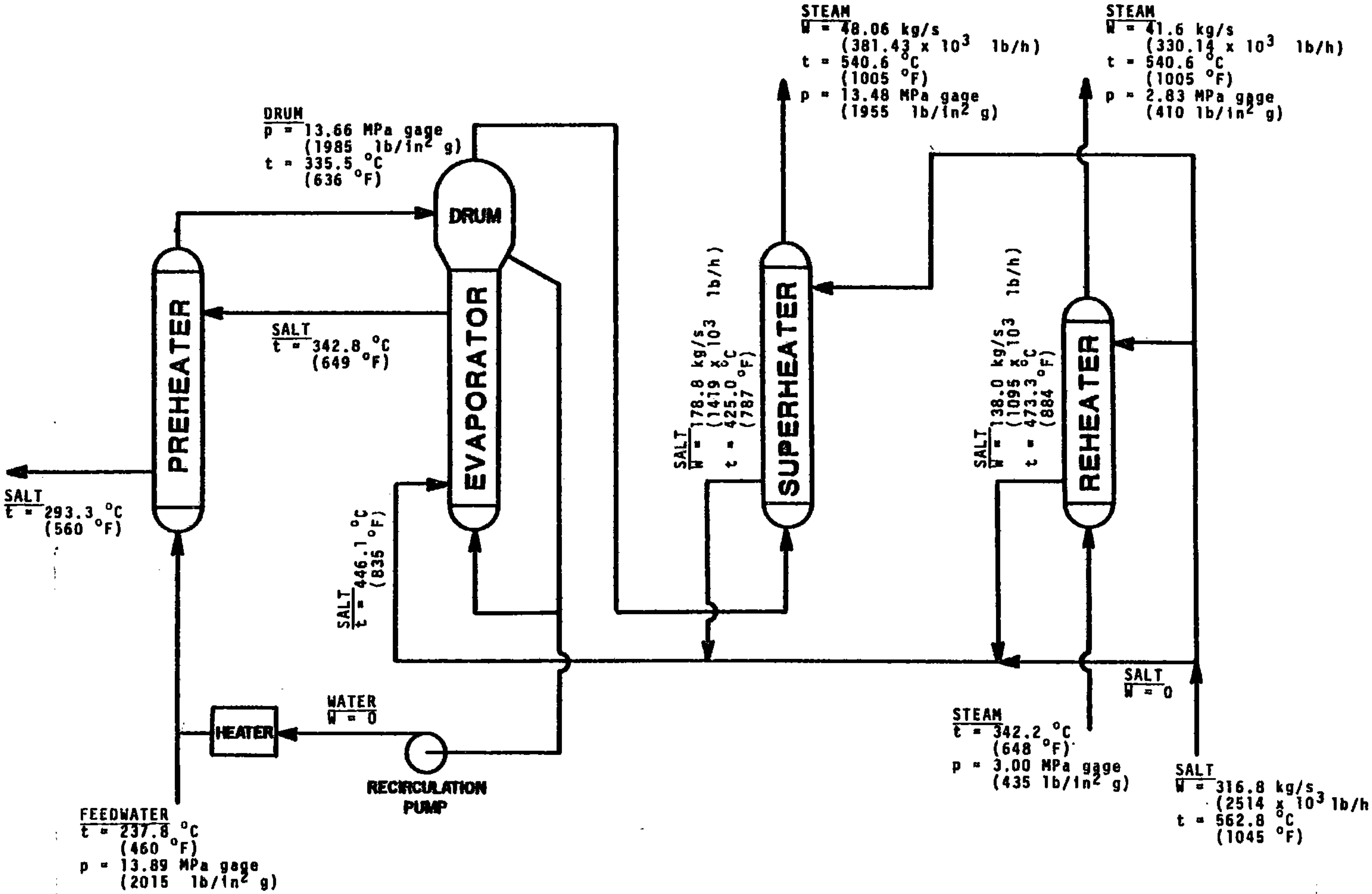


Figure 1.3 Process Flow Diagram--50-MWe Hybrid SGS

A cold salt recirculation pump is also provided to control the salt temperature entering the subsystem during unit start-up and shutdown.

Final main steam temperature is controlled by a valve at the superheater outlet which controls the salt flow rate through the superheater. Saturated steam from the steam drum is bypassed to the superheater outlet for emergency control. Reheat steam temperature is controlled by a valve at the reheater outlet which controls the salt flow rate through the reheater. A spray attemperator is located at the reheater steam inlet for emergency control. The quantity of steam generated is determined by the salt flow rate and temperature entering the evaporator. A salt line which bypasses hot salt around the superheater and reheater to the evaporator is used for this purpose.

The heat exchangers are single pass shell-and-tube exchangers, each with a floating head and double segmental baffles. The 100-MWe units have bellows welded to the lower shell head and the steam/water inlet nozzle which permits differential expansion between the tube bundle and shell. The smaller diameter 50-MWe preheater, superheater, and reheater have the bellows designed in the exchanger shell. The 50-MWe evaporator is similar to the 100-MWe design. The evaporator on both units have a pair of external downcomer pipes and flexible feeders which direct water from the steam drum to the evaporator inlet. The loops in the feeder pipes permit differential growth between the downcomers and the evaporator.

A shroud at the salt inlet to each unit surrounds the tube bundle. The slots in the shroud uniformly distribute salt over the tube bundle. Tie-rods attached to the upper tubesheet support the double segmental baffles, which function as tube-support plates to suppress fibration and buckling. Heat-transfer tubes are welded to the face of each tubesheet using a fillet-type welding technique. Each unit is vertically hung from either a support skirt (superheater, reheater) or lugs (on preheater upper tubesheet; on evaporator, drum shell).

The vertical steam drum, which is designed as an integral part of the evaporator, is equipped with spiral arm separators and box type chevron driers to provide dry saturated steam. Feedwater enters the steam drum through a circular distribution pipe positioned below the drum water level. A blowdown line is provided to control impurity concentration levels in the evaporator water.

Electric trace heaters are provided on the heat exchanger shells as well as on all interconnecting salt piping. The trace heaters are sized to preheat and maintain the salt piping and heat exchanger shells at approximately 288°C (550°F). The heat exchangers and all interconnecting piping are insulated with calcium silicate and covered with aluminum lagging.

Safety valves are located on the steam drum, superheater outlet, reheater inlet, reheater outlet, and preheater outlet. Pressure-relief devices are located in the inlet and outlet salt piping of each heat exchanger to prevent overpressurization of the shell in the event of a tube rupture. A salt drain system is provided to drain the salt from each heat exchanger and the associated interconnecting pipes in 120 minutes. The salt is drained by gravity into a sump tank equipped with a pump which directs the salt to the cold storage tank.

Table 1.1 summarizes significant heat exchanger design features for the 100-MWe solar stand-alone SGS and 50-MWe hybrid SGS; Table 1.2, significant results of the thermal/hydraulic, structural, and subsystem level analyses.

1.4.4 Shop Fabrication and Field Fabrication/Erections Plan

The straight tube heat exchanger design selected for molten salt steam generation is such that standard "state-of-the-art" fabrication techniques can be used. A step-by-step fabrication sequence including all significant fabrication and inspection operations were identified for both the 100-MWe solar stand-alone SGS and 50-MWe hybrid SGS heat exchangers.

Field fabrication and erection of each SGS can be accomplished using standard civil, mechanical, and electrical installation procedures. A schedule that identifies the interrelationships and time periods required for each major installation step was prepared.

1.4.5 Cost Estimates and Schedules

The installed SGS capital costs based on first quarter 1982 dollars are:

	100-MWe Solar Stand-Alone SGS ($)	50-MWe Hybrid SGS ($)
Heat exchangers	3,614,100	2,480,900
Auxiliary systems and equipment	2,994,000	1,942,700
Structure	522,000	475,000
Instrumentation and controls	2,400,000	2,166,000
Subtotal	9,530,100	7,064,000
Contingency at 20 percent of Subtotal	1,906,000	1,412,900
Home office costs	1,715,400	1,271,600
Construction management	914,900	678,200
Total Cost	14,066,400	10,427,300
Fee at 8 percent of Total Cost	1,125,300	834,200
Sell Price	15,191,700	11,261,500

Table 1.1 Heat Exchanger Design Summary

	100-MWe Solar Stand-Alone				50-MWe Hybrid			
	Preheater	Evaporator	Superheater	Reheater	Preheater	Evaporator	Superheater	Reheater
Shell I.D., m (ft)	1.30 (4.10)	1.64 (5.38)	0.91 (3.00)	0.85 (2.80)	0.78 (2.57)	1.00 (3.30)	0.58 (1.93)	0.64 (2.08)
Tube Length, m (ft)	17.5 (57.3)	18.1 (59.5)	18.3 (60.0)	18.9 (62.0)	16.8 (55.0)	18.3 (60.0)	16.8 (55.0)	18.6 (61.0)
Height, m (ft)	20.9 (68.7)	28.0 (91.8)	23.5 (77.1)	24.0 (78.8)	18.2 (59.7)	26.6 (87.3)	18.9 (62.2)	20.7 (67.8)
Tube O.D., mm (in.)	15.9 (0.63)	25.4 (1.0)	15.9 (0.63)	25.4 (1.0)	15.9 (0.63)	25.4 (1.0)	15.9 (0.63)	25.4 (1.0)
Tube MW, mm (in.)	1.47 (0.058)	2.1 (0.083)	1.65 (0.065)	1.24 (0.049)	1.47 (0.058)	2.1 (0.083)	1.65 (0.065)	1.24 (0.049)
Number of Tubes	2325	1359	1049	458	997	629	529	229
Actual Heat Transfer Surface, $10^3 m^2$ ($10^3 ft^2$)	2.03 (21.8)	1.97 (21.2)	0.96 (10.3)	0.69 (7.4)	0.83 (9.0)	0.92 (9.9)	0.44 (4.8)	0.34 (3.7)
Tube Material	CS	1-1/4%Cr-1/2%Mo	304SS	304SS	CS	1-1/4%Cr-1/2%Mo	304SS	304SS
Design Margin, %	15.1	17.64	10.5	9.8	13.1	16.8	9.1	9.5
Dry Weight, 10^3kg (10^3lb)	54.0 (119.0)	122.0 (269.0)	28.5 (62.8)	20.9 (46.0)	18.1 (40.0)	67.1 (148.0)	12.2 (27.0)	10.9 (24.0)
Filled Weight, 10^3kg (10^3lb)	84.8 (187.0)	166.6 (367.2)	44.9 (99.0)	33.1 (73.0)	30.4 (67.0)	104.9 (231.2)	18.1 (40.0)	18.6 (41.0)
Steam Drum I.D., m (ft)	---	2.1 (6.9)	---	---	---	1.5 (5.0)	---	---
Chevron Driers	---	19	---	---	---	10	---	---
Separator Arms	---	27	---	---	---	16	---	---

#/ft² 23.7 38.4 42.7

Table 1.2 Significant Analysis Results

Analysis	Significant Results
Thermal/Hydraulic:	
• Uncertainty	Tube length (i.e., heat transfer surface) was increased to account for uncertainties in heat transfer coefficients, tube material thermal conductivity, variations in wall thickness, and inactive regions. Design margins are listed in Table 1.1. Number of tubes (3%) was increased to account for possibility of tube plugging.
• Circulation	Evaporator, downcomers, feeders, drum internals sized for 4:1 circulation ratio.
• Full- and Part-Load Performance	Variation in steam/water/salt temperature pressure, and flow rate over the operating load range (25 to 100%) were identified.
• Stability	Static (Ledinegg) and dynamic (Nyquist) stability curves plotted for the evaporator indicated that the steam/water flow is stable.
• Critical Heat Flux	The Atomics International and Westinghouse critical heat flux correlations indicate that DNB and/or dryout will not be experienced in the evaporator.
Structural:	
• Fatigue	Allowable temperature variations and ramp rates were determined by steady-state and transient analysis of the tubesheet-shell and tubesheet-head junction for the cyclic nature of SGS operation.
• Buckling	Analysis of a plugged tube (most severe condition) indicated that the baffle spacing is less than the critical buckling length and buckling should not occur.
• Flow-Induced Vibration	The analysis shows that tube vibration and fluid elastic whirling are not significant. The baffle damage and collision damage numbers are significantly below the allowable limit of 1.0.
Subsystem Level:	
• Modes of Operation	Procedures for cold start-up, cold shutdown, diurnal start-up, diurnal shutdown, warm standby, start-up from warm standby, shutdown to cold conditions, and emergency shutdown were identified.
• Dynamic Model	The mathematical model of the SGS for evaluation of response to load changes revealed that in addition to proportional and integral controls, feed-forward and derivative controls are required for quick, stable response; one percent salt flow through the bypass line is required at 100 percent load for pressure control; an evaporator/preheater bypass line under pressure control is required to minimize pressure surges during emergency conditions.

Interest during design, fabrication, and construction, state and local taxes, escalation, and owner's costs were not included.

The time period from start of design to the end of preoperational testing is 32 months for the 100-MWe solar stand-alone SGS and 29-months for the 50-MWe hybrid SGS. The schedule for both units are included in Figure 1.4.

1.4.6 Subsystem Research Experiment (SRE)

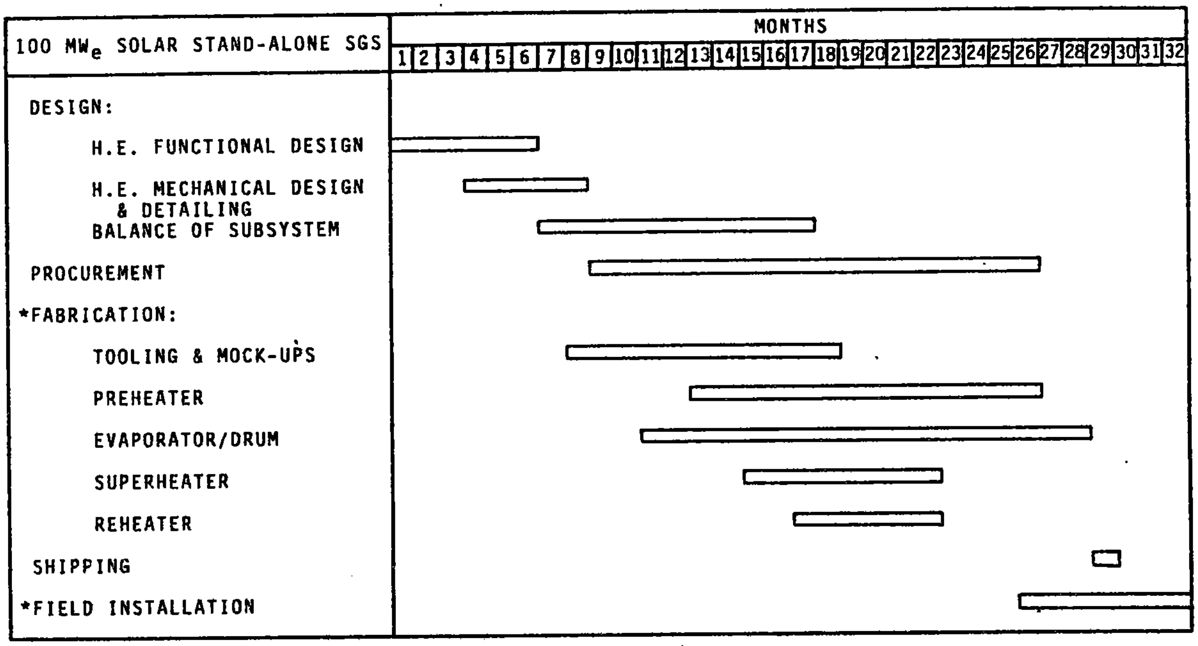
As part of Phase 1 a proposal was prepared for the design, construction, testing, and evaluation of a SRE of sufficient scale to ensure successful operation of the full-size subsystem designed in Phase 1. The objectives of the SRE specified by SNLL are as follows:

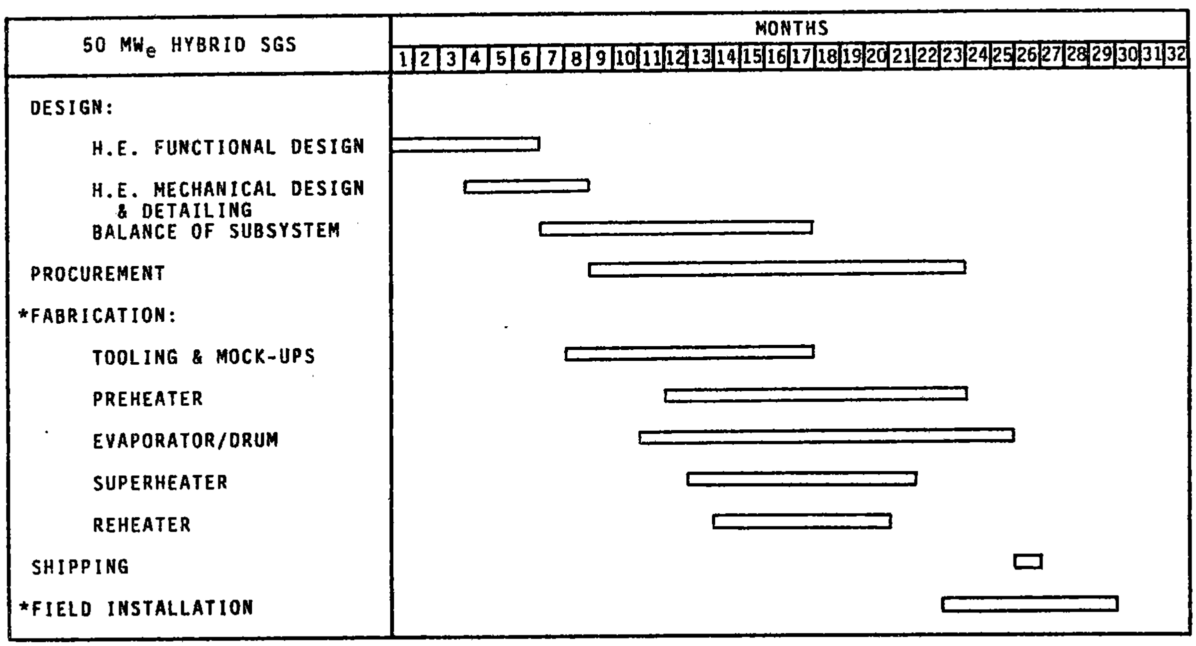
- Demonstrate the ability to design, construct, and operate a molten salt steam generator for generating high-pressure [13.48 MPa gage (1955 lb/in^2g)], high-temperature [541°C (1005°F)] steam for power generation
- Resolve all critical design fabricating, performance, operating, and cost-estimating uncertainties associated with the full-scale SGS designs developed during Phase 1
- Provide a molten salt steam generator that can be utilized for a future full-system experiment (FSE).

As directed by SNLL, our proposed field tests are compatible with the CRTF equipment and are limited to the thermal capacity of the thermal storage SRE.

Specific areas that will be investigated in the SRE to meet the aforementioned objectives are:

- Demonstrate the performance of the various heat exchangers and correlate it with analytical predictions. This will include:
  - Thermal duty
  - Pressure drop
  - Shell-side heat-transfer coefficient
  - Steam/water-side flow distribution
  - Evaporator circulation
  - Tube-side flow stability
  - Absence of departure from nucleate boiling (DNB)





*See Appendix D for Details.

Figure 1.4 SGS Schedules

- Demonstrate the behavior of the SRE control system and correlate it with the computer predictions obtained from the control system dynamic computer model
- Verify procedures established to start up and shut down the SGS during normal and emergency operating conditions
- Demonstrate the ability to design against tube vibration
- Demonstrate the absence of gross structural deformations
- Demonstrate the compatibility of materials of construction with the molten salt.

Although a molten salt SGS for a high-temperature, high-pressure reheat power cycle has never been designed, fabricated, and operated, it is our assessment (based on our design, fabrication, and testing of similar heat exchangers in the past) that there are no critical design or fabrication uncertainties that must be resolved. However, since the proposed operating procedures have never been demonstrated, we believe that an SRE is needed primarily to demonstrate the ability to operate a molten salt steam generator designed for high-temperature, high-pressure applications. Therefore, our proposed SRE is oriented toward a complete subsystem rather than individual components and contains all essential features of the full-size subsystem.

To meet the objectives of the Phase 2 program, we propose an SRE and a simpler, less-expensive alternative SRE. Both will be designed to be compatible with an FSE and with other necessary equipment at the CRTF.

<u>Proposed SRE</u>. The proposed SRE SGS, like the full-scale 100- and 50-MWe subsystems, has four heat exchangers--preheater, natural-circulation evaporator, superheater, and reheater. All heat exchangers are oriented vertically with all heated steam/water flowing upward. The preheater, superheater, and reheater are

counterflow; the evaporator is parallel flow to improve natural circulation. The proposed SRE SGS is designed to generate 1.57 kg/s (12,500 lb/h) superheated steam at 541°C (1005°F) and 13.5 MPa gage (1955 lb/in^{2}g), and 0.68 kg/s (5413 lb/h) simulated reheat steam at 541°C (1005°F) and 2.9 MPa gage (425 lb/in^{2}g).

The proposed SRE includes 11 test series to demonstrate subsystem-level as well as component-level features of the full-scale SGS. Subsystem-level aspects of the SRE involve demonstration of all operating modes of the full-scale system to show that the individual components can be operated as an integrated system in a safe and stable manner. Component-level aspects of the SRE involve monitoring each heat exchanger for specific data that will verify the ability of developed analytical methods to predict heat exchanger performance characteristics.

Subsystem-level testing will include demonstration of the following operating modes:

- Cold start-up
- Full- and part-load steady-state operation
- Load changes
- Diurnal shutdown
- Diurnal start-up
- Shutdown to warm standby
- Warm standby
- Start-up from warm standby
- Shutdown to cold conditions
- Emergency shutdown

While demonstrating the SRE SGS operating modes, data will also be obtained to evaluate component-level considerations such as the following:

- Salt-side film coefficients
- Dynamic flow stability of evaporator
- Absence of DNB/dryout in the evaporator

- Superheater flow stability
- Fouling
- Ambient heat losses
- Absence of tube vibration
- Absence of gross structural deformation
- Tubesheet temperature response to load change.

The estimated cost for the proposed SRE is $1,999,424. The time period from start of SRE designs to completion of data evaluation would be 24 months.

Alternative SRE. The alternative SRE reduces the SRE cost 21 percent, shortens the SRE schedule from 24 months to 21 months, and still includes 9 of the 11 test series planned for the proposed SRE.

The alternative SRE SGS includes only the evaporator and superheater; the preheater and reheater are eliminated. The superheater is identical to the superheater in the proposed SRE. The evaporator is also essentially the same as the proposed SRE evaporator except for increased pressure-part thickness.

Elimination of the superheater and reheater means that the following full-scale SGS features cannot be demonstrated:

- Evaporator drum-water recirculation to preheater inlet
- Use of main steam to preheat reheater
- Establishing main steam flow to reheater before admission of salt to reheater on start-up
- Control response of superheater/reheater bypass combination
- Demonstration of emergency shutdown procedures for complete subsystem.

We do not believe that these omissions pose serious development risks.

SECTION 2

INTRODUCTION

Section 2

INTRODUCTION

2.1 STUDY OBJECTIVES

Studies funded by DOE have shown that molten salt (60% $NaNO_3$, 40% KNO_3) is the most economical receiver coolant and storage fluid for solar plants requiring large amounts of thermal storage. Although molten salt has been used successfully in process heat applications for many years, it has not been used as a heat-transfer medium for generating high-pressure [13.48 MPa gage (1955 lb/in^{2}g)], high-temperature [541°C (1005°F)] steam for power generation. There have been a number of studies of solar thermal power plants using molten salt, but these were primarily concerned with the design and operation of the overall plant.[1-5]* Consequently, important steam generator design and operating considerations were not covered in depth.

This study, conducted by the Foster Wheeler Solar Development Corporation, is Phase 1 of a two-phase project whose objectives are:

- Develop a reliable, cost-effective molten salt steam generating subsystem for solar thermal plants
- Minimize uncertainty in steam generator subsystem capital, operating, and maintenance costs
- Demonstrate the ability of molten salt to generate high-pressure, high temperature steam.

The Phase 1 study involved the conceptual design of molten salt steam generating subsystems for a nominal 100-MWe net solar central receiving electric

*Numbers designate references listed in Section 8.

generating plant (100-MWe solar stand-alone) and a nominal 100-MWe net fossil-fueled electric power generating plant which 50 percent repowered by a solar central receiver system (50-MWe hybrid). As part of Phase 1, a proposal was prepared for Phase 2--design, construction, testing, and evaluation of a Subsystem Research Experiment (SRE) of sufficient size to ensure successful operation of the full-size subsystem designed in Phase 1.

2.2 SGS DEFINITION AND INTERFACE REQUIREMENTS

The nominal parameters defining the molten salt SGS were incorporated in the SGS Definition and Interface Requirements prepared by SNLL and included in the Phase 1 RFQ. The requirements established a starting point for preparation of the SGS specifications to which the SGS was designed. The SGS Definition and Interface Requirements as defined by SNLL are listed below. Also listed are changes recommended by FWSDC and approved by SNLL that form the basis for design of the molten salt SGS.

2.2.1 SGS Definition and Interface Requirements as Defined by SNLL

General. The present program is directed toward developing a molten salt SGS that utilizes modern state-of-the-art steam cycles and heat exchangers. Typical components which may comprise the SGS are:

- Feedwater preheating heat exchanger
- Evaporating heat exchanger
- Steam drum
- Boiler water recirculation pumps, piping and valves
- Superheating heat exchanger
- Steam reheating heat exchanger
- Attemperators
- Salt, feedwater, and steam piping and valves
- Thermal insulation and trace heating
- Controls and instrumentation
- Foundations, component structural supports, and dikes
- Drain tanks, lines, and pumps
- Salt recirculation lines and pumps.

The molten salt SGS must be capable of functioning in the following operating modes:

- Start-up from a cold, dry condition
- Diurnal start-up
- Steam generating and steam reheating at the maximum and minimum rates
- Load changing as required by the turbine generator set (typically ±10 percent/min from 10 to 100% of rated turbine output)
- Sustained operation at any load required by the turbine generator set
- Diurnal shutdown
- Sustained warm standby
- Shutdown to cold, dry condition.

The molten SGS must be capable of safe, controlled shutdown resulting from upset and emergency conditions caused by:

- Turbine trip
- Loss of feedwater flow
- Loss of salt flow
- Any water/steam/salt pipe rupture
- A water/steam heat exchanger tube rupture that allows significant mixing of water/steam with salt
- Loss of pneumatics
- Failure of control system
- Loss of all station power.

The molten salt SGS must be designed and costed to meet the following interface condition for a 100-MWe net solar central receiver electric generating plant and must also be designed and costed for a 100-MWe fossil-fueled electric generating plant that is 50 percent repowered by a solar central receiver system.

Nominal Interface Conditions. Following is a list of the nominal interface conditions:

• Receiver/Thermal Energy Storage Working Fluid	Molten Nitrate Salt 60 wt% $NaNO_3$/40 wt% KNO_3
• Working Fluid Inlet Temperature	
- To SGS, °C (°F)	565 (1050)
- From SGS, °C (°F)	290 (550)
• Turbine-Generator	
- Power, MWe	110 (100 net)
- Throttle Temperature, °C (°F)	540 (1000)
- Throttle Pressure, MPa absolute (lb/in²a)	12.5 (1815)
- Design Inlet Steam Flow Rate, kg/s (lb/h)	96.5 (764,000)
- Design Reheat Steam Flow Rate, kg/s (lb/h)	83 (661,000)
- Cold Reheat Steam Temperature, °C (°F)	340 (645)
- Hot Reheat Steam	
-- Pressure, MPa absolute (lb/in²a)	3.5 (500)
-- Temperature, °C (°F)	540 (1000)
- Last Feedwater Heater (FHW 5) Discharge Temperature, °C (°F)	235 (460)

Applicable Documents. The equipment, materials, design, and construction of the SGS must comply with all Federal, State, local, and user standards, regulations, codes, laws, and ordinances currently applicable for the specific site and the user. These include, but are not limited to, the Government and nongovernment documents listed in this section. If there is an overlap in, or conflict between the requirements of these documents and the applicable Federal, State, County, or Municipal codes, laws, or ordinances, that applicable requirement which is the most stringent takes precedence.

The following documents of the issue in effect on the date of RFQ form a part of this specification to the extent specified. In the event of conflict between the documents referred to and the contents of this specification, the contents of this specification are the superseding requirement.

- Government Documents
  - Specifications
    - Regulations of the Occupational Safety and Health Administration (OSHA)
    - International System of Units, 2nd Revision, NASA SP-7012
  - Standards
    - Applicable Human Engineering Design Criteria
- Nongovernment Documents
  - Specifications (List of subsystem and interface specifications to be prepared by the contractor)
  - Standards and Codes
    - Uniform Building Code--1976 Edition by International Conference of Building Officials
    - ASME Boiler and Pressure Vessel Code
    - Institute of Electrical and Electronic Engineers (IEEE) Codes, as applicable
    - National Fire Protection Association (NFPA) National Fire Design, Construction and Fabrication Standards
    - Standards of AISC (American Concrete Institute)
    - Standards of TEMA (Tubular Exchanger Manufacturers Association).

2.2.2 Revisions/Clarifications to SGS Definition and Interface Requirements

Job-Site Location. For the purpose of cost estimating and determining environmental requirements, we have used Yerington, Nevada, for the job-site location.

Plant Life. The SGS will be designed for a 30-year life.

Environmental Requirements. Environmental requirements will be based on the Yerington, Nevada, job-site location.

Terminal Points. Terminal points defining the boundaries of the SGS and components falling within the scope of the SGS are as follows:

- Salt Side (at boundary of SGS support structure)
  - Hot salt line from thermal storage
  - Cold salt line to thermal storage
- Steam/Water Side (at boundary of SGS support structure)
  - Feedwater line
  - Main steam line to high-pressure turbine
  - Cold reheat line from high-pressure turbine
  - Hot reheat line to intermediate-pressure turbine.

Interconnecting pipe within these terminal points is within the scope of the SGS. Measuring devices, control valves, etc., which are outside these terminal points but are required for SGS operation, are also within the scope of the SGS. However, piping in which these devices are located is not within the scope of the SGS.

Salt drain tanks and drain lines from the heat exchangers to the drain tanks are within the scope of the SGS.

Superheater and turbine bypass systems required for start-up control must be defined as part of the SGS. Instruments and controls required in these systems are within the scope of the SGS. Piping in which these devices are located is not within the scope of the SGS.

Feedwater and salt treatment facilities are not within the scope of the SGS.

Salt Inlet Temperature. All performance calculations will be based on 563°C (1045°F) salt entering the SGS. For design purposes (i.e., material selection), the inlet salt temperature to be used is 565°C (1050°F). The lower salt temperature for performance calculations is to compensate for heat losses in the hot salt feed piping and also to provide a design margin.

Salt Outlet Temperature. At the design point condition the temperature of the salt leaving the SGS will be 293°C (560°F) to obtain an acceptable pinch-point temperature difference [7.3°C (13.1°F)]. The pinch-point temperature difference is described in Appendix A.

Main Steam/Reheat Temperature. Based on conventional utility steam generator design practice, the superheater and reheater will be designed for a 541°C (1005°F) outlet steam temperature to ensure 540°F (1000°F) steam at the turbine.

Throttle Steam Pressure. The steam generator will be designed for the turbine "valves wide open" (VWO) condition, which corresponds to the 5-percent throttle overpressure condition [i.e., 13.1 MPa absolute (1903 lb/in^{2}a)].

Reported Part-Load Performance. Detailed performance will be determined at 25-, 50-, 75-, and 100-percent loads for constant throttle pressure (5-percent overpressure). Unit operation between 0 and 25-percent load will be considered the start-up mode.

Steam-Line Pressure Drop. The reheater and the reheater transfer pipe pressure loss will not exceed 10 percent of the pressure leaving the high-pressure turbine. For initial calculations the reheater pressure drop should be approximately 5 percent. The main steam line pressure loss should be approximately 5 percent of the pressure leaving the superheater.

Salt Properties. The salt properties specified in the RFQ will be used for all performance calculations. Salt property variation will not be included as a variable in the uncertainty analysis to determine heat-transfer surface margins. FWSDC will select fouling factors and corrosion allowances based on review of available reports (Carling Study, Martin Marietta Test Report, etc.). SNLL will be consulted on the selection of values. The salt properties as given in the RFQ are:

Temperature (°C)	Specific Heat (cal/g•°C)	Absolute Viscosity (Pa•s) x 10^3	Density (g/cm^3)	Conductivity (W/m•K)
300	0.399	3.22	1.879	0.500
350	0.389	2.29	1.848	0.510
400	0.381	1.80	1.818	0.519
450	0.374	1.43	1.787	0.529
500	0.366	1.21	1.757	0.539
550	0.358	1.05	1.726	0.548
600	0.350	0.93	1.695	0.558

Design Point Conditions. For preliminary unit sizing, the Sierra Pacific Fort Churchill Unit 1 maximum load (5 percent overpressure) turbine cycle data will be used. The nominal net rating of the unit is 100 MWe.

Power, MWe (MWe net)	115.2 (111.0)
Throttle Temperature, °C (°F)	540 (1000)
Throttle Pressure, MPa absolute (lb/in^{2}a)	13.1 (1903)
Design Inlet Steam Flow Rate, kg/s (lb/h)	96.4 (762,860)
Design Reheat Steam Flow Rate, kg/s (lb/h)	83.4 (660,280)
Cold Reheat Steam Temperature, °C (°F)	342 (648)
Hot Reheat Steam Pressure, MPa absolute (lb/in^{2}a)	3.2 (463)
Hot Reheat Steam Temperature, °C (°F)	540 (1000)
Feedwater Temperature, °C (°F)	235 (460)

These data will be refined as better estimates for auxiliary power, spray flows, blowdown, etc., are known.

SECTION 3

SGS REQUIREMENTS AND SPECIFICATION

Section 3

SGS REQUIREMENTS AND SPECIFICATION

The conditions to which the molten salt SGS is designed are defined in the SGS Requirements and Specification included in Appendix B.* The document is an expansion of the SGS Definitions and Interface Requirements prepared by SNLL and defines the following:

- SGS scope
- References (standards, codes, laws, ordinances, etc.) that apply to the design, construction, and operation of the SGS
- Technical requirements for the following:
  - Subsystem level
  - Heat exchangers
  - Auxiliary equipment
  - Support structure, foundation, and dikes
  - Balance of plant interfaces
  - Modes of operation
- Environmental criteria.

For details, refer to Appendix B.

*The molten salt properties identified in Appendix B are updated properties determined by SNLL at the end of the project and are to be used for future analysis. The properties used in this study are those included in the RFQ and are listed in Section 2.2.2 of this report.

SECTION 4

CONCEPT SELECTION

Section 4

CONCEPT SELECTION

The procedure for selecting an SGS concept included the following steps:

- Identifying selection criteria
- Selecting candidate surface arrangements, circulation methods, and orientations
- Reducing the number of concepts to a manageable number for evaluation
- Qualitatively and quantitatively evaluating three concepts
- Evaluating materials suitable for each component.

This section of the report contains a description of these steps.

4.1 BACKGROUND INFORMATION

4.1.1 Selection Criteria

To establish a basis for comparison, evaluation, and selection of SGS concepts for high-temperature molten salt steam generation, pertinent selection criteria were identified and listed in Table 4.1. Criteria 1 through 14 were specified by SNLL. Criteria 15 through 29 were added by Foster Wheeler based on Foster Wheeler's experience in steam generator design and in concept evaluation for sodium and molten salt heated steam generators.[6-9]

4.1.2 Surface Arrangement[10-16]

Critical to the design of heat exchangers for high-temperature molten salt steam generation is the selection of the heat-transfer surface arrangement. The arrangement selected must satisfy the criteria defined in Table 4.1 and, in particular, must provide adequate flexibility for thermal expansion resulting from the high-temperature environment unique to this application. Candidate surface arrangements include the following:

- Straight tube
- Hockey stick
- Helical coil
- Bayonet tube
- Involute tube (Serpentine)
- U-tube (Common tubesheet)
- U-tube (U-shaped shell)
- U-tube (Involute)

These arrangements are illustrated in Figure 4.1 and discussed in the following sections.

Table 4.1 Concept Selection Criteria

RFQ CRITERIA

1. Cost
2. Performance
3. Impacts on salt temperature swing and nominal feedwater temperature
4. Surface area
5. Pinch-point temperature difference
6. Modularity and sizing
7. Reliability
8. Fabricability
9. Responsiveness to thermal transients
10. Feedwater quality
11. Required development
12. Auxiliary power
13. Prior user experience
14. Ease of service, maintenance, and repair

ADDITIONAL CRITERIA

15. Ease of control
16. Good utilization of volume
17. Ease of locating leaks
18. Ease of plugging tubes
19. Ability to drain water side
20. Ability to drain salt side
21. Suitability for scale-up
22. Resistance to handling, shipping, and seismic loads
23. Thermal expansion provision
24. Ability to withstand thermal transients
25. Thermal/hydraulic stability
26. Design simplicity
27. Low development risk
28. Low tube-wear potential
29. Low safety risk

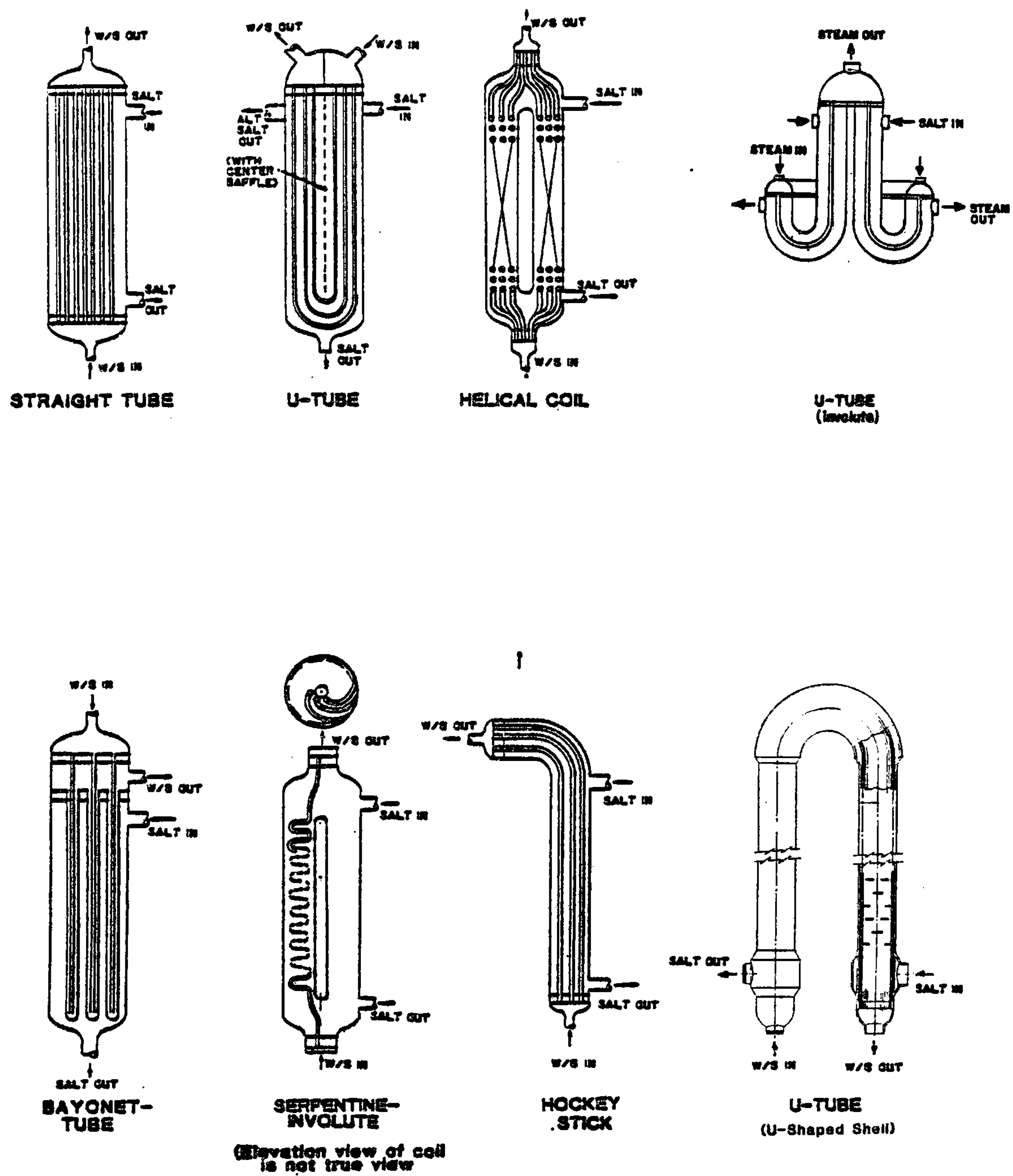


Figure 4.1 Heat Exchanger Surface Arrangements

Straight Tube. Straight tube designs are the simplest heat exchangers from a manufacturing point of view. They have been used extensively as heat exchangers in the process industry, both with fixed and floating heads. As steam generators they have been used in both experimental and power-producing plants. In this design the high-pressure fluid flows inside the tubes and the low-pressure fluid flows on the shell side. For stability the fluid being heated flows upward; the fluid being cooled flows downward.

In chronological order, straight-tube steam generators built by Foster Wheeler were used in the Shippingport nuclear plant and as retrofit steam generators for the USN submarine "Seawolf." Duplex tube evaporators and superheaters are used in the EBR-II Sodium Fast Breeder Facility. A 10-MWt unit was built and tested at the LMEC. Once-through steam generators of this configuration are supplied by Babcock and Wilcox for pressurized water reactor (PWR) power plants. Steam generators for sodium breeder plants for SNR-300 (European consortium) and BN-600 (USSR) are presently being fabricated. These two contain expansion bellows in the shell to compensate for the difference in thermal expansion between the shell and the tubes.

Most of the intermediate heat exchangers used in breeder reactor plants are of straight-tube configuration. Where it is imperative that each tube absorbs its own differential thermal expansion, a straight-tube design can be modified to include a compound bend expansion loop in each tube. Examples of this design are the ALCO-BLH steam generator tested at the LMEC and intermediate heat exchangers for the fast flux test facility (FFTF) and the Prototype Fast Reactor (PFR).

The straight-tube design with a compound bend in each tube is very expensive to manufacture, as it involves drum bending of tubes and row-by-row assembly. Segmented supports are usually required. For these reasons the compound bend design will be excluded from further consideration in our study.

Advantages:

- Operates in natural-circulation and forced-recirculation modes and possibly once-through mode
- Drainable on both sides
- Cheapest to manufacture, employing well-established manufacturing techniques
- Easiest to maintain, including tube inspection and replacement
- Most compact
- Requires least floor space
- Best surface-to-volume ratio.

Disadvantages:

- In once-through mode of operation, possible instability at low loads because of low overall pressure drop resulting from short flow paths.
- Possible need for expansion bellows to compensate for difference in thermal expansion between tube and shell

Applications:

- Preheater
- Superheater
- Evaporator
- Reheater

Hockey Stick. The hockey stick design is the product of strict nuclear requirements for the sodium breeder plant to provide maximum flexibility in each tube and to minimize potential tube failure and thus catastrophic sodium/water reactions. This particular design has been selected for the Clinch River

Breeder Reactor Plant. The first double hockey stick heat exchanger was built for the USAEC in the early 1950s. In the 1970s, a 28-MWt steam generator model was built and tested at the LMEC. Steam generators of this configuration have been fabricated for the Clinch River Breeder Reactor Plant. A model containing a few tubes, using the Clinch River tube configuration, was tested at General Electric. Testing was discontinued because of excessive tube bowing.

In this design each tube is bent like a hockey stick. The assembly of the unit requires that one row of tubes be assembled at a time, followed by a complete nondestructive examination before proceeding to the next row. Expensive internal bore welding of tubes to tubesheet is required at the upper tubesheet. The bend in the shell is assembled in segments. In operation, this design is very similar to the straight-tube design.

Advantages:

- Can operate in all three modes: natural-circulation, forced-recirculation, and possibly once-through
- Can be drained on both sides
- Accessible for inspection of tubes and tube plugging
- Each tube will absorb its own differential expansion.

Disadvantages:

- Very expensive to manufacture, requiring special techniques for welding and nondestructive examination (NDE)
- Bend area not normally exposed to flow--poor utilization of heat-transfer surface
- Bending of tubes and assembly of the bend in the shell increase the cost.

Applications:

- Economizer
- Superheater
- Evaporator
- Reheater

Helical Coil. The development of the helical coil steam generator in support of the fast breeder plants dates back to the early 1960s, when under the auspices of the U.S. Atomic Energy Commission (USAEC), development of a once-through helical coil steam generator was begun by Babcock & Wilcox. The 25-MWt module was to be tested at the Liquid Metal Engineering Center (LMEC), but fabrication of the unit was never completed. In the late 1960s, a three-tube model (1 MWt) was built by Foster Wheeler and tested at General Electric. Testing of models of helical coil configurations was performed by a European consortium (in support of SNR-300) and by Japan (in support of Monju). Presently under construction or awaiting start-up are helical coil steam generators for SNR-300 (one-half are straight tube; the other half helical coil configurations), Monju (Japan--evaporators and superheaters), and Super Phenix (France--once-through). In the United States, extensive design efforts including mock-ups have been done in support of a high-temperature gas-cooled reactor (HTGR) by Foster Wheeler/General Atomics.

To maintain equal-length circuits in the helical coil design, the tube pitch and number of tubes are varied with coil diameter. To minimize the possibility of large water leaks, and thus catastrophic sodium/water reactions in sodium breeder reactor applications, either both tubesheets were kept above the sodium level or the shell was penetrated with individual tubes, with the header located outside the shell. In the first case, tubes could not be drained; in the second case, manufacturing costs were high. Neither method is needed for salt application.

Advantages:

- Length of the circuitry makes it best suited for once-through operation (can also operate in a forced-recirculation mode)
- Tubes are accessible for plugging
- Can be drained on both sides
- Can absorb some differential thermal expansion and thus eliminate the need for expansion bellows
- Relatively small tubesheets
- Relatively short length of exchanger.

Disadvantages:

- Because of high flow resistance, cannot operate in a natural-circulation mode
- Requires special manufacturing techniques that must be developed
- Tubes not replaceable
- Inside of tubes not accessible for inspection
- Potential instability in a once-through mode at low loads.

Applications:

- Preheater
- Superheater
- Evaporator
- Reheater

Bayonet Tube. The bayonet-tube steam generators are not new to the industry. They can be traced back to the early 1930s, when units of this configuration were built in Germany. In the United States, the first model in support of the breeder program was built by ALCO for Atomic Power Development Associates (APDA). It had a rating of 0.5 MWt and was tested successfully for 635 hours. Two bayonet-tube steam generators of 3- and 85-MWt capacity were built by Griscom-Russell for Hallam Nuclear Power Facility (HNPF). In these units the steam was generated on the outside of the tubes. In the late 1960s a seven-tube (2-MWt) model was built by Foster Wheeler and tested extensively at General Electric. On a larger scale, bayonet-tube steam generators were used in the BN-350 (USSR) breeder plant. Advanced studies by Foster Wheeler and Combustion Engineering led to recommending this design for future breeder reactor plants. Similarly, after exhaustive studies of various steam generators, Foster Wheeler has recommended this design for the HTGR plants.

In the bayonet-tube design, the water or steam flows through the inner tube and the evaporation or superheating occurs in the annular space between the inner and outer tubes. Since both tubes penetrate the lower tubesheet, a relatively thick tubesheet is required.

Advantages:

- Differential expansion accommodated by each tube
- Replacement of tubes possible.

Disadvantages:

- Water side cannot be drained
- Regenerative effect in the inner tube increases heat-transfer surface
- Thick tubesheet required
- Surface penalty because inner tube serves only as flow channel
- Operation in a once-through mode questionable.

Applications:

- Preheater
- Evaporator
- Superheater
- Reheater

Involute Tube (Serpentine). This design concept has seen only very limited application as steam generators. It has been proposed for other applications where heat-transfer surface has to be fitted inside a cylindrical vessel. The only known use of this concept was in the Fermi Plant (200 MWt), where three steam generators built by Griscom-Russell operated in a once-through mode. These steam generators experienced a number of problems, such as tube vibration, tube failures and, initially, gross instability. The instability was corrected by installation of tubular flow restrictors [3.18 mm (1/8-in.) dia by 4.88 m (16 ft) long] in the downcomer leg of the steam generator.

In this design the tubes are bent into a serpentine shape and assembled into panels; then the panels are bent into the shape of an involute, resulting in uniform spacing between the tube panels over their full span.

Advantages:

- Both sides can be made drainable
- Inside of tubes accessible for plugging at tubesheets
- Can accommodate differential thermal expansion between tubes and shell.

Disadvantages:

- Because of high tube-side flow resistance, steam generator can operate only in forced-circulation and once-through modes
- Inside of tubes not accessible for inspection

- Potential instability exists in once-through mode
- Involute configuration adds to manufacturing cost
- Replacement of tube is extremely difficult.

Applications:

- Preheater
- Evaporator
- Superheater
- Reheater

U-Tube (Common tubesheet--upright design). The U-tube configuration, as illustrated in Figure 4.1, has the low-pressure fluid (molten salt) on the shell side and the high-pressure fluid (steam/water) on the tube side. This type configuration has been used for the evaporator in the PFR (England).

Advantages:

- Employs proven fabrication techniques
- Accessible for inspection and replacement of tubes
- Can accommodate differential thermal expansion between individual tubes.

Disadvantages:

- Tube side cannot be drained
- Possible instability because of downflow in evaporator
- Requires thick tubesheets
- Can be used only in heat exchangers where temperature change between inlet and outlet is small
- Can operate only in forced-circulation mode.

Applications:

- Evaporator

U-Tube (Common tubesheet--inverted design). The inverted U-tube, with the high-pressure fluid (steam/water) on the shell side and the low-pressure fluid (molten salt) on the tube side, has found wide application in the utility industry, both as feedwater heaters and pressurized water reactor (PWR) steam generators. In feedwater heaters the high-pressure water is on the tube side and the low-pressure steam is on the shell side. Feedwater heaters operate at relatively low temperatures but, at times, at very high pressures (supercritical cycle). In PWR steam generators, evaporation takes place on the shell side, with the steam separation taking place in an integral steam drum located above the tube bundle. A relatively thick shell is required to contain the pressure.

Some of the problems associated with this design are corrosion of the tubes at the tubesheet, tube vibration, and tube wear in the support plates. Some of these problems could be corrected with improved designs and the removal of accumulations of deposit from the face of the tubesheet. The inverted U-tube design is applicable only where the temperature gradient across the two halves of the tubesheet is relatively low. A large temperature gradient across the tubesheet precludes using this configuration in a superheater or reheater and may even preclude its use as a preheater. This must be confirmed by analysis.

Advantages:

- Drainable on both sides
- Accessible for inspection of the tubes and for retubing
- Flexible - absorbs differential expansion built into tubes
- Can be manufactured with proven techniques
- Well-suited for natural-circulation mode of operation.

Disadvantages:

- Can operate in a natural-circulation mode only
- Cannot be used as superheater or reheater because of large temperature gradients across the width of the tubesheet
- For high-pressure steam cycle, requires very thick shells
- Still has some operating problems, such as tube vibration, wear in support plates, and tube failure because of corrosion
- Requires relatively thick tubesheets.

Applications:

- Evaporator.

U-Tube (U-shaped shell). A 30-MWt horizontal, once-through U-tube steam generator with a U-shaped shell was built by B&W for the Sodium Reactor Experiment. A 1-MW prototype unit was also built by B&W and tested at Atomic Power Development Associates. The BOR reactor in the USSR utilized a vertical U-tube design.

Advantages:

- Each tube can expand relative to shell
- Separate tubesheets for hot and cold fluids
- Can be drained in horizontal position.

Disadvantages:

- Less effective heat-transfer surface in return bend
- Differential growth between hot and cold legs
- Temperature stratification on shell side for horizontal units.

Applications:

- Preheater
- Evaporator
- Superheater
- Reheater

U-Tube (Involute). The U-tube configuration with an involute tube pattern has been used as the superheater in the DFR (England). The tubesheets are toroidal. The splitting of the tubesheet was necessitated by a large temperature difference between the inlet and outlet steam.

Advantages:

- Employs proven fabrication techniques
- Accessible for inspection and replacement of tubes.

Disadvantages:

- Tube side cannot be drained
- Possible instability because of downflow in evaporator
- Compound bending of tubes required to accomplish involute pattern
- Poor surface-to-volume utilization because of tubeless center core
- Expensive toroidal headers
- Two passes on shell side.

Applications:

- Superheater
- Reheater
- Preheater

4.1.3 Circulation Methods

Steam generators used for utility and industrial applications use one of the following circulation methods:

- Once-Through:
  - Benson
  - Sulzer
- Recirculation:
  - Natural
  - Forced

Each method is schematically illustrated in Figure 4.2. The circulation method applies to main steam generation for the high-pressure turbine. The reheater, which provides superheated steam for the intermediate- and low-pressure turbines, is shown to identify its impact on the circulation method. Requirements, advantages, and disadvantages for each method are summarized in Table 4.2. Significant features of each method are discussed in the following sections.

Once Through.

General. All once-through steam generators have a heated transition zone where the water in the two-phase steam/water mixture is completely evaporated and saturated steam (100-percent quality) is obtained. Dissolved solids in the feedwater are concentrated in the water phase as steam is generated, since their solubility in steam is very low. When complete evaporation occurs, the solids concentrated in the water plate out on the heated surface. The fouled heat-transfer surface impedes heat transfer and can potentially fail from excessive corrosion, depending on the type of deposit.

Modern utility steam generators, whether once-through or recirculation type, have feedwater treatment facilities that provide highly purified

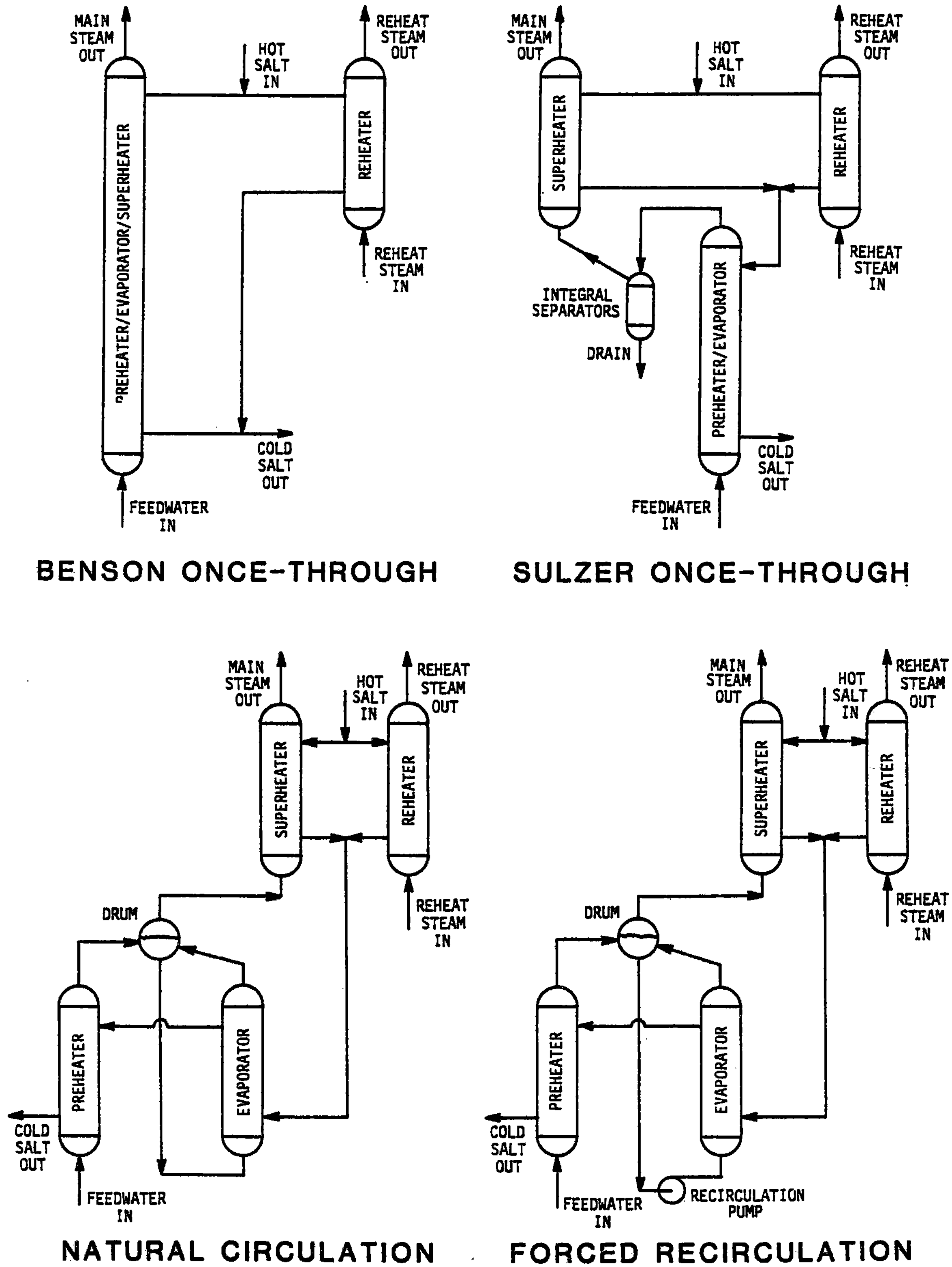


Figure 4.2 Circulation Methods

Table 4.2 Comparison of Circulation Methods

	Once-Through	Natural Circulation	Forced Circulation
REQUIREMENTS	High mass flow rate - small tubes	Predictable heat absorption and distribution	Continuous pump operation
	Minimum feedwater flow limit for stability and tube protection	Saturated on subcooled water in downcomer	Saturated or subcooled water in downcomer
	Start-up system with breakdown valves	Avoid DNB	Avoid DNB
	Balanced heat absorption and flow distribution		
	High feedwater purity		
ADVANTAGES	Low weight	Simplest design	Freedom in selection of tube size
	Minimum number of heat exchanger shells	Simple start-up system	Simple start-up system
		Easily adaptable to reheat cycle	Easily adaptable to reheat cycle
		Ability to maintain feedwater above salt freezing point by recirculating drum water	Ability to maintain feedwater above salt freezing point by recirculating drum water
		Dry steam - steam/water separation in drum	Dry steam - steam/water separation in drum
		Self-compensating for absorption rate variations (increased heat absorption increases flow)	Accepts wide imbalances in heat absorption
			Positive circulation
DISADVANTAGES	Elaborate start-up system requiring breakdown valves and steam bypass	Extra weight of drum and large tubes	Extra weight of drum, pump, valves
	Minimum flow	Sluggish response to fast heat absorption variations	Additional pump power
	DNB and film boiling	Stagnation, DNB, chemical attack	Pump availability
	Flow instabilities	Flow imbalances with unbalanced heat absorption	
	Large enthalpy pickup per circuit		
	Not self-compensating or absorption rate variations (increased heat absorption decreases flow)		
	Multiple valves and controls		
	Large pressure drop required for stability		

water. Because of the aforementioned characteristic of once-through units, a condensate polishing system must be provided to protect the steam generator in case of a condenser leak that would result in contaminated water entering the system. In a recirculation unit, the steam drum provides a reservoir of water from which feedwater impurities can be removed (blowdown). In addition, the steam-separating capability of the steam drum prevents impurities from being carried into the high-temperature superheater and turbine. Whatever enters a once-through unit will plate out on the heat-transfer surface or be carried through to the turbine.

The condensate polishing system provides final cleanup of the condensate entering the feedwater loop by reducing suspended and dissolved solids. The typical condensate polishing system, shown in Figure 4.3, consists of several mixed, deep-bed ion exchanger vessels containing a mixed charge of cation and anion resin. Condensate from the condenser flows through each of the polisher vessels in parallel and then is transferred to the feedwater heaters by individual condensate booster pumps. When the ion exchange resin is exhausted, it is sluiced out to an external regeneration facility consisting of three vessels in which the resin is separated, individually regenerated, and remixed before its return to service. There are individual acid and caustic regeneration facilities for the cation and anion resin. Each regeneration facility consists of a day tank receiving concentrated regenerant from a bulk storage tank, regenerant pumps, dilution water, and mixing tee, (and, in the case of caustic only, a heating tank) to enable measured quantities of dilute regenerant solution to be delivered to the external regeneration facility.

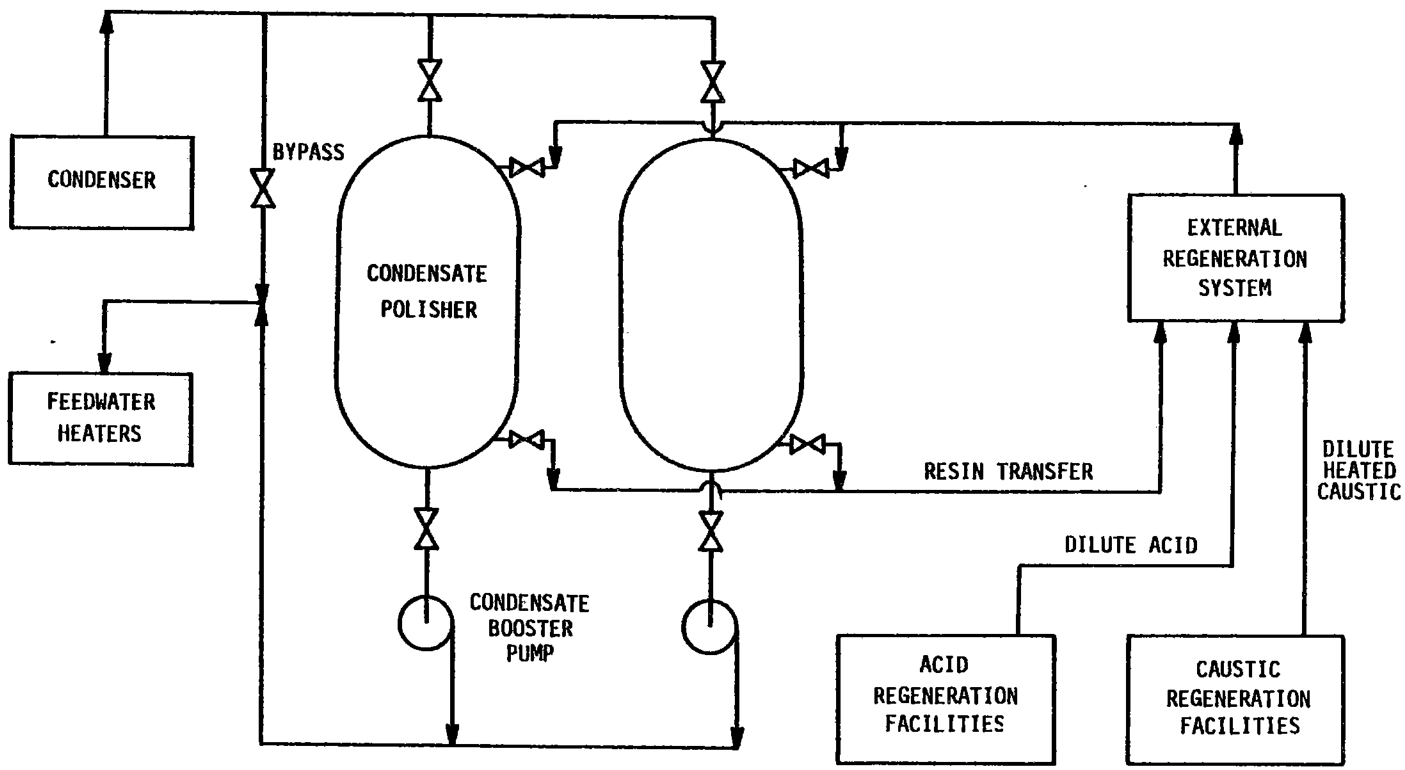


Figure 4.3 Feedwater Treatment Facility

In addition to the condensate polishing system, other once-through steam generator characteristics of special note, identified in Table 4.2, include the following:

- At low-load conditions, the feedwater entering the system will be below the freezing point of molten salt [221°C (430°F)]. Consequently, an auxiliary feedwater heater must be provided to maintain the feedwater above the salt freezing point.
- To minimize steam/water flow instabilities, once-through units must be designed with high pressure drop. With the frictional pressure drop as the main component of the total pressure drop, increased heat absorption in a given tube will decrease the flow rate through that tube and increase the temperature of the exiting steam.
- With feedwater entering subcooled and leaving superheated, there is a large enthalpy change per flow circuit. A small percentage variation in absorbed heat or flow from tube to tube can result in a significant difference in the temperature of steam leaving the various tubes.

Benson. The once-through Benson circulation method preheats, evaporates, and superheats main steam in a single vessel. Each of these steam/water phases has its own thermal/hydraulic characteristics. With each phase occurring in all tubes, the regions in which they occur cannot be optimized simultaneously. One region will dictate the design. In addition, the high temperature superheat region will dictate material selection and tube wall thickness, resulting in overdesign of the preheat and evaporator regions.

For unit start-up, high-pressure water flow is established at a selected minimum flow rate to prevent the two-phase flow instabilities that can occur at low pressure. The high pressure subcooled water leaving the unit is diverted to a high-pressure flash tank or throttled and passed through an integral separator start-up system (ISSS), which separates the saturated steam from the saturated water. The steam is directed to the condenser; the drain water is recycled to the deaerator, the condenser, or both. As water temperature in the unit increases, a point is reached where the pressure is lowered and steam generation is allowed to occur in the unit. Further salt heating produces superheated steam at the unit outlet. The steam can be directed to the turbine when the steam/turbine metal temperature difference is within acceptable limits.

From a control standpoint, the physical location of the point in the heat-transfer path where superheating begins may shift axially along the tubes and the unit is therefore characterized by variable areas in the preheating, evaporating and superheating sections. A slight change in feedwater flow will cause a change in the amount of heat-transfer area devoted to each section. The resultant change will cause a fluctuation in steam temperature. Even with very precise feedwater controls, the steam temperature in the Benson cycle will tend to move outside the acceptable range for the turbine.

Feedwater flow perturbations will also tend to upset the static stability of the steam generator. Static instability represents that condition where the pressure drop across the tube does not increase with increased flow or decrease with decreased flow. Therefore, in a statically unstable situation, a decrease in flow will be accompanied by no change in pressure drop (or possibly by an increase in pressure drop), which would promote an additional reduction in flow and ultimately lead to voiding the unit--or at least to preventing the desired superheated steam temperature from being achieved. In the Benson-type system, most of the pressure drop is in the superheating region. If the reduction in pressure drop associated with the decrease in superheating length is equal to or greater than the increased pressure drop associated with increased flow, an unstable situation will exist. Likewise, other perturbations, such as salt flow or salt flow maldistribution, will cause changes in superheating length which may initiate instabilities. These unstable tendencies are more pronounced at part-load operation.

The specified steam cycle in this study incorporates steam reheat. Therefore, a Benson-type steam generator requires a reheater in parallel with the main vessel. Salt enters the system at 563°C (1045°F) to produce 541°C (1005°F) main and reheat steam. Since steam enters the reheater at approximately 342°C (648°F), the temperature of the salt leaving the reheater must be higher than this value. It is not practical to attempt admitting this salt at an intermediate point within the main steam generating vessel or to cool it before it is admitted to the cold-salt storage tank. Consequently, it must be blended with the salt leaving the main vessel. Over the load range, the temperature of the steam entering the reheater and the temperature of the salt leaving the reheater will vary. The variation will affect the temperature of the salt admitted to the cold-salt storage tank.

Sulzer. The once-through Sulzer circulation method preheats, evaporates, partially superheats main steam in one vessel (preheater/evaporator), and completes superheating in a second vessel (superheater). Steam separators (integral separators) are located downstream of the preheater/evaporator to provide dry, saturated steam to the superheater during start-up. During normal operation superheated steam simply passes through the in-line steam separators before it enters the superheater.

With preheating/evaporation and superheating in separate vessels, each vessel can be optimized for its own thermal/hydraulic performance. Materials and wall thicknesses can be selected for the specific temperatures and pressures each vessel will encounter.

The procedure for start-up of the Sulzer steam generator is similar to that described for the Benson-type unit. The main difference (and advantage) is that the saturated steam discharged from the steam separators can be passed through the superheater and heated to whatever temperature is required by the turbine. It is not necessary to wait until superheated steam conditions are established in the steam generating vessel.

From a control standpoint, a continuous spray station can be located upstream of the superheater for final steam temperature control. With steam generation decoupled from final steam temperature control, system control is more stable than for the Benson-type unit. The potential for static instabilities is also reduced by separating the high-pressure-drop superheat region from the two-phase steam-water evaporating region.

With a separate superheater and reheater, the salt streams leaving each unit can be combined before they pass through the steam generating vessel. A single flow leaving the system at a uniform temperature enables temperature variations to the cold-salt tank to be minimized.

Recirculation.

General. Recirculation steam generators, whether the natural- or forced-circulation type, are characterized by a steam drum that provides dry, saturated steam to the superheater. The steam/water separating capability of the steam drum permits evaporation with low-quality steam leaving the evaporator. Low-quality steam maintains a wet tube surface and nucleate boiling, which has a high heat-transfer rate. The possibility for dryout and film boiling, which has a low heat-transfer rate, is minimized.

Other recirculation steam generator characteristics of special note, identified in Table 4.2 include the following:

- Saturated water in the steam drum maintains the feedwater temperature above the salt freezing point [221°C (430°F)] during part-load operation. Drum water can be blended with the incoming feedwater to achieve the required temperature.
- The steam drum provides a reservoir for collection of feedwater impurities, as noted in the earlier discussion on once-through steam generators.
- A preheater is required to permit cooling of the salt to below the water saturation temperature [336°C (636°F)] before it is returned to the cold-salt storage tank.

Natural Circulation. Natural-circulation steam generators use the difference in density between the steam/water mixture in the heated evaporator tubes and the saturated water in the downcomers to provide the driving force for circulation. The evaporator tube bundle is thus designed with a small frictional pressure-drop component. If one tube were heated more than the others, flow in that tube would increase and therefore would compensate for the increased heat absorption.

Forced Recirculation. Forced recirculation steam generators use a pump to maintain circulation in the evaporator. Because of the high-temperature, high-pressure environment, capital and maintainenance costs for the pump are high, as are the daily power consumption costs. However, the pump does ensure circulation through the evaporator and permit more flexibility in tube bundle design because a range of tube-side pressure drops can be tolerated.

4.1.4 Orientation

The molten salt heat exchangers can be oriented either vertically or horizontally. The selected orientation is strongly dependent upon the heat-transfer surface arrangement and circulation method used. Consequently, advantages and disadvantages of a particular orientation must be evaluated on a case-by-case basis along with the heat-transfer surface arrangement and circulation method selected, as is done during concept elimination and evaluation.

In general, the following items apply to exchanger orientation:

- Vertical units with steam/water flowing upward are more stable than horizontal units.
- Piping and structural support can be minimized with horizontal units.
- Access for maintenance and repair is easier with horizontal units.
- Horizontal units have the potential for shell-side flow stratification and the resultant thermally induced stresses in the shell and tubesheets.
- Field erection and installation are less difficult with horizontal units.

Figures 4.4 and 4.5 illustrate vertical orientations for natural- and forced-circulation, straight-tube arrangements for 100-MWe units. For visual comparison, a horizontal, forced-circulation, straight-tube arrangement for the same capacity unit is illustrated in Figure 4.6.

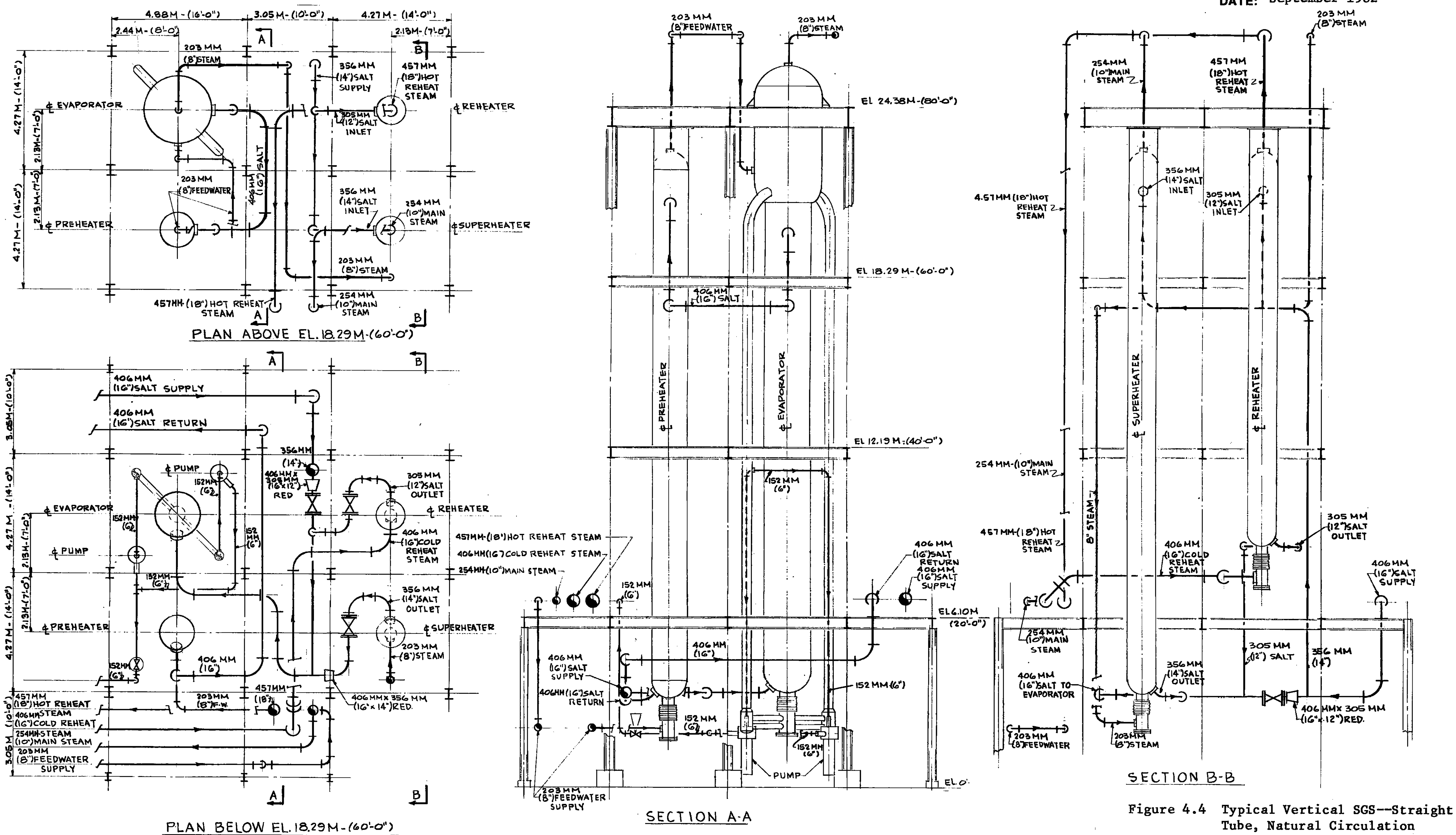
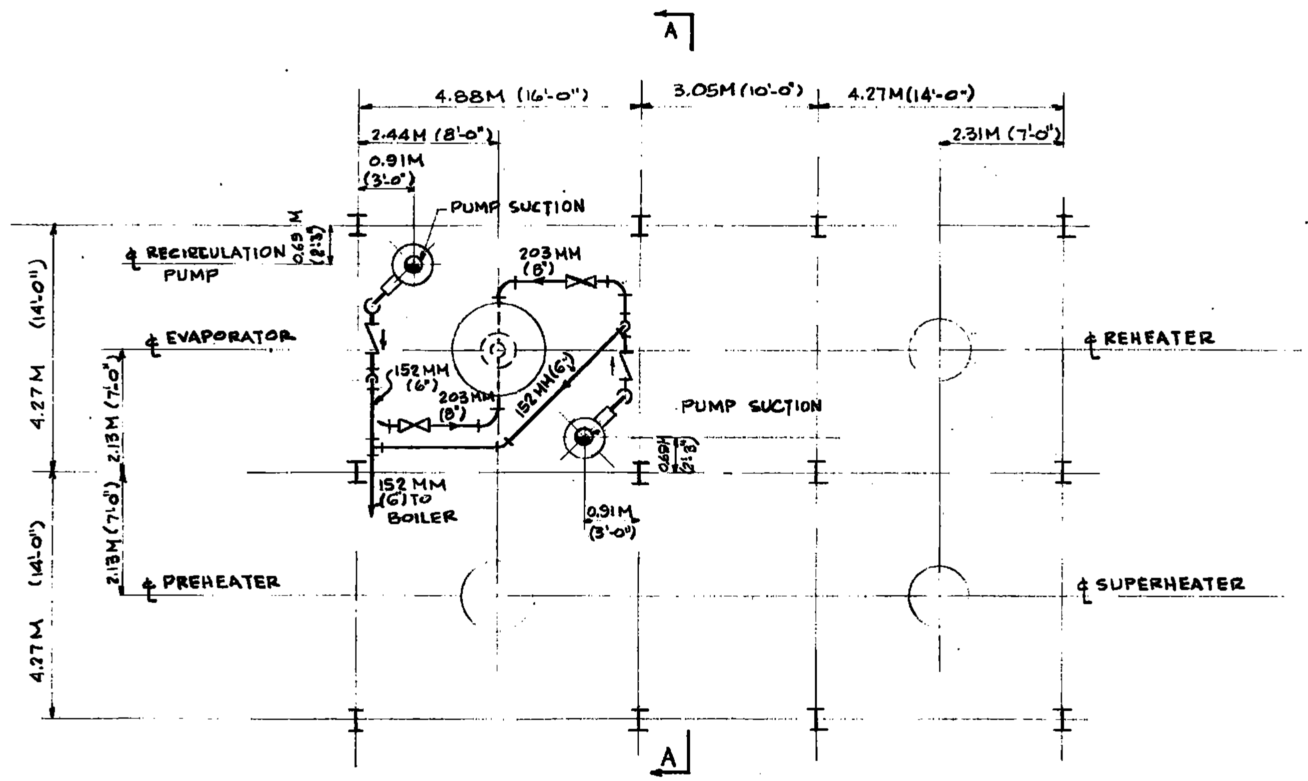
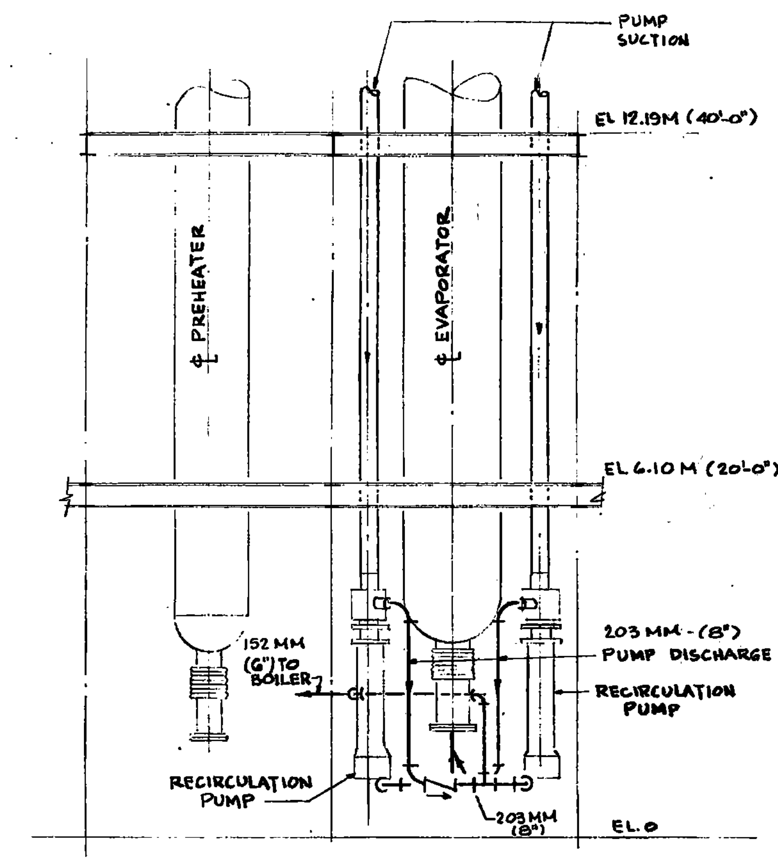


Figure 4.4 Typical Vertical SGS--Straight Tube, Natural Circulation



PLAN



SECTION A-A

Figure 4.5 Typical Vertical SGS--Straight Tube, Forced Recirculation

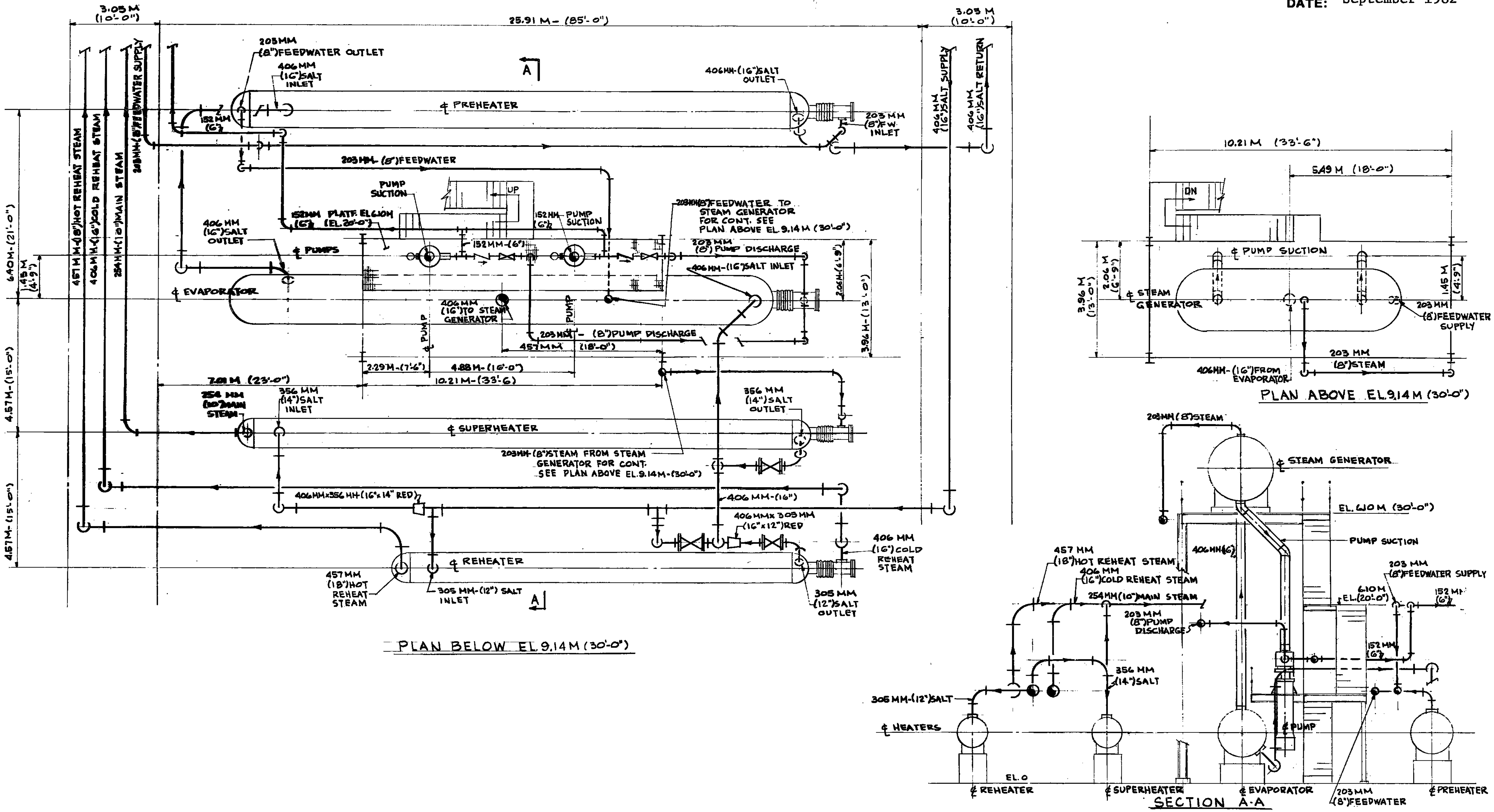


Figure 4.6 Typical Horizontal SGS--Straight Tube, Forced Recirculation

4.2 PRELIMINARY CONCEPT ELIMINATION

With eight candidate surface arrangements, three circulation methods, two orientations, and potentially four heat exchanger components, there were numerous possible combinations of features to consider in selecting an SGS concept. To reduce the candidate concepts to a manageable number, a preliminary concept elimination process was followed.

Figure 4.7 identifies the surface arrangements, circulation methods, orientations, and heat exchanger components. Elimination criteria were established based on the specific concept selection criteria defined in the previous section. Concepts meeting the elimination criteria were considered no further. Because of the obvious disadvantages of the Benson once-through circulation method (noted in Section 4.1.3), the once-through column in Figure 4.7 refers to the Sulzer circulation method only. Items of special note in this figure include the following:

Straight Tube

- The horizontal natural-circulation evaporator was eliminated because it is difficult to design for adequate circulation. To provide the driving force for natural circulation, the steam drum must be elevated high above the exchanger steam/water discharge. Also, the steam/water pressure drop through the exchanger tubing must be minimized to provide the required circulation. Consequently, the low mass-flow rates required for horizontal two-phase flow result in flow stratification between the steam and water. With steam in contact with the upper surface of the tube and water in contact with the lower surface, the difference in film conductance between the two phases causes differences between the top and bottom tubewall temperatures and thus tube warping and potential tube failure.

- The horizontal once-through preheater/evaporator was also eliminated because of the potential for two-phase flow stratification. The unit must be designed with a high pressure drop at the design point to ensure that the mass flow rate at the minimum load condition is sufficient to obtain the proper flow regimes.

ELIMINATION CRITERIA:

A. DIFFICULT TO DESIGN FOR ADEQUATE CIRCULATION [1,2,7,11,13,25,26,27,29]*

B. TWO-PHASE FLOW STRATIFICATION AT LOW LOADS [2,7,10,25,27,29]

C. DIFFICULT TUBE BUNDLE SUPPORT [1,8,11,13,26,27,28,29]

D. TUBE SIDE NON-DRAINABLE [7,10,19,29]

E. DOWNFLOW BOILING CIRCUIT [2,7,25,27,29]

F. EXCESSIVE TEMPERATURE GRADIENT ACROSS TUBE SHEET [3,9,24,27,29]

G. DIFFICULT TO DESIGN FOR STEAM SIDE PRESSURE DROP LIMIT [2]

H. REGENERATION EFFECTS REQUIRE EXCESSIVE SURFACE AREA [1,4,5,16]

*NOTE: SELECTION CRITERIA NOT SATISFIED

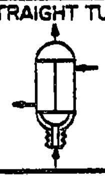
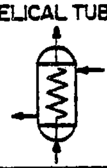
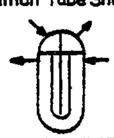
		NATURAL CIRCULATION		FORCED RECIRCULATION		ONCE-THROUGH	
CONFIGURATION	COMPONENT	VERT.	HOR.	VERT.	HOR.	VERT.	HOR.
STRAIGHT TUBE 	PREHEATER						B
	EVAPORATOR		A,B				
	SUPERHEATER						
	REHEATER						
HELICAL TUBE 	PREHEATER		C,D		C,D		C,D
	EVAPORATOR	A	C,D,E		C,D,E		
	SUPERHEATER		C,D		C,D		C,D
	REHEATER		C,D		C,D		C,D
U-TUBE (Common Tube Sheet) 	PREHEATER	D,F	F	D,F	F	D,E,F	B,F
	EVAPORATOR	D,E,F		D,E,F			
	SUPERHEATER	D,F	F	D,F	F	D,F	F
	REHEATER	D,F	F	D,F	F	D,F	F
BAYONET TUBE 	PREHEATER	D		D		D	B
	EVAPORATOR	D	A,B	D			
	SUPERHEATER	D,H	H	D,H	H	D,H	H
	REHEATER	D,G,H	G,H	D,G,H	G,H	D,G,H	G,H

Figure 4.7 Preliminary Concept Elimination

ELIMINATION CRITERIA:

A. DIFFICULT TO DESIGN FOR ADEQUATE CIRCULATION [1,2,7,11,13,25,26,27,29]*
B. TWO-PHASE FLOW STRATIFICATION AT LOW LOADS [2,7,10,25,27,29]
C. DIFFICULT TUBE BUNDLE SUPPORT [1,8,11,13,26,27,28,29]
D. TUBE SIDE NON-DRAINABLE [7,10,19,29]
E. DOWNFLOW BOILING CIRCUIT [2,7,25,27,29]
F. EXCESSIVE TEMPERATURE GRADIENT ACROSS TUBE SHEET [3,9,24,27,29]
G. DIFFICULT TO DESIGN FOR STEAM SIDE PRESSURE DROP LIMIT [2]
H. REGENERATION EFFECTS REQUIRE EXCESSIVE SURFACE AREA [1,4,5,16]

*NOTE: SELECTION CRITERIA NOT SATISFIED

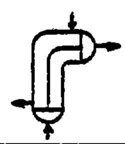
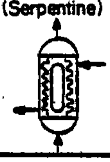
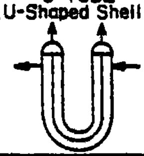
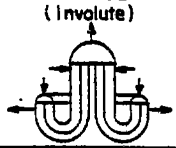
		NATURAL CIRCULATION		FORCED RECIRCULATION		ONCE-THROUGH	
CONFIGURATION	COMPONENT	VERT.	HOR.	VERT.	HOR.	VERT.	HOR.
HOCKEY STICK 	PREHEATER						B
	EVAPORATOR		A,B				
	SUPERHEATER						
	REHEATER						
INVOLUTE TUBE (Serpentine) 	PREHEATER		C		C	B	C,D
	EVAPORATOR	A,B	A,C,D		C,D		
	SUPERHEATER		C		C		C
	REHEATER	G	C,G	G	C,G	G	C,G
U-TUBE (U-Shaped Shell) 	PREHEATER	D		D		D,E	B
	EVAPORATOR	D,E	A,B	D,E			
	SUPERHEATER	D		D		D	
	REHEATER	D		D		D	
U-TUBE (Involute) 	PREHEATER	D	C	D	C	D,E	C
	EVAPORATOR	D,E	A,B,C,E	D,E	C,E		
	SUPERHEATER	D	C	D	C	D	C
	REHEATER	D	C	D	C	D	C

Figure 4.7 Preliminary Concept Elimination (Cont)

Hockey Stick

- The horizontal natural-circulation and once-through evaporators were eliminated for the same reasons identified for the straight-tube design.

Helical Coil

- All horizontal units were eliminated, primarily because of the difficulty in providing adequate tube bundle support and the inability to drain the tubes.
- Horizontal evaporators have tube-side boiling with alternating downflow and upflow with the potential for two-phase flow instability.
- Because of the high pressure-drop characteristic of a coiled arrangement, design of a natural-circulation unit would be difficult.
- Horizontal natural-circulation and once-through evaporators have the tendency for two-phase flow stratification at low loads and the potential for thermally-induced stresses.

Bayonet Tubes

- All vertical units were eliminated because the tube side cannot be drained.
- All superheater and reheater units were eliminated because additional heat-transfer surface is required to negate the effects of regeneration. Since incoming steam in the inner tube is heated by outgoing steam in the outer tube, there is a reduction in effective temperature difference for heat transfer and an increase in heat-transfer area required. The thermal resistance resulting from the additional tubewall and annular stream of steam also increases the heat-transfer area required.
- The physical arrangement of the tube and the fact that high mass-flow rates are required for optimum heat transfer because of the regeneration effects make design of the reheater within acceptable steam and pressure drop limitations a difficult task. High reheat pressure drops significantly affect turbine power output.

Involute Tube (Serpentine)

- All horizontal units were eliminated, primarily because of the difficulty in providing adequate tube bundle support and the inability to drain the tubes.
- The numerous bends in each tube result in high tube-side pressure drop, which makes designing the reheater within the acceptable pressure-drop range difficult.

U-Tube (Common tubesheet)

- All vertical units were eliminated because the tube side cannot be drained. By inverting the unit to make the tube side drainable, the outlet tube leg would be downflow, which is not desirable from a flow-stability standpoint.
- Except for the recirculation (forced, natural) evaporators which operate at a constant tube-side temperature, all arrangements would have excessive temperature gradients across the tubesheet resulting from the variation between inlet and outlet steam/water temperature.

U-Tube (U-shaped shell)

- Similar to the U-tube arrangement with a common tubesheet.
- Vertical units cannot be drained.

U-Tube (Involute)

- All horizontal units were eliminated because of the difficulty in providing adequate tube bundle support.
- All vertical units were eliminated because the tube side cannot be drained.

The concepts surviving the preliminary concept elimination are listed in Table 4.3. If one component required vertical orientation, all components were assumed vertical. The reasoning behind this assumption was that if a support structure must be built for one component, the expense to expand it for all vessels would not be significant.

Because 11 concepts survived preliminary elimination, further elimination was required to reduce the concepts to a more manageable number (three) for detailed evaluation. The four hockey stick concepts identified in Table 4.3 were eliminated for the following reasons:

- The thermal/hydraulic characteristics of the hockey stick design are comparable to the straight-tube design.

- Fabrication of the hockey stick design is costly. Tube bundle assembly is difficult, and the upper tubesheet requires internal bore welds.

- The U-tube (U-shell) design provides individual tube expansion comparable to the hockey stick design and eliminates some of the fabrication difficulties.

Table 4.3 Concepts Surviving Preliminary Concept Elimination

Configuration	Circulation Mode	Orientation*
Straight tube	Natural circulation	Vertical
Straight tube	Forced recirculation	Vertical
Straight tube	Forced recirculation	Horizontal
Straight tube	Once-through	Vertical
Helical coil	Forced recirculation	Vertical
Helical coil	Once-through	Vertical
U-tube (U-shaped shell)	Forced recirculation	Horizontal
Hockey stick	Forced recirculation	Vertical
Hockey stick	Forced recirculation	Horizontal
Hockey stick	Once-through	Vertical
Hockey stick	Natural circulation	Vertical

*If one component requires vertical orientation, all components are assumed vertical.

For further evaluation the concept comparison sequence chart shown in Figure 4.8 was prepared. In addition to the heat exchanger concept comparison, a comparison was made between a separate horizontal steam drum and a vertical steam drum mounted atop the evaporator for the recirculation concepts. Results of the comparisons follow.

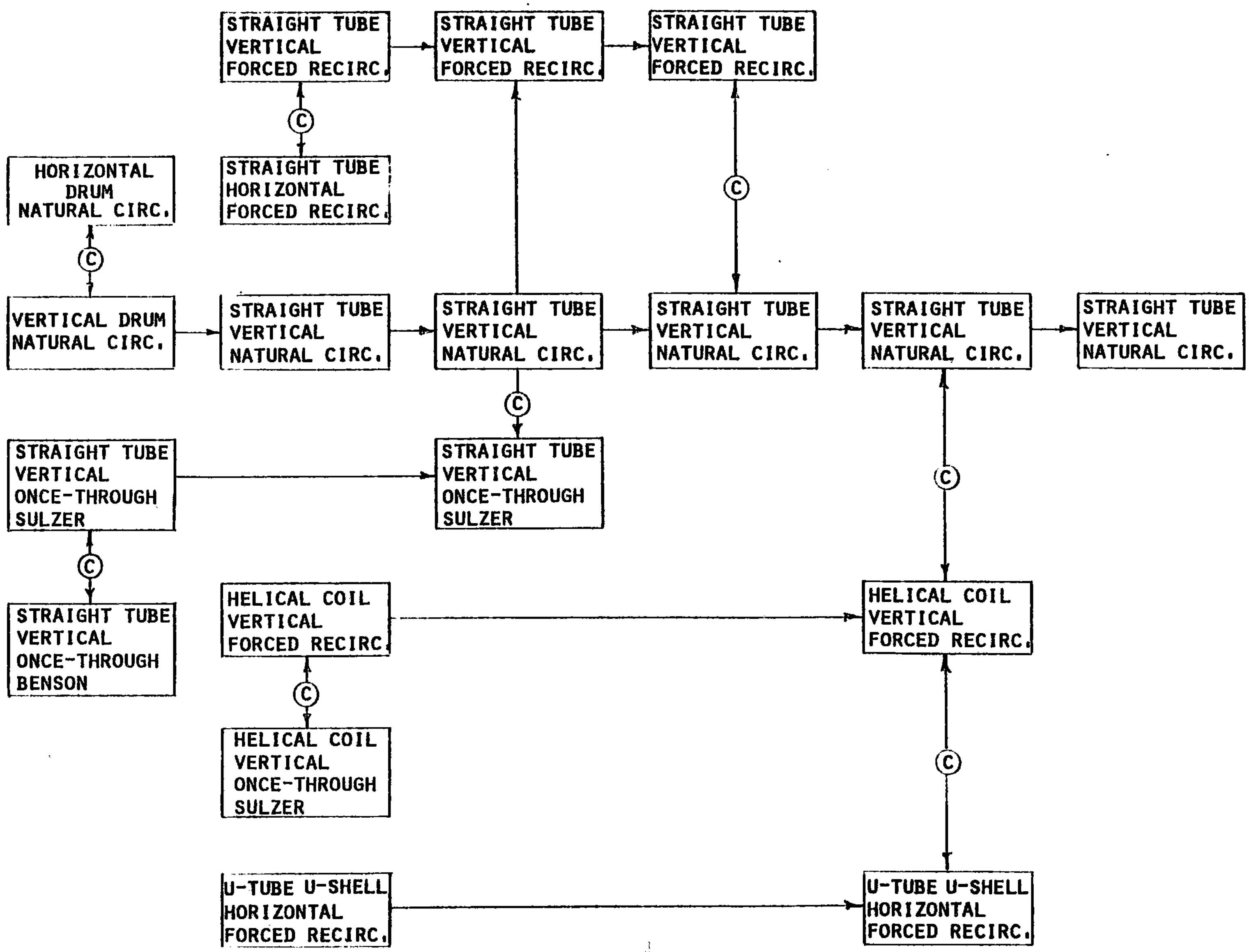


Figure 4.8 Concept Comparison Sequence

4.2.1 Steam Drum Orientation--Horizontal vs. Vertical (Natural circulation)

Conventional fossil-fuel-fired natural-circulation utility steam generators use a horizontal steam drum arrangement to supply dry saturated steam to the steam superheaters. The arrangement, as manufactured by Foster Wheeler, includes a horizontal drum equipped with internals (horizontal separators, chevron driers, feed pipe, etc.), downcomers to direct recirculated water from the drum to the heated elements, and risers which direct the steam/water mixture from the heated elements to the steam drum. As shown in Figure 4.9, this type of arrangement can also be applied to the molten salt SGS evaporator.

Saturated water from the steam drum recirculates through two 460-mm (18-in.) downcomers to the evaporator inlet elevation. Seven 150-mm (6-in.) feeders extend from each downcomer to direct the water to the evaporator inlet. The feeders are looped to provide sufficient flexibility for differential thermal growth between the downcomers and evaporator vessel. The steam/water mixture leaving the evaporator passes through two 406-mm (16-in.) 45-deg elbows into a 406-mm (16-in.) manifold. The steam/water mixture is directed to the steam drum through 16 risers [150 m (6 in.)] extending from the manifold. Since the steam drum and the top of the evaporator are rigidly supported, the risers are looped to provide flexibility for thermal growth.

The number and size of risers, downcomers, and feeders were dictated by the pressure drop required to obtain a 4:1 circulation ratio. The circuitry arrangement illustrated in Figure 4.9 was selected to meet this requirement and also provide adequate flexibility for thermal expansion. The use of fewer, large pipes requires significantly more pipe length to provide flexibility and complicates the mechanical design.

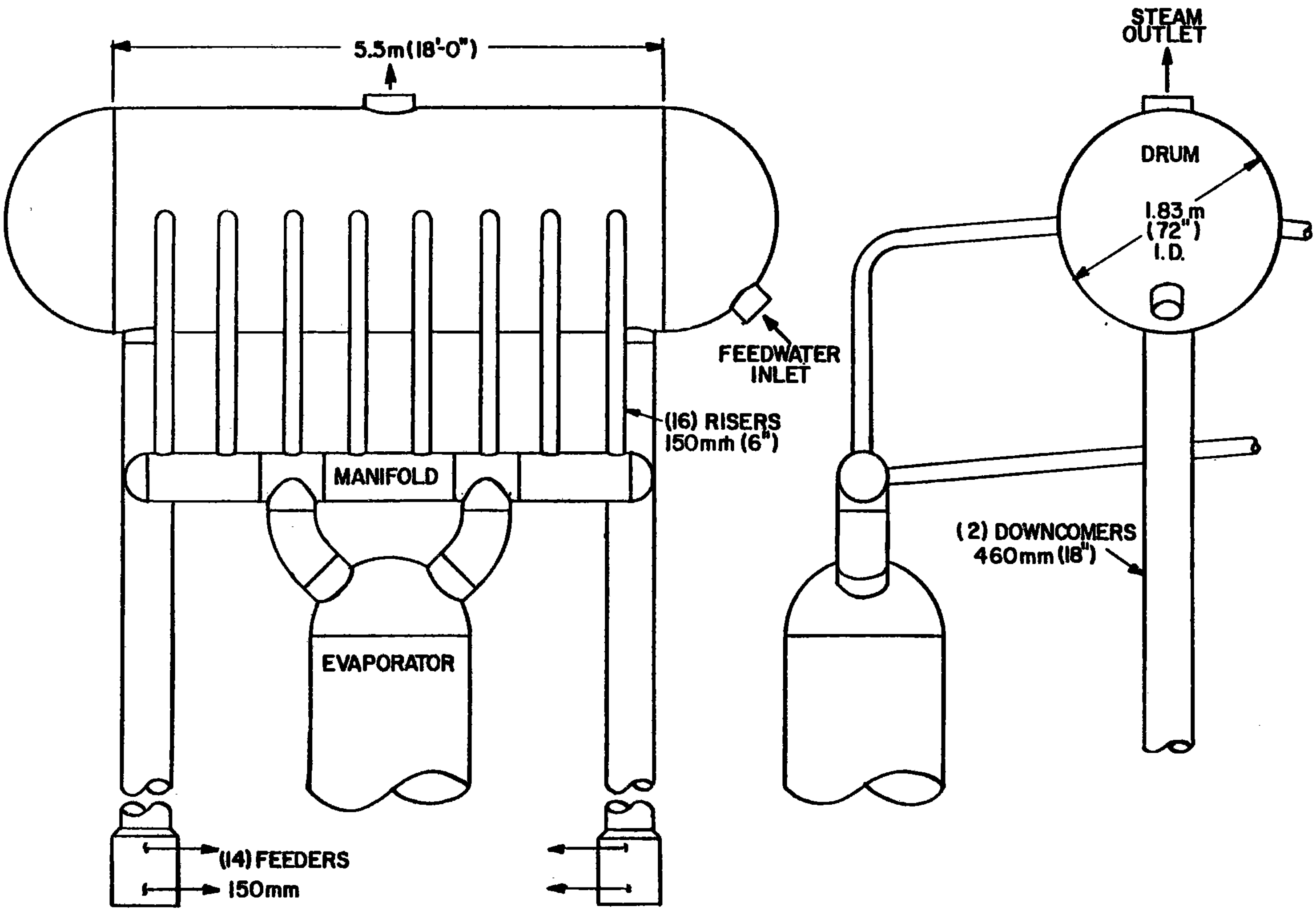


Figure 4.9 Horizontal Steam Drum Arrangement

The size of the steam drum was dictated by the length needed to accommodate the number of tandem separators and chevron driers (Figure 4.10) required to provide dry saturated steam with less than 1 ppm carryover. Figure 4.11 shows their arrangement within the steam drum.

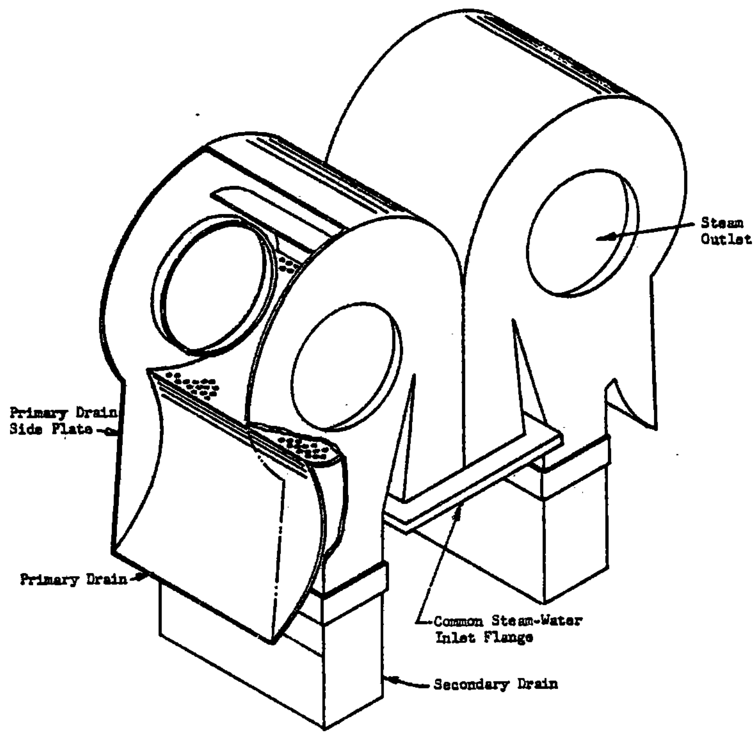
The vertical steam drum illustrated in Figure 4.12 is commonly used in nuclear steam generators and is also very similar to the flash tanks provided on once-through steam generators for start-up control. The drum is designed as an integral component of the evaporator and is welded to the upper tubesheet. The downcomer/feeder arrangement is similar to that used on the aforementioned horizontal steam drum. However, because of the elimination of risers and the lower pressure-drop characteristic of the drum internals, additional pressure drop can be taken in the downcomers and feeders. Consequently, the downcomers can be reduced to 406 mm (16 in.) and the number of 150-mm (6-in.) feeders can be reduced to 12.

The drum is equipped with 27 spiral-arm steam separators arranged in two concentric rows and 19 box-type chevron driers that provide dry, saturated steam with less than 1 ppm carryover. Figure 4.13 illustrates typical separator arms.

Both the vertical and horizontal steam drum arrangements meet the functional performance requirements. However, as noted in Table 4.4, the material and fabrication costs for the horizontal arrangement are 35 percent higher than for the vertical arrangement. The costs for horizontal drum and internals are 6 percent lower than for the vertical drum. The savings for the vertical arrangement results from the elimination of the steam outlet head on the evaporator and the reduction in piping. Although quantitatively not determined, the

TANDEM SEPARATORS

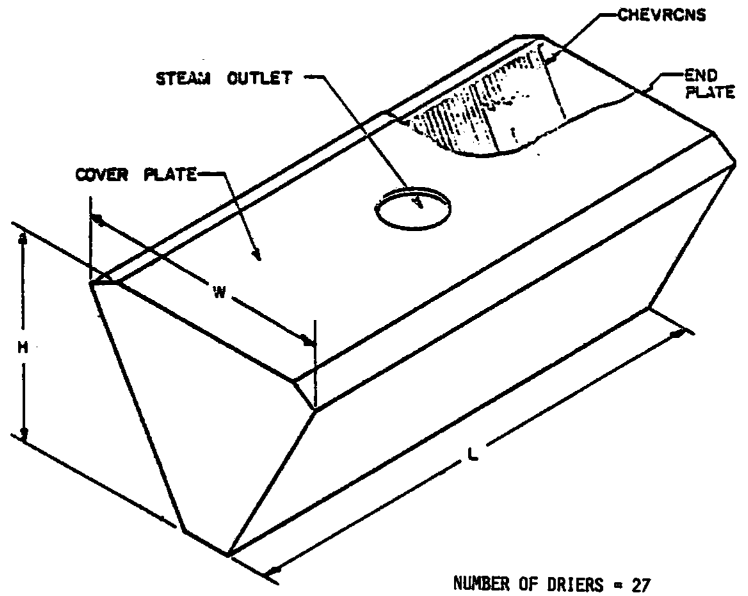
HORIZONTAL STEAM DRUM



ROWS = 5
NUMBER OF SEPARATORS = 40
CENTERLINE SPACING = 690MM (27")

CHEVRON DRIERS

HORIZONTAL STEAM DRUM



NUMBER OF DRIERS = 27
L = 1.14M (45")
H = 180MM (7")
CENTERLINE SPACING = 205MM (8.1")

Figure 4.10 Tandem Separators and Chevron Driers

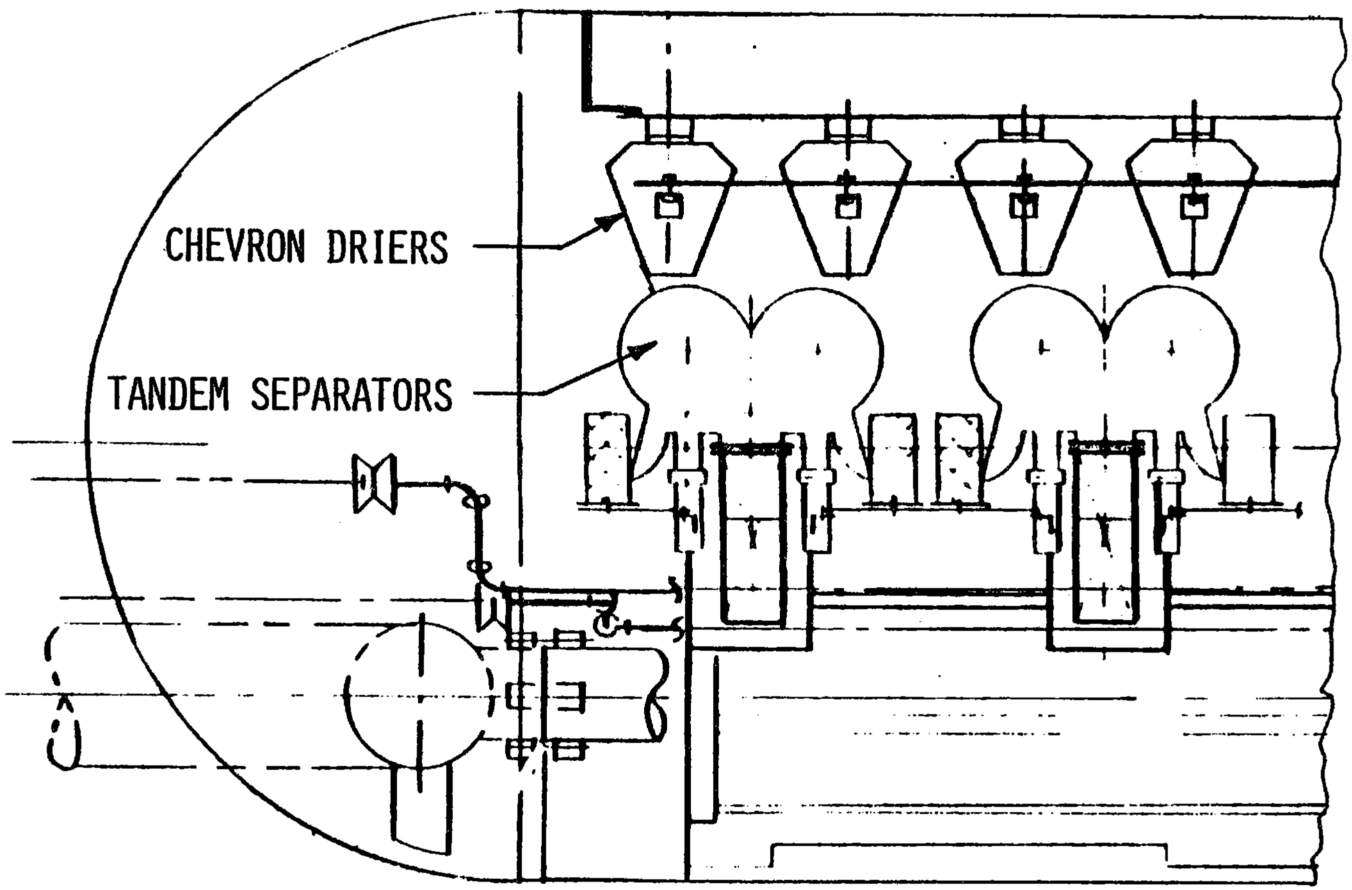


Figure 4.11 Drum Internals Arrangement--Horizontal Steam Drum

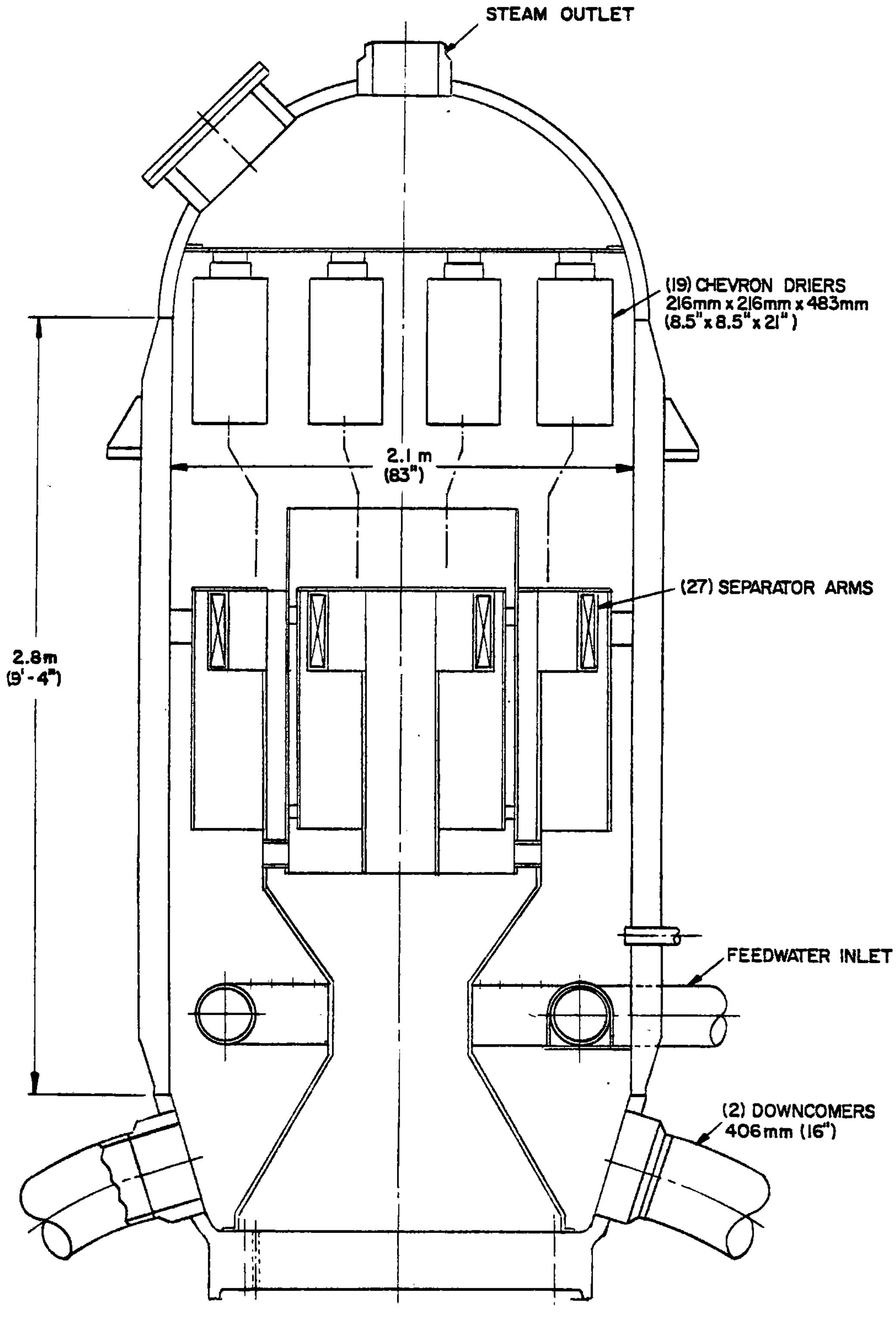


Figure 4.12 Vertical Steam Drum Arrangement

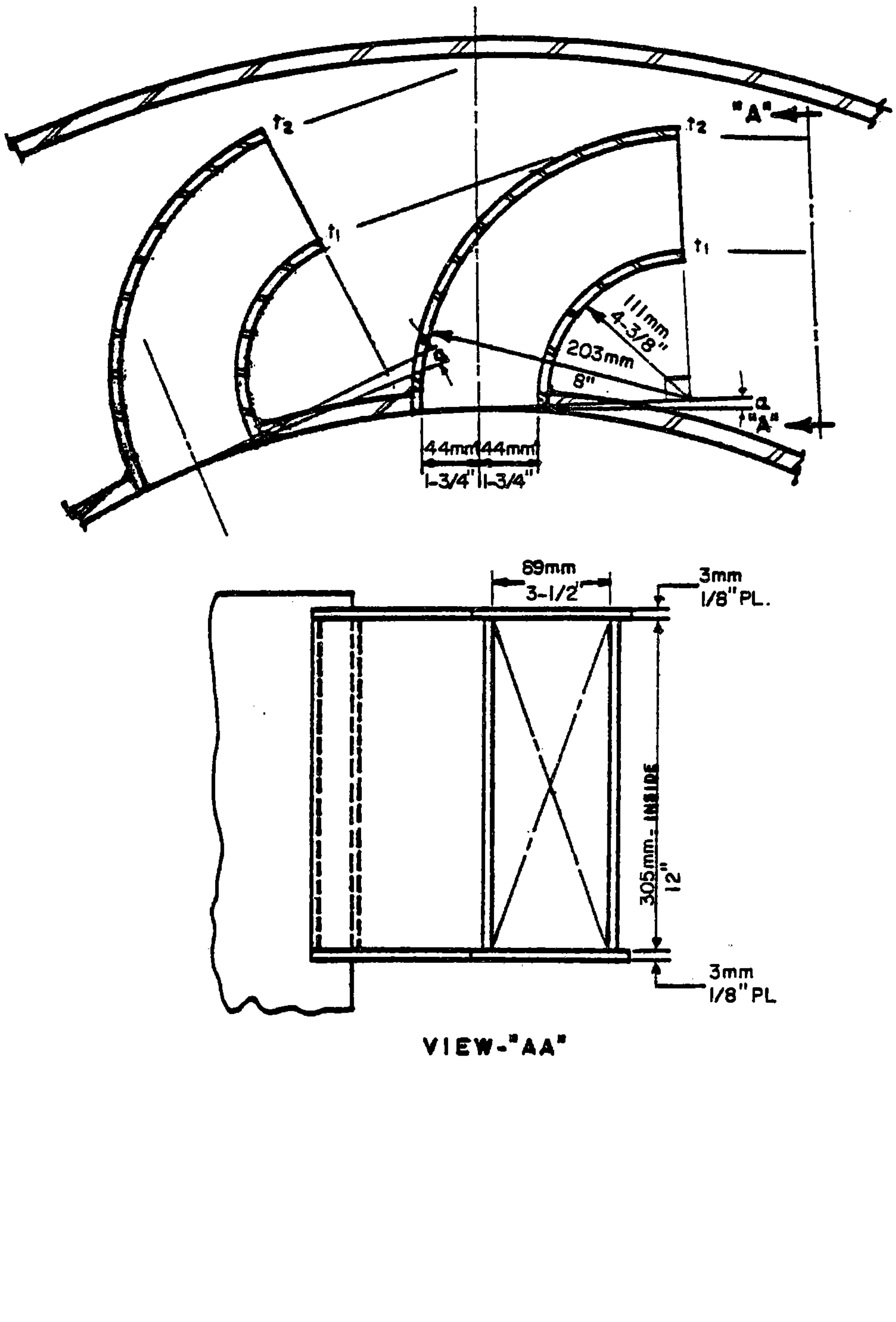


Figure 4.13 Separator Arms

costs of field erection and support structure for the horizontal arrangement will also add to the overall cost. By designing the vertical drum as an integral component of the evaporator, it does not have to be separately installed and supported. Consequently, the vertical drum arrangement was selected for all vertical evaporator recirculation arrangements.

Table 4.4 Cost Comparison for Horizontal vs. Vertical Steam Drum (Natural-circulation design)

	Cost ($)		
Shop Material and Fabrication	Vertical	Horizontal*	
Drum and internals:			
Material	58,700	74,000	
Labor and overhead	109,100	84,000	
Subtotal	167,800	158,000	(-6%)
Piping:			
Material	20,500	57,400	
Labor and overhead	18,100	39,000	
Subtotal	38,600	96,400	(+150%)
Evaporator steam outlet head:			
Material	---	16,300	
Labor and overhead	---	8,600	
Subtotal		24,900	
Total	206,400	274,300	(+35%)

*Both erection and support are more expensive for the horizontal drum.

4.2.2 Once-Through Benson vs. Sulzer (Vertical, straight tube)

The description of Benson and Sulzer once-through circulation methods included in Section 4.1.3 identified features of the Sulzer design which are superior to the Benson design. These include the following:

- Flow Stability. By having the high pressure drop superheating region in a separate vessel, the Sulzer design is less likely to experience static flow instability.

- Start-Up Steam Temperature Control. Before generating steam in the evaporator, the Sulzer design can provide superheated steam at whatever temperature is required by the turbine by flashing steam upstream of the integral steam separators and superheating it in the superheater vessel. Since the Benson design does not include a separate superheating vessel, the required steam conditions must be established in the evaporator before steam can be admitted to the turbine.

- Heat-Transfer Surface Optimization. Preheating, evaporation, and superheating occur within each tube of a Benson type unit. Consequently, simultaneous optimization of each of these regions is not possible. By having a separate superheater, the Sulzer type design permits optimization of the superheater without affecting the preheat and evaporation regions.

- Material Selection. With preheating, evaporation, and superheating in one vessel, the Benson type unit must be designed with material satisfactory for the high temperature end. Consequently, the cold end will be overdesigned. By having a separate superheater, the Sulzer preheater/evaporator can be designed with less expensive materials.

- Cold Salt Temperature. The specified reheat steam cycle requires a separate reheater vessel. Since steam enters the reheater at a temperature of 342°C (648°F) the salt must leave at a temperature greater than this value. In the Sulzer design the salt leaving the reheater is blended with the salt leaving superheater, passed through the preheater/evaporator, and directed to the cold salt storage tank at a uniform temperature. In the Benson design, salt leaving the reheater is blended with salt leaving the preheater/evaporator/superheater vessel and directed to the cold salt storage tank. Variation in reheater inlet steam temperature will result in salt temperature variations to the cold salt storage tank.

Based on the aforementioned features of the Sulzer design, the Benson type once-through (vertical, straight tube) was eliminated from further consideration.

4.2.3 Horizontal vs. Vertical (Straight tube, forced recirculation)

Advantages and disadvantages for horizontal and vertical orientation of the straight tube, forced recirculation concept are summarized in Table 4.5.

Although there is the potential for cost savings with horizontal orientation, the horizontal concept was eliminated from further consideration primarily for functional reasons. These include the following:

- Salt Stratification. In the horizontal position there is a tendency for salt stratification on the shell side. Cold salt will settle in layers at the bottom of the vessel and hot salt at the top. A large temperature difference from top to bottom will impose severe thermal stresses on the shell and tubesheets. Heat transfer performance will also be lowered because of the stratification. This problem has been identified for sodium-heated nuclear steam generators.
- Salt Drainability. In order to minimize the aforementioned salt stratification and to improve heat transfer, baffle plates can be used to create an up and down crossflow. However, in order to drain salt from the unit, drainage taps may have to be located between alternate baffles. Also, in the horizontal concept the expansion bellows are not fully drainable.
- Water/Steam Maldistribution. The potential for water/steam maldistribution is greatest in horizontal units. Gravity acts to create the maldistribution as compared to vertical upflow units in which it tends to stabilize the flow.

4.2.4 Once-Through (Sulzer) vs. Recirculation (Vertical, straight tube)

Requirements, advantages, and disadvantages of once-through and recirculation circulation methods were summarized in Table 4.2 and briefly described in Section 4.1.3. Features of the recirculation methods (natural and forced) that make them superior to the once-through method, for this application, include the following:

- Feedwater Treatment. The steam drum in the recirculation methods with its blowdown and steam separating capability protects the unit if a condenser leak permits untreated water to enter system. Once-through units require a

Table 4.5 Horizontal vs. Vertical Orientation (Straight tube, forced recirculation)

	Horizontal	Vertical
Advantages:	Minimizes interconnecting piping	Hanging arrangements permits unrestricted thermal expansion
	Minimizes support structure	Can utilize integral vertical or separate horizontal steam drum
	Minimizes erection costs	
	Easier to service and repair	
Disadvantages:	Salt stratification	Additional interconnecting pipe
	Shell-side draining more difficult	Additional support structure
	Expansion bellows cannot be drained	Additional erection costs
	Steam/water maldistribution	
	Shell requires intermediate supports	
	Tube/baffle wear and binding potential	

condensate polishing system. Feedwater treatment facility capital cost differences between once-through and recirculation steam generators are identified in Table 4.6.

- Feedwater Inlet Temperature Control. Recirculation units have a reservoir of saturated water in the steam drum which can be recirculated to the preheater inlet to keep the feedwater inlet temperature above the salt freezing point.

- Flow Stability. Recirculation units are less likely to experience two-phase flow instabilities.

- Control System. The simple start-up control system for recirculation units is better suited for the daily cycling required by a solar steam generator.

- Unit Size. The preliminary size of the preheater/evaporator for a 100 MWe once-through (Sulzer) steam generator is identified in Figure 4.14. By incorporating preheat and evaporation in a single vessel, the tube length [38.1 m (125 ft)] required to obtain reasonable shell and tubeside flow characteristics is excessively long.

Based on these features and the items identified in Table 4.2, the once-through Sulzer (vertical, straight tube) concept was eliminated form further consideration.

4.2.5 Once-Through (Sulzer) vs. Forced Recirculation (Vertical, helical coil)

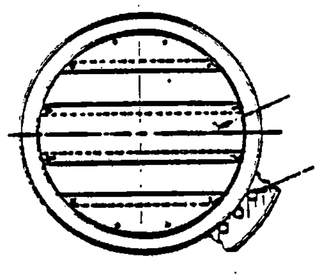
The once-through (Sulzer) vertical, helical coil concept was eliminated from further consideration for the same reasons identified above for the once-through (Sulzer) vertical, straight tube concept.

4.2.6 Natural Circulation vs. Forced Recirculation (Vertical, straight tube)

As noted in Table 4.2, the characteristics of natural-circulation and forced-recirculation steam generators are very similar. However, natural circulation was selected over forced recirculation primarily because of cost and maintenance.

Table 4.6 Feedwater Treatment System--Once-Through vs. Drum Type

Description	Cost ($)	
Once-Through System		
Condensate Polishing System (Powdered resin type):	225,000	
2 - 1.1-m (42-in.)-dia service vessels		
1 - Precoat tank with pump		
2 - Holdinng pumps and sample rack		
Chemical Feed System:	15,000	
1 - 100-gal stainless steel ammonia solution tank		
2 - Metering pumps with accessories		
1 - 100-gal stainless steel hydrazine solution tank		
2 - Metering pumps with accessories		
1 - Panel		
Total	240,000	
Drum-Type System		
Chemical Feed System:	22,500	
1 - 100-gal stainless steel ammonia solution tank		
2 - Metering pumps with accessories		
1 - 100-gal stainless steel hydrazine solution tank		
2 - Metering pumps with accessories		
1 - 100-gal stainless steel phosphate solution tank		
2 - Metering pumps with accessories		
1 - Panel		
Total	22,500	
Cost Differential		$217,500



TUBE O.D.	15.88 mm (5/8")
MIN. WALL	1.65 mm (0.065")
NUMBER OF TUBES	1593
TUBE LENGTH, L	38.1 m (125')
HEIGHT, H	41.5 m (136')
SHELL I.D., D	826 mm (33")

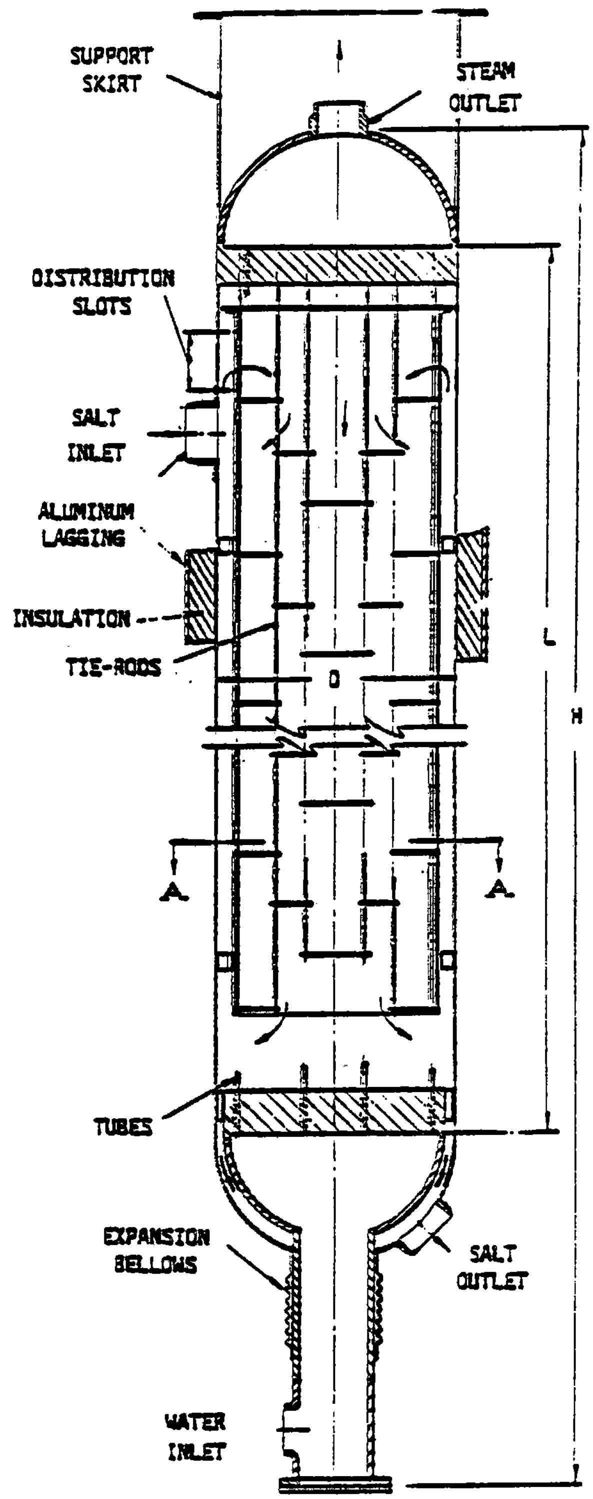


Figure 4.14 Once-Through Preheater/Evaporator--Straight Tube Design

Table 4.7 indicates that the preheater, superheater, and reheater for both systems will be identical and that material and fabrication costs will be the same. The cost of the evaporator for the forced-recirculation concept can potentially be reduced because high tube-side mass flow rates can be used; they result in increased heat-transfer rates, reduced heat-transfer surface, and fewer, smaller tubes. However, to achieve these savings, recirculation pumps capable of high-temperature, -pressure, and -flow-rate operation are required. The capital cost differential between these pumps and the low-capacity drum water recirculation pumps required for feedwater temperature control in the natural-circulation concept is approximately $204,300. If the greater operating and maintenance costs for the larger capacity recirculation pumps are not considered, the forced-recirculation evaporator would have to be 40 percent less expensive than the natural-circulation evaporator to compensate for the increased pump capital cost. Since the evaporator tube-side heat-transfer coefficient is only a small fraction of the overall heat-transfer coefficient, it is not conceivable that a 40-percent savings in evaporator cost can be achieved.

We thus eliminated the forced recirculation (vertical, straight tube) concept from further consideration.

Table 4.7 Natural Circulation vs. Forced-Recirculation Costs
(Straight tube, vertical)

	Component Material and Fabrication Costs ($)	
Description	Natural Circulation	Forced Recirculation
Preheater	395,300	395,300
Evaporator	505,700	*
Superheater	394,300	394,300
Reheater	402,600	402,600
Recirculation pumps (2)	183,100	387,400
Pump cost differential	---	+204,300
Pump operating and maintenance costs	---	Greater

*Forced-recirculation evaporator material and fabrication costs must be 40 percent lower than those for the natural-circulation evaporator to compensate for recirculation pump capital costs.

4.3 CONCEPT EVALUATION

The three SGS concepts surviving the preliminary concept elimination process were:

- Straight tube, vertical, natural circulation
- Helical coil, vertical, forced recirculation
- U-tube (U-shaped shell), horizontal, forced recirculation.

For final concept selection, the heat exchangers for each concept were sized in sufficient detail to obtain approximate cost estimates. Unit sizing was based on:

- Temperature, pressure, and flow requirements as stated in the Phase 1 proposal and RFQ for the 100-MWe solar stand-alone SGS
- Carbon steel preheater, alloy steel (1%Cr-1/2%Mo) evaporator, and stainless steel (Type 304) superheater and reheater
- Tube- and shell-side fouling resistances of 90 $m^2 \cdot {}^\circ C/MW$ (0.0005 $h \cdot ft^2 \cdot {}^\circ F/Btu$)
- No surface margins
- Tube- and shell-side pressure drops of approximately 207 kPa (30 lb/in^2)
- Reasonable length-to-diameter ratio.

Results of unit sizing for the straight tube, helical coil, and U-tube designs are included in Figures 4.15, 4.16, and 4.17 respectively. The estimated shop material and fabricating costs for each component are listed in Table 4.8. The material and, in particular, fabrication costs for the helical coil and U-tube designs are considerably more than for the straight tube design.

	PREHEATER	EVAPORATOR	SUPERHEATER	REHEATER
SHELL I.D., D	1219mm (48.0")	1524mm (60.0")	991mm (39.0")	1003mm (39.5")
HEIGHT, H	21.0m (69.1')	*23.9m (78.3')	20.8m (68.3')	17.2m (56.4')
TUBE LENGTH, L	18.3m (60')	16.8m (55')	18.3m (60')	14.6m (48')
TUBE O.D.	15.9mm (5/8")	25.4mm (1")	15.9mm (5/8")	19.1mm (3/4")
NUMBER OF TUBES	1816	1361	1108	1028
TUBE MATERIALS	CS	1 Cr-½ Mo	304SS	304SS
PRESSURE DROP: KPa (lb/in^2)				
SHELL SIDE	207 (30)	166 (24)	200 (29)	55 (8)
TUBE SIDE	14 (2)	CR=4.2	124 (18)	179 (26)

* NOTE: TO TOP OF VERTICAL STEAM DRUM

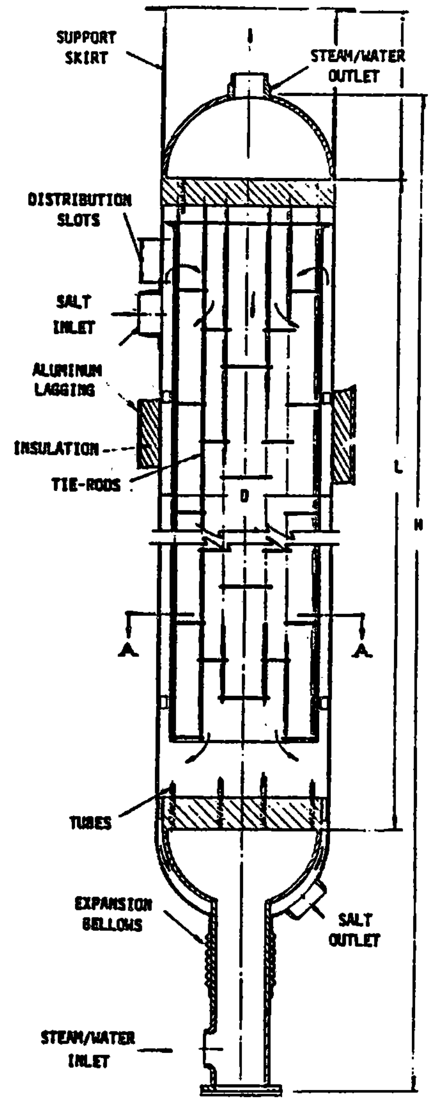


Figure 4.15 Straight Tube (Vertical, natural circulation)--Preliminary Design

	PREHEATER	EVAPORATOR	SUPERHEATER	REHEATER
SHELL I.D.,D	1270mm (50.0")	1397mm (55.0")	1270mm (50.0")	1397mm (55.0")
HEIGHT, H	35.7m (117')	31.7m (104')	26.2m (86')	20.1m (66')
TUBE LENGTH	51.2m (168')	49.4m (162')	35.4m (116')	28.7m (94')
TUBE O.D.	25.4mm (1")	31.8mm (1-¼")	25.4mm (1")	38.1mm (1-½")
NUMBER OF TUBES	370	352	370	252
TUBE MATERIAL	CS	1 Cr-½ Mo	304SS	304SS
PRESSURE DROP: KPa (lb/in^2)				
SHELL SIDE	310 (45)	179 (26)	47 (14)	21 (3)
TUBE SIDE	41 (6)	186 (27)	207 (30)	159 (23)

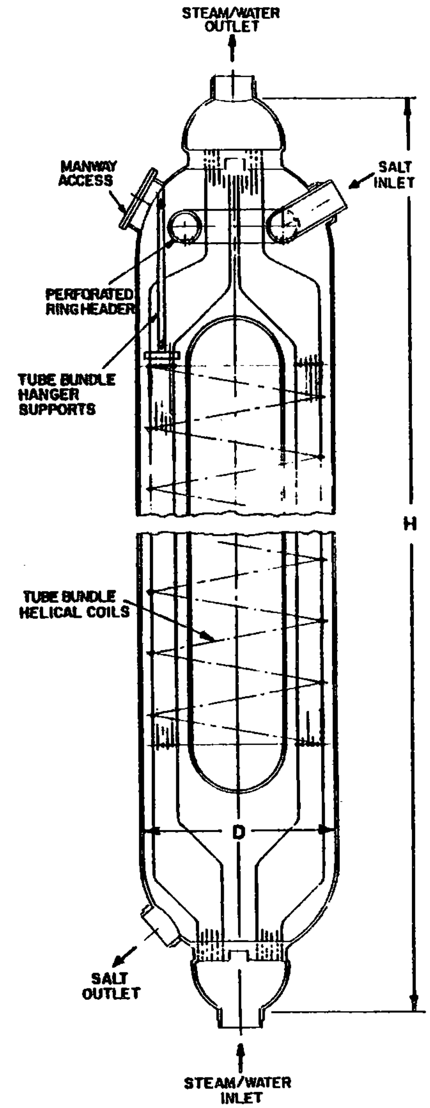


Figure 4.16 Helical Coil (Vertical, forced recirculation)--Preliminary Design

	PREHEATER	EVAPORATOR	SUPERHEATER	REHEATER
SHELL I.D., D	1118mm (44.0")	1067mm (42.0")	864mm (34.0")	889mm (35.0")
UNIT LENGTH, L	12.7m (41.6')	12.6m (41.5')	11.8m (38.7')	13.8m (45.3')
TUBE LENGTH	24.4m (80')	24.4m (80')	22.9m (75')	26.8m (88')
TUBE O.D.	15.9mm (5/8")	15.9mm (5/8")	15.9mm (5/8")	31.8mm (1-¼")
NUMBER OF TUBES	1587	1403	914	355
TUBE MATERIAL	CS	1 Cr-½ Mo	304SS	304SS
WIDTH, W	2.2m (7.3')	2.1m (7.0')	1.7m (5.7')	1.8m (5.8')
PRESSURE DROP: KPa (lb/in^2)				
SHELL SIDE	145 (21)	200 (29)	145 (21)	152 (22)
TUBE SIDE	14 (2)	186 (27)	214 (31)	145 (21)

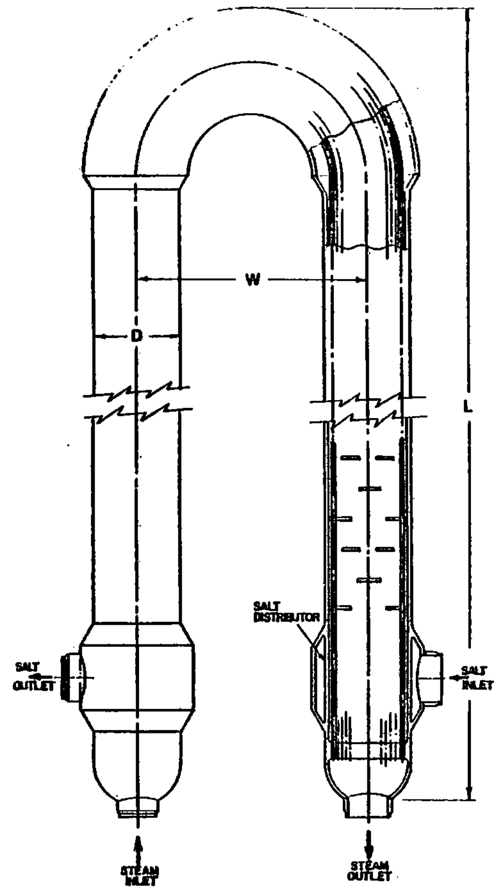


Figure 4.17 U-Tube, (U-shaped shell)(Horizontal, forced recirculation)--Preliminary Design

Table 4.8 Component Material and Fabrication Cost Comparison ($)

	Straight Tube Vertical Natural Circulation		Helical Coil Vertical Forced Recirculation		U-Tube (U-shell) Horizontal Forced Recirculation	
	Material	Fabrication	Material	Fabrication	Material	Fabrication
Preheater	188,900	206,400	143,800	573,200	211,500	402,800
Evaporator	245,700	260,000	347,300	771,200	291,900	391,400
Superheater	177,000	217,300	291,600	734,100	202,100	380,000
Reheater	211,400	191,200	252,000	571,300	209,400	332,600
Subtotal	823,000	874,900	1,034,700	2,649,800	914,900	1,507,800
			(+26%)	(+203%)	(+11%)	(+72%)
Total	1,697,900		3,684,500 (+117%)		2,421,700 (+43%)	

Consequently, the helical coil and U-tube concepts were eliminated for the following reasons:

Helical Coil

- Costly fabrication
- Expensive recirculation pumps required.

U-Tube (U-shaped shell)

- There are problems associated with horizontal orientation (salt stratification, and draining, tube/baffle wear potential, etc.).
- Return bend support is difficult.
- Return bend heat-transfer surface is not as effective as in the straight section because of baffle location.
- Differential expansion between hot and cold legs is a problem.
- Expensive recirculation pumps are required.
- Fabrication is more costly than for the straight-tube design.

4.4 MATERIALS

4.4.1 Selection

Evaluation of materials for the molten salt SGS heat exchangers was based on the following preliminary salt, steam, and water temperatures and pressures identified in the proposal:

Conditions		Preheater	Evaporator	Superheater	Reheater
Design Temperature, °C (°F)		371 (700)	371 (700)	566 (1050)	566 (1050)
Design Pressure, MPa gage (lb/in^2g)		14.7 (2132)	14.4 (2082)	14.4 (2082)	3.8 (550)
Fluid:					
Tube Side		Water	Steam/ Water	Steam	Steam
Shell Side		Molten Salt: 60%$NaNO_3$-40%KNO_3			
Fluid Temperature, °C (°F):					
Molten Salt:	In	341 (645)	447 (836)	566 (1050)	566 (1050)
	Out	288 (550)	341 (645)	447 (836)	447 (836)
Steam/Water:	In	238 (460)	336 (636)	336 (636)	342 (648)
	Out	336 (636)	336 (636)	541 (1005)	541 (1005)

The microscopic examination and chemical identification of scales occurring on the outside surfaces of specimens subjected to nitrate-nitrite salts (40% $NaNO_2$-7% $NaNO_3$-53% KNO_3) and nitrate salts (60% $NaNO_3$-40% KNO_3) in Sandia and Martin Marietta laboratory test programs indicate that the molten salt corrosion mechanism results primarily from oxidation processes.[17,18]

Oxide flaking has been observed on carbon steel and 2-1/4%Cr-1%Mo steel alloys subject to relatively short-term molten salt exposure. However, evidence indicates that the films formed on these alloys becomes more stable after longer term exposure, producing an oxidation behavior similar to that observed on boiler tubing that has been exposed to steam/water environments.

A review of molten salt technology shows that carbon steel has been a satisfactory tubing material in over 500 commercial units where 40% $NaNO_2$-7% $NaNO_3$-53% KNO_3 salt has been used as the heat-transfer medium.[19] While most of these industrial units operate in the 371 to 454°C (700 to 850°F) range, a number operate at temperatures as high as 499°C (930°F), and at least two are still operating satisfactorily after 10 years between 510 and 538°C (950 and 1000°F).

In extended open-air immersion tests conducted by Martin Marietta, carbon steel suffered minor metal loss after 16,000 hours exposure at 399°C (750°F) in 60% $NaNO_3$-40% KNO_3 salt. The surface loss was calculated at 165 μm (6.5 mil) based on a 30-year extrapolation of specimen weight-change data.[18]

In open beaker immersion tests conducted at Sandia, both carbon steel and 2-1/4%Cr-1%Mo steel suffered weight changes equivalent to slightly less than 25 μm (1 mil) after 4560 hours (190 days) of exposure to 40% $NaNO_2$-7% $NaNO_3$-53% KNO_3 maintained at 550°C (1022°F).[17] This value compares favorably with the 25 to 38 μm (1.0 to 1.5 mil) suffered by ferritic alloys exposed to steam/water environments for similar times and temperatures.

The 25 to 38 μm (1 to 1.5 mil) surface recession resulting from oxidation is derived from the Larson-Miller Plot shown in Figure 4.18,[20,21] which is based on more than 30 years' experience with boilers operating under steady-state and peaking service similar to the thermal cycling encountered in solar-powered components.

On the basis of Foster Wheeler's review of laboratory test results and industrial experience, as well as consideration of the cost, ease of fabrication, strength, and oxidation resistance, the following materials should be satisfactory for molten salt/steam/water service:

- Carbon steel for 427°C (800°F) maximum temperature (Tubewall temperature)
- 1-1/4%Cr-1%Mo steel for 510°C (950°F) maximum temperature
- 2-1/4%Cr-1%Mo for 566°C (1050°F) maximum temperature
- Type 304SS for a service temperature above 574°C (1075°F), if good boiler steam is used
- Incoloy 800 for molten salt service above 574°C (1075°F), if steam contains some traces of chloride (Cl^-) contamination.

The maximum tubewall temperature limits for the molten salt/steam service are slightly more conservative than those shown in Table 4.9[22] for steam/water service.

Although the aforementioned materials should be satisfactory for the molten salt SGS, a conservative approach was taken in the final selection of

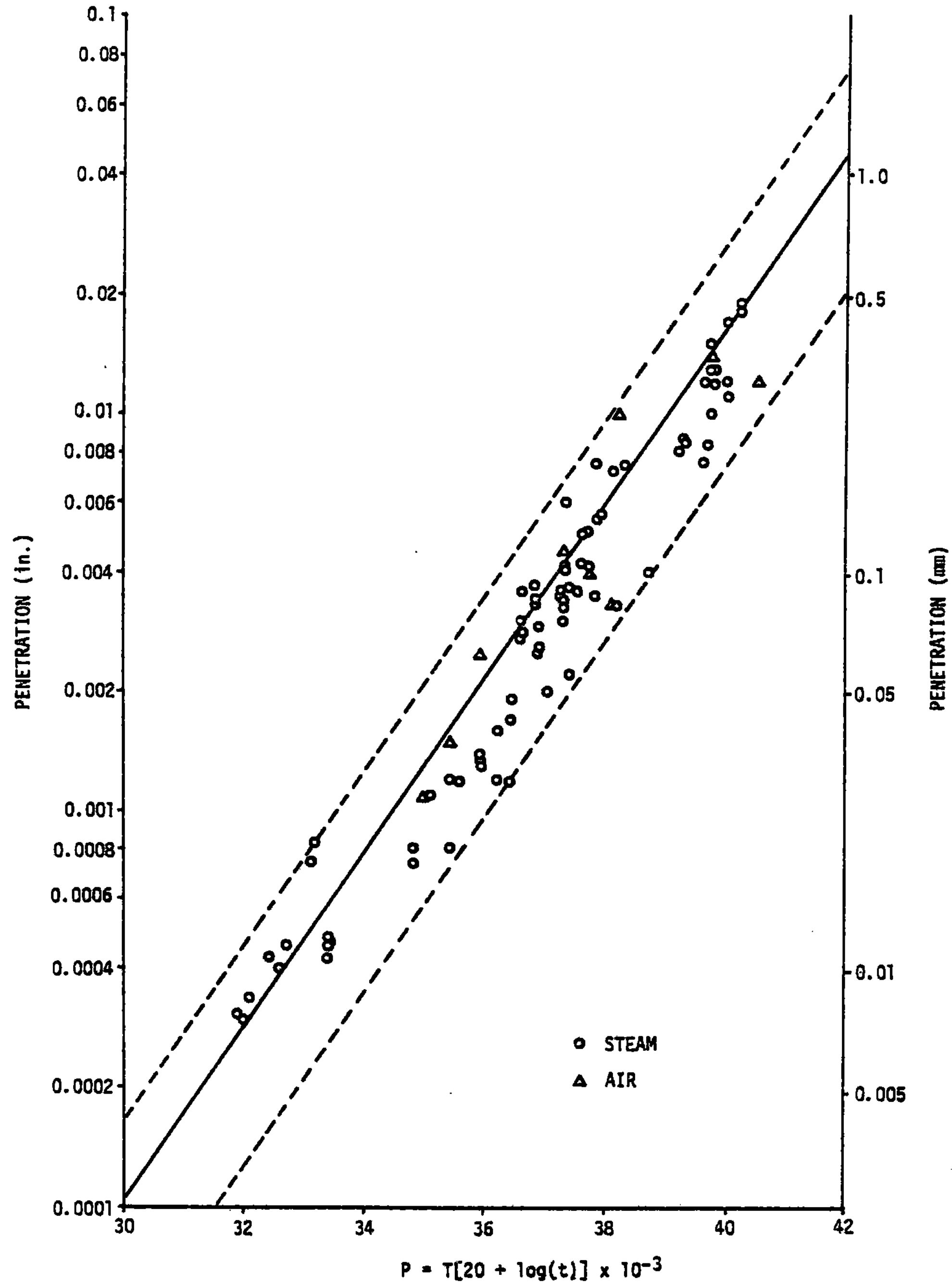


Figure 4.18 Steam/Air Corrosion of Ferritic Steels--Larson-Miller Plot

Table 4.9 Tube Material and Temperature Limitations*

	Specification	Grade	Material	Mean Metal Temperature °C (°F)
Seamless Carbon Steel Tube	SA-210	A-1	Carbon Steel	454 (850) Maximum
	SA-210	C	Carbon Steel	454 (850) Maximum
	SA-192	---	Carbon Steel	454 (850) Maximum
Electric-Resistance Welded Carbon Steel Tubes	SA-178	A	Carbon Steel	454 (850) Maximum
	SA-178	C	Carbon Steel	454 (850) Maximum
	SA-226	---	Carbon Steel	454 (850) Maximum
Seamless Alloy Steel Tubes	SA-209	T-1a	C-1/2%Mo	468 (875) Maximum
	SA-213	T-2	1/2%Cr-1/2%Mo	510 (950) Maximum
	SA-213	T-12	1%Cr-1/2%Mo	538 (1000) Maximum
	SA-213	T-11	1-1/4%Cr-1/2%Mo	566 (1050) Maximum
	SA-213	T-22	2-1/4%Cr-1%Mo	593 (1100) Maximum
	SA-213	T-9	9%Cr-1%Mo	621 (1150) Maximum
Seamless Stainless Steel Tubes	SA-213	321H	18%Cr-10%Ni-Ti	Above 593 (1100)
	SA-213	347H	18%Cr-10%Ni-Cb	Above 593 (1100)
	SA-213	304H	18%Cr-8%Ni	Above 593 (1100)
	SA-213	316H	16%Cr-13%Ni-3%Mo	Above 677 (1250)

*Materials from ASME Code, Section I, 1971 (Table p. 23.1).

materials. Because of limited high-temperature molten salt corrosion data, the following materials were selected for the SGS design:

Component	Material
Preheater	Carbon Steel
Evaporator	1-1/4%Cr-1%Mo
Superheater	Type 304 Stainless Steel
Reheater	Type 304 Stainless Steel

Use of austenitic stainless steel for the superheater and reheater is desirable from oxidation and scaling-resistance standpoints. However, if contaminants (i.e., chloride ions) are inadvertently entrained in the system, the stainless steel is susceptible to stress-corrosion cracking. The entrainment of contaminants can occur during storage periods or at various stages of fabrication, including hydrostatic testing and drying operations, as well as from the use of chloride-containing insulation materials. Of particular concern is the transition zone between the evaporator and superheater sections. This zone is subject to alternate wetting/drying processes and the formation/concentration of boiler water contaminants known to induce stress-corrosion cracking of austenitic stainless steel materials.

4.4.2 Corrosion Allowance

The corrosion data listed in Table 4.10 illustrate metal surface recession and oxide thickness for the ferritic alloys after exposure in steam/water environments for predetermined times and temperatures. (The oxide thickness is approximately 2.2 times metal penetration.)

Table 4.10 Steam/Water Corrosion Data

Unit		Exposure Time		Parameter*	Metal Penetration		Oxide Thickness	
		Years	Hours		μm	mil	μm	mil
Preheater (Carbon steel)								
Tube O.D.:	336°C (636°F) Molten Salt	30	262,800	27.9	<2.5	<0.1	<25.4	<1
Tube I.D.:	238°C (460°F) Water	30	262,800	27.9	<2.5	<0.1	<25.4	<1
Evaporator (1-1/4%Cr-1/2%Mo; carbon steel)								
Tube O.D.:	447°C (836°F) Molten Salt	30	262,800	33	12.4	0.49	25.4 - 50.8	1 - 2
Tube I.D.:	341°C (645°F) Steam/Water	30	262,800	27.9	<2.5	<0.1	<25.4	<1
Superheater/Reheater (2-1/4%Cr-1%Mo):								
Tube O.D.:	566°C (1050°F) Molten Salt	5	43,800	37.2	101.6	4	223.5	8.8
		11.4	100,000	37.8	142.2	5.6	312.4	12.3
		30	262,800	38.4	182.9	7.2	401.3	15.8
Tube I.D.:	538°C (1000°F) Steam	5	43,800	36.0	55.9	2.2	121.9	4.8
		11.4	100,000	36.4	66.0	2.6	144.8	5.7
		30	262,800	37.1	94.0	3.7	205.7	8.1

*Parameter = T(°R)[20 + Log Time (Hours)](10^{-3})

The corrosion allowance recommended by Foster Wheeler for the preheater, evaporator, superheater, and reheater components shown in Table 4.11 reflects minimum and maximum metal surface recession rates based on scale exfoliation processes occurring every 5 years at the higher temperature of 566°C (1050°F) when the scale reaches a thickness of approximately 229 μm (9 mil). The cumulative effect of the higher oxidation rates produced on the outside surfaces of the tubing as a consequence of scale flaking is based on six exfoliation cycles totaling 610 μm (24 mil) during the 30 years of operation at the maximum operating temperature of 566°C (1050°F). Scale exfoliation is not considered a problem at lower tubewall temperatures.

The lower design corrosion allowance of 375 μm (15 mil) recommended for 2-1/4%Cr-1%Mo material for the superheater and reheater components listed in Table 4.11 is based on a consideration that the units will be operated only half the time at the maximum design temperatures.

Foster Wheeler's review of available corrosion data indicates that carbon steel should be a satisfactory material for both the preheater and evaporator units. However, since the evaporator may be subjected to temperature excursions, 1-1/4%Cr-1/2%Mo material is recommended.

Table 4.11 Recommended Corrosion Allowances

					Corrosion Allowances			
					5-Year Exfoliation Periods		Design	
Unit	Tube-side	Fluid	Temperature °C (°F)	Material	μm	mil	μm	mil
Preheater	I.D.	Steam	238 (460)	Carbon Steel	Nil		125	5*
	O.D.	Molten Salt	336 (636)		Nil			
Evaporator	I.D.	Steam	341 (645)	Carbon Steel	Nil		125	5*
	O.D.	Molten Salt	447 (836)		Nil			
			>454 (850)	1-1/4%Cr-1/2%Mo†	Nil		125	5*
Superheater	I.D.	Steam	541 (1005)	2-1/4%Cr-1%Mo	100	4	375	15
	O.D.	Molten Salt	566 (1050)		600	24		
Reheater	I.D.	Steam	541 (1005)	2-1/4%Cr-1%Mo	100	4	375	15
	O.D.	Molten Salt	566 (1050)		600	24		

*Unit projected to operate at maximum design temperature for one-half time.
†Alternative alloy.

SECTION 5

SGS DESIGN

Section 5

SGS DESIGN

5.1 DESIGN ANALYSIS PLAN

Before starting the detailed design and analysis of the SGS and its components, a Design Analysis Plan was prepared for review and approval by SNLL. The plan identifies the various thermal/hydraulic and structural analyses, the sequence in which they are performed, and their interrelationships. Governing design criteria are identified along with computer codes, correlations, fouling factors, etc. Appendix C is the design analysis plan.

5.2 THERMAL/HYDRAULIC DESIGN

The first step in the design of the molten salt SGS heat exchanger components is the thermal/hydraulic design, which consists of basic thermal sizing and an uncertainty analysis. Basic thermal sizing establishes the minimum heat-transfer area required to meet the specified thermal duty based on nominal thermal design parameters (film coefficients, tube thermal conductivity, average tubewall thickness, etc.). However, there are uncertainties associated with each thermal design parameter because of the spread in data upon which they are based. Therefore, to be confident that the selected heat-transfer area will meet the specified thermal duty, we performed an uncertainty analysis to establish surface margins. The end results of the thermal/hydraulic design are the number, size, and length of tubes required for each heat exchanger as well as the tube pitch pattern, baffle geometry, and approximate shell I.D.

The design using the least number of tubes can reduce the cost associated with tube-to-tubesheet welding and tubesheet and tube-support-plate drilling. Therefore, design parameters such as number of tubes; tube size, pitch, and thickness; and number of baffles were varied to obtain technically feasible designs that meet the specified performance with the least number of tubes and at a reasonable length.

The thermal/hydraulic designs of the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS were based on the performance parameters identified in Table 5.1. These are the conditions necessary for meeting the turbine requirements identified in the SGS Requirements and Specification (Appendix B).

5.2.1 Basic Thermal Sizing

Computer Codes. Basic thermal sizing of each heat exchanger was performed using two computer codes. The Heat Transfer Research, Inc. (HTRI) computer code ST-4[23] was used to calculate shell-side heat transfer coefficient. This value was then put into the Foster Wheeler in-house thermal performance computer code MSSG to compute the overall heat transfer coefficient and the required heat transfer area. The HTRI ST-4 computer code uses the stream analysis method to determine the flow distribution over the baffled tube bundle from which the shell-side heat-transfer coefficient can be computed. However, the capacity of the program for computing tube-side heat-transfer coefficients is limited to single-phase flow and its stepwise calculation is limited to a maximum of 10 nodes, which is insufficient for the wide temperature range over which thermal properties of steam/water vary. Consequently, the MSSG code was used to compensate for the limitations of the ST-4 program.

Table 5.1 Design-Point Performance Parameters

Performance Parameters	100-MWe SGS	50-MWe SGS
Temperature, °C (°F)		
Steam/Water:		
Feedwater	237.8 (450)	237.8 (460)
Superheater Inlet	335.6 (636)	335.6 (636)
Final Steam	540.6 (1005)	540.6 (1005)
Reheater Inlet	342.2 (648)	342.2 (648)
Reheater Outlet	540.6 (1005)	540.6 (1005)
Salt:		
Superheater Inlet	562.8 (1045)	562.8 (1045)
Superheater Outlet	425.0 (797)	425.0 (797)
Reheater Inlet	562.8 (1045)	562.8 (1045)
Reheater Outlet	473.3 (884)	473.3 (884)
Evaporator Inlet	446.1 (835)	446.1 (835)
Evaporator Outlet	342.8 (649)	342.8 (649)
Preheater Inlet	342.8 (649)	342.8 (649)
Preheater Outlet	293.3 (560)	293.3 (560)
Flow, kg/s (10^3 lb/h)		
Steam/Water:		
Feedwater	96.6 (766.7)	48.3 (383.3)
Blowdown	0.48 (3.8)	0.24 (1.9)
Main Steam	96.12 (762.9)	48.06 (381.4)
Reheater	83.2 (660.3)	41.6 (330.1)
Salt:		
Preheater	633.52 (5028)	316.8 (2514)
Evaporator	633.52 (5028)	316.8 (2514)
Superheater	357.58 (2838)	178.8 (1419)
Reheater	275.94 (2190)	138.0 (1095)
Pressure, MPa gage (lb/in^2g)		
Steam/Water:		
Drum	13.69 (1485)	13.69 (1985)
Final Steam	13.48 (1955)	13.48 (1455)
Reheater Inlet	3.03 (440)	3.00 (435)
Reheater Outlet	2.86 (415)	2.83 (410)

The MSSG code is a one-dimensional thermal sizing computer code that determines the required heat-transfer tube length by breaking the tube lengthwise into a series of 304-mm (1-ft)-long nodes. Input data required by the program include the following:

- Tube O.D.
- Tube average wall thickness
- Number of tubes
- Tube pitch
- Tube-side fouling resistance
- Shell-side fouling resistance
- Shell-side film coefficient*
- Salt flow rate
- Steam/water flow rate
- Salt inlet temperature
- Steam/water inlet temperature
- Steam/water inlet pressure
- Steam/water outlet temperature

The conservation equations governing energy, mass, and momentum for the series of nodes form a series of 1st-order ordinary differential equations. With inlet and outlet conditions specified, this system of coupled nonlinear equations is integrated using a 4th-order Runge-Kutta scheme. The integration terminates when the computed outlet steam/water outlet temperature reaches the specified value. During the integration local fluid thermal properties, overall heat-transfer coefficient, and logarithmic mean temperature difference (LMTD); tube-side pressure drop; and the threshold for departure from nucleate boiling (DNB) are computed for each node.

*Obtained from HTRI ST-4 computer code for double segmental baffles.

Fouling Resistance. For conservatism, the effect of fouling was considered in the thermal sizing. Fouling resistances used are as follows:

	$m2 \cdot °C/MW$ ($h \cdot ft^2 \cdot °F/Btu$)	
Component	Shell Side	Tube Side
Preheater	90 (5×10^{-4})	21 (1.5×10^{-4})
Evaporator	90 (5×10^{-4})	54 (3×10^{-4})
Superheater	90 (5×10^{-4})	21 (1.5×10^{-4})
Reheater	90 (5×10^{-4})	21 (1.5×10^{-4})

The shell-side salt fouling resistance was obtained from the Standards of the Tubular Exchanger Manufacturers Association (TEMA).[24] The tube-side steam/water fouling resistances were selected based on Foster Wheeler heat exchanger design experience.

Heat-Transfer Correlations. Relative to tubeside steam/water heat transfer, the regions of forced-convection heat transfer in a vertical heated tube can be classified as subcooled liquid, subcooled nucleate boiling, nucleate boiling, DNB, film boiling, and superheated steam. The correlations used in the MSSG program and their associated uncertainties are as follows:

- Subcooled Liquid. For fully developed turbulent flow in smooth tubes, the following relationship is recommended by Dittus and Boelter:[25]

$$Nu = 0.023\ Re^{0.8} Pr^{n} \tag{1}$$

where

Nu = Nusselt number

Re = Reynolds number

Pr = Prandtl number

n = 0.4 for heating

n = 0.3 for cooling.

The physical properties are evaluated at the fluid bulk temperature. This empirical correlation represents the test data within ±25% (σ = standard deviation = 8.3%).[26]

- Subcooled Nucleate Boiling. In the subcooled nucleate boiling region, the wall temperature remains essentially constant at a few degrees above the saturation temperature, while the mean bulk fluid temperature is below the saturation temperature, corresponding to the fluid pressure. For fully developed subcooled boiling, Thom[27] suggested the following correlation for water:

$$\Delta t_{sat} = 0.072Q^{0.5} \Big/ \left(e^{P} \; 1260 \right) \tag{2}$$

where

t_{sat} = Amount by which the wall temperature exceeds the saturation temperature

Q = Heat flux

P = Pressure.

The equation correlated almost 90 percent of the points within ±15 percent (σ = 9.1 percent).

- Nucleate Boiling. Chen's correlation[28] was adopted for the pre-CHF/DNB bulk boiling heat transfer. The correlation covers both the saturated nucleate boiling region and the two-phase forced convective region. Both the macroconvective heat-transfer mechanism associated with ordinary forced convection and the microconvective mechanism attributed to bubble nucleation were assumed to occur to some degree in the heat-transfer process for the boiling of saturated liquid. The contributions made by the two mechanisms are additive:

$$h = h_{mac} + h_{mic} \tag{3}$$

The convective contribution h_{mac} is given by the modified Dittus-Boelter equation as:

$$h_{mac} = \left[F(0.023) \right] \left(Re_L^{0.8} \right) \left(Pr_L^{0.4} \right) \left(k_L/D \right)$$

where

k_L = Liquid thermal conductivity

D = Tube I.D.

The parameter F is a function of the Martinelli parameter X_{tt} as shown in Figure 5.1.[29] The thermal properties are evaluated for liquid.

The contribution by bubble nucleation, h_{mic}, is given by:

$$h_{mic} = S(0.0012) \frac{k_L^{0.79} c_{pL}^{0.45} \rho_L^{0.49} g_c^{0.25}}{\sigma^{0.5} \mu_L^{0.29} h_{fg}^{0.24} \rho_g^{0.24}} \left(\Delta T_{sat}^{0.24} \Delta P_{sat}^{0.75} \right) \qquad (4)$$

where

c_{pL} = Liquid specific heat

ρ_L = Liquid density

g_c = Gravitational constant

σ = Vapor-liquid surface tension

μ_L = Liquid viscosity

h_{fg} = Latent heat

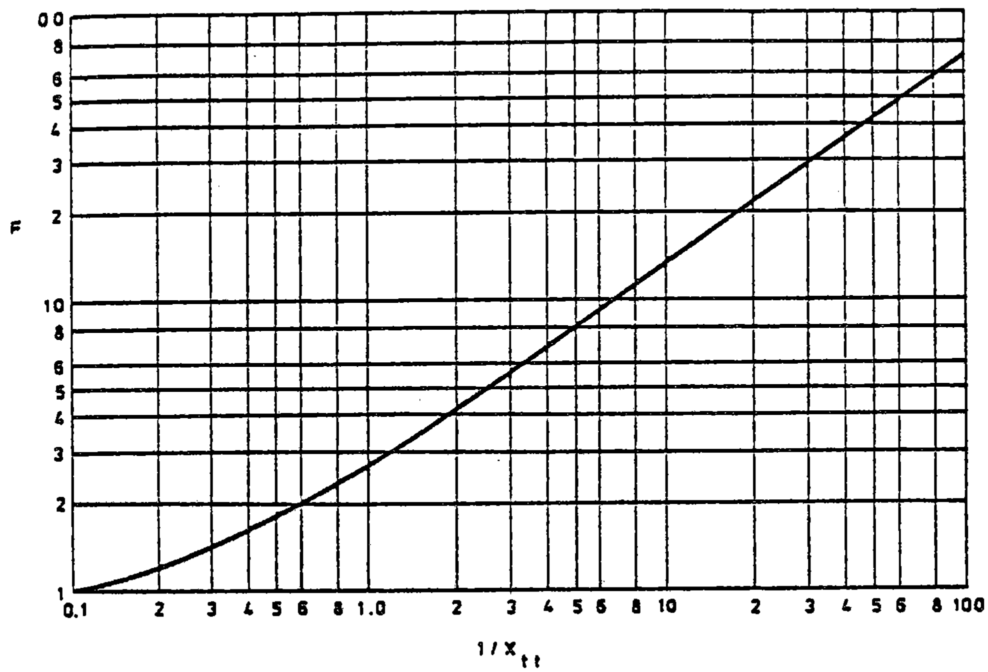
ρ_g = Vapor density

ΔP_{sat} = Difference in vapor pressure corresponding to ΔT_{sat}.

The parameter S is a suppression factor and is represented as a function of the local two-phase Reynolds number $\left(Re_L \times F^{1.25} \right)$ in Figure 5.1.

Chen compared his correlation with other experimental data. The average deviation for Dengler and Addoms's[30] water data was 14.7 percent; for Schrock and Grossman's water data,[31] 15.1 percent. Therefore, for this study a standard deviation (σ) of 15 percent was used.

- Departure from Nucleate Boiling. DNB is associated with the transition from the nucleate boiling regime to the film boiling regime. Nucleate boiling is characterized by a high heat-transfer coefficient; film boiling exhibits a low heat-transfer coefficient. The location in the tube at which the DNB occurs can fluctuate with time and can cause temperature cycling of the tubewall. The fluctuation in tubewall I.D. temperature is of principal concern in the evaporator. If the amplitude of the temperature oscillation is sufficiently large, prolonged temperature cycling can cause tubewall fatigue.



$$X_{tt} = \left(\frac{\mu L}{\mu g}\right)^{0.1} \left(\frac{\rho g}{\rho L}\right)^{0.5} \left(\frac{1-x}{x}\right)^{0.9}$$

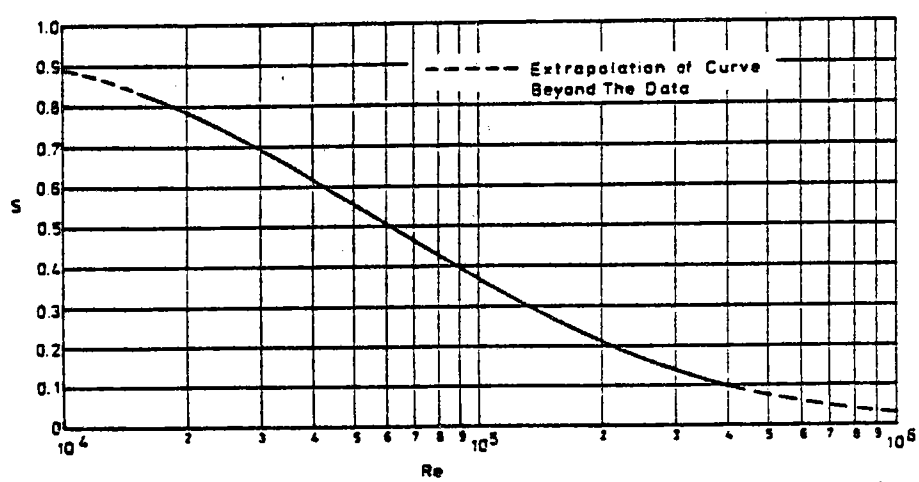


Figure 5.1 Convective Boiling Factor F and Suppression Factor S

Experiments were performed by Westinghouse and Atomics International on the evaluation of DNB for nuclear heat exchangers at high steam/water pressures. The correlation developed by Westinghouse[32] is given by:

$$X_{DNB} = 415.3/h_{fg}\left(G/10^6\right)^{0.445}\left[\left(P/1000\right)^{-1.05} - 1.17D^{1.2}\left(Q/10^6\right)\right] \qquad (5)$$

where

X_{DNB} = Steam quality at DNB.

The correlations developed by Atomics International[33] are:

$$X_{DNB} = 18.85/h_{fg}\left(\rho_v/\rho_L\right)\sqrt{G/10^6} \quad \text{for } Q > 0.2 \times 10^6 \text{ Btu/h}\cdot\text{ft}^2 \qquad (6)$$

$$X_{DNB} = 18.85\left[0.2/\left(Q/10^6\right)^{1.5}\right]\Big/h_{fg}\left(\rho_v/\rho_L\right)\sqrt{G/10^6} \quad \text{for } Q < 0.2 \times 10^6 \text{ Btu/h}\cdot\text{ft}^2 \qquad (7)$$

where

G = Mass velocity

Q = Heat flux.

Both the Westinghouse and Atomics International correlations were implemented in the MSSG computer code, providing users the option of using either one of the correlations.

- Film Boiling. The heat-transfer mechanism of a vapor-liquid mixture in which the critical heat flux has been exceeded can be classified as film boiling. The heat-transfer correlation suggested by Bishop, et al.[34] is given by:

$$Nu_f = 0.0193\, Re_f^{0.80}\, Pr_f^{1.23}\left(\rho_v/\rho_b\right)^{0.68}\left(\rho_v/\rho_L\right)^{0.068} \qquad (8)$$

where

Nu_f = Nusselt number at film temperature

$\rho_b = \rho_v\alpha + \rho_L(1 - \alpha)$

α = Void fraction.

The fluid properties are evaluated at the average film temperature. The correlation has a 2σ probability limit of ±19 percent (σ = 9.5 percent).

- Superheated Steam. At high steam pressure, the heat-transfer correlation suggested by Bishop[35] is given by:

$$Nu_f = 0.0073\ Re_f^{0.886}\ Pr_f^{\ 0.61} \tag{9}$$

The thermal properties of steam are evaluated at the film temperature. The correlation has a standard deviation of 5.9 percent.

- Pressure Drop Correlations. For tube-side pressure drop, the total pressure drop can be divided into three components: friction, gravitation, and acceleration. For single-phase flow (subcooled water or superheated steam), the friction factor is determined according to the conventional Moody diagram using empirical correlations. The friction factor for laminar flow is:

$$f = 64/R_e \tag{10}$$

where

f = Friction factor

R_e = Reynolds number.

For turbulent flow in rough pipes, Colebrook[36] developed the following empirical equation from experimental data:

$$1/\sqrt{f} = 1.74 - 2.0\ \log_{10}\left[1 + 18.7\ \left(r_o/e\right)/\sqrt{f}\ \left(R_e\right)\right] \tag{11}$$

where

r_o = Pipe inside radius

e = Pipe roughness.

These friction factors are for isothermal conditions. Since the steam/water in the heat exchangers is heated, the friction factors were modified as suggested by Mendler, et al.,[37] to take into account the temperature difference between the fluid and the tubewall.

For regions of local and bulk boiling, pressure drop was computed by the Martinelli-Nelson method[38] modified by Mendler, et al.,[37] to account for the effect of mass velocity. Because the correlations are quite extensive, they are not presented here.

- Results of Basic Thermal Sizing. The results of basic thermal sizing for the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS are identified in Tables 5.2 and 5.3. Figures 5.2 through 5.5 illustrate the salt and steam/water temperature profiles through the heat exchangers for both subsystems as computed by the MSSG computer code.

5.2.2 Uncertainty Analysis

With the basic size of each heat exchanger determined from the aforementioned thermal/hydraulic correlations, appropriate surface margins were added. The surface margins account for uncertainties associated with each thermal design parameter because of the spread in data from which they are based.

The statistical Root of Sum Square (RSS) method was used to determine appropriate surface margins to achieve a 90-percent confidence level in the design. A 90-percent confidence level corresponds to a thermal design parameter variation of 1.282 standard deviations (σ). The RSS method determines surface margin by increasing the basic tube length while maintaining the same number of tubes. The additional tube length is computed from the following equation:

$$\Delta L = \sqrt{(\Delta \ell_i)^2 + (\Delta \ell_o)^2 + (\Delta \ell_k)^2 + (\Delta \ell_t)^2} \tag{12}$$

where

ΔL = Total additional tube length of the unit

$\Delta \ell_i$ = Tube length increment resulting from uncertainty of tube-side heat-transfer coefficient

$\Delta \ell_o$ = Tube length increment resulting from certainty of shell-side heat-transfer coefficient

$\Delta \ell_k$ = Tube length increment resulting from uncertainty of tube thermal conductivity

$\Delta \ell_t$ = Tube length increment resulting from uncertainty of tube thickness.

Each individual Δ_ℓ is based on the same 90-percent confidence level.

Table 5.2 Thermal Sizing Results--100-MWe Solar Stand-Alone SGS

Description	Preheater	Evaporator	Superheater	Reheater
Tube O.D., mm (in.)	15.9 (0.625)	25.4 (1.0)	15.9 (0.625)	25.4 (1.0)
Tubewall Thickness, mm (in.) -0%/+20%	1.47 (0.058)	2.11 (0.083)	1.65 (0.065)	1.24 (0.049)
Tube Material	SA-210, Gd. C	SA-213-T11	304SS	304SS
Basic Sizing				
Number of Tubes	2250	1320	1010	440
Pitch/Tube O.D.	1.50	1.40	1.60	1.40
Basic Tube Length, m (ft)	15.18 (49.80)	15.42 (50.58)	16.53 (54.24)	17.03 (55.88)
Design Tube Length, m (ft)*				
Basic Tube Length	15.18 (49.80)	15.42 (50.58)	16.53 (54.24)	17.03 (55.88)
Uncertainties, 90% Confidence Level	17.00 (55.77)	17.66 (57.93)	17.83 (58.49)	18.26 (59.91)
Inactive Region of Support Plates	17.21 (56.47)	17.87 (58.63)	18.05 (59.23)	18.50 (60.68)
Partially Inactive Regions in Inlet and Exit Bundle Area	17.47 (57.30)	18.14 (59.50)	18.26 (59.92)	18.70 (61.36)
Number of Tube Support Plates†	45	45	57	59
Design Margin, %	15.06	17.64	10.47	9.81
Tube Plugging Allowance, Tubes	68	40	30	14
Total Number of Tubes	2318	1360	1040	454

*Length shown is accumulated length resulting from all preceding effects.
†Double segmental type.

Table 5.3 Thermal Sizing Results--50-MWe Hybrid SGS

Description	Preheater	Evaporator	Superheater	Reheater
Tube O.D., mm (in.)	15.9 (0.625)	25.4 (1.0)	15.9 (0.625)	25.4 (1.0)
Tubewall Thickness, mm (in.) -0%/+20%	1.47 (0.058)	2.11 (0.083)	1.65 (0.065)	1.24 (0.049)
Tube Material	SA-210, Gd. C	SA-213-T11	304SS	304SS
Basic Sizing				
Number of Tubes	960	600	505	220
Pitch/Tube O.D.	1.40	1.40	1.40	1.40
Basic Tube Length, m (ft)	14.81 (48.60)	15.62 (51.25)	15.42 (50.36)	16.96 (55.65)
Design Tube Length, m (ft)*				
Basic Tube Length	14.81 (48.60)	15.62 (51.25)	15.42 (50.36)	16.96 (55.65)
Uncertainties, 90% Confidence Level	16.33 (53.58)	17.77 (58.29)	16.38 (53.74)	18.18 (59.64)
Inactive Region of Support Plates	16.50 (54.14)	17.98 (58.99)	16.55 (54.29)	18.36 (60.25)
Partially Inactive Regions in Inlet and Exit Bundle Area	16.76 (55.0)	18.25 (59.86)	16.75 (54.97)	18.57 (60.93)
Number of Tube Support Plates†	43	45	53	59
Design Margin, %	13.11	16.8	9.15	9.49
Tube Plugging Allowance, Tubes	29	18	15	7
Total Number of Tubes	989	618	520	227

*Length shown is accumulated length resulting from all preceding effects.
†Double segmental type.

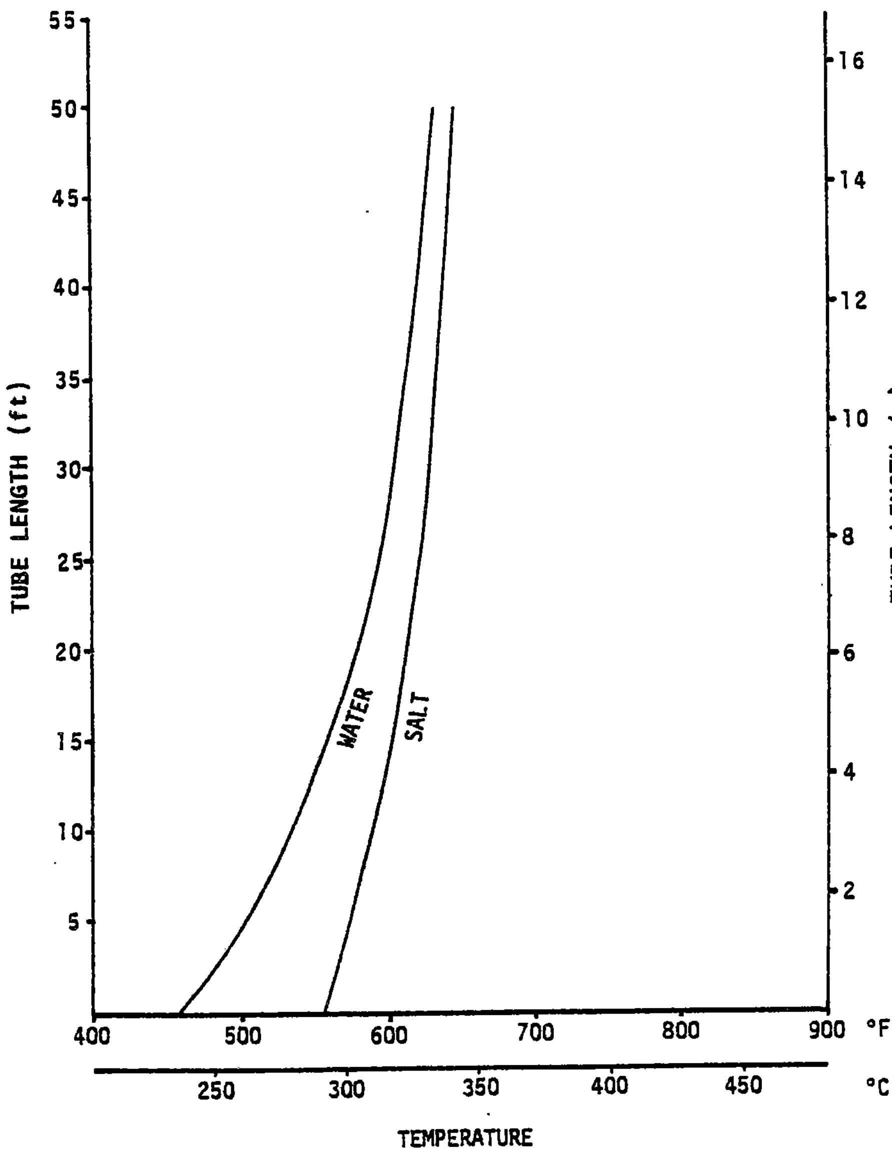


Figure 5.2 Preheater Temperature Profile--100-MWe Solar Stand-Alone SGS

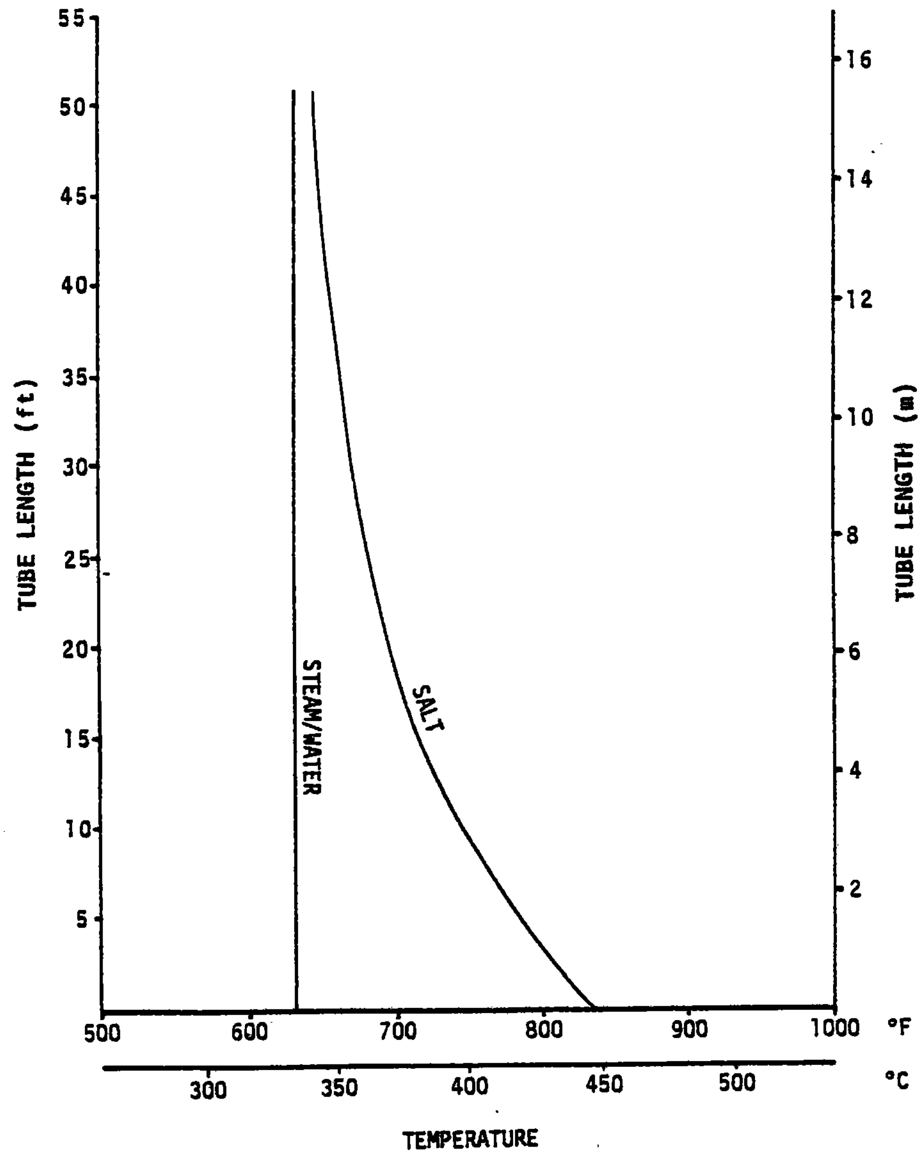


Figure 5.3 Evaporator Temperature Profile--100-MWe Solar Stand-Alone SGS

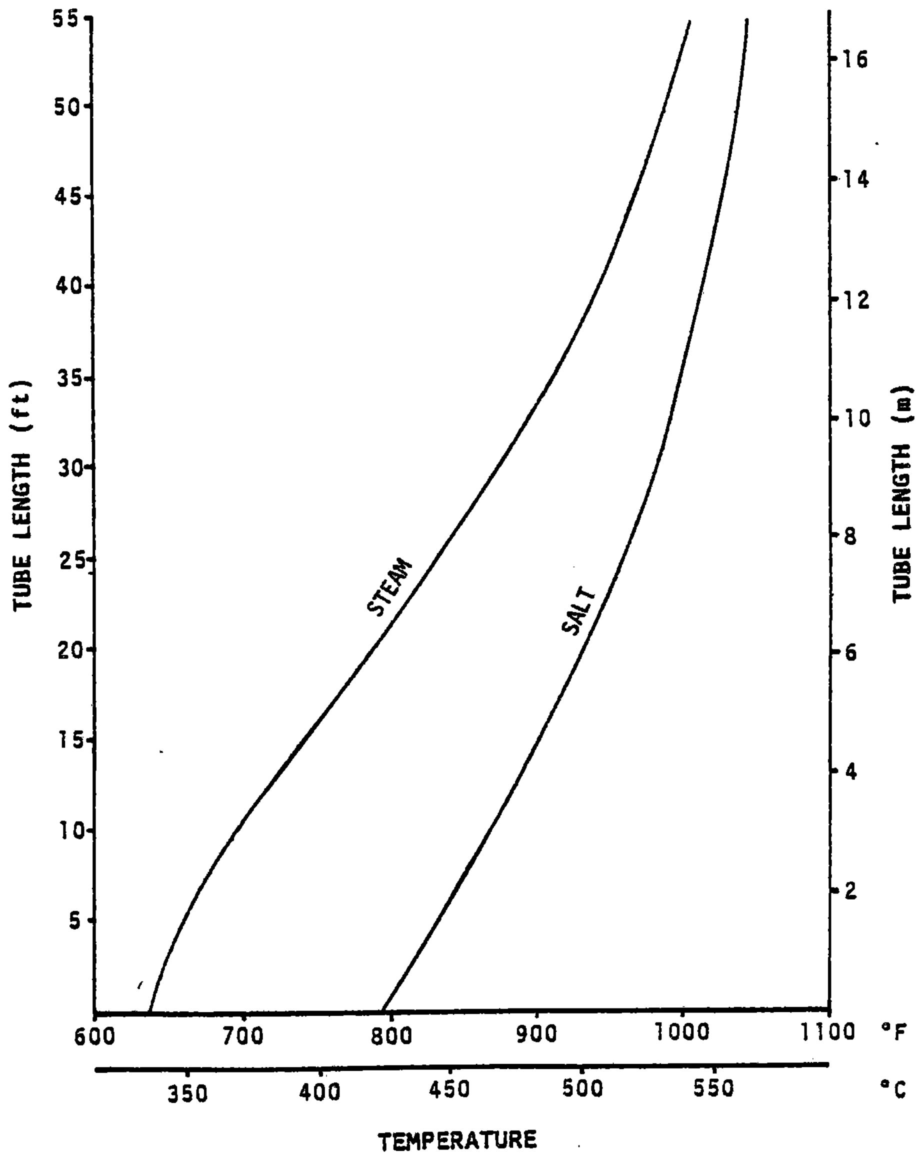


Figure 5.4 Superheater Temperature Profile--100-MWe Solar Stand-Alone SGS

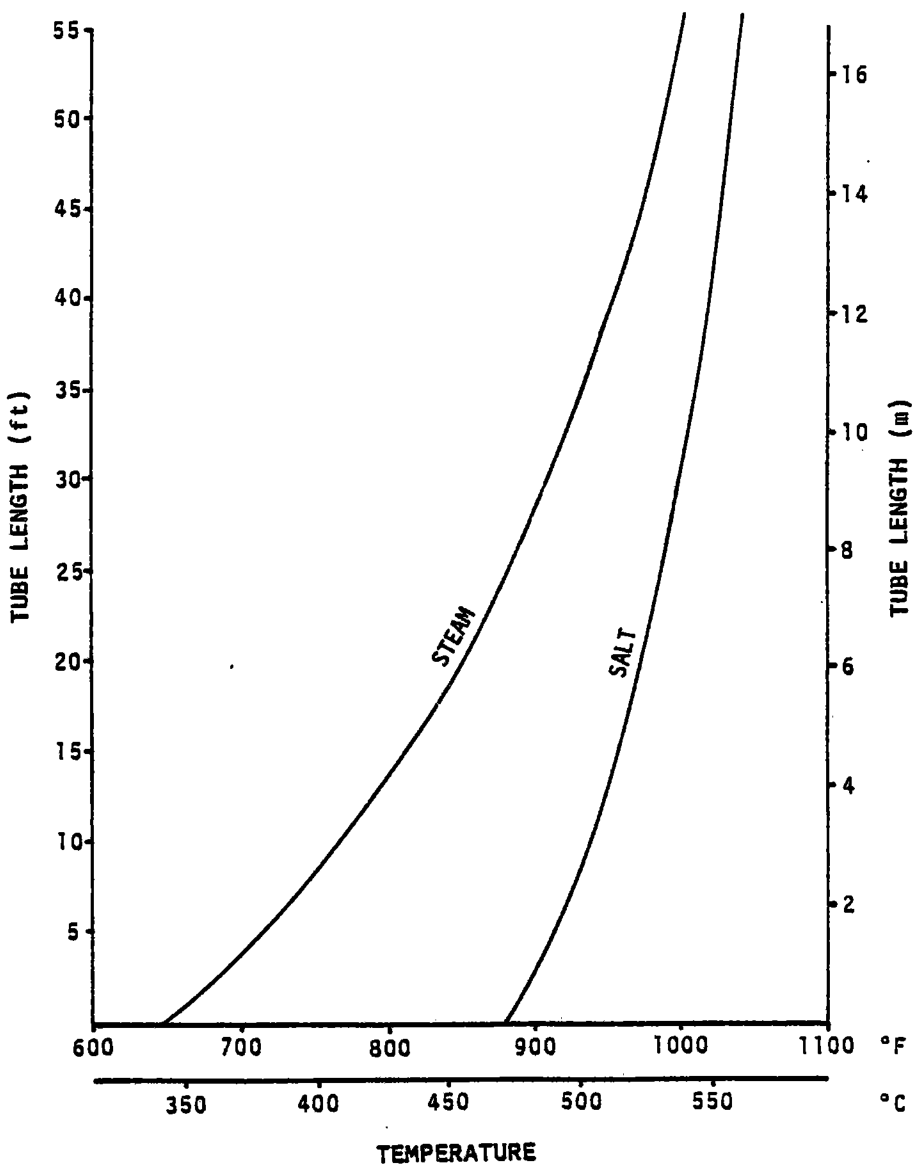


Figure 5.5 Reheater Temperature Profile--100-MWe Solar Stand-Alone SGS

Thermal design parametric variations used in the uncertainty analysis are as follows:

- Tube-Side Heat-Transfer Coefficient. The discussion on heat-transfer correlations identified the following variations in data for the regions of forced convection of steam/water flowing inside a tube:

Region	Standard Deviation (%)
Subcooled liquid	8.3
Subcooled nucleate boiling	9.1
Nucleate boiling	15.0
Film boiling	9.5
Superheated steam	5.9

- Shell-Side Heat-Transfer Coefficient. Approximately 80 percent of the data from which the HTRI ST-4 correlations for double segmental baffles are based are within ±20 percent of the predicted value. Approximately 90 percent of the data are predicted within ±25 percent.[23] A standard deviation of 16.5 percent was used.

- Tube Material Thermal Conductivity. For Type 304 stainless steel, the thermal conductivity uncertainty is approximately ±5 percent.[37] A standard deviation of 1.67 percent was used. The standard deviation for carbon steel is 0.48 W/m•°C (0.28 Btu/h•ft) over the expected operating temperature range.[37] The thermal conductivity variation for SA 213-T11 (1-1/4%Cr-1/2%Mo) was not directly available. The thermal conductivities reported from the ASME Boiler and Pressure Vessel Code, Section VIII-Division 2[39] were used in the MSSG computer code. However, higher values were reported by Boure, Bergles, and Tong.[40] To estimate the thermal conductivity uncertainty for SA-213-T11, the Code was assumed to be the lower limit and the values from Boure were assumed to be the mean value.

- Tubewall Thickness. The minimum tubewall thickness is specified with a 20-percent tolerance. The mean tubewall thickness was used for basic thermal sizing. A standard deviation of 3.03 percent was used for the uncertainty analysis.

In addition to the aforementioned uncertainties associated with thermal design parameters, the effects of tube support plates and the partially inactive regions of the shell-side inlet/exit bundle areas were individually assessed. Tube length was increased to account for these effects. As a final margin, the number of tubes in each unit was increased by 3 percent to account for the possibility of tube plugging.

The results of the uncertainty analysis for the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS are identified in Tables 5.2 and 5.3.

5.3 MECHANICAL DESIGN

As a result of the thermal/hydraulic design, the number, size, and length of tubes required for each heat exchanger, as well as the tube pitch pattern and baffle geometry, were determined. The next step in the design of the molten salt SGS heat exchanger components was mechanical design. Mechanical design involves sizing of all pressure parts such as shells, steam and shell heads, nozzles, tubes, reinforcements, etc., so that they will withstand the environment to which they will be exposed and so that they can be fabricated with the least amount of difficulty.

All heat exchangers were designed in accordance with the requirements of the ASME Boiler and Pressure Vessel Code, Section VIII-Division 1,[41] and the Standards of the Tubular Exchanger Manufacturers Association (TEMA).[24] The exchangers were also designed for a 0.25-g seismic load vertically and horizontally (applied simultaneously) while pressurized to operating pressure at the full-load operating temperature. The design pressures and temperatures for both the 100-MWe solar stand-alone and the 50-MWe hybrid heat exchangers are identical and are identified for each unit on the drawings in subsequent sections.

Tubesheet hole pattern and shell I.D. were determined using Foster Wheeler's computer code TUBELAY. TUBELAY determines the radius of the enclosing circle for any given number of tubes arranged in a triangular (equilateral) or square pitch pattern. It provides a row-by-row tube count and calculates the maximum radius to the center of the outermost tube in each row. An approximate plot of a typical quadrant of the tube pattern is provided to assist in the selection of the formal design configuration. The program assumes complete filling of the tube pattern, which originates from a central tube and extends outward until the input requirement for the total number of tubes has been filled. If necessary to fill the specified pattern, a few tubes were added, increasing the number of tubes beyond that specified by the thermal/hydraulic analysis.

5.3.1 100-MWe Solar Stand-Alone Heat Exchangers

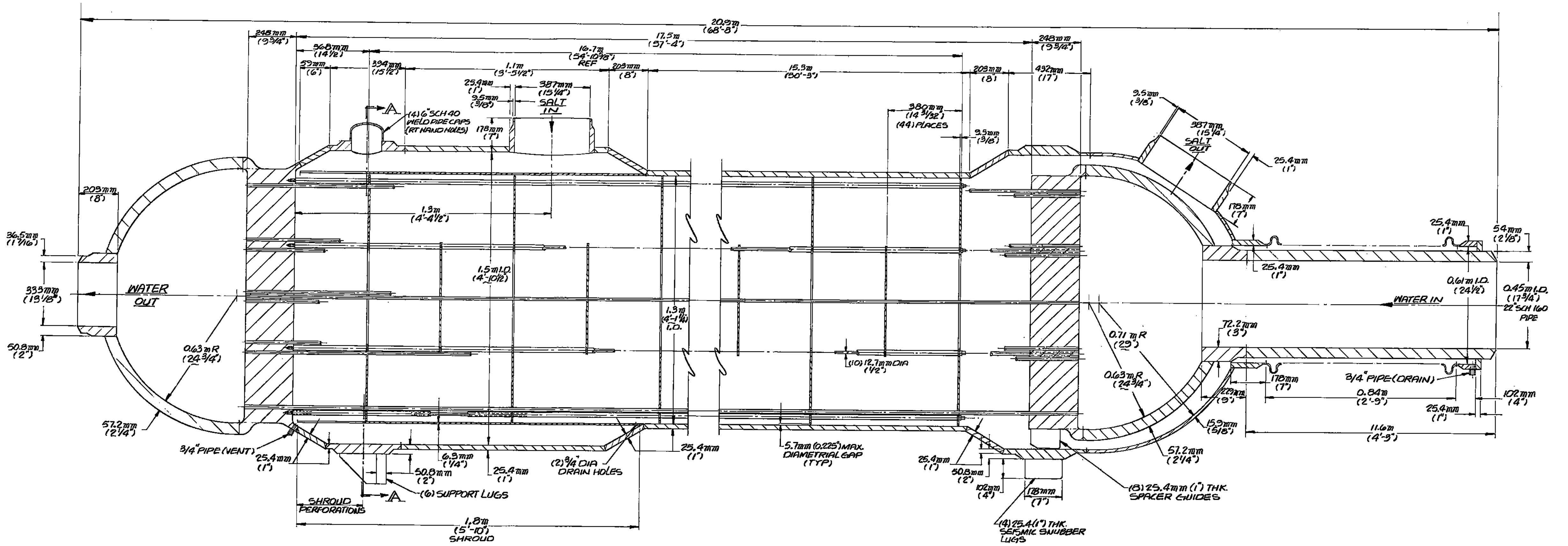
The 100-MWe solar stand-alone preheater, evaporator/drum, superheater, and reheater are illustrated in Figures 5.6, 5.7, 5.8, and 5.9. Features of interest include the following:

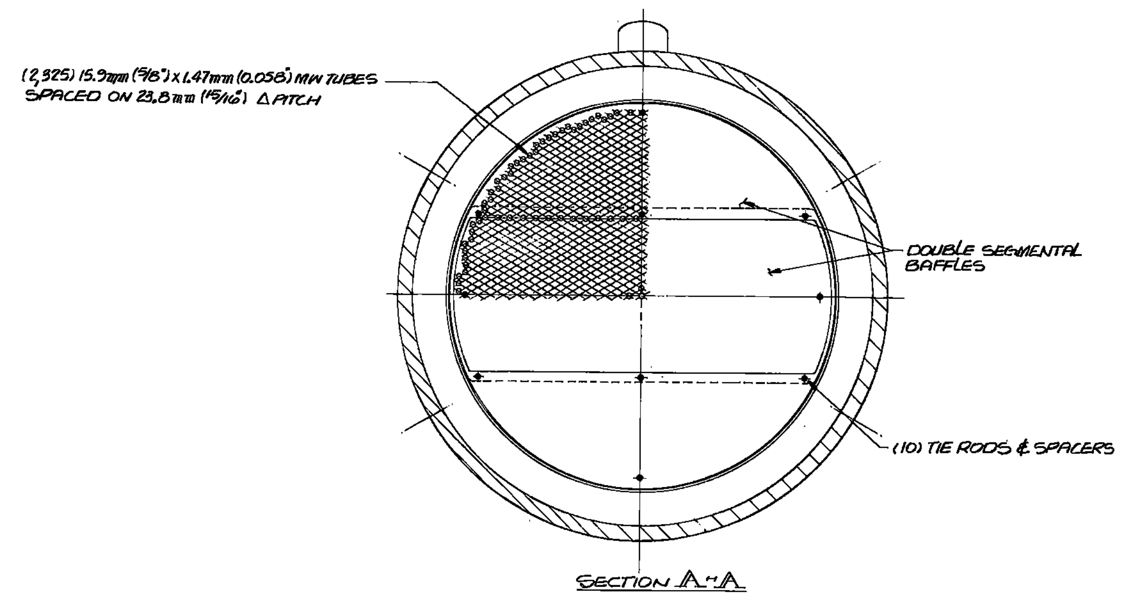
Superheater and Reheater.

(1) These units are fabricated from Type 304 stainless steel. For a 566°C (1050°F) design temperature, the ASME Code specifies two different values for allowable stress. The higher value [Sa = 84.1 MPa (12.2 x 10^3 lb/in^2)] was used in all calculations. Use of the lower value [Sa = 65.5 MPa (9.5 x 10^3 lb/in^2)] is required where slight distortions could cause leaks in such areas as gasketed joints. These vessels were designed with welded connections so that slight distortions would not be detrimental.

(2) The steam inlet nozzles were sized to permit access to the tubesheet for tube plugging. The connecting pipe must be removed to gain access to the tubesheet. Several alternative types of maintenance openings were considered—including a standard bolted-weld neck flange, a diaphragm seal, and a standard hand hole with a welded cap. They were not considered practical for this application. Our conclusion is based on Foster Wheeler's past experience with them for the high operating temperatures involved and the limited area available on the head for placing such an opening.

(3) The steam outlet nozzle sizes were determined by the overall unit pressure-drop requirement. As in the steam inlet nozzles, the connecting pipe must be cut and removed to gain access to the tubesheet for tube plugging. Tube plugging, for both the upper and lower tubesheets, would be done by explosively expanding plugs in the tubes. (The limited accessibility to the tubesheet precludes use of welded plugs.)





NOTES:

1- VESSEL DESIGNED PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION, W/1980 WINTER ADDENDUM

2- DESIGN TEMP:
TUBE SIDE – 371°C (700°F)
SHELL SIDE – 371°C (700°F)
DESIGN PRESSURE:
TUBE SIDE – 15,512 KPAG (2250 PSIG)
SHELL SIDE – 2,068 KPAG (300 PSIG)

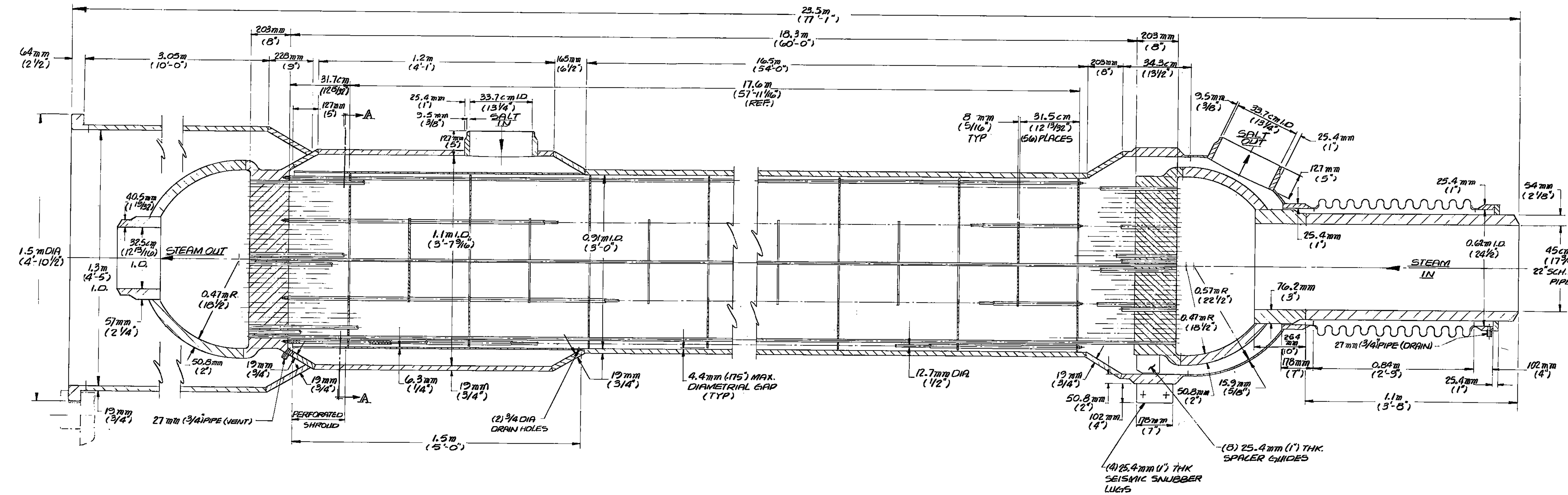
3- MATERIAL SPECS: CARBON ST'L

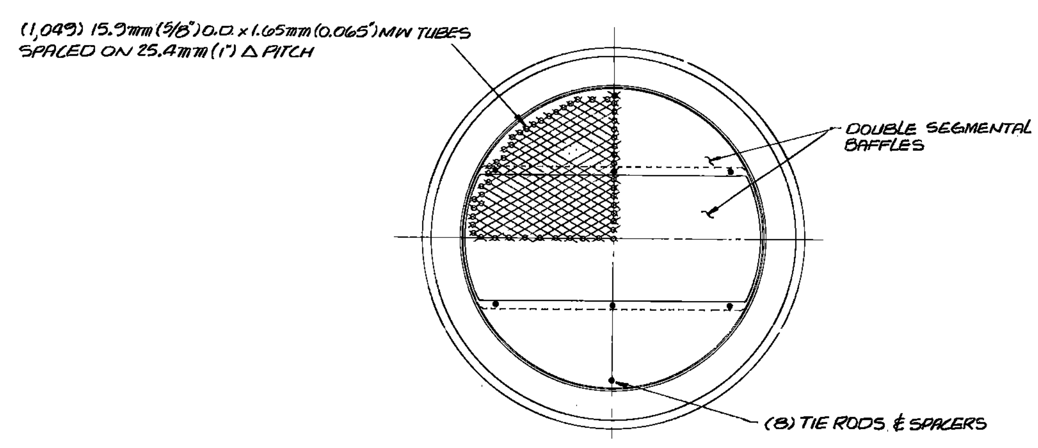
MAJOR PARTS	ASME SPEC.
FORGINGS	SA-266 GR4
HEADS	SA-515 GR 65
PLATES	SA-515 GR 65
NOZZLES	SA-266 GR4
TIE RODS	SA-193 GR B7
TIE ROD NUTS	SA-194 GR 7
TIE ROD SPACERS	SA-106 GR B
TUBING	SA-556 GR B2
BELLOWS	SA-B76 TY 304 SS

4- WEIGHTS:
DRY (NO INSUL.) – 53,978 KG (119,000 LBS) APPROX.
OPERATING WEIGHT, NO INSUL. & WITH MOLTEN SALT @ 248°C (480°F) & WATER @ 492°C (550°F) – 84,823 KG (187,000 LBS) APPROX.

Figure 5.6 Preheater--100-MWe Solar Stand-Alone SGS

Figure 5.7 Evaporator--100-MWe Solar Stand-Alone SGS





SECTION A-A

NOTES:

1- VESSEL DESIGNED PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION, W/1980 WINTER ADDENDUM

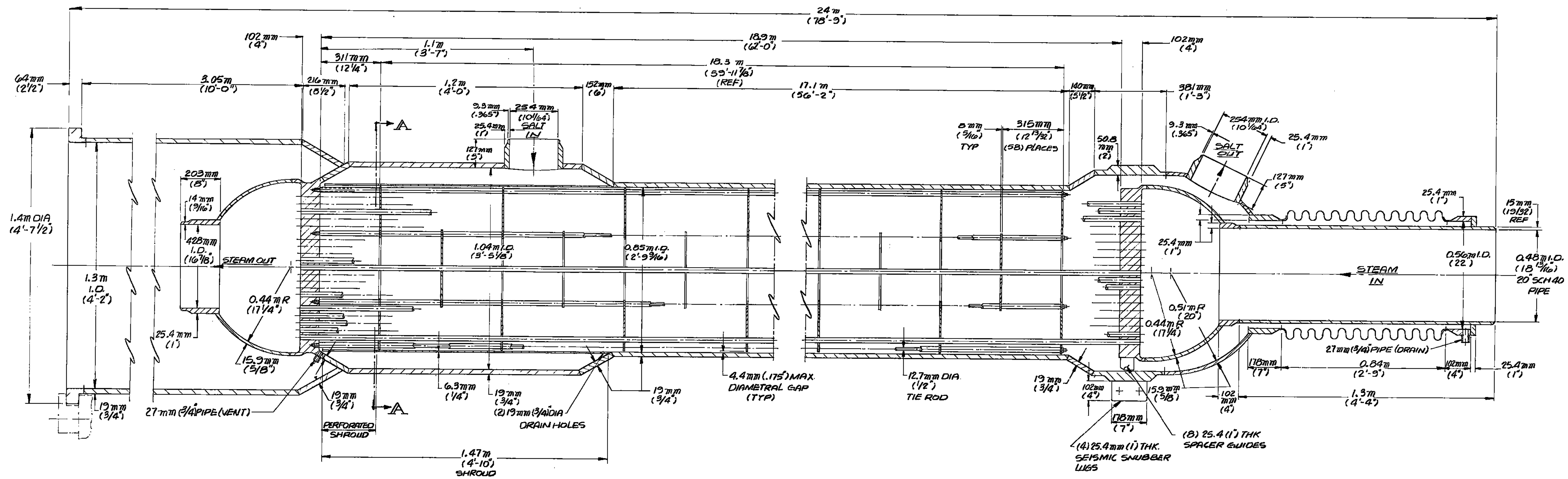
2- DESIGN TEMP:
   TUBE SIDE - 565.6°C (1050°F)
   SHELL SIDE - 565.6°C (1050°F)
   DESIGN PRESSURE:
   TUBE SIDE - 15,340 KPAG (2225 PSIG)
   SHELL SIDE - 2,068 KPAG (300 PSIG)

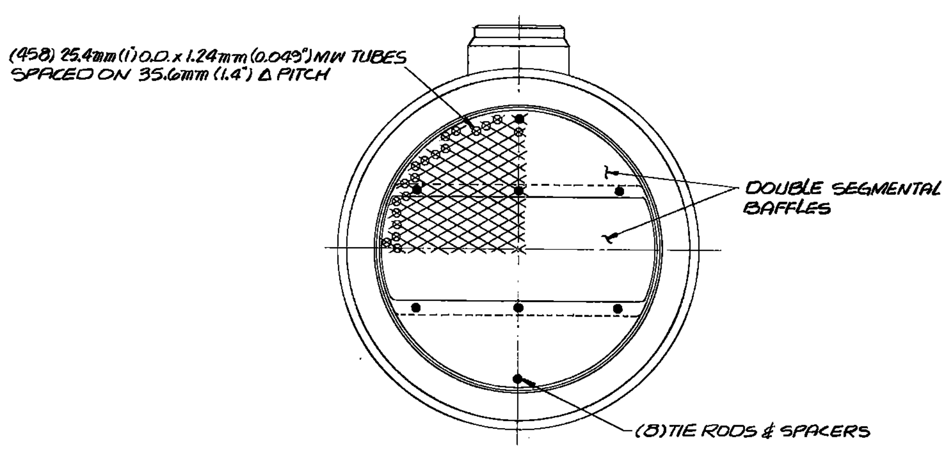
3- MATERIAL SPECS: TYPE 304 SS

MAJOR PARTS	ASME SPEC.
FORGINGS	SA - 182
HEADS	SA - 240
PLATES	SA - 240
NOZZLES	SA - 182
TIE RODS	SA - 193
TIE. ROD NUTS	SA - 194
TUBING	SA - 213
TIE ROD SPACERS	SA - 376
BELLOWS	SA - 376

4- WEIGHTS:
   DRY (NO INSUL.) - 28,486 KG (62,800 LBS)
   OPERATING WEIGHT, NO INSUL. & WITH MOLTEN SALT @ 248°C (480°F) - 44,895 KG (98,975 LBS)

Figure 5.8 Superheater--100-MWe Solar Stand-Alone SGS





SECTION A-A

NOTES:

1- VESSEL DESIGNED AS PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION, W/1980 WINTER ADDENDUM

2- DESIGN TEMP:
   TUBE SIDE – 565.6°C (1050°F)
   SHELL SIDE – 565.6°C (1050°F)
   DESIGN PRESSURE:
   TUBE SIDE – 3,964 KPAG (575 PSIG)
   SHELL SIDE – 2,068 KPAG (300 PSIG)

3- MATERIAL SPECS: TYPE 304 SS

MAJOR PARTS	ASME SPEC
FORGINGS	SA - 182
HEADS	SA - 240
PLATES	SA - 240
TUBING	SA - 213
NOZZLES	SA - 182
TIE RODS	SA - 193
TIE ROD NUTS	SA - 194
TIE ROD SPACERS	SA - 376
BELLOWS	SA - 376

4- WEIGHTS:
   DRY (NO INSUL.) – 20,865 KG (46,000 LBS) APPROX.
   OPERATING WEIGHT, NO INSUL. & WITH MOLTEN SALT @ 248°C (480°F) – 33,113 KG (73,000 LBS) APPROX.

Figure 5.9 Reheater--100-MWe Solar Stand-Alone SGS

(4) The hanging support cylinder provides a support attachment point that has a much lower temperature than that of the heat exchanger. This temperature differential permits use of a carbon steel support structure and also reduces the differential thermal expansion between the vessel and support.

(5) The outer support plates have an integral ring that connects each set of outer plates. This ring aids in positioning the plates during fabrication.

(6) The expansion bellows permits differential expansion between the tubes and the shell. The bellows is located in the low-pressure salt side of the heat exchanger.

(7) The tubesheets are contoured at the perimeter to minimize differential thermal expansion in the tubesheet.

(8) The seismic snubber lugs at the lower tubesheet will be connected to hydraulic or mechanical snubbers. The inner spacer lugs prevent relative lateral movement between the tubesheet and the outer shell. The snubbers prevent lateral movement from seismic loading while permitting thermal expansion of the unit.

(9) All nozzle connections are standard pipe sizes.

(10) Vent and drain nozzles permit complete venting or draining of each heat exchanger.

(11) The heat exchangers, designed to operate vertically, will be fabricated in a horizontal position.

(12) The heat exchangers will be shipped, completely assembled, on a single railroad car using support at two points that will take lateral and vertical loads. One of the supports will also take axial loads.

(13) At the construction site, the unit will be turned vertically with a specially designed lifting ring bolted to the hanging support flange. While the heat exchanger is being upended by a second crane, the lower end will be supported at one of the snubber lugs.

(14) The nozzles will be sealed with a welded closure for pressure testing. For this purpose the nozzles will be fabricated an extra 25 mm (1 in.) long. Following testing, the welded closure will be removed and the nozzle will be final machined to the proper contour.

(3) The steam outlet nozzle, water and chemical feed nozzles, downcomers, and recirculation pump feed line sizes were determined by the pressure-drop requirements of the unit.

5.3.2 50-MWe Hybrid Heat Exchangers

The 50-MWe hybrid preheater, evaporator/drum, superheater, and reheater are illustrated in Figures 5.10, 5.11, 5.12, and 5.13. The mechanical designs of these units are very similar to the mechanical designs of the 100-MWe solar stand-alone units. Differences are as follows:

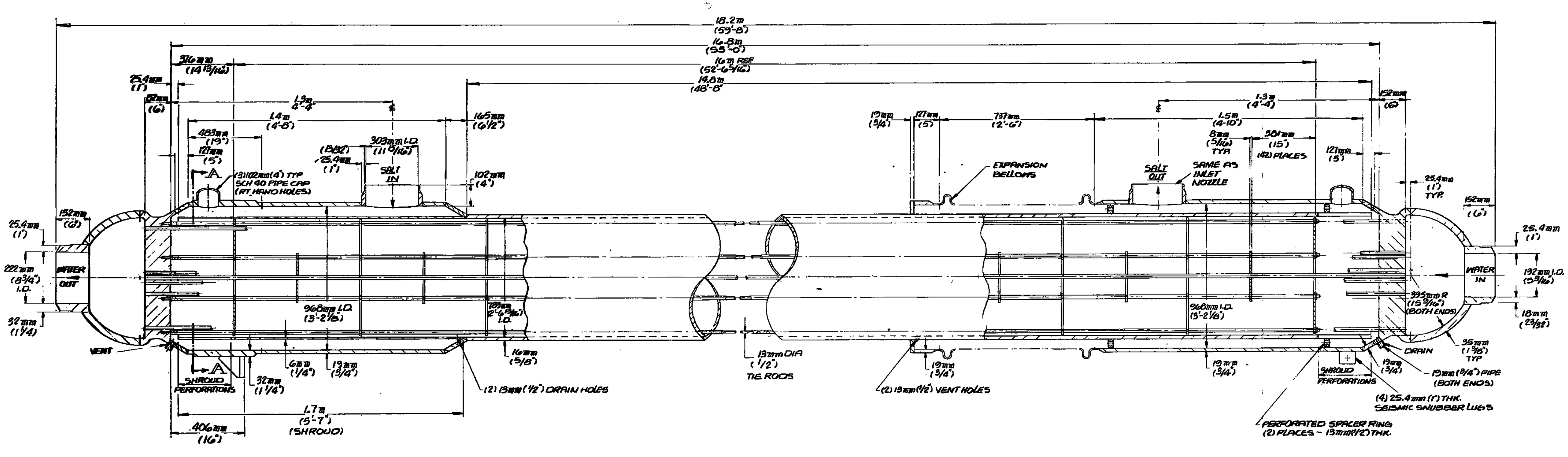
Superheater, Reheater, and Preheater.

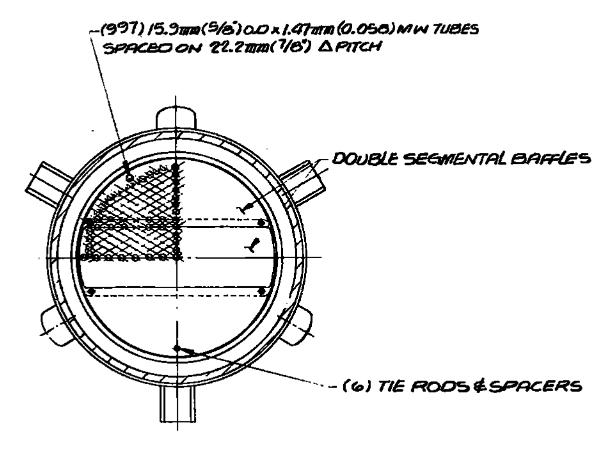
(1) These heat exchangers have the expansion bellows in the shell. The bellows in the 100-MWe units was located at the bottom of the vessel. Bellows located in the shell is preferable for fabrication; however, the 100-MWe shell sizes would have necessitated a much larger bellows. The hydrostatic end load (which is supported entirely by the tubes) would, therefore, be much greater for the 100-MWe units. The bellows for both the 100-MWe and 50-MWe units are removable without dismantling the tube bundle.

(2) All nozzles were sized to satisfy the overall unit pressure-drop requirements. Tubes can be plugged by reaching through the nozzles; the heads are not large enough for a man to enter them.

(3) The two spacer rings at the bellows end of the shell prevent relative lateral movement between the lower tubesheet and the shell. These rings also aid in stiffening the inner and outer shells.

Evaporator. The manway opening, which will be used to gain access to the upper tubesheet, is located on the cylindrical part of the steam drum. The spherical part of the drum is not large enough to accommodate the 406-mm (16-in.) opening. Because of this change in location, water and chemical feed pipes sections must be removed to gain access to the internal cover. Because the pipe sections that must be removed are internal, a slight leak in the connections will not be a problem; therefore, gasketed flanges are not required.





SECTION A-A

NOTES:

1- VESSEL DESIGNED AS PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION W/1980 WINTER ADDENDUM

2- DESIGN TEMP:
TUBE SIDE - 371°C (700°F)
SHELL SIDE - 371°C (700°F)

DESIGN PRESSURE:
TUBE SIDE - 15,512 KPAG (2,250 PSIG)
SHELL SIDE - 2,068 KPAG (300 PSIG)

3- MATERIAL SPEC. - CARBON STL.

MAJOR PARTS	ASME SPEC
FORGINGS	SA-266 GR 4
HEADS	SA-515 GR 65
PLATES	SA-515 GR 65
NOZZLES	SA-266 GR 4
TIE RODS	SA-193 GR B7
TIE ROD NUTS	SA-194 GR 7
TIE ROD SPACERS	SA-106 GR B
TUBING	SA-556 GR B2
BELLOWS	SA-376 TY 304 SS

4- WEIGHTS:
DRY (NO INSUL.) - 18,144 KG (40,000 LBS) APPROX.
OPERATING WEIGHT, NO INSUL. WITH MOLTEN SALT @ 248°C (480°F) - 30,391 KG (67,000 LBS) APPROX.

Figure 5.10 Preheater--50-MWe Hybrid SGS

FOSTER WHEELER SOLAR DEVELOPMENT CORPORATION

REF.: 20-9909B
DATE: September 1982

NOTES:

1- VESSEL DESIGNED PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION W/1980 WINTER ADDENDUM

2- DESIGN TEMP:

STEAM DRUM & PARTS NOT IN CONTACT WITH MOLTEN SALT – 371°C (700°F)

HEAT EXCHANGER:

SHELL SIDE – 468°C (875°F)
TUBE SIDE – 371°C (700°F)

DESIGN PRESSURE:

STEAM DRUM – 15,344 KPAG (2,225 PSIG)

HEAT EXCHANGER:

SHELL SIDE – 2,068 KPAG (300 PSIG)
TUBE SIDE – 15,344 KPAG (2,225 PSIG)

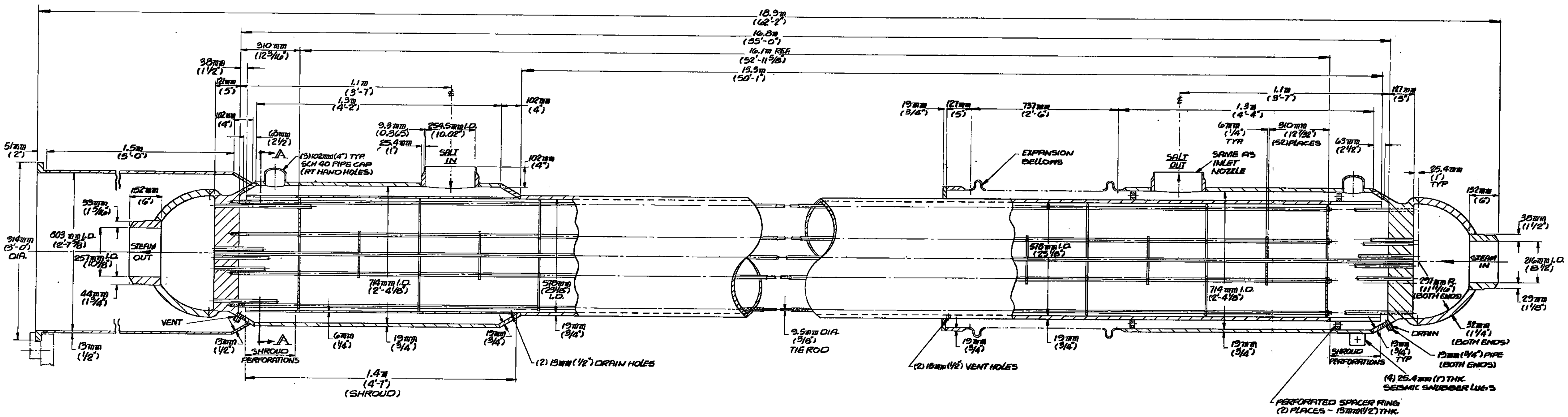
3- MATERIAL SPECS:

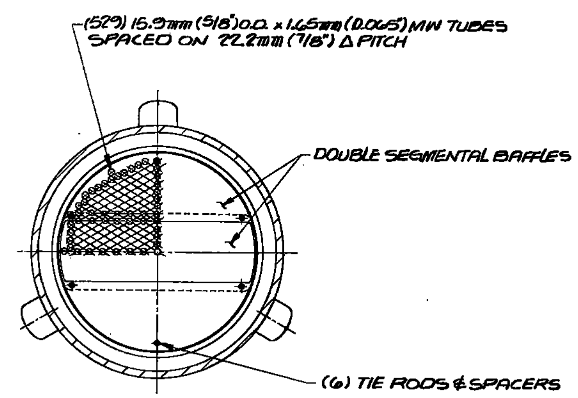
HEAT EXCHANGER-1¼% CR-½% MO	
MAJOR PARTS	ASME SPEC.
FORGINGS	SA-182-F11
HEADS	SA-387-GR11-CL1
PLATES	SA-387-GR11-CL1
NOZZLES	SA-182-F11
BELLOWS	SA-376 TY 304 SS
TUBING	SA-213-T11
PIPE	SA-335-P11
TIE ROD NUTS	SA-194-7
TIE ROD SPACERS	SA-335-P11
TIE RODS	SA-193-B7
STEAM DRUM-CARB. STL.	
MAJOR PARTS	ASME SPECS.
FORGINGS	SA-266 GR4
HEADS	SA-515 GR65
PLATES	SA-515 GR65
NOZZLES	SA-266 GR4
PIPE	SA-106 GR B

4- WEIGHTS:

DRY (NO INSUL.) – 67,132 KG (148,000 LBS) APPROX.
OPERATING WEIGHT, NO INSUL., WITH MOLTEN SALT @ 248°C (480°F) & WATER @ 336°C (636°F) – 104,871 KG (231,200 LBS.) APPROX.

Figure 5.11 Evaporator--50-MWe Hybrid SGS





SECTION A-A

NOTES:

1- VESSEL DESIGNED AS PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION W/1980 WINTER ADDENDUM

2- DESIGN TEMP:
TUBE SIDE - 565.6°C (1,050°F)
SHELL SIDE - 565.6°C (1,050°F)

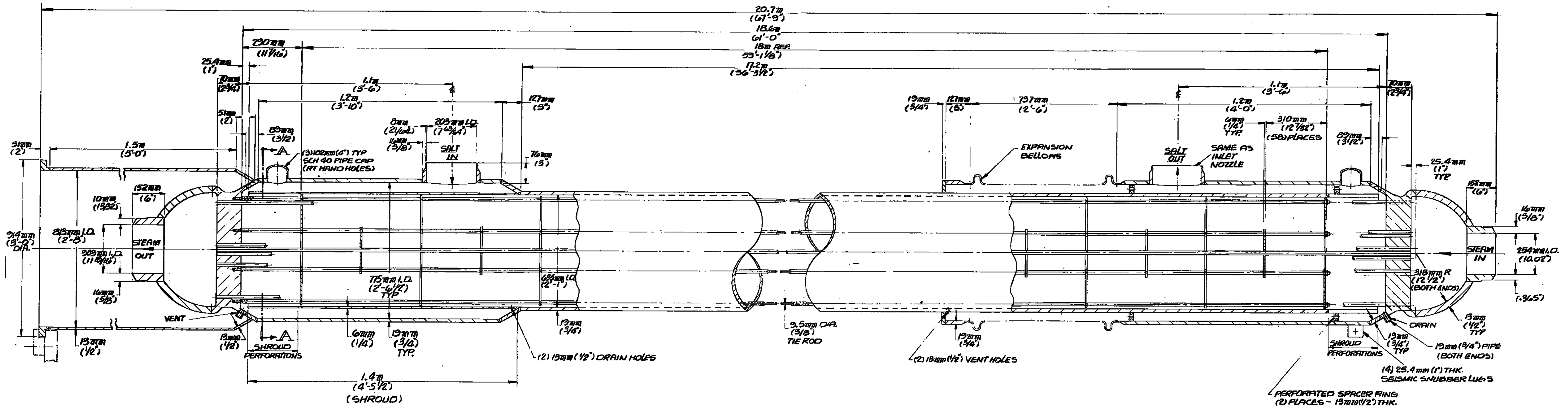
DESIGN PRESSURE:
TUBE SIDE - 15,340 KPAG (2,225 PSIG)
SHELL SIDE - 2,068 KPAG (300 PSIG)

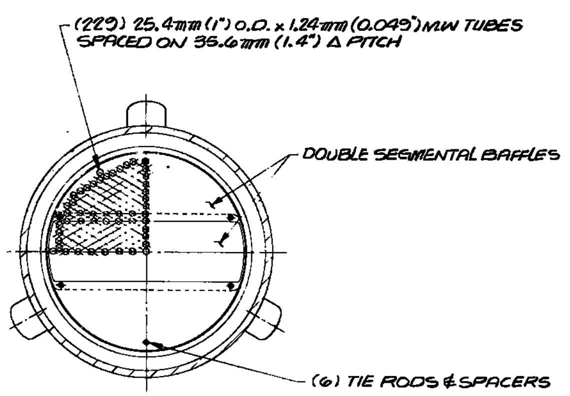
3- MATERIAL SPEC. - TYPE 304 SS

MAJOR PARTS	ASME SPEC.
FORGINGS	SA - 182
HEADS	SA - 240
PLATES	SA - 240
NOZZLES	SA - 182
TIE RODS	SA - 193
TIE ROD NUTS	SA - 194
TIE ROD SPACERS	SA - 376
TUBING	SA - 213
BELLOWS	SA - 376

4- WEIGHTS:
DRY (NO INSUL.) - 12,247 KG (27,000 LBS.) APPROX.
OPERATING WEIGHT, NO INSUL. WITH MOLTEN SALT @ 248°C (480°F) - 18,144 KG (40,000 LBS) APPROX.

Figure 5.12 Superheater--50-MWe Hybrid SGS





SECTION A-A

NOTES:

1- VESSEL DESIGNED AS PER ASME BOILER & PRESSURE VESSEL CODE SECT VIII, DIV 1 1980 EDITION W/1980 WINTER ADDENDUM

2- DESIGN TEMP:
TUBE SIDE – 565.6°C (1,050°F)
SHELL SIDE – 565.6°C (1,050°F)

DESIGN PRESSURE:
TUBE SIDE – 3,964 KPAG (575 PSIG)
SHELL SIDE – 2,068 KPAG (300 PSIG)

3- MATERIAL SPEC. – TYPE 304 SS

MAJOR PARTS	ASME SPEC
FORGINGS	SA-182
HEADS	SA-240
PLATES	SA-240
NOZZLES	SA-182
TIE RODS	SA-193
TIE ROD NUTS	SA-194
TIE ROD SPACERS	SA-376
TUBING	SA-213
BELLOWS	SA-376

4- WEIGHTS:
DRY (NO INSUL.) – 10,886 KG (24,000 LBS) APPROX.
OPERATING WEIGHT, NO INSUL. WITH MOLTEN SALT @ 248°C (480°F) – 18,598 KG (41,000 LBS) APPROX.

Figure 5.13 Reheater--50-MWe Hybrid SGS

5.4 THERMAL/HYDRAULIC ANALYSIS

5.4.1 Evaporator Circulation Analysis

The selected molten salt SGS concept utilizes natural circulation to maintain flow through the evaporator. The density difference between the steam/water mixture in the heated evaporator tubes and the saturated water in the downcomers provides the driving force for circulation.

The natural-circulation arrangement contains two circuits. The downcomer circuit starts at the bottom of the steam drum and consists of the downcomer and feeder pipes, which terminate at the evaporator water-inlet nozzle as shown in Figure 5.7 for the 100-MWe solar stand-alone SGS and Figure 5.11 for the 50-MWe hybrid SGS. The riser circuit consists of the evaporator inlet nozzle, evaporator tubes, and steam drum steam separators. Proper design of these circuits is required to obtain a circulation rate that will prevent the onset of DNB, dryout, or both.

A circulation ratio of 4:1 was selected based on Foster Wheeler's experience in the design of utility steam generators. The circulation ratio is defined as the total flow rate relative to the steam generated. Results of the DNB analysis (Section 5.4.4) indicate that DNB should not occur with a 4:1 circulation ratio. The size of the steam drum and the number and size of steam separator arms and Chevron driers were based on Foster Wheeler design standards for the operating drum pressure and steam generation rate.

An iterative procedure was used to design the evaporator tube bundle and the downcomer circuit. Flow balancing calculations between the riser and downcomer circuits were performed using Foster Wheeler's Computer Code EQPRSDRP, which computes total pressure drop from the acceleration, frictional, and gravitational components for each circuit. Each circuit should have the same absolute value of total pressure drop. Given the steam generation rate, the circulation ratio, and the preliminary configuration of the evaporator, we determined the configuration of the downcomer circuit. If the resultant design was not attractive either technically or economically, we modified the configuration of the evaporator, downcomer, or both until the desired configuration was obtained. Variables for the downcomer circuit were the number and size of downcomers and feeders.

The results of the evaporator circulation analysis are shown in Table 5.4.

5.4.2 Full- and Part-Load Performance

Estimated full- and part-load performance for the 100-MWe solar standalone SGS and 50-MWe hybrid SGS is listed in Tables 5.5 and 5.6. Predicted temperature and flow distributions over the load range for the 100-MWe solar stand-alone SGS are plotted in Figure 5.14. As noted in the SGS Requirements and Specifications (Appendix B), there are numerous possible operating conditions for the 50-MWe hybrid SGS, depending on the operating condition of the fossil-fuel-fired steam generator. Consequently, performance was estimated solely for the extremes in operating conditions.

The salt static pressure in the SGS must stay above atmospheric pressure to prevent dissolved gases from coming out of solution and forming a gas pocket within the heat exchangers. Consequently, the salt static pressure at the preheater outlet is regulated to be at or above 0.34 MPa gage (50 lb/in^2g).

Table 5.4 Results of Evaporator Circulation Analysis

Performance Parameters	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Circulation Ratio	4:1	4:1
Number of Downcomers	2	2
Size of Downcomers, mm (in.)	406 (16) Nominal pipe Sch. 140	305 (12) Nominal pipe Sch. 140
Number of Feeders	12	12
Size of Feeders, mm (in.)	152 (6) Nominal pipe Sch. 120	127 (5) Nominal pipe Sch. 120
Number of Pipes Between Evaporator and Steam Drum	0	0
Size of Pipe Between Evaporator and Steam Drum	N/A	N/A
Absolute Value of Circuit Total ΔP, kPa (lb/in^2)	132 (19.1)	131 (19.0)
Steam Drum Shell I.D., m (in.)	2.11 (83)	1.52 (60)
Number of Separator Arms	27	16
Inlet Converging-Diverging Section of Steam Drum, m (in.)	0.86 (34)	0.64 (25)

Table 5.5 Estimated Full- and Part-Load Performance—
100-MWe Solar Stand-Alone SGS

Performance Parameters	Load, % 25	50	75	100
Temperature, °C (°F)				
Steam/Water:				
Feedwater	201.7 (395)	210.3 (4105)	222.5 (432.5)	237.8 (460)
Superheater Inlet	331.7 (629)	332.8 (631)	333.9 (633)	335.6 (636)
Final Steam	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)
Reheater Inlet	298.9 (570)	308.9 (588)	324.4 (616)	342.2 (648)
Reheater Outlet	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)
Salt:				
Superheater Inlet	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)
Superheater Outlet	374.4 (706)	391.7 (737)	407.2 (765)	425.0 (797)
Reheater Inlet	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)
Reheater Outlet	398.9 (750)	430.0 (806)	454.4 (850)	473.3 (884)
Evaporator Inlet	443.3 (830)	443.9 (831)	445.0 (833)	446.1 (835)
Evaporator Outlet	332.2 (630)	334.4 (634)	337.8 (640)	342.8 (649)
Preheater Inlet	332.2 (630)	334.4 (634)	337.8 (640)	342.8 (649)
Preheater Outlet	275.6 (528)	281.1 (538)	287.2 (549)	293.3 (560)
Flow, kg/s (10^3 lb/h)				
Steam/Water:				
Feedwater	24.86 (198.1)	48.29 (383.3)	72.45 (575)	96.6 (766.7)
Blowdown	0.13 (1.0)	0.25 (2.0)	0.37 (2.9)	0.48 (3.8)
Main Steam	24.03 (190.7)	48.06 (381.4)	72.08 (572.1)	96.12 (762.9)
Reheater	21.26 (168.7)	42.37 (336.3)	63.0 (500)	83.2 (660.3)
Recirculation	8.01 (63.6)	11.72 (93)	9.61 (76.3)	— —
Salt:				
Preheater	167.58 (1330)	330.0 (2619)	490.26 (3891)	633.52 (5028)
Evaporator	167.58 (1330)	330.0 (2619)	490.26 (3891)	633.52 (5028)
Superheater	65.01 (516)	143.76 (1141)	237.51 (1885)	357.58 (2838)
Reheater	43.47 (345)	105.71 (839)	184.46 (1464)	275.94 (2190)
Bypass	59.09 (469)	80.51 (639)	68.29 (542)	— —
Pressure, MPa gage (lb/in^{2}g)				
Steam/Water:				
Feedwater	13.1 (1900)	13.27 (1925)	13.51 (1960)	13.89 (2015)
Drum	13.07 (1895)	13.2 (1915)	13.41 (1945)	13.69 (1985)
Final Steam	13.05 (1893)	13.1 (1800)	13.27 (1925)	13.48 (1955)
Reheater Inlet	0.83 (120)	1.59 (230)	2.34 (340)	3.03 (440)
Reheater Outlet	0.82 (119)	1.48 (215)	2.21 (320)	2.86 (415)
Salt:				
Superheater Inlet	0.18 (26.8)	0.36 (52.1)	0.68 (98.0)	1.15 (167.4)
Reheater Inlet	0.19 (26.9)	0.36 (51.7)	0.66 (96.4)	1.13 (164.0)
Preheater Outlet	0.34 (50)	0.34 (50)	0.34 (50)	0.34 (50.0)

Table 5.6 Estimated Full- and Part-Load Performance--50-MWe Hybrid SGS

Performance Parameters	Load, Solar/Fossil (%) 50/50	50/25	50/0	25/0	12.5/87.5	12.5/25
Temperature, °C (°F)						
Steam/Water:						
Feedwater	237.8 (460)	218.5 (425.3)	210.3 (410.5)	182.7 (360.8)	237.8 (460)	193.3 (380)
Superheater Inlet	335.6 (636)	335.4 (635.8)	335.4 (635.8)	332.7 (630.8)	331.8 (629.2)	231.8 (629.2)
Final Steam	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)
Reheater Inlet	342.2 (648)	318.3 (605)	302.2 (576)	298.3 (569)	341.1 (646)	284.4 (544)
Reheater Outlet	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)	540.6 (1005)
Salt:						
Superheater Inlet	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)
Superheater Outlet	425 (787)	421.1 (790)	421.1 (790)	381.4 (736.5)	346.1 (655)	346.1 (655)
Reheater Inlet	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)	562.8 (1045)
Reheater Outlet	473.3 (884)	468.9 (876)	464.4 (868)	424.2 (795.6)	367.5 (693.5)	324.7 (616.5)
Evaporator Inlet	446.1 (835)	446.7 (836)	446.1 (835)	445 (833)	442.5 (828.5)	440 (824)
Evaporator Outlet	342.8 (649)	341.7 (647)	341.7 (647)	333.9 (633)	332.2 (630)	332.2 (630)
Preheater Inlet	342.8 (649)	341.7 (647)	341.7 (647)	333.9 (633)	332.2 (630)	332.2 (630)
Preheater Outlet	293.3 (560)	289.4 (553)	287.8 (550)	276.7 (530)	286.1 (547)	275.0 (527)
Flow, kg/s (10^3 lb/h)						
Steam/Water:						
Feedwater	48.3 (383.34)	48.3 (383.34)	48.3 (383.34)	25.0 (198.13)	12.1 (95.83)	12.1 (95.83)
Blowdown	0.24 (1.91)	0.24 (1.91)	0.24 (1.91)	0.93 (7.41)	0.06 (0.47)	0.06 (0.47)
Main Steam	48.06 (381.43)	48.06 (381.43)	48.06 (381.43)	24.0 (190.67)	12.0 (95.36)	12.0 (95.36)
Reheater	41.6 (330.14)	41.98 (333.17)	42.38 (336.32)	21.25 (168.67)	10.4 (82.41)	10.7 (84.65)
Recirculation	8.03 (63.7)	8.03 (63.7)	11.39 (90.4)	9.22 (73.2)	0 (0)	4.76 (37.81)

Table 5.6 Estimated Full- and Part-Load Performance--50-MWe Hybrid SGS (Cont)

Performance Parameters	Load, Solar/Fossil (%) 50/50	50/25	50/0	25/0	12.5/87.5	12.5/25
Temperature, °C (°F)						
Preheater	316.8 (2514)	333.3 (2645)	338.4 (2686)	167.7 (1331)	78.7 (625)	83.9 (666)
Evaporator	316.8 (2514)	333.3 (2645)	338.4 (2686)	167.7 (1331)	78.7 (625)	83.9 (666)
Superheater	178.8 (1419)	174.0 (1381)	174.0 (1381)	70.9 (563)	27.7 (220)	27.7 (220)
Reheater	138.0 (1095)	146.0 (1159)	147.5 (1171)	51.6 (410)	15.5 (123)	15.9 (126)
Bypass	0 (0)	13.2 (105)	16.9 (134)	45.1 (358)	35.5 (282)	40.3 (320)
Pressure, MPa gage (lb/in²g)						
Steam/Water:						
Feedwater	13.89 (2015)	13.89 (2015)	13.89 (2015)	13.27 (1925)	13.1 (1900)	13.1 (1900)
Drum	13.69 (1985)	13.69 (1985)	13.69 (1985)	13.20 (1915)	13.07 (1895)	13.07 (1895
Final Steam	13.48 (1955)	13.48 (1955)	13.48 (1955)	13.1 (1900)	13.05 (1893)	13.05 (1893)
Reheater Inlet	3.00 (435)	2.24 (325)	1.41 (205)	0.76 (110)	3.03 (440)	1.17 (170)
Reheater Outlet	2.83 (410)	2.00 (290)	1.03 (150)	0.60 (87)	3.02 (438)	1.14 (166)
Salt:						
Superheater Inlet	1.00 (145.1)	1.00 (145.1)	1.00 (145.1)	1.00 (145.1)	1.00 (145.1)	1.00 (145.1)
Reheater Inlet	1.02 (147.8)	1.02 (147.8)	1.02 (147.8)	1.02 (147.8)	1.02 (147.8)	1.02 (147.8)
Preheater Outlet	0.34 (50.0)	0.34 (50.0)	0.34 (50.0)	0.34 (50.0)	0.34 (50.0)	0.34 (50.0)

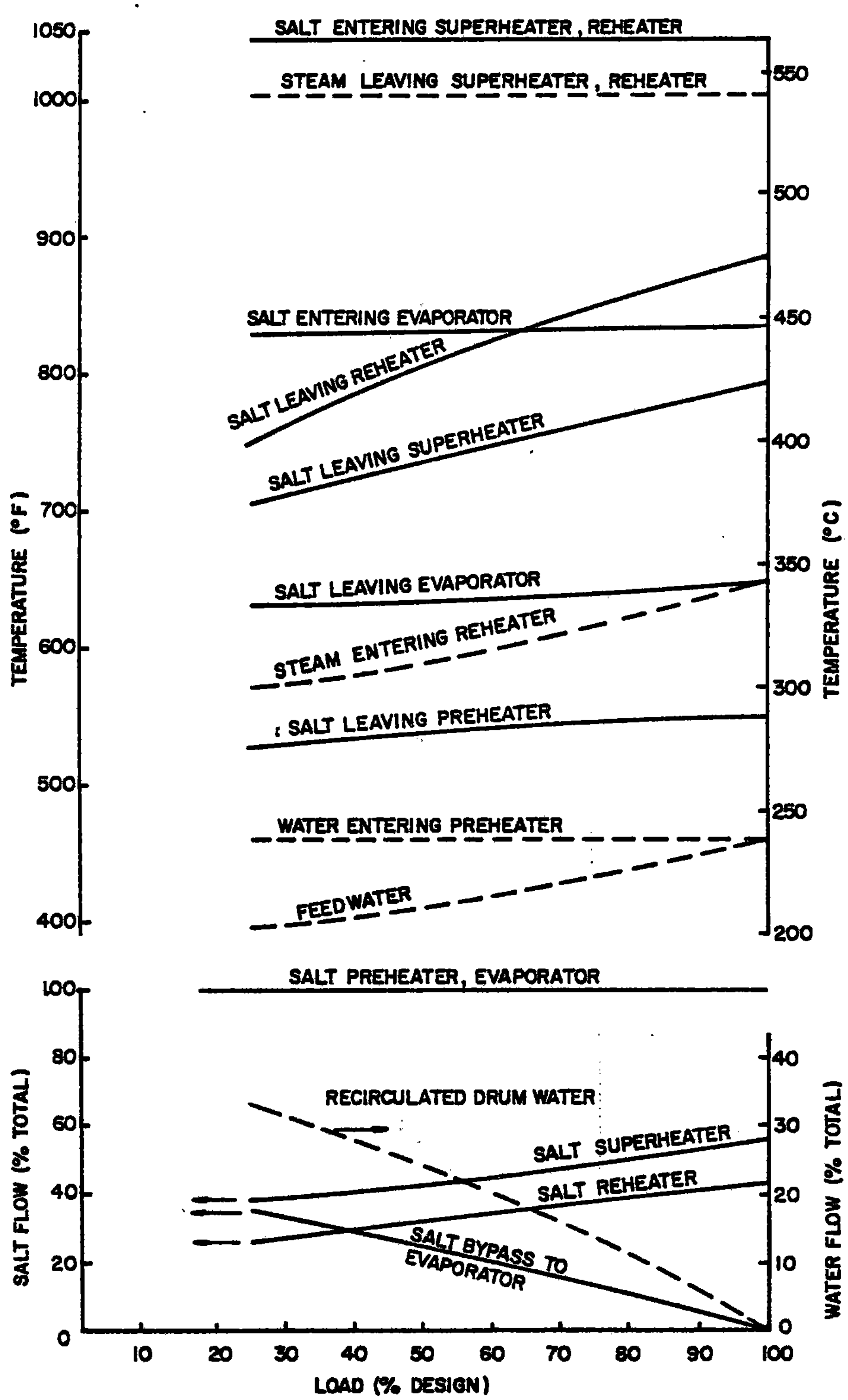


Figure 5.14 Predicted Performance Over Load Range--
100-MWe Solar Stand-Alone SGS

5.4.3 Stability Analysis

Heat exchangers with boiling, condensing, or other two-phase flow are susceptible to flow instabilities that may cause flow excursions or flow oscillations. These undesirable phenomena could cause forced mechanical vibrations of components; disturb system control; or induce DNB, dryout, or both.

In a boiling channel, small fluctuations are present because of variations in bubble formation rate and population, flow regimes, heat-transfer mechanism, turbulence, etc. These fluctuations may induce flow instabilities, which can be either static or dynamic in nature. The threshold of flow instability can be analyzed by conservation equations of mass, momentum, energy, and the proper equation of state. Static instability lies in the steady-state laws, while dynamic instability is a time-variant phenomenon.

Static Instability. A flow is subjected to a static instability when the flow conditions, changed by a small perturbation, will not return to original steady-state conditions. Most static instabilities are induced by primary phenomena such as the Ledinegg flow excursion, DNB, maximum superheat for vapor bubble burst, and flow pattern transitions.

The static instability of primary concern is the Ledinegg flow excursion. The significance of the flow excursion is best observed by plotting pressure-drop/flow-rate characteristics, as schematically shown in Figure 5.15. A system of parallel heated tubes is considered, with attention focused solely on one tube, where various levels of heat input are allowed. The quantity of heat input depends on the shell-side heating medium flow distribution surrounding heated

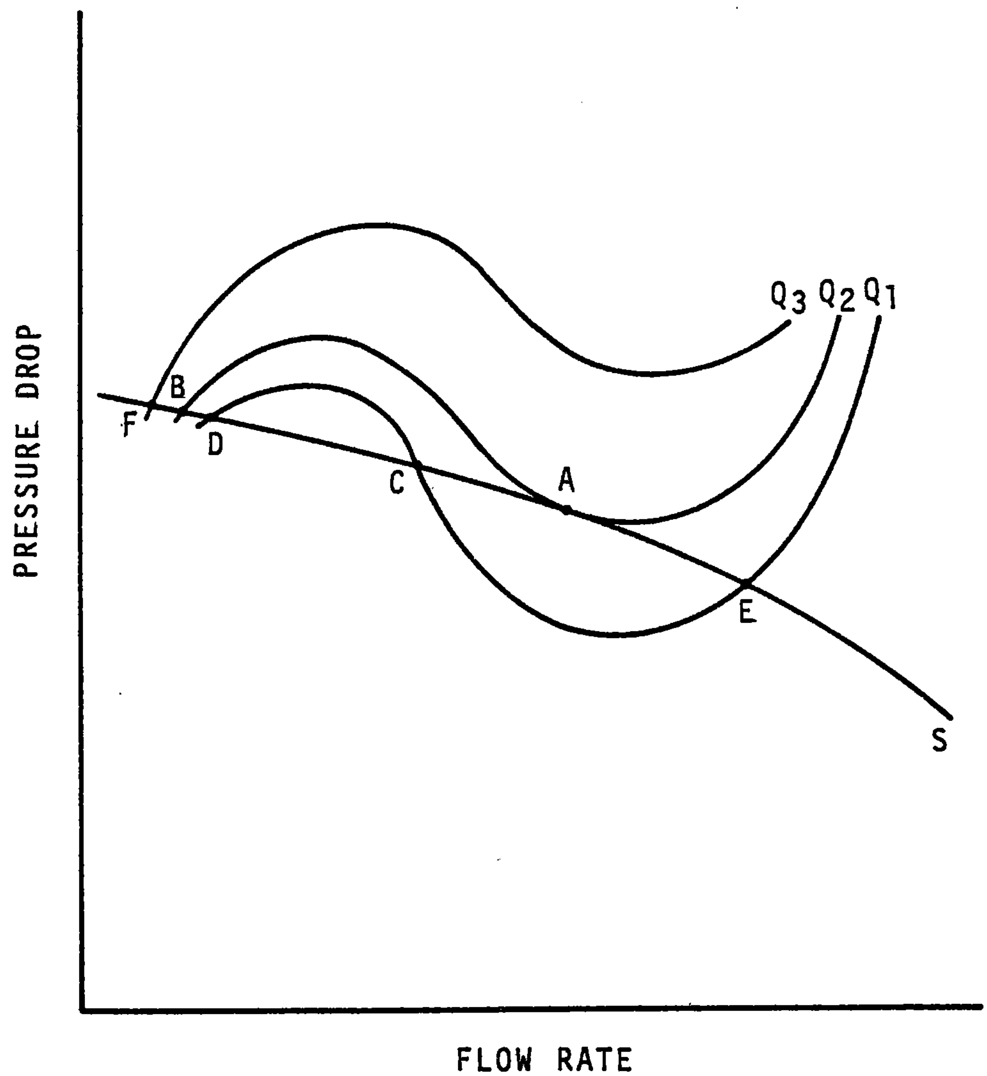


Figure 5.15 Static Instability--Ledinegg Flow Excursion[40]

tubes. Demand curves (internal characteristics of tube) Q1, Q2, and Q3 denote increased quantities of heating medium surrounding the concerned heated tube. Curve S denotes the loop supply pressure-drop/flow-rate characteristics external to the tube. Intersections with curve Q1 are the possible operating points as indicated by C, D, or E. Operation at Point D or E will be stable; operation at Point C, unstable. For example, if at either Point D or E the flow is perturbed to increase (decrease), the pressure drop of the heated tube increases (decreases) [i.e., the demand of the system is larger (less) than the external supply, and consequently the flow will return to its original value]. However, if the flow is perturbed to increase (decrease) at Point C, the external system supplies more (less) than that required to maintain the flow. Consequently, the flow rate will increase (decrease) until the new operating point, E (D), is reached. Therefore, the shape of Curve Q1--especially at Point C--as shown in Figure 5.15, should be avoided to ensure static stability within the possible range of load operations. Figure 5.15 also indicates the sensitivity of tube-side flow maldistribution in the same system. For instance, the operating point is assumed at Point E. With an increased heating medium flow around the local tube to Q2, the flow decreases monotonically to Point A. Perturbations in any of the system variables can cause a flow excursion or rapid deceleration to a stable point, B. A further increase in heating medium surrounding the tube to Q3 results in operation at Point F. Therefore, the heating imbalance among circuits will induce flow maldistribution in a system of parallel heated tubes.

The general criterion for the Ledinegg instability is:

$$\partial \Delta P/\partial W \Big|_{internal} < \partial \Delta P/\Delta w \Big|_{external} \tag{13}$$

or simply:

$$\partial \Delta P/\partial W \Big|_{internal} < 0 \text{ for a fixed } \Delta P \text{ between inlet and outlet plenums} \tag{14}$$

where

W = Flow rate

ΔP = Pressure drop.

This criterion for the flow excursion instability is well documented, and the prediction techniques have been developed using the steady-state conservation equations of mass, momentum, energy, and the equations of state.

The potential static instability of the evaporator designed for the 100-MWe solar stand-alone SGS was analyzed at full-load conditions. The inlet conditions of water and salt were kept constant, and the flow rate of water was perturbed to establish the pressure-drop/flow-rate characteristics. The effect of shell-side salt flow distribution, resulting in a variation of heat input to different tubes, was also investigated to check the potential tube-side flow maldistribution in parallel heated tubes of the evaporator. The molten salt steam generator performance MSSG computer code was used to compute the total steam/water pressure drop (including friction, acceleration, gravitation) under various steam/water and salt flow rates.

The results of the static instability analysis are presented in Figure 5.16. Used in this figure is the relative flow rate, which is the ratio of flow rate under perturbation to that under normal operating condition at full load. Curve Q denotes the condition of salt surrounding a local tube under the normal flow distribution condition, and Curves Q+ and Q- denote 110 percent and 90 percent of the normal salt distribution surrounding the tube.

As seen in Figure 5.16, the slopes of all the curves are positive, thereby ensuring the static stability of the unit. These curves also indicate that the potential tube-side maldistribution (resulting from variations in heat input caused by uneven shell-side flow distribution) is negligible.

The thermal/hydraulic design criteria used for sizing both the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS were the same except for the reduction in flow resulting from the difference in thermal rating. Consequently, the results of the 100-MWe solar stand-alone evaporator static stability analysis give confidence that the 50-MWe hybrid evaporator is statically stable. Thus further analysis is precluded.

Dynamic Instability. Dynamic instabilities can be classified according to the propagation of disturbances such as pressure-wave (acoustic) and density-wave instabilities. Pressure-wave instability is characterized by a high-frequency (10 to 100 Hz), the period being the same order of magnitude as the time required for a pressure wave to travel through the system. Density-wave instability has low frequency (0.1 to 1 Hz), with a period which is approximately one to two times the time required for a fluid particle to travel through the system.

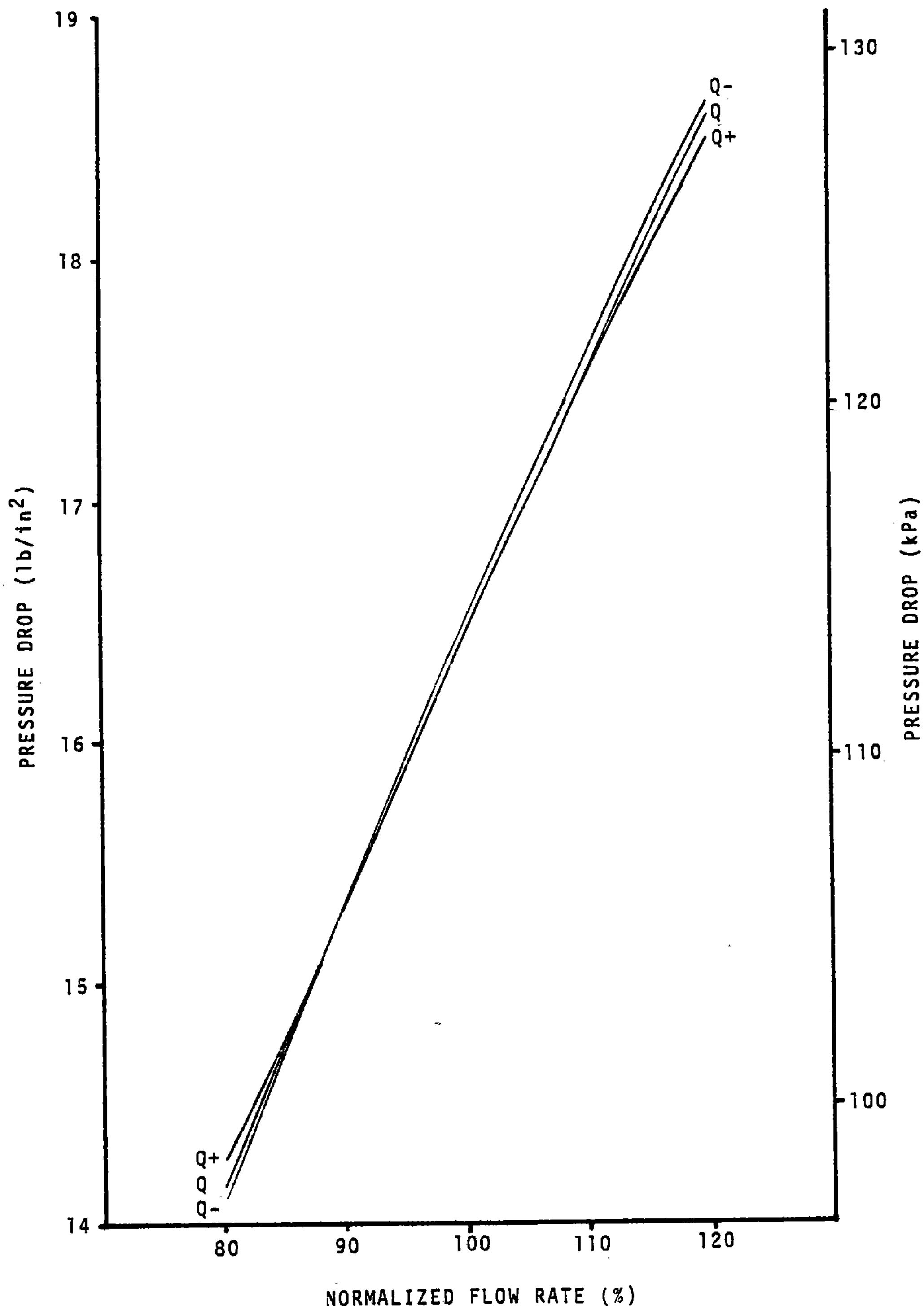


Figure 5.16 Static Instability Analysis--100-MWe Solar Stand-Alone Evaporator

Pressure-wave instabilities have been observed in subcooled, bulk, and film boiling. The threshold of instability is in the negative sloping region of the pressure drop-flow rate curve as Point C in Figure 5.15.[40] Since the pressure-drop flow-rate curves for the 100-MWe solar stand-alone evaporator (as shown in Figure 5.16) have positive slopes, we have concluded that the evaporator is not susceptible to pressure-wave instabilities.

Density-wave instability is the most common type of two-phase flow instability in a boiler tube. It is defined as sustained (or growing) oscillation of flow variables such as flow rate, vapor-generation rate, fluid density, and pressure drop within a tube.

Density-wave instability is attributed to the feedback and interaction of flow variables. A temporary reduction of subcooled flow entering a heated tube will create propagating enthalpy and density perturbations in the subcooled region, thereby creating the oscillation of boiling boundary. Changes in flow and the length of single-phase region result in an oscillatory single-phase pressure drop. At the boiling boundary, the enthalpy perturbations are transformed into quality (or void fraction) perturbations that travel up the heated tube with the flow. The combined effects of flow and void fraction perturbations and variations of the two-phase length create a two-phase pressure-drop perturbation. However, the loop supply pressure drop is maintained constant. Therefore, the two-phase pressure-drop perturbation produces a feedback perturbation of the opposite sign in the single-phase region, which can either enforce or accentuate the imposed oscillation.

Density-wave oscillation can be analyzed by the conventional linear feedback theory in the frequency domain. For this study, the Foster Wheeler computer code NUFREQ-FW was used to analyze the density-wave instability for the evaporator at full-load conditions. NUFREQ-FW is an updated and extended version of the NUFREQ computer code developed by General Electric for the investigation of density-wave instability in a heated two-phase channel (boiling water reactor applications).[4,2] Foster Wheeler has modified the General Electric version to include the superheated region, permitting application to once-through steam generators. The model for the superheated region includes both the heated region and an adiabatic riser. In the computer code, the equations governing the conservation of mass, momentum, and energy were first linearized about the normal operating condition. The dynamic analysis solved the linearized partial differential equations using Laplace transformation of the temporal terms and integration of the spatial variations. The resultant equations represented the transfer function for each spatial node. The computer code was then written in complex variable notation and employed frequency response techniques to develop the system transfer function.

The Nyquist stability criteria from control system theory were applied to determine whether the boiling channel was stable. Nyquist's theorem can be phrased as follows: The necessary condition for a linear system to be unstable is that the complex locus of the open-loop transfer function passes through or encircles in a clockwise manner the unity point on the negative real axis.

The 100-MWe solar stand-alone evaporator was analyzed at full-load conditions using NUFREQ-FW. The resultant Nyquist diagram is shown in Figure 5.17.

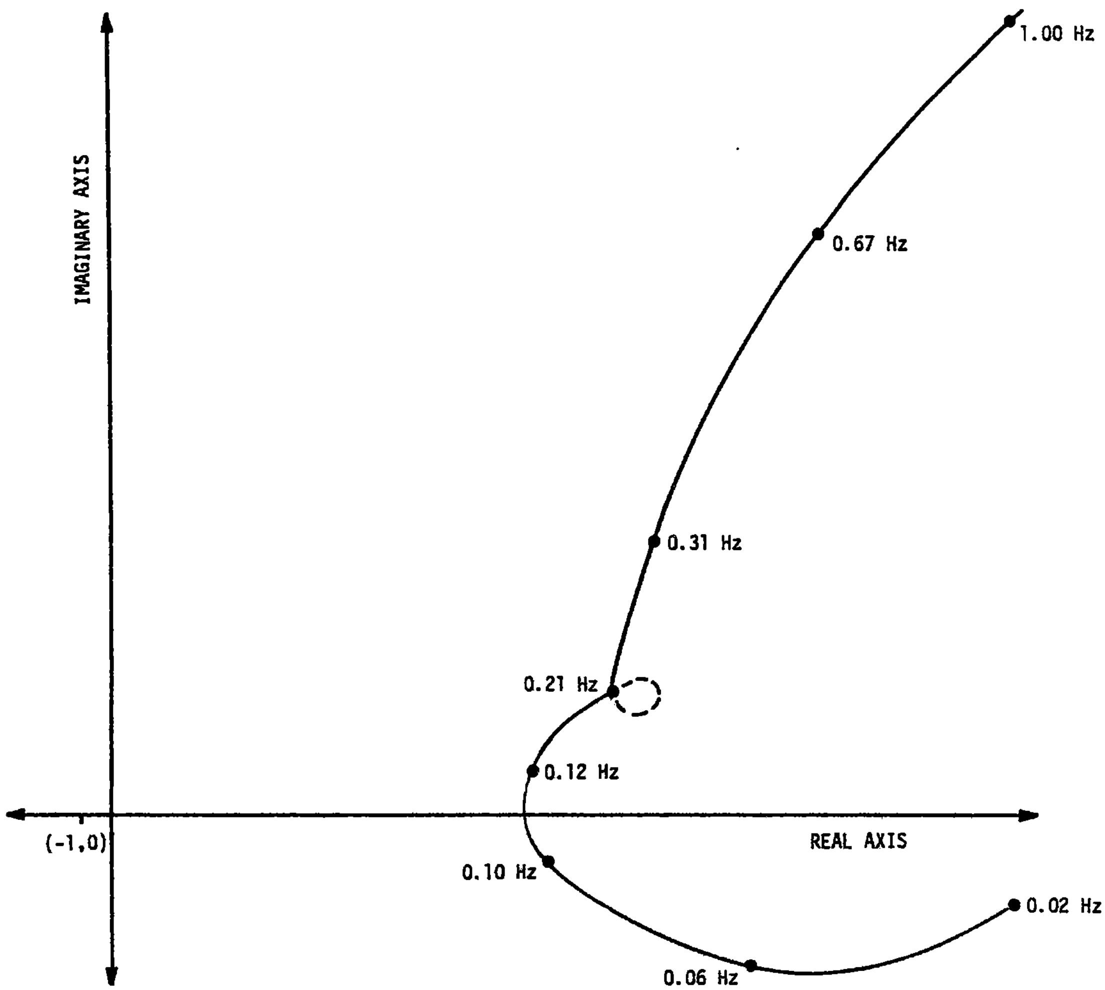


Figure 5.17 Dynamic Instability Analysis--100-MWe Solar Stand-Alone Evaporator

From Figure 5.17, the complex locus of the open-loop transfer function does not pass through or encircle the unity point on the negative real axis. Therefore, we have concluded that density-wave instability is not likely to occur in the evaporator at the full-load condition.

As noted for the static stability analysis, the design of the 100-MWe solar stand-alone evaporator and the 50-MW hybrid evaporator are similar; thus the results of the dynamic instability analysis for the 100-MWe unit can be applied to the 50-MWe unit without further analysis.

5.4.4 Critical Heat-Flux Analysis

Critical heat flux is defined as the maximum heat flux occurring just before a change in boiling heat-transfer mode, as indicated by a tubewall temperature excursion. The mechanism for transition varies depending upon the two-phase flow pattern within the tube. In subcooled or low-steam-quality bubble-type flow, excessively high heat fluxes can create a vapor blanket between the inner tubewall and the steam/water core. This type of occurrence is called DNB. At somewhat lower heat-flux levels and higher steam qualities, there can be transition from an annular flow regime to a dispersed (mist) flow regime. Water droplets in the flow core cannot maintain a wet tube surface because the heat-flux level is such that the droplets vaporize before they reach the surface. This type of occurrence is called dryout. When DNB or dryout occur, the heat-transfer coefficient deteriorates sharply and the tubewall temperature increases sharply. Figure 5.18 illustrates DNB and dryout.

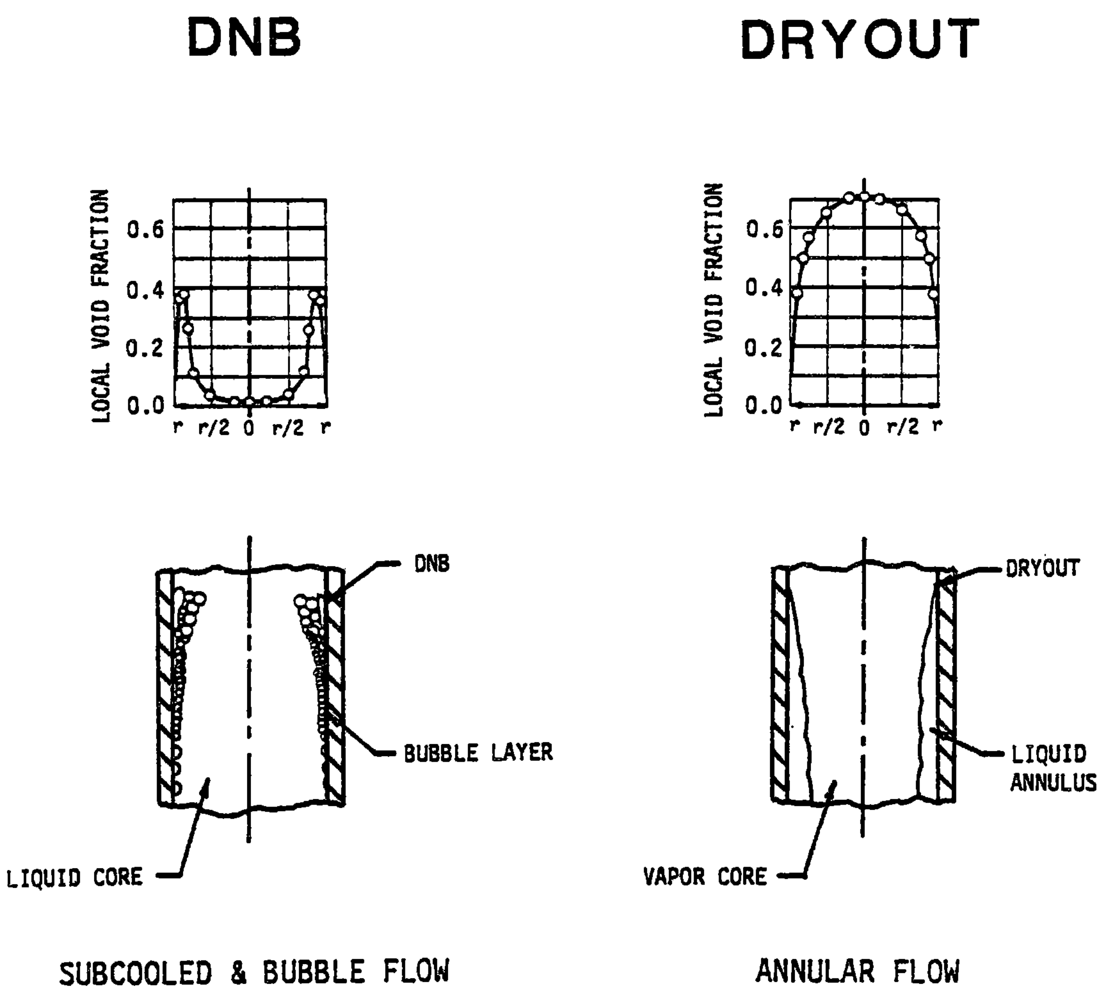


Figure 5.18 DNB and Dryout

For the evaporator design considered in this study, the tubewall temperature cannot exceed the local salt temperature. Consequently, tube burnout is not expected to be a problem. However, tube fatigue attributed to the tubewall thermal oscillation and dissolved solids deposition/corrosion at the critical heat-flux point can be a potential problem.

To identify whether DNB or dryout will occur, the Atomics International and Westinghouse critical heat-flux correlations described in Section 5.2[32,33] were incorporated in the MSSG thermal sizing computer code. The critical heat-flux steam qualities predicted by the Westinghouse correlation for the 100-MWe solar stand-alone evaporator are plotted in Figure 5.19. As can be seen, the correlation predicts that DNB and dryout will not occur.

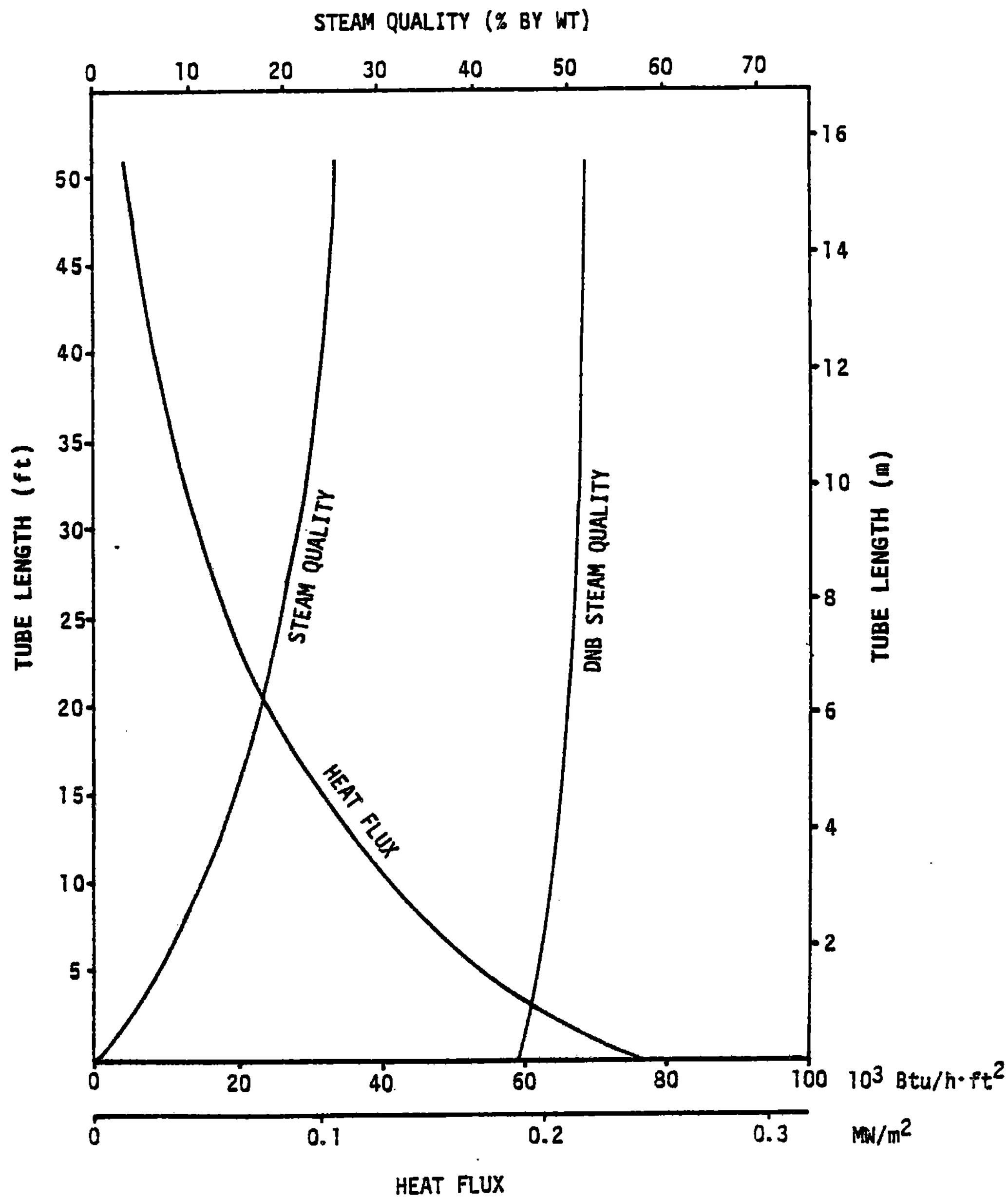


Figure 5.19 Critical Heat Flux Analysis--100-MWe Solar Stand-Alone Evaporator

5.5 STRUCTURAL ANALYSIS

The steam generator structural sizing calculations were performed according to the requirements of Section VIII-Division 1 of the ASME Boiler and Pressure Vessel Code[41] and TEMA Standards.[24] Since the steam generator is subjected to daily start-up and shutdown cycles as well as other transients, fatigue and creep-fatigue are important failure modes that should be precluded. These failure modes are not explicitly addressed in Section VIII-Division 1; thus we did additional thermal stress analyses. These stresses were then evaluated by the criteria set forth in Section VIII-Division 2.[39] The elevated-temperature fatigue curves of Code Case N-47 were also used in this evaluation.[43] In addition, tube buckling and flow-induced vibration analyses were done. The criteria used to evaluate tube buckling were those of Code Case N-47, Appendix T-1500 (Buckling and Instability).[43] Flow-induced vibrations were evaluated according to the criteria set forth in the HTRI Report.[44]

This section summarizes the stress analyses and evaluation of all critical elements of the steam generator. Specifically, the following items are discussed:

- Steady-state analysis of the tubesheet
- Fatigue evaluation of tubesheet junctions
- Transient analysis
- Buckling analysis
- Flow-induced vibration analysis.

5.5.1 Steady-State Tubesheet Analysis

One of the critical components of the steam generator is the tubesheet. To determine the steady-state temperature and stress distributions in the tubesheet, we performed a simplified analysis. The model is shown in Figure 5.20. Taking advantage of the symmetry of the tube arrangement, a representative cylinder is isolated from the tube as shown. The steady-state heat-transfer equation is solved for this cylinder using the following boundary conditions.

$$k\ (\partial T/\partial z) = h_m(T - T_m) \text{ at } z = 0 \tag{15}$$

$$k\ (\partial T/\partial z) = -h_s(T - T_s) \text{ at } z = t \tag{16}$$

$$k\ (\partial T/\partial r) = h_s(T - T_s) \text{ at } r = r_i \tag{17}$$

$$\partial T/\partial r = 0 \text{ at } r = r_o \tag{18}$$

where

h_s = Heat-transfer coefficient on the steam side

h_m = Heat-transfer coefficient on the salt side

T_s, T_m = Steam and salt temperatures

k = Thermal conductivity

r_i, r_o = Inner and outer radii of the representative cylinder.

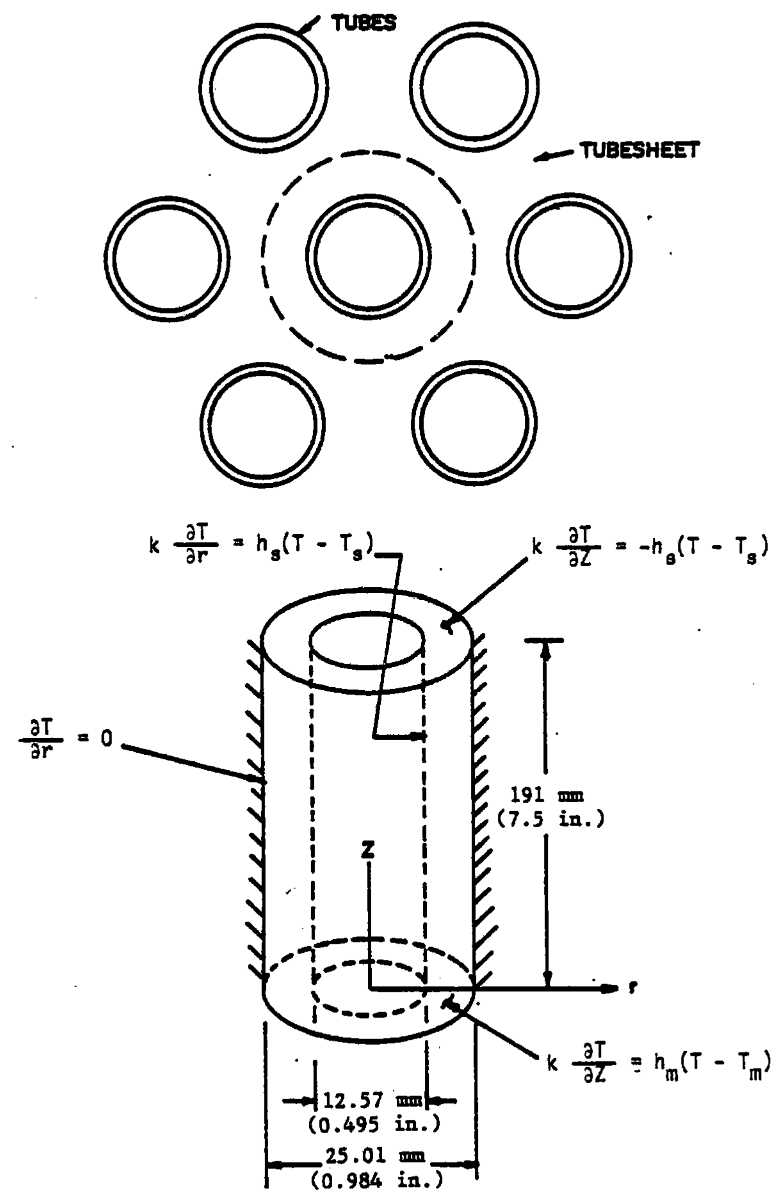


Figure 5.20 Tubesheet Model

A computer program was written to solve the problem. A typical set of input conditions is shown below:

$$h_s = 3.725\ \text{kW/m}^2\text{K}\ (656\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})$$

$$h_m = 5.394\ \text{kW/m}^2\text{K}\ (950\ \text{Btu/h}\cdot\text{ft}^2\cdot{}^\circ\text{F})$$

$$\alpha = 0.0177\ \text{m}^2\text{/h}\ (0.19\ \text{ft}^2\text{/h})$$

$$k = 21.97\ \text{W/mK}\ (12.7\ \text{Btu/h}\cdot\text{ft}\cdot{}^\circ\text{F})$$

$$T_m = 565.6^\circ\text{C}\ (1050^\circ\text{F})$$

$$T_s = 540.6^\circ\text{C}\ (1005^\circ\text{F})$$

For these conditions the temperature distribution in the tubesheet is shown in Table 5.7. The average, maximum, and minimum temperatures are as follows:

Average metal temperature, $T_{av} = 541.1^\circ\text{C}$ (1006°F)

Maximum metal temperature, $T_{max} = 560.6^\circ\text{C}$ (1041°F)

Minimum metal temperature, $T_{min} = 540.6^\circ\text{C}$ (1005°F)

The average and minimum metal temperatures are very close. The maximum thermal stress is given by:

$$\sigma = \alpha E/1 - \nu\ (T_{max} - T_{av}) \qquad (19)$$

and is equal to 78.6 MPa (11.4 kip/in^2), which is substantially lower than the allowable stress.

Table 5.7 Temperature Distribution in Tubesheet, °C (°F)

$\bar{z}$*	$\bar{r}$†			
	0.330	0.0439	0.0548	0.0656
0.0	555.0 (1031.2)	558.9 (1038.0)	560.2 (1040.3)	560.5 (1040.9)
0.2	540.6 (1005.0)	540.6 (1005.0)	540.7 (1005.2)	540.7 (1005.2)
0.4	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)
0.6	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)
0.8	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)
1.0	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)	540.6 (1005.0)

*$\bar{z}$ = z/t
†$\bar{r}$ = r/t

Assuming that the tubesheet has to withstand 10,000 daily cycles, the allowable strain range, $\Delta\epsilon$, at 566°C (1050°F) is 0.00221, (from the fatigue curves of Code Case N-47) corresponding to an allowable ΔT (i.e., $T_{max} - T_{av}$) of 81.67°C (147°F) in the tubesheet. From the foregoing analysis, the actual ΔT is only 19.4°C (35°F). Corresponding to the ΔT of 19.4°C (35°F), the elastic stress intensity is 78.6 MPa (11.4 kip/in²). At this stress level, the creep rupture life is more than 300,000 hours--considered to be large enough so that a complete creep-fatigue analysis is deemed unnecessary. Hence there is a substantial margin of safety in the fatigue life of the tubesheet in addition to the factors of safety that are already built into the fatigue curves of Code Case N-47.

5.5.2 Fatigue Evaluation of Tubesheet Junctions

The analysis described in Section 5.5.1 considered the tubesheet as an equivalent plate with fixed end conditions. However, the junctions of the upper and lower tubesheets with the head and shell are more critical areas. We analyzed these junctions for stresses to determine the maximum allowable salt and steam temperature variations.

Upper Tubesheet Cylindrical-Shell/Spherical-Head Junction. The exact geometry of this junction is shown in Figures 5.8 and 5.9 for the 100-MWe solar stand-alone superheater and reheater. For analytical purposes, we used the simplified model shown in Figure 5.21. In this model the hanger was ignored. The average temperature of the spherical head and the tubesheet, at any given time, was assumed to be equal to the steam temperature. The average temperature of the cylindrical shell was assumed to be that of the salt. The three

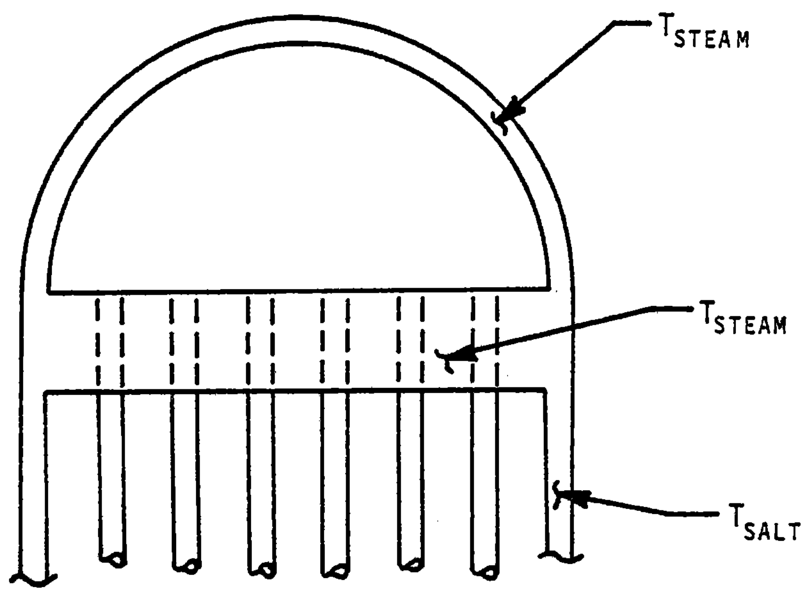


Figure 5.21 Upper Tubesheet/Head/Shell Junction Model

elements (i.e., the tubesheet, the head, and the shell) were first allowed to expand freely, and the compatibility at the junction was satisfied by introducing forces and moments. The stresses and strains caused by these forces were obtained by the formulas from Roark.[45] A computer program was written to perform the numerical calculations. The displacements and strain ranges can be expressed in the following way:

$$u_r = k_1 \alpha\Delta T \quad (20)$$

$$(\Delta\varepsilon)_{ts} = k_2 \alpha\Delta T \quad (21)$$

$$(\Delta\varepsilon)_{cs} = k_3 \alpha\Delta T \quad (22)$$

$$(\Delta\varepsilon)_{ss} = k_4 \alpha\Delta T \quad (23)$$

where

u_r = Radial displacement

$(\Delta\varepsilon)_{ts}$, $(\Delta\varepsilon)_{cs}$, $(\Delta\varepsilon)_{ss}$ = Strain ranges in tubesheet, cylindrical shell, and spherical shell

k_1, k_2, k_3, k_4 = Constants

$\alpha\Delta T = (\alpha_{salt}T_{salt} - \alpha_{steam}T_{steam})$

α_{salt}, α_{steam} = Coefficients of thermal expansion of the metal at temperatures equal to T_{salt} and T_{steam}.

Using the computer program, a parametric study determined the effect of varying the thicknesses of the tubesheet and the cylindrical shell. The thicknesses of the head and shell were assumed to be equal. The results of this study are shown in Table 5.8, which clearly shows that the radial displacement and the strain ranges are not sensitive to the variations in the thicknesses of the shell and tubesheet. The strain ranges in the shell (represented by k_3) are two orders of magnitude higher than those in the head and tubesheet. Table 5.8 also clearly shows that the strain ranges can be conservatively estimated as:

Tubesheet:	0.002 $\alpha\Delta T$
Cylindrical shell:	1.83 $\alpha\Delta T$
Head:	0.003 $\alpha\Delta T$

Table 5.8 Parametric Study--Upper Tubesheet Junction

Cylindrical Shell Thickness, t_{cs}	Tubesheet Thickness, t_{ts}: 190.5 mm (7.5 in.)	304.8 mm (12 in.)
19.1 mm (0.75 in.)	k_1 = 0.01911	k_1 = 0.01212
	k_2 = 0.00112	k_2 = 0.00071
	k_3 = 1.82652	k_3 = 1.82713
	k_4 = 0.00166	k_4 = 0.00105
25.4 mm (1 in.)	k_1 = 0.02882	k_1 = 0.01841
	k_2 = 0.00170	k_2 = 0.00109
	k_3 = 1.82567	k_3 = 1.82658
	k_4 = 0.00251	k_4 = 0.00160

Using these strain values and the maximum allowable strain values for a fatigue life of 10,000 cycles, the allowable steady-state variation of (T_{salt} - T_{steam}) was calculated as a function of the maximum metal temperature and is plotted in Figure 5.22. The maximum metal temperature was conservatively assumed to be equal to the salt temperature. The fatigue curves of Code Case N-47 were used in this analysis. The sudden drop in the allowable temperature variations at higher metal temperatures results from the deterioration of fatigue properties of the material at those temperatures.

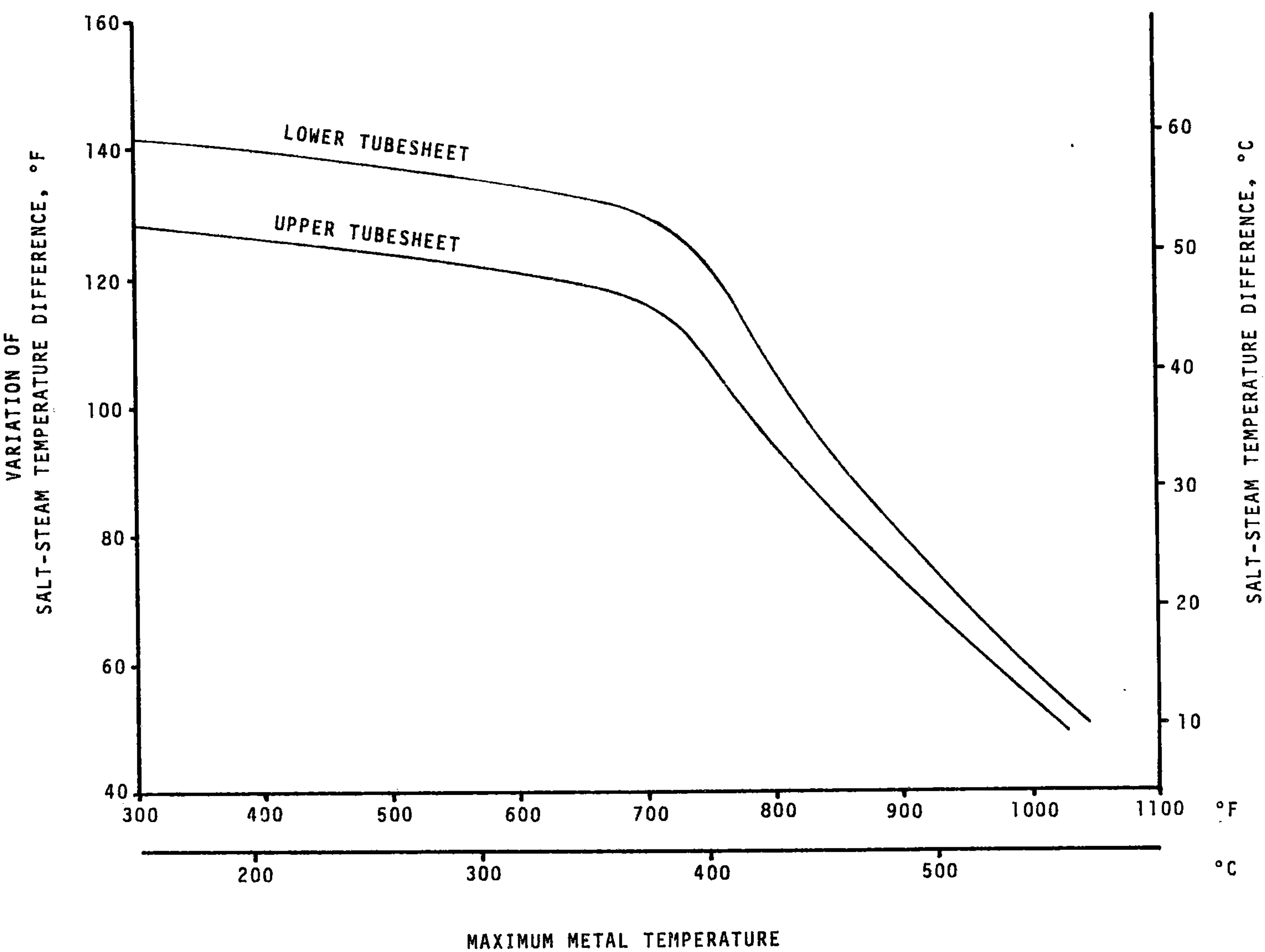


Figure 5.22 Allowable Salt/Steam Steady-State Temperature Variation

Lower-Tubesheet/Spherical-Head Junction. This analysis was similar to the upper-tubesheet analysis previously discussed. The model used is shown in Figure 5.23. We assumed the tubesheet temperature to be the same as the steam temperature--the outside temperature of the head to be T_{steam} and the inside temperature, T_{salt}. The displacements and strain ranges can be expressed as follows:

$$u_r = k_1 \alpha\Delta T \quad (24)$$

$$(\Delta\varepsilon)_{ts} = k_2 \alpha\Delta T \quad (25)$$

$$(\Delta\varepsilon)_{ss} = k_4 \alpha\Delta T \quad (26)$$

Using a computer program specially written for this analysis, we conducted a parametric study by varying the head and tube thicknesses. The results of this study are shown in Table 5.9. From this table, the predominant strain range is clearly the one in the spherical shell, and the value of this strain range is not

Table 5.9 Parametric Study--Lower Tubesheet Junction

Spherical Shell Thickness, t_{ss}	Tubesheet Thickness, t_{ts}: 190.5 mm (7.5 in.)	Tubesheet Thickness, t_{ts}: 304.8 mm (12 in.)
19.1 mm (0.75 in.)	$k_1 = 0.00974$ $k_2 = 0.00057$ $k_4 = 1.64854$	$k_1 = 0.00613$ $k_2 = 0.00036$ $k_4 = 1.64885$
25.4 mm (1 in.)	$k_1 = 0.01484$ $k_2 = 0.00087$ $k_4 = 1.64809$	$k_1 = 0.00938$ $k_2 = 0.00053$ $k_4 = 1.64857$

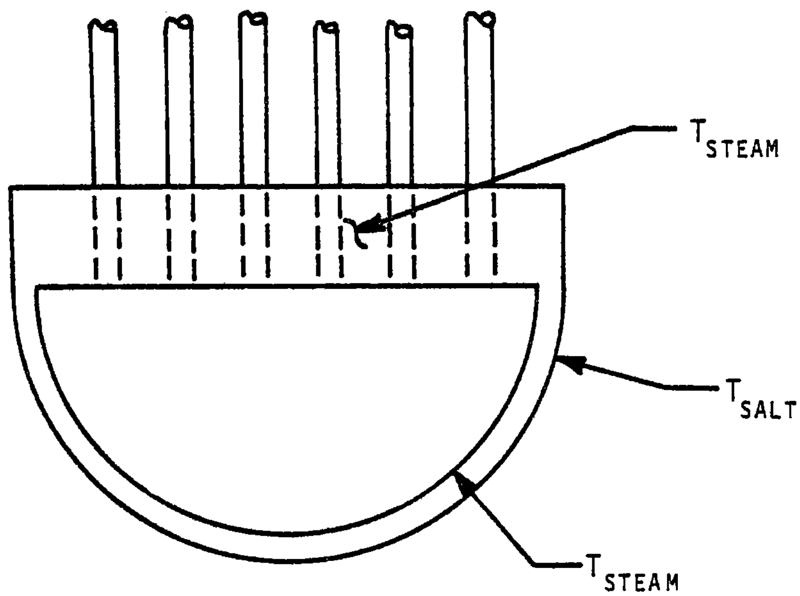


Figure 5.23 Lower Tubesheet/Head Junction Model

sensitive to variations in the thicknesses of the tubesheet and shell. Thus the maximum strain ranges can be approximated as follows:

Tubesheet: $0.0009\ \alpha\Delta T$

Head: $1.65\ \alpha\Delta T$

The allowable variations of (T_{salt} - T_{steam}) are plotted in Figure 5.22.

5.5.3 Transient Analysis

Transient analyses of the tubesheet and tubesheet junction were done to determine the allowable ramp rates (i.e., the rate at which the salt temperature is increased from the hot standby conditions to the peak operating condition). The hot standby temperatures, T_{hsb}, and the tubesheet thicknesses were varied parametrically. We assumed no steam flow, making this analysis very conservative. The allowable ramp rates as a function of (T_{salt} - T_{hsb}) are shown in Figure 5.24, where the shell/tubesheet junction can be seen as the most critical area. In this analysis the temperature distributions in the tubesheets were determined by using the Schneider Charts.[46] The computer models mentioned in Section 5.5.2 were used in analyzing the shell junction.

5.5.4 Buckling Analysis

When some of the tubes of the steam generator are plugged, the average temperature of these tubes is considerably higher than that of the unplugged tubes; consequently, these tubes are subjected to high compressive stresses, which may cause buckling. To preclude such a possibility, we did a buckling analysis. The average metal temperature of the unplugged tubes (T_{up}) was first

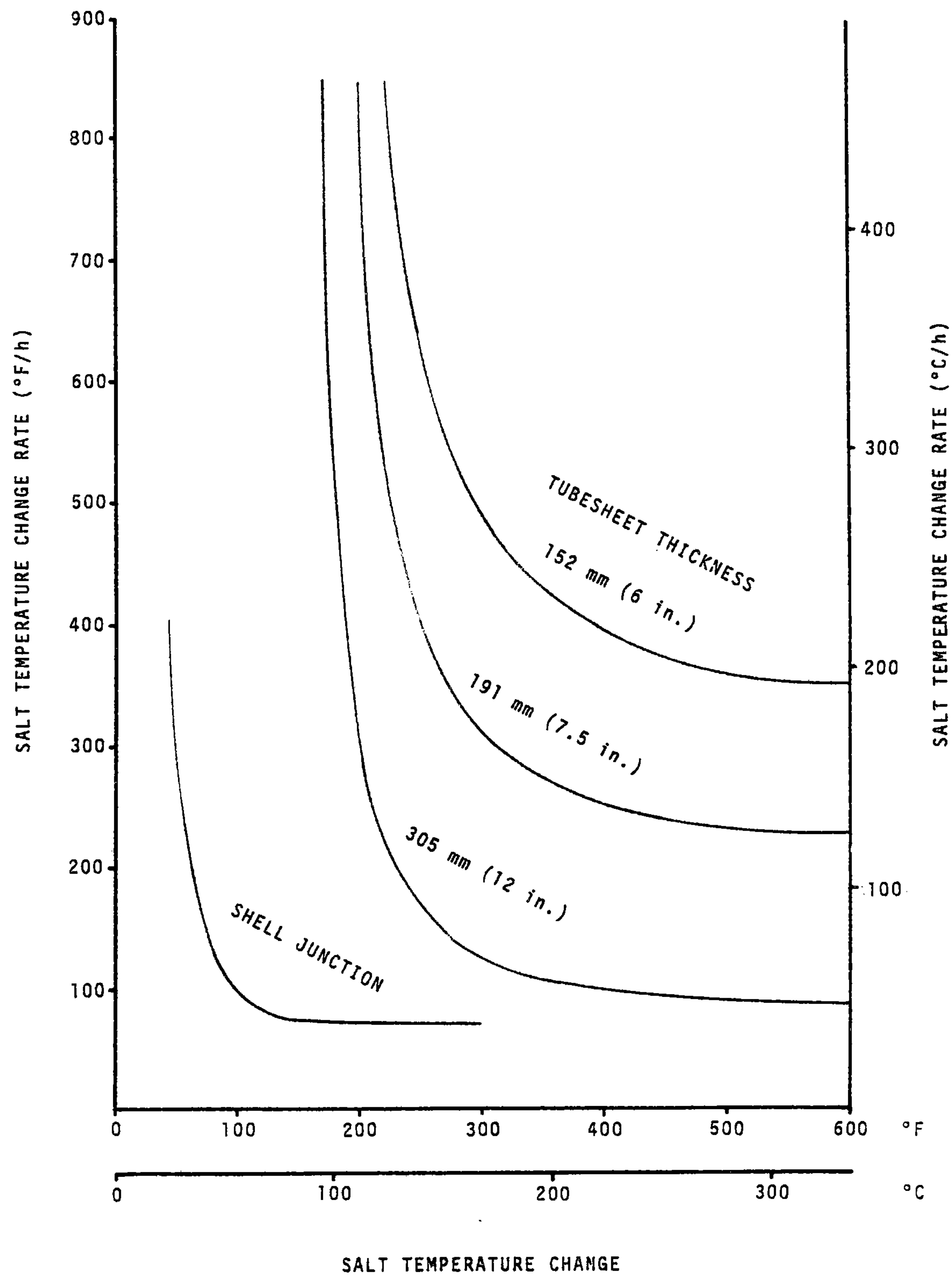


Figure 5.24 Allowable Rates of Heating

determined. The average metal temperature of the plugged tubes was T_{pl}. If p is the fraction of plugged tubes (i.e., number of plugged tubes divided by the total number of tubes), the compressive load on the plugged tubes is:

$$P = (1 - p)AE\alpha(T_{pl} - T_{up}) \tag{27}$$

The Euler buckling load for a column is given by:

$$P_c = \pi^2 EI/L^2 \tag{28}$$

where

L = Unsupported length of the tubes

Using the buckling criteria of Code Case N-47 and a strain factor of 1.67:

$$(1 - p)AE\alpha(T_{pl} - T_{up}) = P_c\ 1.67$$

or (29)

$$(T_{pl} - T_{up}) = P_c/1.67(1-p)AE\alpha$$

For p << 1:

$$(T_{pl} - T_{up}) = P_c/1.67AE\alpha \tag{30}$$

Substituting for P_c:

$$L = 1.216\ \ (r_i^2 + r_o^2\ \ a(T_{pl} - T_{up}) \tag{31}$$

The maximum allowable unsupported lengths for the reheater and superheater are 1.04 m (40.8 in.) and 0.52 m (20.6 in.) respectively.

5.5.5 Flow-Induced Tube Vibrations

In general, there are two major types of vibrations in a shell-and-tube heat exchanger. One involves exciting the natural frequency of the tubes and the other involves exciting the acoustic frequency of the shell. The natural frequency of the tube depends primarily on the span length, with long spans having low natural frequencies. The acoustic frequency depends upon the velocity of sound in the shell-side fluid and the shell I.D. Only when the shell-side fluid is a gas or vapor are acoustic frequencies of concern in shell-and-tube heat exchangers. Therefore, for present molten salt heat exchangers, only flow-induced tube vibrations were examined.

Preliminary vibration analyses were performed for all heat exchangers at full-load conditions using the HTRI ST-4 computer code.[23] This computer program first determines the shell-side flow distribution and then computes the natural frequency of tubes and various vibration prediction criteria such as vortex shedding frequency,[47] critical velocity,[48] damage numbers,[49] parallel flow vibration,[50] and vibration amplitudes of cross flow and parallel flow. The major results of the analyses are tabulated in Tables 5.10 and 5.11 for the 100-MWe solar stand-alone SGS and 50-MWe hybrid SGS. For preliminary analyses, the axial stress loadings on tubes were not considered. In general, the compressive stress reduces the natural frequency of the tube while the tensile stress increases the frequency. Tubes in heat exchangers can experience either tension or compression, depending on the operating conditions. Once the magnitude and direction of the load, compression, or tension are known, the effect of this loading can be further examined. However, the preliminary results indicate that all vibrational parameters are well below the criteria for vibration onset.

Table 5.10 Flow-Induced Tube Vibration Analysis--100-MWe Solar Stand-Alone SGS

Description	Preheater	Evaporator	Superheater	Reheater
Length for Natural Frequency, m (ft)	0.760 (2.49)	0.786 (2.58)	0.629 (2.06)	0.622 (2.04)
Tube Natural Frequency, Hz	41.80	61.17	61.35	81.16
Vortex Shedding Frequency, Hz	18.27	12.05	14.97	12.66
Cross-Flow Velocity, m/s (ft/s)	0.665 (2.18)	0.576 (1.89)	0.652 (2.14)	0.597 (1.96)
Critical Velocity, m/s (ft/s)	0.738 (2.42)	1.664 (5.46)	1.122 (3.68)	1.443 (6.54)
Vortex Shedding Frequency/ Tube Natural Frequency	0.437	0.197	0.244	0.156
Damage Numbers				
Baffle	0.3835	0.2469	0.1069	0.1589
Collision	0.1680	0.0346	0.0549	0.0259
Vibration Amplitudes, μm (in. x 10^6)				
Parallel Flow	4.37 (172)	1.55 (61)	4.72 (186)	3.96 (156)
Cross Flow	214.9 (8460)	38.35 (1510)	85.04 (3350)	28.45 (1120)

Table 5.11 Flow-Induced Tube Vibration Analysis--50-MWe Hybrid SGS

Description	Preheater	Evaporator	Superheater	Reheater
Length for Natural Frequency, m (ft)	0.761 (2.50)	0.800 (2.62)	0.629 (2.06)	0.620 (2.03)
Tube Natural Frequency, Hz	40.16	59.01	57.90	81.56
Vortex Shedding Frequency, Hz	17.75	8.62	16.91	8.24
Cross-Flow Velocity, m/s (ft/s)	0.604 (1.98)	0.433 (1.42)	0.573 (1.88)	0.390 (1.28)
Critical Velocity, m/s (ft/s)	0.695 (2.28)	1.606 (5.27)	1.009 (3.31)	2.006 (6.58)
Vortex Shedding Frequency/ Tube Natural Frequency	0.442	0.146	0.292	0.101
Damage Numbers				
Baffle	0.4743	0.1434	0.1506	0.0835
Collision	0.2173	0.0208	0.0928	0.0108
Vibration Amplitudes, μm (in. x 10^6)				
Parallel Flow	7.14 (281)	1.88 (74)	5.84 (230)	3.68 (145)
Cross Flow	179.30 (7060)	22.61 (890)	67.56 (2660)	11.68 (460)

Tables 5.10 and 5.11 indicate that, in all cases where the ratios of vortex shedding frequency to the natural frequency of the tube are far less than 1.0 and critical velocities are much larger than cross-flow velocities, vibration and fluid elastic whirling vibration are not significant. Tube failure caused by loosening at tube joints and cutting at baffles would not occur, as indicated by damage numbers which are much less than 1.0. The vibrational amplitudes attributed to either the parallel flow or the cross flow are much less than half the minimum gap between adjacent tubes. Therefore, the potential vibration would not lead to collisions of tubes at operating conditions.

In conclusion, the analyses indicate that all units are free of flow-induced vibrations at zero axial stress loading. The effect of axial loading on the natural frequency of the tube may be further analyzed. The onset of significant vibration could occur only when the natural frequency of the tube is drastically reduced.

5.6 SUBSYSTEM LEVEL ANALYSIS

This section defines the subsystem level analyses that were performed to ensure a safe, stable, and operationally functional steam generator system under start-up, shutdown, steady-state, and system upset conditions. The analyses considered all SGS components, piping, controls, and interface requirements with the balance of the plant. Both the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS were considered. Based on the analyses, we selected a control scheme and defined the modes of operations of the SGS subsystem. A dynamic model of the SGS system was developed to optimize the control scheme and predict subsystem response to load changes.

5.6.1 100-MWe Solar Stand-Alone Control and Operating Modes

Description. The 100-MWe solar stand-alone SGS receives hot, molten salt from the hot-salt storage tank to produce superheated steam for use in an Electric Power Generating System (EPGS). The SGS uses four separate heat exchangers to perform the preheating, evaporating, superheating, and reheating duties. A schematic of the heat exchanger arrangement and interconnecting piping, valves, and controls is shown in Figure 5.25.

All four heat exchangers are of the straight-tube type, with salt on the shell side and steam/water on the tube side. The preheater, superheater, and reheater are counterflow heat exchangers; the evaporator is a parallel-flow heat exchanger. A steam drum is located atop the evaporator to separate evaporated steam from the water.

Hot salt from a storage tank is pumped in parallel to the reheater and superheater vessels. After transferring heat to the reheat and main steam, the

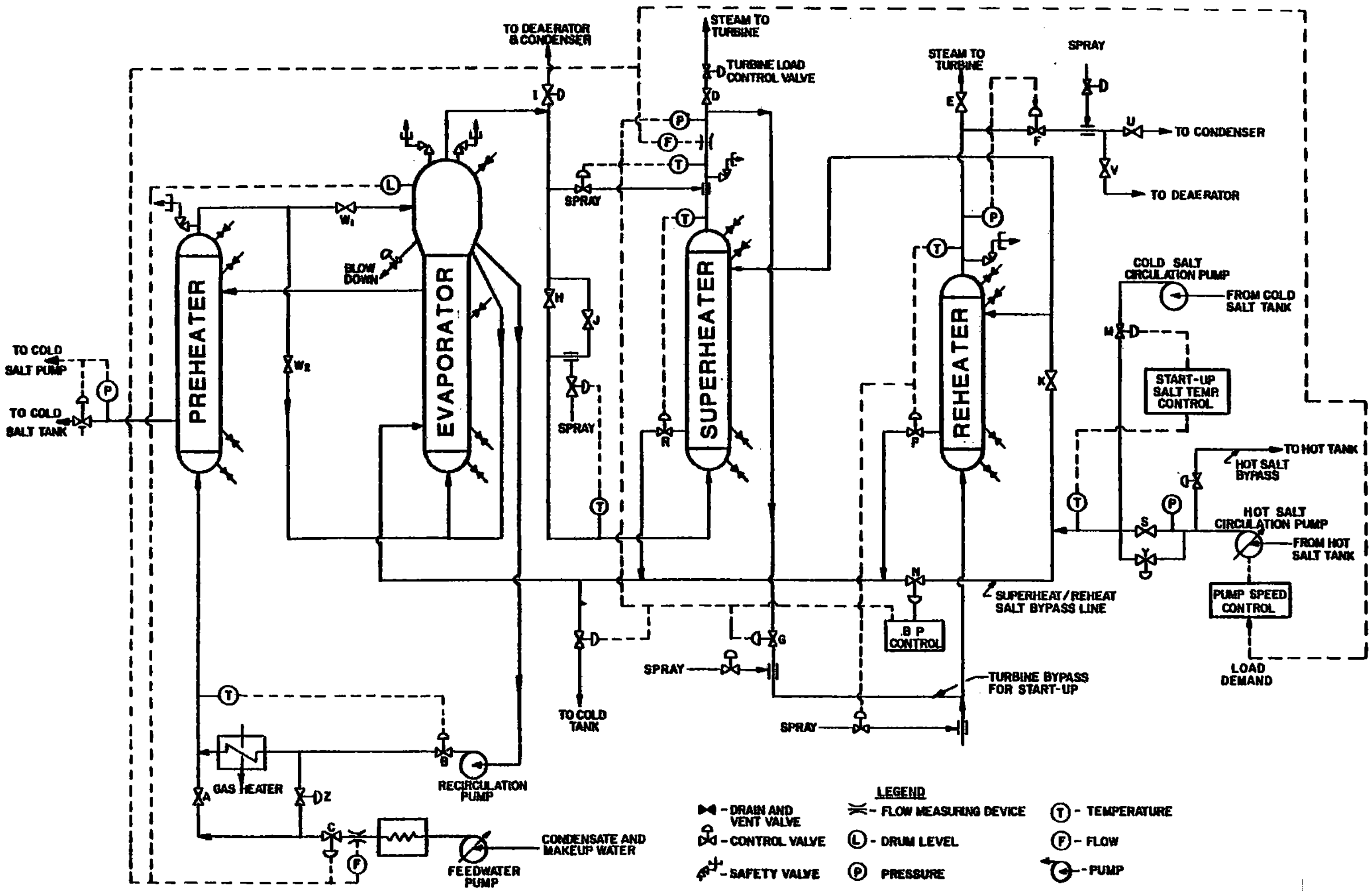


Figure 5.25 Control Diagram--100-MWe Solar Stand-Alone SGS

salt streams exiting from the reheater and superheater combine with a bypass hot-salt stream and enter the evaporator, where the hot salt gives up heat to evaporate water. The salt is then routed to the preheater, where the feedwater is heated. The salt exiting from the preheater is sent to a cold-salt storage tank for cycling to the solar receiver system.

A feedwater pump supplies treated water to the preheater, where it is heated before entering the evaporator. Saturated steam is generated in the natural-circulation evaporator and routed to the superheater, where it is superheated before passing to the high-pressure turbine for power generation. Intermediate-pressure steam from the turbine is brought to the reheater for further superheating and sent to the low-pressure turbine for additional work. The exiting steam goes to a condenser, and the condensed water is then recycled through the feedwater pump and preheated.

A recirculation pump and a gas heater in the evaporator circuit are used during part-load operation and system start-up. Table 5.12 lists the steam/water and salt valves and their primary functions. There are also vent, drain, and safety valves in the system.

Control System. The control system for the SGS (shown in Figure 5.25) uses interlocking controls to ensure safe and stable performance of the unit during experimental testing. System flow is regulated through a series of closed loops, which actuate control valves at designated locations. The control valves

Table 5.10 Flow-Induced Tube Vibration Analysis--100-MWe Solar Stand-Alone SGS

Description	Preheater	Evaporator	Superheater	Reheater
Length for Natural Frequency, m (ft)	0.760 (2.49)	0.786 (2.58)	0.629 (2.06)	0.622 (2.04)
Tube Natural Frequency, Hz	41.80	61.17	61.35	81.16
Vortex Shedding Frequency, Hz	18.27	12.05	14.97	12.66
Cross-Flow Velocity, m/s (ft/s)	0.665 (2.18)	0.576 (1.89)	0.652 (2.14)	0.597 (1.96)
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Hot salt from a storage tank is pumped in parallel to the reheater and superheater vessels. After transferring heat to the reheat and main steam, the

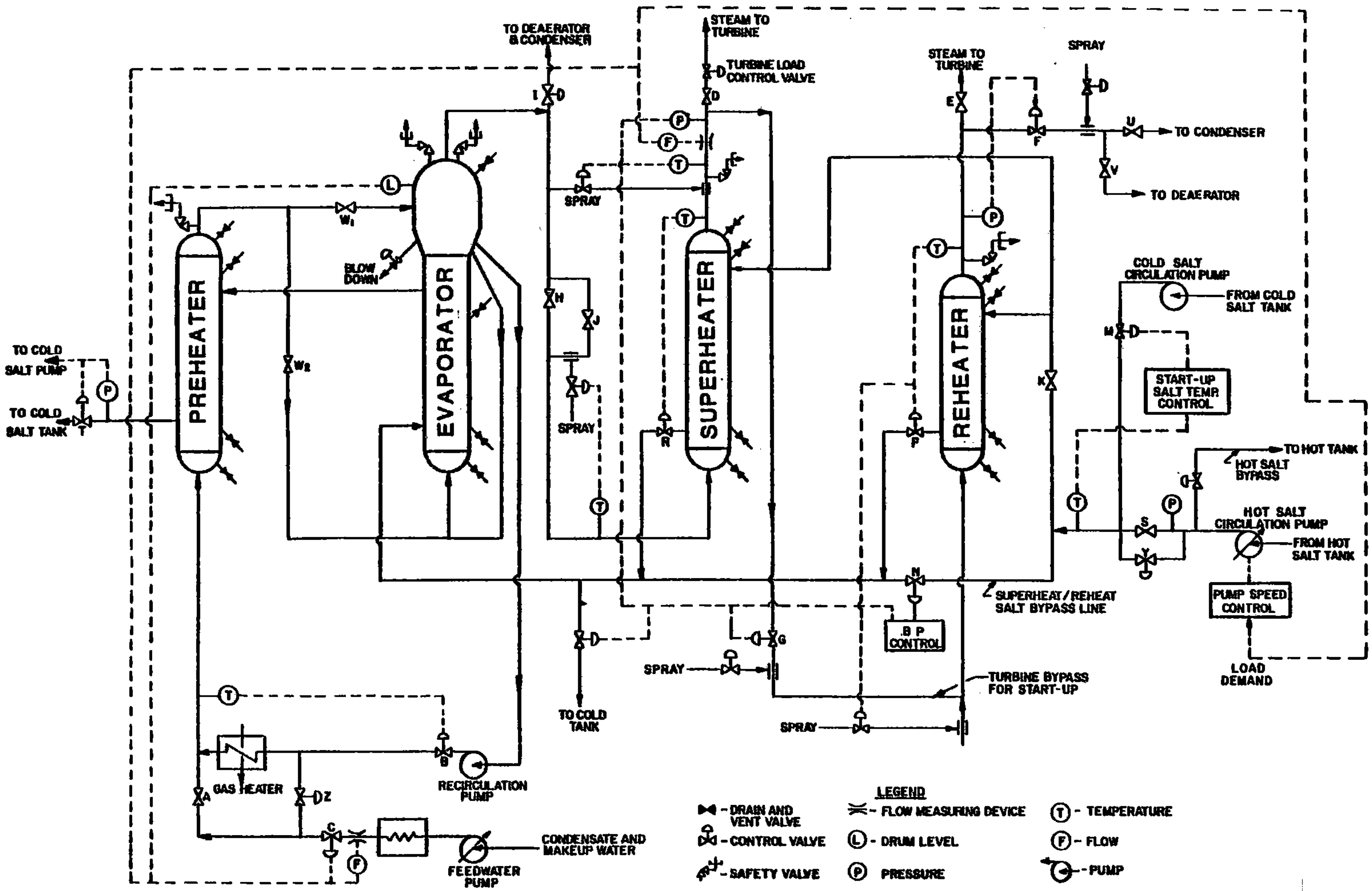

Figure 5.25 Control Diagram--100-MWe Solar Stand-Alone SGS

salt streams exiting from the reheater and superheater combine with a bypass hot-salt stream and enter the evaporator, where the hot salt gives up heat to evaporate water. The salt is then routed to the preheater, where the feedwater is heated. The salt exiting from the preheater is sent to a cold-salt storage tank for cycling to the solar receiver system.

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Control System. The control system for the SGS (shown in Figure 5.25) uses interlocking controls to ensure safe and stable performance of the unit during experimental testing. System flow is regulated through a series of closed loops, which actuate control valves at designated locations. The control valves

Table 5.12 100-MWe Solar Stand-Alone SGS Valves*

Valve	Function
A	Feedwater Shutoff
B	Water Recirculation Control
C	Feedwater Flow Control
D	Superheater Steam Shutoff
E	Reheater Steam Shutoff
F	Reheater Turbine Bypass
G	Superheater Turbine Bypass
H	Drum Steam Shutoff
I	Drum Steam Condenser Bypass
J	Drum Steam Letdown
K	Salt Shutoff to Reheater and Superheater
M	Cold Salt Flow Control
N	Salt Bypass Control
P	Reheater Salt Flow Control
R	Superheater Salt Flow Control
S	Hot Salt Shutoff
T	Cold Salt Flow Shutoff
U	Condenser Bypass
V	Deaerator Bypass
W_1, W_2	Start-Up Evaporator Water Flow
Y	Hot Salt Start-Up Bypass
Z	Feedwater Start-Up Bypass

*See Figure 5.25 for valve location.

are identified in Table 5.12. A summary of the control philosophies for the SGS follows:

- Drum Water Level Loop. The flow of feedwater to the heat exchanger drum will be under three-element control.. The loop utilizes measurements of feedwater flow, steam flow, and drum-water level and feeds these parameters to a Program and Logic Controller (PLC). The output from this controller regulates the position of the feedwater control valve.

- Superheater Outlet Temperature Loop. A temperature indicator located on the outlet side of the superheater senses the outlet temperature. The signal is interfaced with the PLC to determine the proper analog setting for the superheater salt flow-control valve. Regulation of the salt flow controls outlet temperature over a specified range. Beyond this range, the PLC activates the saturated steam spray mechanism, which further reduces any excess temperature. A temperature indicator at the outlet of the superheater, upstream of the junction with the saturated steam spray line, controls this spray mechanism.

- Reheater Outlet Temperature Loop. The reheater outlet temperature loop and the superheater control loop are similarly controlled; that is, the temperature indicator on the outlet side of the reheater sends signals to the PLC which, in turn, positions the reheater salt flow-control valve. However, the spray-control mechanism injects spray water on the inlet side of the reheater in contrast to injecting it on the outlet side of the superheater.

- Superheater Pressure Loop. Steam pressure is regulated by controlling the salt flow through the bypass valve. A transducer located on the outlet side of the superheater monitors superheater pressure for controlling the bypass valve. A pressure drop causes the salt bypass valve to open and admit more salt and hence more heat to the evaporator. The result is increased steam generation and the required pressure rise.

- Trace Heater Loop. Trace heaters utilizing single-element control are used on all heat exchangers and salt lines to maintain a 260°C (500°F) temperature during cold start-up.

- Recirculation Pump. A drum-water recirculation pump maintains the feedwater at a temperature above the salt freezing point [238°C (460°F)] during part-load operation.

- Feedwater Pump. A bypass line maintains minimum flow through the feedwater pump during periods of low-load demand.

- Pressure-Reducing and Spray Station. Located in the steam line between the superheater and reheater, this station ensures that the steam entering the reheater during start-up and emergency shutdown is at proper pressure and temperature. A pressure-reducing and spray station in the steam bypass line between the evaporator and the superheater ensures that proper steam temperature and pressure enter the superheater during start-up.

- Start-Up Salt Temperature Control Loop. Proper salt temperature during start-up is ensured by manipulating the valves at the exit of the hot- and cold-salt storage tanks.

Operating Modes. The SGS is designed for various steady-state and transient modes of operation. A typical operating cycle for the SGS is illustrated in Figure 5.26. In describing the SGS operating cycle, we have assumed the starting point to be the Cold Shutdown Mode (State 7), where the whole SGS (including the connecting piping) is at ambient condition. We have also assumed that at least 1 hour of full-load-operation hot salt is available in the hot-salt storage tank before a cold start-up can begin.

As described later in detail, the SGS will be brought from the Cold Shutdown Mode to Full-Load Operation (State 1) using the hot salt from the hot-salt storage tank. At this time, the solar receiver system will begin supplying fresh hot salt to the storage tank for continued operation at full load. The solar receiver system and the hot-salt storage tank are designed so that the quantity of hot salt produced during 10 hours of daily sunshine is sufficient to operate the SGS at full load for 10 hours and provide 6 hours of full-load hot salt in the tank at the end of the 10-hour Full-Load Operation. Beyond 10 hours, the SGS continues to operate at full load for 5 more hours before beginning Diurnal Shutdown to the Hot Standby Mode (State 5). The remaining supply of hot salt in the storage tank is used for Start-Up, initial Full-Load Operation, and Shutdown. During daily operation the SGS may operate at any part-load condition.

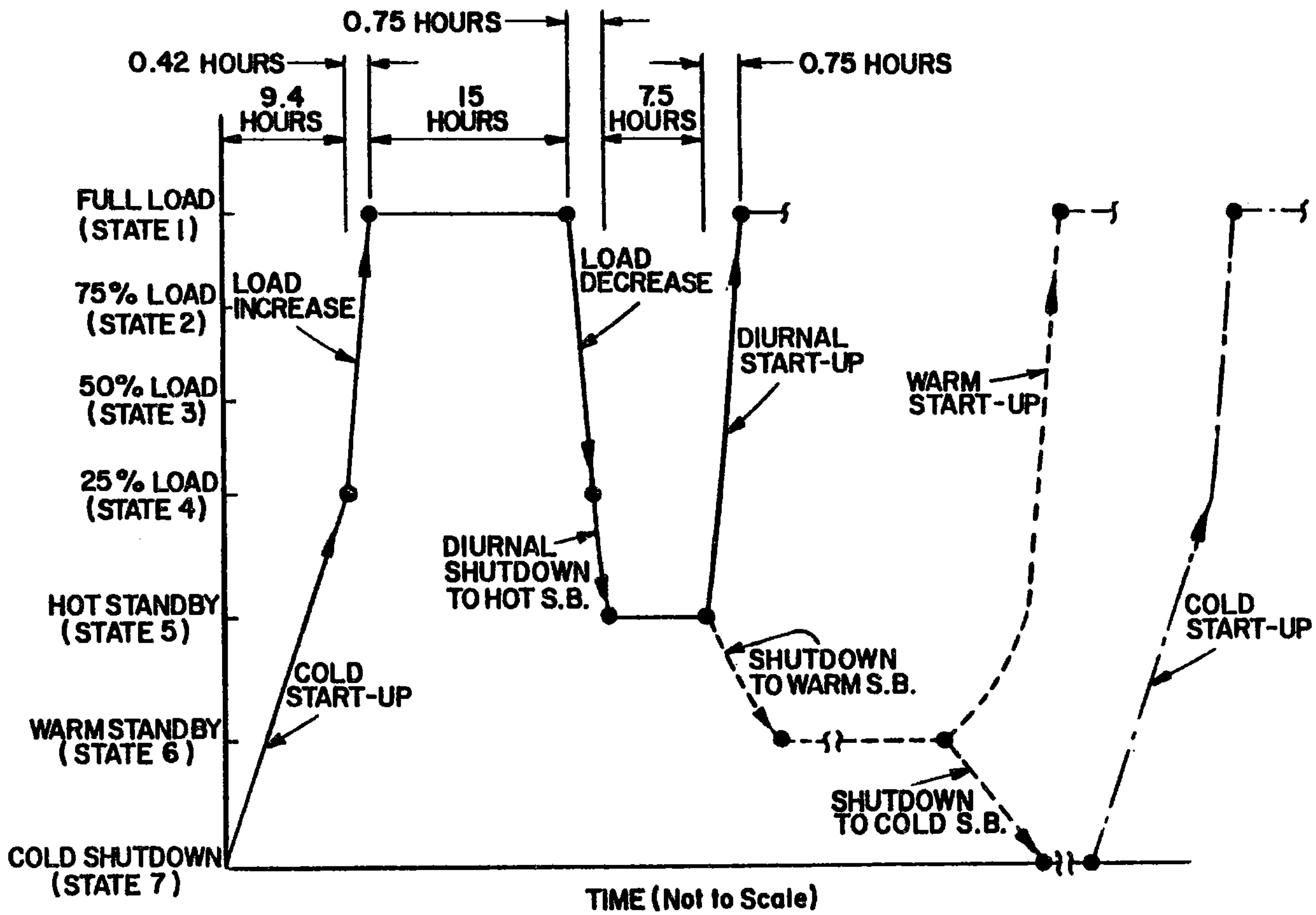

Figure 5.26 Operating Modes--100-MWe Solar Stand-Alone SGS

The Hot Standby Mode consists of bottling the SGS on the steam/water and salt sides so that all flows to and from the SGS stop. The SGS is allowed to remain under these conditions for 7.5 hours, after which Diurnal Start-Up can begin. Diurnal Start-Up takes the SGS from the Hot Standby Mode to the Full Power Mode. This daily operating cycle is continually repeated unless an extended shutdown or cold shutdown is necessary.

It may be necessary to bring the SGS to a Warm Standby Mode (State 6) from the Hot Standby Mode because of extended cloud cover. In the Warm Standby Mode, the SGS continues to cool from the Hot Standby Mode, losing heat to ambient. The SGS remains in this mode until either a fresh supply of hot salt becomes available (in which case the SGS is brought to full load) or the SGS is drained of salt and water and cooled down to the dry, ambient long-term Cold Shutdown Mode for maintenance or other reasons. From the Cold Shutdown Mode, the SGS is brought to full load as described earlier.

Table 5.13 predicts how the 30-year life of the SGS is apportioned among various SGS operating modes. Table 5.14 gives predicted data on "state" changes (transitions) between various operating modes during the SGS lifetime. The data in Tables 5.13 and 5.14 will be used in the SGS design.

The following sections present detailed descriptions of the various operating modes and describe the procedures for transition between them. The systems diagram (Figure 5.25) is used to describe various sequences of events and control parameters.

Table 5.13 Operating Modes Time Distribution*

Operating Mode	State	Time (Hours)
Full Load	1	80,000
75% Load	2	20,000
50% Load	3	20,000
25% Load	4	20,000
Hot Standby	5	77,000
Warm Standby	6	20,000
Cold Shutdown	7	26,000
Total		263,000

*Durations of transition operating modes are lumped with the steady-state SGS modes.

Table 5.14 Operating Mode Transition Occurrences

Transition Mode	Number
Load Change	18,120*
Diurnal Start-Up	9,000
Diurnal Shutdown	9,000
Warm Start-Up	167
Warm Shutdown	167
Cold Start-Up	60
Cold Shutdown	60

*Depends on load demand.

Cold Shutdown (State 7). The SGS is under dry, ambient conditions with steam/water and salt sides filled with nitrogen at slightly above atmospheric pressure. All drain and vent valves are closed, and power to the trace heaters is off. The system is isolated from the Electric Power Generation System (EPGS) by closing Valves D and E. The water- and salt-supply systems are isolated by closing Valves A, B, C, M, S, Y, and T. Control Valves F and I to the condenser are also closed, as is the blowdown valve from the drum. All water and salt pumps are off.

Cold Start-Up (State 7 to State 1). This procedure describes the transition between the Cold Shutdown Mode (State 7) and Full-Load Operation (State 1). At the beginning of this transition, the valve positions are as described in the Cold Shutdown Mode. All controls are set for manual operation.

The procedure begins by closing Valve H, opening Valves A and C, and starting the feedwater pump which supplies 4.68 MPa absolute (681 lb/in^2a) 27°C (80°F) feedwater to the preheater and the evaporator. Start-up Valve W_1 is closed and Valve W_2 is opened to route water to the evaporator from below. The preheater and evaporator are filled with water until the desired drum-water level is achieved. The vent valves are opened to expel nitrogen. Valves B and Z are opened and Valve A is closed. Valve Z controls feedwater flow so that the drum-water level is maintained. The recirculation pump is started to circulate ambient water through the preheater and evaporator at approximately 12-percent flow. At this point a gas heater at the discharge of the water recirculation pump is used to heat the water. The heated water passes through the preheater and the evaporator. Water from the drum is routed back to the recirculation pump.

Trace heaters on the entire SGS are started and controlled so that the magnitude of the temperature difference between the shell and tubes of any heat exchanger does not exceed 56°C (100°F). Heating of water using the gas heater continues until the water temperature reaches 238°C (460°F), which is 22°C (40°F) below the saturation temperature. By this time trace heating has brought the preheater and evaporator shell and associated piping temperatures to 288°C (550°F). Valve I to the deaerator is opened, Valve K is closed, and 343°C (650°F) salt from the salt blending system is admitted to the evaporator and preheater by opening bypass Valve N. (Low-load salt blending is obtained by regulating Valves Y and M; Valve S remains closed.) Opened Valve T allows approximately 260°C (500°F) salt to flow from the preheater exit to the cold-salt tank. The gas heater and the trace heaters in the preheater-evaporator subsystem are shut off and Valve B is controlled to ensure 238°C (460°F) water to the preheater. Valve W_1 is opened and W_2 is closed to allow a normal flow of water to the evaporator.

Salt flow is adjusted to obtain 5-percent steam flow at approximately 260°C (500°F). The vent valves are closed once all nitrogen has been expelled. Valve C is regulated to maintain drum-water level. Steam from the drum is used for heating feedwater and for initial turbine warm-up. (Initial turbine warm-up takes approximately 6 hours.) Letdown Valve J and associated water spray in the steam bypass line to the superheater are used to allow reduced-pressure superheated steam at 34.5 kPa absolute (5 lb/in²a) and 116°C (240°F) to enter the superheater and reheater. The superheater and reheater shell temperatures should be at approximately 60°C (140°F) before admitting steam. The steam from the superheater goes to the reheater through Valve G.

Some steam condenses in the superheater and reheater and is removed using drain valves. The entrained nitrogen is expelled by opening vent valves which are closed once that act is completed. Opened Valve F allows steam to exit for feedwater heating and turbine warm-up.

Water spray downstream of Valve J and trace heating on the superheater and reheater are controlled so that the temperature of low-pressure steam to the superheater and reheater is increased gradually to approximately 260°C (500°F) and the shell temperature reaches 288°C (550°F). The steam flow to the superheater is then switched from the bypass line (Valve J) to the normal line (Valve H), raising the pressure in the superheater to approximately 4.68 MPa absolute (681 lb/in^2a). Letdown Valve G ensures that steam pressure to the reheater does not exceed 3.34 MPa gage (485 lb/in^2g) at any time. By opening Valves K, P, and R, 343°C (650°F) salt at 5-percent flow is admitted to the reheater and superheater. Valve N is adjusted so that the total salt flow and temperature to the evaporator ensure continued generation of 260°C (500°F) saturated steam at 5-percent flow. The salt temperature from the blending system is increased at the rate of 83°C (150°F)/h.

Steam flow at 5 percent is established, and the steam temperature increases as a result of the increased salt temperature. Water sprays downstream of Valve G and at the inlet to the reheater ensure that the temperature of steam entering the reheater does not exceed 288°C (550°F) at any time. When the main steam temperature reaches 399°C (750°F), the turbine load-control valve admits main steam for turbine rolling by opening Valve D and closing Valve G. High-pressure turbine exit steam enters the reheater and reheated steam enters the low-pressure turbine by opening Valve E and closing Valve F. Rolling is established in approximately 15 minutes. The turbine is synchronized when the main steam temperature reaches 468°C (875°F), and it remains so for approximately 30 minutes.

Having established 5-percent load with 468°C (875°F) steam, the load and steam temperature and pressure are increased linearly to 25 percent, 541°C (1005°F), and 13.46 MPa gage (1955 lb/in²g) by increasing the salt temperature to 563°C (1045°F) and salt flow to its 25-percent value. The increased feedwater and salt flow are accommodated by opening Valves A and S and closing Valves Z and Y. The load should be increased at the rate of 2 percent/min while making sure that the turbine steam-to-metal temperature mismatch remains within -28°C (-50°F) and 56°C (100°F). All controls are now set on automatic, as described in the section on full- and part-load operation which follows. Beyond the 25-percent load level, the load is increased to full load at the rate of 3 percent/min using the procedure described in the section on load changes. The cold start-up procedure is shown schematically in Figure 5.27.

Full- and Part-Load Operation (States 1-4). The normal SGS operating load range will be from 25 to 100 percent. Below 25-percent load, the SGS will be in a start-up or shutdown mode. Steam/water and salt parameters at full and part load for the four heat exchangers are given in Figure 5.14.

For part- or full-load operation, the feedwater is pumped to the preheater, the flow being controlled by the superheated steam flow and drum-water level. Any deviations from these parameters are eliminated by feedwater flow adjustment. To preclude salt freeze-up in the preheater, the temperature of

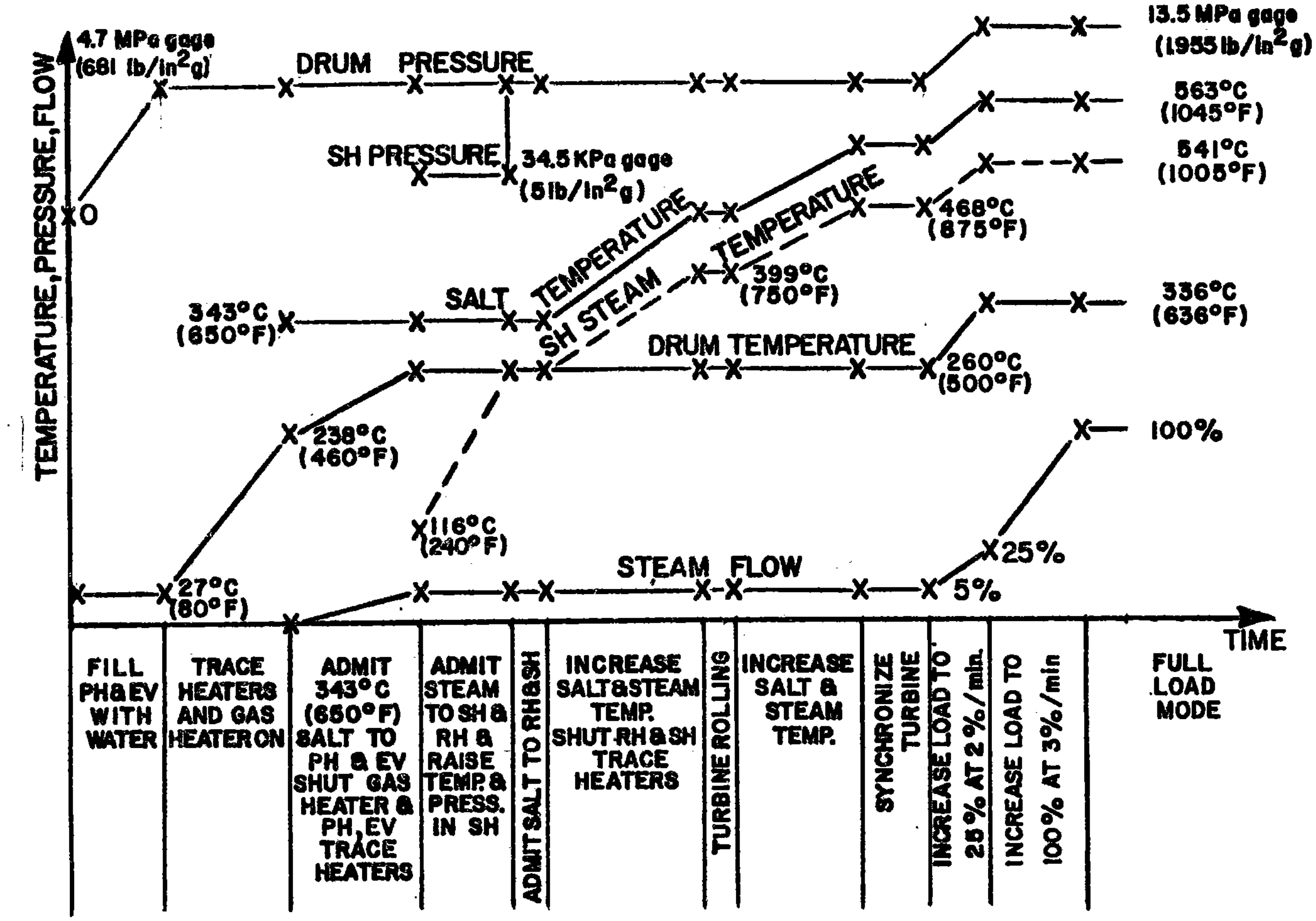

Figure 5.27 Cold Start-Up--100-MWe Solar Stand-Alone SGS

the feedwater entering the preheater is kept at 238°C (460°F) minimum, using the recirculation pump from the evaporator. The preheated water enters the steam drum in the evaporator and is circulated, through the downcomer, by natural circulation. Saturated steam from the steam drum at 336°C (636°F) enters the superheater, where it is superheated to 541°C (1005°F). The superheated steam enters the high-pressure turbine to do work. A saturated steam spray from the steam drum exit to the superheater steam exit is used for emergency temperature control. The superheated steam temperature is controlled primarily by regulating the salt flow through the superheater (by adjusting Valve R). Superheater outlet pressure is maintained at the set point by controlling Valve N on the superheater/reheater salt bypass line.

Steam from the high-pressure turbine at reduced temperature and pressure enters the reheater. The reheated steam exit temperature is controlled by salt flow through the reheater (adjustment of Valve P on the reheater salt line). Spray control at the reheat inlet is also available to moderate the reheat steam exit temperature. All controls are on automatic during full- and part-load operation.

Load Changes. The SGS will be designed to operate at any load between full and 25 percent. Various operating parameters at 25-percent load are defined in Figure 5.14. The load changes between these operating points will be achieved by adjusting the feedwater and salt flows using the automatic control logic.

For an increased load demand, the main turbine throttle valve opens further, resulting in increased steam flow. This increase is the signal to the feedwater system for increased feedwater flow. The higher steam flow also causes reduction in steam pressure and temperature and leads to a reduction in drum-water level. (The drum-water level drops after an initial rise caused by the reduced pressure.) The reduced drum-water level combines with the increased steam flow to demand higher feedwater flow. The reduced-pressure signal adjusts the salt bypass control (Valve N) in anticipation of the increased load. The reduced steam temperature signal adjusts the superheater salt flow (Valve R), allowing higher salt flow. A similar adjustment in temperature is also made on the reheater. As a result of these control events, the SGS approaches a condition of thermal equilibrium at a higher load after some initial transient behavior.

Similar logic in reverse order is used for reduction in load demand. All load changes are limited to 3 percent/min.

Diurnal Shutdown (State 1 to State 5). When the hot-salt storage tank level reaches the 1-hour mark, a procedure is initiated for daily shutdown. The SGS is brought down at 3 percent/min to 25-percent load and at 2 percent/min beyond that. The steam generator is tripped at 15-percent load. At the trip point, the turbine first-stage temperature drops to 482°C (900°F). The superheater salt inlet and steam outlet temperatures are 563 and 541°C (1045 and 1005°F). The steam generator is isolated by shutting off Valves C, D, B, E, A, S, K, N, and T. The reheater bypass Valve G is set on automatic and at 14.63/3.34 MPa gage (2125/485 lb/in²g). The condenser bypass Valve F is set on automatic and at 3.44/0 MPa gage (500/0 lb/in²g). The valves keep the pressure surge in the system from exceeding these limits. Over a 7.5-hour shutdown period (hot standby mode) caused by heat distribution, the salt and steam reach a common temperature at every level in the system. The temperatures at the top and bottom of the superheater are 544 and 410°C (1011 and 770°F) respectively after the 7.5-hour cooldown period. The temperatures at the other levels vary linearly between these values. During the same time, the reheater salt and steam temperatures reach 540°C (1003°F) at the top and 405°C (761°F) at the bottom. The drum cools down to 13.09 MPa gage (1900 lb/in^2g), corresponding to a saturated steam temperature of 331°C (628°F). The turbine first-stage temperature cools down to 441°C (825°F) during this period. The temperature of salt and water in the preheater is 269°C (517°F) at the bottom and 337°C (640°F) at the top as a result of cooldown.

Diurnal Start-Up (State 5 to State 1). The hot-salt storage level should be at approximately the 45-minute mark from the previous day's shutdown procedure. When the hot-salt storage tank level reaches the 1-hour mark, daily start-up procedures begin. We assume that feedwater at only 105°C (220°F) is available at the outlet of the deaerator.

For start-up, all valves are on manual. Salt is brought in at 443°C (830°F) through the bypass line into the evaporator. Salt flow should be at 4 percent of full-load flow. The desired temperature will be obtained by mixing the salt from the hot- and cold-salt storage tanks using Valves M and Y. Salt is circulated through the evaporator and the preheater. Valve W_1 should be open; Valve W_2 should be shut. Steam generated is sent to the deaerator (by opening Valve I) and heats the feedwater. Valve B is opened and the recirculation pump is started. To balance the drum-water level, feedwater enters the system through Valves C and Z and is heated to 238°C (460°F) by an auxiliary heater and by mixing it with hot water at 331°C (628°F) drawn from the drum through Valve B. Valve B is regulated so that the feedwater temperature is maintained at 238°C (460°F) before it enters the preheater.

When approximately 4-percent steam is generated, Valves I and N are shut and Valve H is opened. The cold-salt circulation pump is off and Valve M is shut. Valves R, P, and N are opened to allow 563°C (1045°F) salt to enter the reheater and superheater; 443°C (830°F) salt enters the evaporator. Valves F, G, and V are opened. Steam now flows through the superheater and reheater and through the bypass line to the deaerator to heat the feedwater. Salt flow is increased to generate 5-percent steam. By adjusting the spray at the outlet of the superheater and the inlet of the reheater, the steam temperature at the superheater and reheater outlets is brought to 496°C (925°F) so that the steam entering the turbine matches the turbine first-stage temperature of 441°C (825°F). Valves G, F, and V are closed; Valves D and E and the turbine load-control valve are opened. Steam enters the turbine for rolling, which is established in 15 minutes. The 5-percent steam is held at 496°C (925°F) for 15 minutes to synchronize the turbine. Having established the 5-percent load with 496°C (925°F) steam, the load and steam temperature are increased linearly to 25 percent and 541°C (1005°F) by increasing the salt flow to its 25-percent value and by decreasing the sprays. The increased feedwater and salt flows are accommodated by opening Valves A and S and closing Valves Z and Y. The load should be increased at 3-percent/min while making sure that the turbine steam-to-metal temperature mismatch remains within -28°C (-50°F) and 56°C (100°F).

When the feedwater temperature reaches 238°C (460°F), Valve B is closed and the recirculation pump is shut off. Valve C is on automatic.. The steam generator is now fully on automatic for load-change to a higher load.

Shutdown to Warm Standby Mode (State 5 to State 6). As described earlier, the SGS is brought daily to an overnight hot standby shutdown after the hot-salt level in the storage tank reaches a 1-hour supply. Using part of this salt supply, the SGS is started the next day (assuming that fresh salt supply from the solar receiver system is available for the daily full-load operation). However, cloud cover may prevent the solar receiver system from supplying the required hot salt. If cloud cover extends for a day, a few days, or weeks, the SGS will have to be brought to a warm standby mode, where it will remain until sufficient fresh, hot salt is available.

The warm standby mode is simply an extension of the hot standby mode. The valve positions are the same as for hot standby. The SGS continues to lose heat to the atmosphere so that the shell temperatures for the preheater and evaporator drop at 0.55°C (1°F)/h and the superheater and reheater shell temperatures drop at 1.7°C (3°F)/h and 2.2°C (4°F)/h respectively. In addition, slow heat conduction in the longitudinal direction within each heat exchanger continues, tending to bring each vessel to an isothermal condition. The reduced temperatures lead to steam/water-side pressure reduction and some steam condensation takes place. The SGS remains in a transient cooling mode for a long time, and the final SGS conditions before restart are dependent upon the time the SGS has been in this transient mode.

From a design viewpoint, the limiting condition is when transient cooldown is given sufficient time to reach a steady-state condition. Another variable used in arriving at the warm standby steady-state condition is that the trace heaters are turned on if the metal temperature drops below 288°C (550°F), keeping the system warm and preventing freezing of salt in the SGS. To compensate for any salt contraction, cold salt is pumped into the system through the preheater salt outlet using a salt-pressure signal.

Warm Standby Mode (State 6). Based on the previous discussion, the whole SGS is at a uniform 288°C (550°F) in this mode. This temperature is maintained by trace heaters on the SGS heat exchangers and piping.

As a result of the reduced [288°C (550°F)] temperature, the pressure on the main steam/water-side drops to the saturation value of 7.23 MPa gage (1050 lb/ in^{2}g). The steam in the reheater remains in the superheated state. Some condensation occurring in the superheater is removed by drain valves.

The SGS remains in this state until either a start-up for full- or part-load operation is desired or shutdown to cold, dry conditions is necessary.

Start-Up From Warm Standby Mode (State 6 to State 1). The initial conditions for start-up from warm standby are that the whole SGS is at 288°C (550°F) and Valves A, B, C, D, E, I, S, and T are closed. Valves G and F are at their pressure settings of 14.63/3.34 MPa gage (2125/485 lb/in^{2}g) and 4.58/0 MPa absolute (665/0 lb/in^{2}a). The turbine is cold at ambient conditions. This start-up procedure begins when at least a 1-hour supply of hot salt is available in the hot-salt tank.

Valve H is closed to isolate the superheater and reheater; Valve C is opened, and treated water at 27°C (80°F) is admitted through Valve Z to the gas heater for heating to 238°C (460°F). Supplemental heating to reach 238°C (460°F) is supplied by controlling Valve B to circulate water in the evaporator. The purpose is to ensure that the feedwater entering the preheater is at least 238°C (460°F) to preclude salt freeze-up. Simultaneously, Valve T is opened and, by blending hot and cold salt using Valves M and Y, 343°C (650°F) salt is admitted to the evaporator/preheater through Valve N. Valve I is opened to release generated steam to the deaerator for feedwater heating and turbine warm-up. The drum-water level is maintained by admitting 238°C (460°F) feedwater to the preheater by adjusting Valve Z. This mode continues until 4-percent steam generation at 288°C (550°F) to the deaerator is achieved by adjusting salt flow.

Valve H is opened and Valve I closed, allowing the steam to flow through the superheater and reheater and then to the condenser or deaerator. Valves P and R are adjusted and 343°C (650°F) salt at 5-percent flow is admitted through the reheater and superheater. Trace heaters are shut off. The temperature of the salt to the SGS is increased by 83°C (150°F)/h using the hot- and cold-salt blending system. The steam temperature continues to increase as a result of the increased salt temperature. [Water sprays downstream of Valve G and at the inlet of the reheater ensure that the temperature of steam entering the reheater does not exceed 288°C (550°F) at any time.] The steam flow is established at 5 percent. The steam from the SGS is available during this start-up for turbine warm-up, as required.

When the main steam temperature reaches 399°C (750°F), the turbine load-control valve admits steam for turbine rolling. (Valve D is opened and Valve G is closed.) Valve E is opened and Valve F is closed for normal reheat steam flow. [We have assumed that the turbine has been warmed up and is ready to admit the 399°C (750°F) steam.] Turbine rolling is established in about 15 minutes. The turbine is synchronized when the main steam temperature reaches 468°C (875°F) and is held for approximately 30 minutes. Valves A and S are opened so that the feedwater and salt flow can be increased beyond 5 percent. The gas heater is off and Valves Z and Y are closed.

Beyond this, the steam temperature is linearly ramped to 541°C (1005°F) at 25 percent of steam flow by increasing the salt temperature to 563°C (1045°F) and salt flow to its 25-percent value. The load is increased at 2 percent/min while making certain that the turbine steam-to-metal temperature mismatch remains within -28°C (-50°F) and 56°C (100°F). The cold-salt pump is shut off and Valve M is closed. The recirculation pump continues to monitor and control the temperature of the feedwater entering the preheater at 238°C (460°F). Beyond 25-percent load, the load is increased to full load at the rate of 3 percent/min using the load-change procedure described earlier.

Shutdown to Cold Conditions (State 6 to State 7). This shutdown begins at warm standby and terminates in long-term cold shutdown of the SGS in a cold, dry, ambient state.

The procedure begins when trace heaters are shut off and salt is drained from the SGS. All salt-side drain and vent valves in the heat exchangers and the piping are opened. The salt is collected and routed to the cold-salt tank. The salt side of the SGS is then purged with nitrogen and drain and vent valves are closed, leaving nitrogen entrained in the SGS.

The second step is to drain the water from the SGS, but only after the salt-side draining has been completed. This is accomplished by opening Valves F and I to allow steam to go to the condenser. The system pressure will thus continue to drop, resulting in generation of more steam, which will be routed to the condenser. After atmospheric pressure is achieved, Valves F and I are closed, nitrogen is admitted through the vents, and the drain valves on the water side of the SGS are opened to drain any water.

The system is then purged with nitrogen and all drain and vent valves are closed, leaving nitrogen entrained on the steam/water-side of the SGS heat exchangers and piping. The system is allowed to cool down to ambient conditions. More nitrogen is supplied to the SGS to keep pressure slightly above atmospheric.

Emergency Transient Conditions. Described here are the procedures and safeguard mechanisms built into the control system for safe, controlled shutdown resulting from the following events:

Event	Occurrences
Turbine trip	30
Loss of feedwater flow	0
Loss of salt flow	0
Rupture of any steam/water/salt pipe	0
Rupture of a steam/water tube	2
Loss of pneumatic pressure	5
Failure of control systems	2
Loss of all station power	30

- Turbine Trip. When the turbine trips, the turbine control valve automatically slams shut, resulting in a sudden surge of pressure and temperature in the steam generator. A signal from the master control shuts the salt pump off when the turbine trips. The pressure surge is sustained by the safety valve, relief valves, and Valves G and F. These bypass valves (G and F) open when the pressure reaches 14.63 MPa gage (2125 lb/in^2g) at the superheater outlet and 3.2 MPa gage (465 lb/in^2g) at the reheater outlet. (These valves are designed for 25-percent of full-load steam flow.) The surge in steam temperature is sustained by Valves R and P, which close automatically, and also by sprays. The feedwater pump under the three-element control shuts off when the steam flow reaches 15-percent load. The steam generator is isolated by closing Valves A, D, E, S, Y, and T.

- Loss of Feedwater Flow. There are many fail-safe mechanisms built into the SGS to prevent this from happening. There are three 50-percent-capacity feedwater pumps that can be operated by both motor and steam turbine. A low-pressure switch on the pump discharge activates the feedwater steam turbine in the event of pressure drop. These design features should ensure that feedwater is available to the evaporator at all times. However, in the event of loss of feedwater flow, the master controller sensing the loss shuts the turbine load-control valve and turns off the salt pump, causing a sudden surge in pressure that is relieved by the safety valve and the turbine bypass Valves G and F. Any steam generated as a result of stored energy flows through the bypass line to the condenser. The drum-water level acts as a backup system by tripping the steam generator and turbine in the event the level reaches the low limits. One of the advantages of the natural-circulation system is the reservoir of water in the drum to cool the tubes and prevent overheating.

- Loss of Salt Flow. The event will not occur because of the special design features of the SGS. Three 50-percent-capacity salt pumps that can be operated by motor and steam turbine ensure salt flow whenever needed. However, in the event of loss of salt flow, the master controller sensing the loss closes the turbine load-control valve. Any surge in pressure causes the bypass Valves G and F to open, and steam flows to the condenser. The pressure surge is further sustained by the safety valve. The feedwater pump, under three-element control, shuts off when flow reaches 15 percent. The steam generator is isolated by closing Valves A, D, E, S, Y, and T.

- Rupture of Any Steam/Water/Salt Pipe. Proper piping design and early leak detection should prevent this event from happening. Such a rupture would cause a sudden drop in system pressure, in turn causing the master control to trip the turbine and the boiler. Rupture of a salt line results in a similar reaction from the master controller, and the trace heater is turned on automatically to prevent the salt from freezing.

- Loss of Pneumatic Pressure. The air compressors are designed to have at least 1/2 hour of instrument air storage, allowing enough time to shut the steam generator down in an orderly fashion.

- Failure of Control System. A backup system takes over in case of failure of the main control system.

- Loss of All Station Power. An uninterrupted power source (UPS) with battery backup provides 1/2 hour of power so that the steam generator can be brought to an orderly shutdown.

- Rupture of Steam/Water Tube. Tube rupture would result in a mix of high-pressure water or steam with hot salt, causing a sudden surge in pressure in the salt lines. Relief valves open to mitigate the pressure surge. Rupture discs break open if the pressure increases beyond the relief-valve settings. Drains collect the salt flow from the ruptured discs.

5.6.2 50-MWe Hybrid SGS

Description. In the solar/fossil-fuel hybrid configuration, a 50-MWe molten salt solar Steam Generator Subsystem (SGS) operates in combination with a 100-MWe fossil-fuel-fired boiler. The hybrid system supplies steam to a 100-MWe Electric Power Generating System (EPGS). The purpose of the hybrid configuration is to supplement fossil-fuel steam generation by solar steam generation as much as possible within the EPGS load-demand range.

A schematic of the heat exchanger arrangement along with interconnecting piping, valves, and controls is shown in Figure 5.28. The configuration of the solar SGS heat exchangers was described in detail earlier. The only difference is that the steam-generating capacity of the solar SGS in the hybrid system is half that of the solar stand-alone SGS.

Control System. The controls and valves in the hybrid solar SGS are the same as in the solar stand-alone SGS. Added to this are additional valves and controls required for hybrid operation. These controls are described in detail

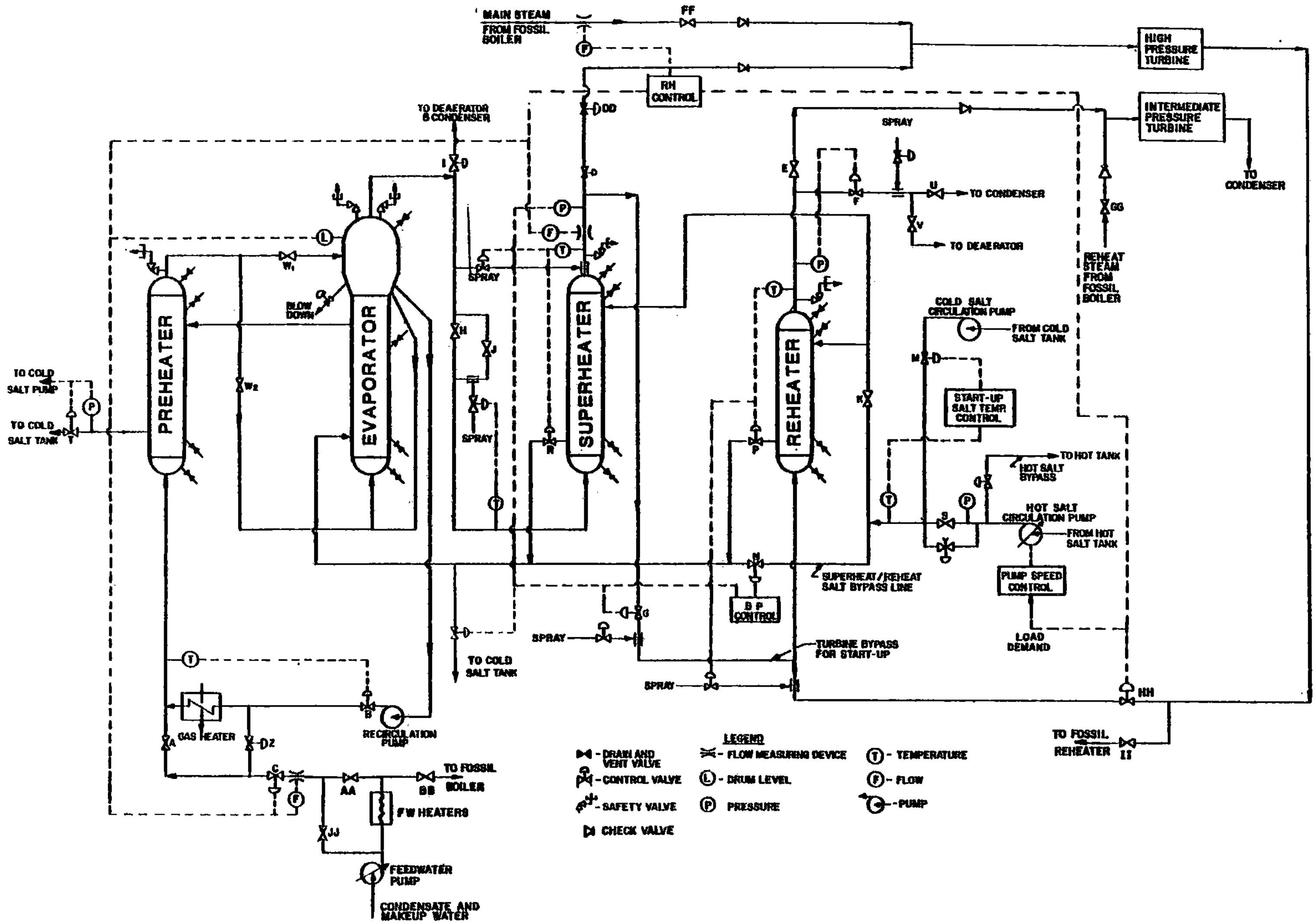

Figure 5.28 Control Diagram--50-MWe Hybrid SGS

in the following sections, along with various modes of operation. No attempt has been made to describe the configuration and internal controls of the fossil-fuel-fired boiler.

Operating Modes. The 50-MWe solar SGS in the solar/fossil hybrid configuration has the same operating modes as those for the 100-MWe solar stand-alone SGS, which was described earlier. The predicted apportionment of the 30-year life of the hybrid solar SGS among various operating modes and the predicted number of transitions between various operating modes are also the same as for the solar stand-alone SGS configuration. In the following paragraphs, the details of all solar hybrid operating modes are provided by making reference to stand-alone operating modes and describing any differences. The following operating modes are considered:

- Cold Shutdown (State 7)
- Cold Start-up (State 7 to State 1)
- Full- and Part-Load Operation (States 1, 2, 3, and 4)
- Load Changes (Between States 1 and 4)
- Diurnal Shutdown (State 1 to State 5)
- Diurnal Start-up (State 5 to State 1)
- Shutdown to Warm Standby (State 5 to State 6)
- Warm Standby (State 6)
- Start-up From Warm Standby (State 6 to State 1)
- Shutdown to Cold Conditions (State 6 to State 7).

In describing the solar hybrid operating modes, we have assumed that the fossil-fuel-fired boiler is always operating at some load between 25 and 100 MWe or is in a start-up or shutdown mode (below 25 MWe). For conditions where the fossil-fuel-fired boiler is not operating, the operating modes for the solar SGS are identical to those described for the solar stand-alone configuration. The only difference is that the fossil-fuel-fired boiler is isolated from the solar SGS and EPGS by closing appropriate valves.

Cold Shutdown. This mode is the same as that for the solar stand-alone except Feedwater Isolation Valves AA and JJ are closed and the feedwater pump is operating and supplying feedwater to the fossil-fuel-fired boiler through open Valve BB. Closed Valves D, E, and HH isolate the solar SGS on the steam side.

Cold Start-Up. The operating feedwater heaters make hot feedwater available for hybrid cold start-up. However, the tube-side temperature in the heat exchangers must be increased gradually in step with the shell temperature. This is accomplished by heating the preheater and evaporator shells to 82°C (180°F) and then filling the two heat exchangers with the available feedwater, which can range from 121°C (250°F) to 149°C (300°F), depending upon the fossil-fuel-fired boiler load. This feedwater is admitted to the preheater and evaporator through Bypass Valve JJ and Valve C. Valve AA remains closed. Valve JJ is a letdown valve set to reduce the pressure downstream to 4.59 MPa gage (666 lb/in^2g). The feedwater at 4.59 MPa gage (666 lb/in^2g) is routed through Valve Z and the nonoperating gas heater to the preheater and evaporator. Valve H to the superheater is closed, Valve W, is closed, and Valve W_2 is open.

The remaining procedure for cold start-up is the same as that for the solar stand-alone SGS [i.e., the water is circulated in the preheater-evaporator circuit and heated to approximately 238°C (460°F) using the gas heater]. The differences in the hybrid solar SGS cold start-up are:

- Once 5-percent steam flow at 468°C (875°F) from the superheater is established, Valves A and AA are opened and Valves Z and JJ are closed before ramping the load to 25 percent. This allows hot feedwater to flow through Valves A and C to maintain drum-water level.
- All steam produced before reaching 15-percent flow is routed to the condenser; no steam for turbine warm-up, rolling, or synchronizing is needed. Valves D, E, and HH are opened at 15-percent steam flow to allow constant steam flow to the turbines.

The cold start-up schematic for the hybrid solar SGS is the same as that for the solar stand-alone except that the hold times for turbine rolling and synchronizing are eliminated.

Full- and Part-Load Operation. Steady-state operation of the hybrid solar SGS is identical to that described earlier for the solar stand-alone. The major difference is in the interaction of the hybrid solar SGS with the fossil-fuel-fired boiler to meet the turbine load demand.

In determining the various full- and part-load hybrid modes, we based the following guidelines on Sierra Pacific's Ft. Churchill Unit 1:

- The turbine is rated at 100 MWe and the minimum steam flow to the turbine corresponds to 25 MWe.
- The fossil-fuel-fired boiler is rated at 100 MWe and its minimum part-load operation is at 25 MWe.
- The 50-MWe solar SGS can operate at a minimum part load of 12.5 MWe.

- The fossil-fuel-fired boiler is operational at all times to minimize cycling.
- The purpose of the hybrid operating mode is to achieve maximum solar SGS utilization.

Using these guidelines, the solar/fossil hybrid operating modes shown in Figure 5.29 are possible. The fossil-fuel-fired boiler operates alone to meet turbine load demand from 25 to 37.5 MWe. At this point, the solar SGS is brought on-line and operates at 12.5 MWe to supplement the fossil-fuel-fired boiler operating at 25 MWe to meet the 37.5-MWe turbine demand. From 37.5- to 75-MWe turbine load demand, the fossil-fuel-fired boiler remains at 25 MWe and the solar SGS ramps to 50 MWe. Beyond a 75-MWe load, the solar SGS continues to operate at 50 MWe, and the fossil-fuel-fired boiler ramps to 50 MWe to meet the maximum 100-MWe turbine load demand.

There are other modes of operation possible where fossil-fuel alone or solar-energy alone satisfy the turbine demand. This, however, is not hybrid operation; it is accomplished simply by isolating the fossil-fuel-fired boiler or solar SGS from the EPGS. Isolation valves on the feedwater and steam side are provided for that purpose.

For a given hybrid operating mode, main steam from the solar SGS superheater and the fossil-fuel-fired boiler reach the high-pressure turbine through a common header. Steam exiting from the high-pressure turbine is routed back to the solar SGS and fossil-fuel-fired boiler/reheaters. The main steam flow signal from the solar SGS superheater controls Valve HH to admit a proportional amount of steam to the solar SGS reheater. The remaining steam goes to the fossil-fuel-fired boiler reheater through Valve II. Standard three-element

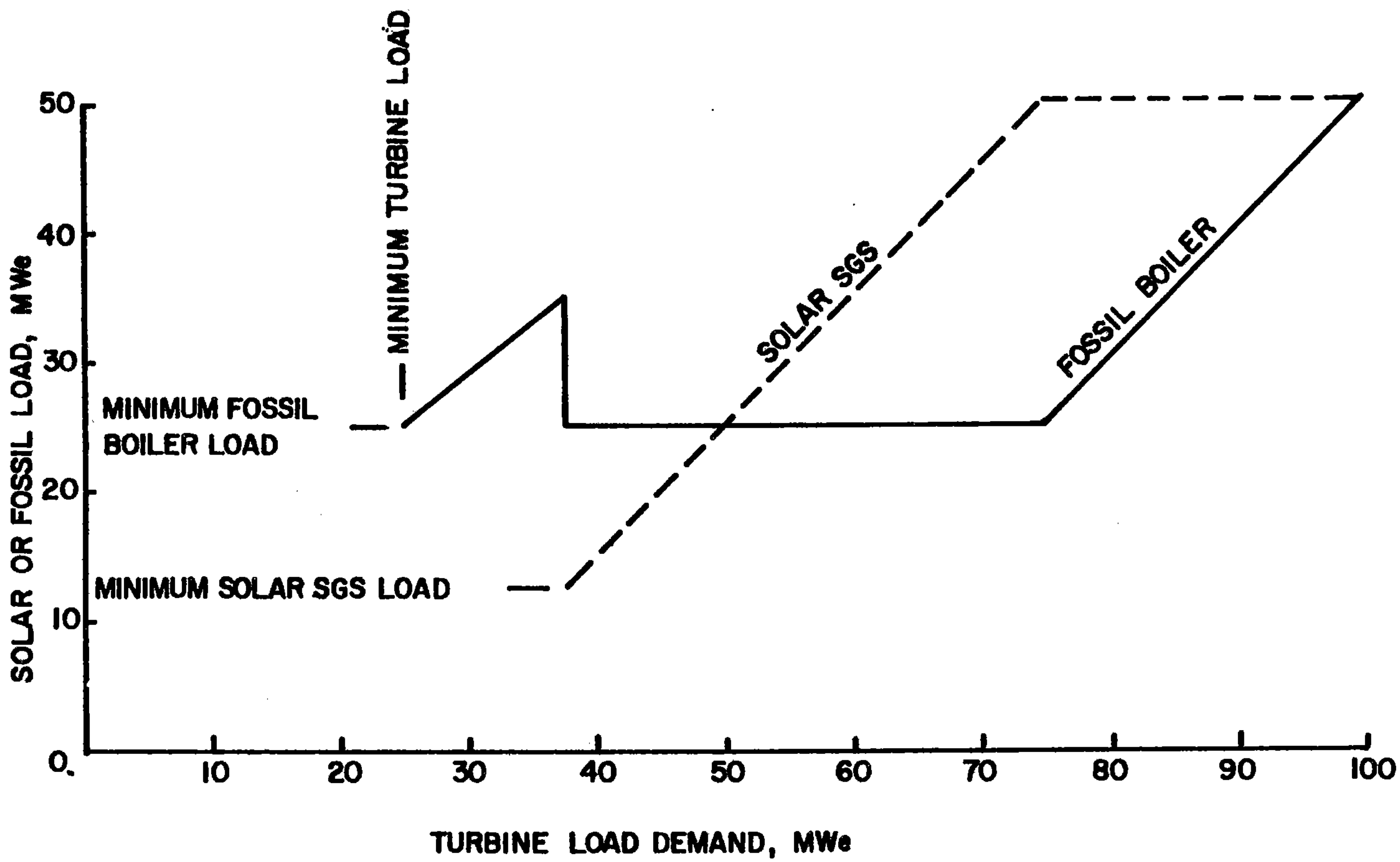

Figure 5.29 Operating Modes--50-MWe Hybrid SGS

feedwater controls on the solar SGS and fossil-fuel-fired boiler are used for feedwater flow to each circuit from the feedwater heaters.

Load Changes. The change in load demand from the EPGS is accommodated by the solar SGS or the fossil-fuel-fired boiler, depending upon the load on the EPGS. For a total EPGS load between 37.5 and 75 MWe, the fossil-fuel-fired boiler supplies a constant steam flow corresponding to 25 MWe, while the solar SGS steam generation is varied from 12.5 to 50 MWe to meet the EPGS demand. Similarly, for an EPGS load above 75 MWe, the solar SGS continues to supply its maximum steam rate, corresponding to 50 MWe, while the steam rate from the fossil-fuel-fired boiler is varied from 25 to 50 MWe to satisfy the total load demand. This operational interaction between the solar SGS and the fossil-fuel-fired boiler is shown in Figure 5.29.

The following illustrates the load change procedure (load increase and decrease from 75-MWe operating point): At 75-MWe EPGS load, the solar SGS will be supplying steam corresponding to 50 MWe and the fossil-fuel-fired boiler will be supplying the remaining amount, corresponding to 25 MWe. For an increased steam demand from this condition, the turbine throttle valve will open further, leading to reduction in pressure in the common steam mixing header, which will be served by both the solar SGS and the fossil-fuel-fired boiler. The programmed supervisory control, however, will instruct the solar SGS to ignore this signal and the solar SGS will continue to operate unchanged. The fossil-fuel-fired boiler will react to the reduced pressure signal, and its control system will take appropriate action to maintain pressure in the common header.

Similarly, for a load decrease from the 75-MWe condition, the supervisory control will instruct the fossil-fuel-fired boiler to ignore the pressure increase signal, and it will be the responsibility of the solar SGS to pull back to maintain reduced-load conditions. Below 37.5-MWe EPGS load demand, the solar SGS begins a shutdown procedure and the fossil-fuel-fired boiler alone satisfies the load demand.

The steam from the high-pressure turbine is split to the solar SGS and fossil-fuel-fired boiler reheaters using the main steam flow signals from the solar SGS and fossil-fuel-fired boiler. The load-change procedure within the solar SGS is identical to that described earlier for the solar stand-alone SGS, with the difference that the solar SGS responds to the supervisory control instead of to the turbine throttle valve.

Diurnal Shutdown. When the hot-salt storage tank reaches the 1-hour mark, the procedure for diurnal shutdown of the solar SGS begins. Control Valve DD on the main steam line is manually controlled to reduce the steam flow from the solar SGS at 3 percent/min. At the same time, a signal from the supervisory control instructs the fossil-fuel-fired boiler to supply more steam so that the total steam demand continues to be satisfied. Below 25-percent steam flow, the solar SGS is brought down to the 15-percent load level at 2 percent/min. At the 15-percent load level, the solar SGS is isolated from the fossil-fuel-fired boiler and EPGS by closing Valves, AA, D, E, and HH.

As described in the solar stand-alone diurnal shutdown procedure, the automatic settings on Valves G and F allow any high-pressure steam to exit to the condenser. The whole SGS remains in a bottled-up state overnight (for 7.5 hours). The temperature distribution within each heat exchanger in the solar SGS is as described earlier for the solar stand-alone SGS.

<u>Diurnal Start-Up</u>. The diurnal start-up procedure for the hybrid solar SGS is the same as for the solar stand-alone SGS with the following differences:

- Hot feedwater is available by opening Valve JJ or AA from the beginning, since the fossil-fuel-fired boiler is operating.
- Because the turbine is already generating power, no rolling or synchronizing steps are necessary. The steam produced in the solar SGS is routed to the condenser until 15-percent steam flow is achieved. Valves D, E, and HH are opened to bring the solar SGS on-line. The load is increased to 25-percent steam flow at 2 percent/min.

<u>Shutdown to Warm Standby</u>. This transition takes place because of extended cloud cover, changing the system from the hot standby to the warm standby mode. The valve positions remain the same as they do in the hot standby mode, and the SGS is simply allowed to cool down until the heat exchanger shell or salt piping temperature reaches 288°C (550°F), at which time trace heaters are turned on to maintain that temperature. The procedure is identical to that described for the solar stand-alone SGS shutdown to warm standby.

<u>Warm Standby</u>. The whole SGS is uniformly at 288°C (550°F). The pressure on the main steam/water side drops to the saturation value of 7.14 MPa gage (1036 lb/in^2g). Details of warm standby are the same as those described for the solar stand-alone operating mode.

Start-Up From Warm Standby. Start-up of the solar SGS from warm standby for hybrid operation is essentially the same as that for the solar stand-alone. Only the following differences exist:

- Hot feedwater is available by opening Valve AA, since the feedwater heaters are operating. Thus a gas heater is not required.
- The turbine is operating, using steam from the fossil-fuel-fired boiler; therefore, the turbine warm-up, rolling, and synchronizing steps described for the solar stand-alone SGS are eliminated. The steam produced in the solar SGS is routed to the condenser until 15-percent steam flow is established. Valves D, E, and HH are opened to bring the solar SGS on-line. The load on the solar SGS is increased to 25-percent steam flow at 2 percent/min.

Shutdown to Cold Conditions. This shutdown begins from Warm Standby and terminates in long-term cold shutdown of the SGS in a dry, ambient state. The procedure for shutting down the hybrid solar SGS to cold conditions is identical to that described for the solar stand-alone SGS.

5.6.3 Dynamic Simulation Model

To optimize the control scheme and predict the subsystem response, a mathematical model of the entire SGS system was prepared. All components that affect the transient response of the SGS (preheater, evaporator, superheater, reheater, pumps, valves, and piping) were modeled. Using Laplace transform, a block diagram of the various components was developed. The block diagram models the known physics and thermodynamics as part of the SGS subsystem. The same block diagram was used to model the sensing elements and controller. The Bode plot technique was used to determine the type of controller and its gain for stable response. From the block diagram, differential equations were developed.

These equations were solved using the computer program DSS/2 (Dynamic Simulation System, Version 2).[51] The model can be segregated into three main loops. These are:

- Superheater (SH) outlet temperature loop
- Reheater (RH) outlet temperature loop
- Throttle pressure loop.

Although the three loops are intercoupled, valid transient and stability analyses can be separately performed on each loop. This approach permits first-order design parameters to be established and allows the analyst to gain an insight into system performance. The loops are intercoupled and ensuing results are more reliably interpreted.

The dynamic model shown in Figures 5.30 and 5.31 represent the dynamic characteristics of the various components in the SGS, including the controllers, in terms of Laplace transform notation. The coefficients used in Figures 5.30 and 5.31 are defined in Table 5.15. An analytical solution to the equation resulting from this model predicts the response of the following to a specific change in steam flow rate:

- Drum pressure
- Steam line pressure losses from superheater to turbine
- Salt flow through superheater, reheater, evaporator, and salt bypass line
- Superheater outlet steam temperature with and without spray
- Superheater salt outlet temperatures
- Reheater outlet steam temperature with and without spray

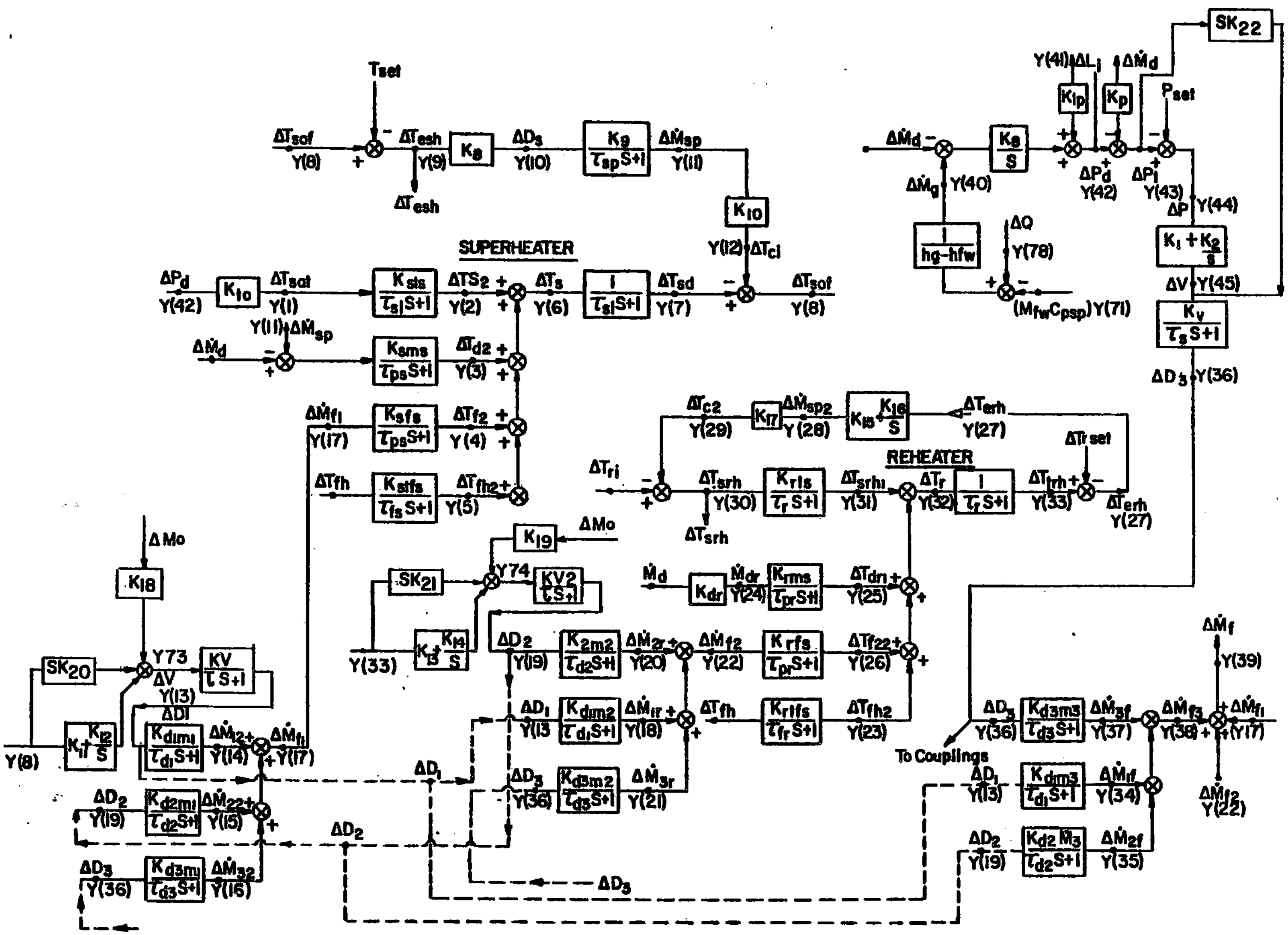

Figure 5.30 Dynamic Model--Steam Side

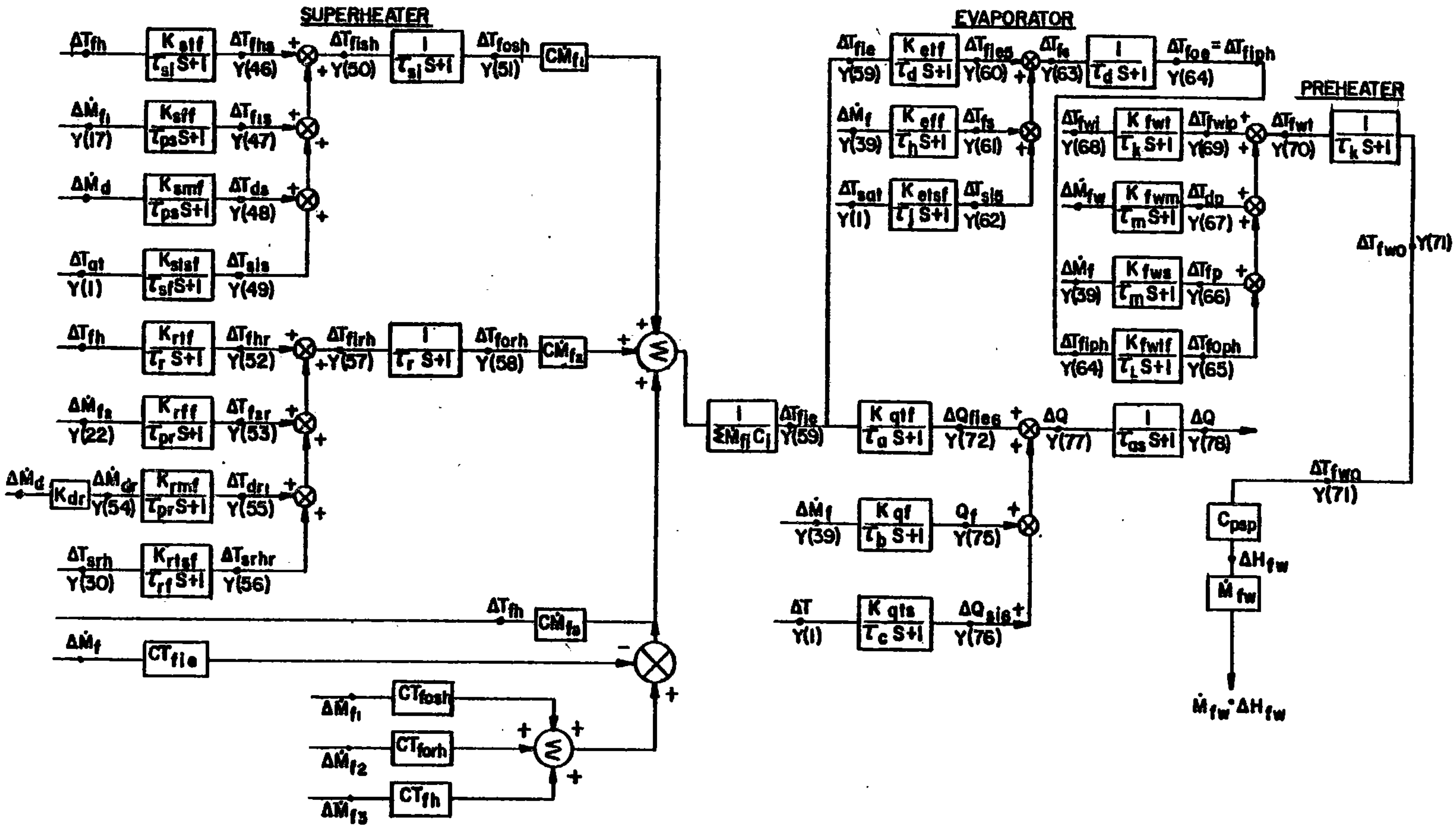

Figure 5.31 Dynamic Model--Salt Side

Table 5.15 Definition of Variables

K_{to}	= Change saturation temperature/unit change in pressure
K_{rts}, K_{Sts}, K_{etsf}, K_{fwt}	= Change steam outlet temperature to unit
K_{sms}, K_{rms}, K_{fwm}	= Change in steam outlet temperature to unit; change in salt flow
K_{sfs}, K_{rfs}, K_{effee}, K_{fws}	= Change in steam outlet temperature to unit; change in salt flow
K_{stfs}, K_{rtfs}, K_{eft}, K_{fwtf}	= Change in steam outlet temperature to unit; change in salt inlet temperature
$K_{d^1m^1}$, $K_{d^2m^1}$, $K_{d^3m^1}$, $K_{d^2m^2}$, $K_{d^1m^1}$, $K_{d^3m^2}$, $K_{d^3m^3}$, $K_{d^1m^1}$, $K_{d^2m^3}$	= Change in salt flow resulting from unit change in valve openings
K_B	= Change in drum pressure resulting from unit change in steam flow
K_V	= Change in valve openings/unit; signal change
K_P	= Coefficient of steam pressure drop
K_{LP}	= Coefficient of drum pressure from drum-water level change
K_1, K_2, K_3	= Controller characteristic
τ_{SI}, τ_{PS}, τ_{fs}, τ_{di}	= Time constants (All τ)
K_{stf}, K_{rtf}	= Change in salt outlet temperature resulting from unit change salt inlet temperature
K_{sff}, K_{rff}	= Change in salt outlet temperature resulting from unit change in salt flow
K_{smf}, K_{rmf}	= Change in salt outlet temperature resulting from unit change in steam flow rate
K_{stsf}, K_{rtsf}	= Change in salt outlet temperature resulting from unit change in steam inlet temperature
K_{qtf}	= Change in heat to evaporator resulting from unit change in evaporator salt inlet temperature
K_{qf}	= Change in heat to evaporator resulting from unit change in salt flow
K_{qts}	= Change in heat to evaporator resulting from unit change saturation temperature
T_{fosh}	= Superheater salt outlet temperature
T_{forh}	= Reheater salt outlet temperature
T_{fie}	= Evaporator salt inlet temperature

Table 5.15 Definition of Variables (Cont)

Q	= Heat to evaporator
P_D	= Drum pressure
T_{sof}	= Superheater steam outlet temperature
T_{erm}	= Reheater steam outlet temperature
M_f	= Evaporator salt flow
M_{3f}	= Bypass salt flow
M_{1f}	= Superheater salt flow
M_{2f}	= Reheater salt flow
M_D	= Steam demand
M_g	= Steam generated
D_1, D_2, D_3	= Salt control valve opening
T_{SET}, P_{set}, T_{rset}	= Set points
hg-hfw	= Heat of vaporization

- Reheater outlet salt temperature
- Evaporator inlet and outlet salt temperature
- Preheater outlet water temperature
- Total heat input to evaporator.

Numerous cases were run to optimize the control scheme. Figure 5.25 incorporates the results of this optimization. Figure 5.32 shows the SGS response to load change of 10 percent/min, the maximum load-up/load-down rate. The steam demand was changed from 100 to 90 percent flow in 60 seconds. A transient increase in steam pressure and a small transient decrease in superheater and reheater outlet temperatures are indicated. Very quick and stable responses are obtained for the reheater and superheater temperature with transient pressure response ceasing after 800 seconds. The dynamic response of the SGS to the emergency shutdown condition is shown in Figure 5.33. In this case steam flow was ramped down from 100 to 50 percent flow in 10 seconds. Stable transient response of the superheater and reheater outlet temperatures, shown in Figure 5.33, indicates that the transient behavior ceases within 300 seconds. The transient decrease in the superheater and reheater outlet temperatures occurs because the feed-forward control signal immediately adjusts the salt flow through the reheater and superheater in proportion to the steam demand while the steam generated still exceeds the steam demanded. The excess steam generated causes the transient increase of main steam pressure, resulting in opening of the safety valves--an emergency response, not an everyday occurrence. More detailed analysis is necessary to predict transient pressure surge accurately during the rapid emergency condition. Figure 5.33 shows the evaporator inlet salt temperature transient response. Since the transient pressure surges were the result of the

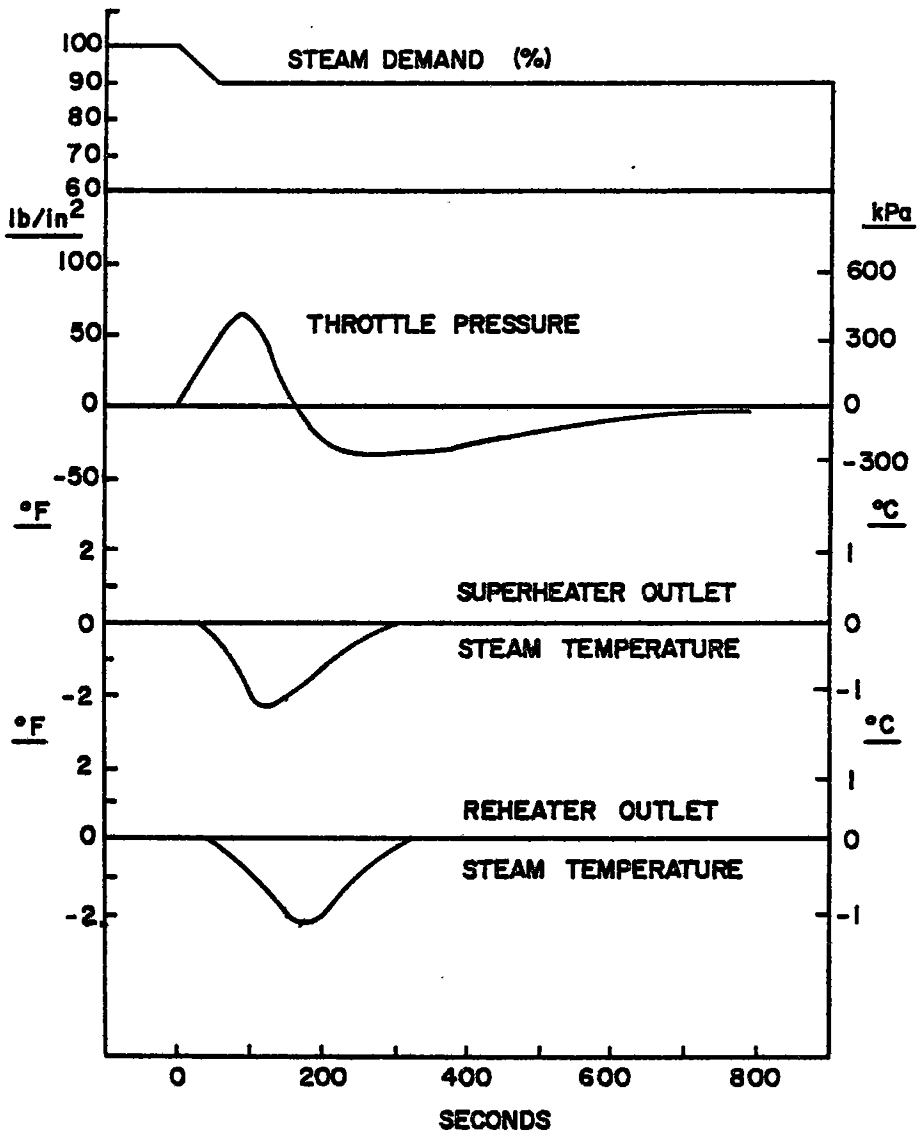

Figure 5.32 Transient Response for 10 Percent Load Change

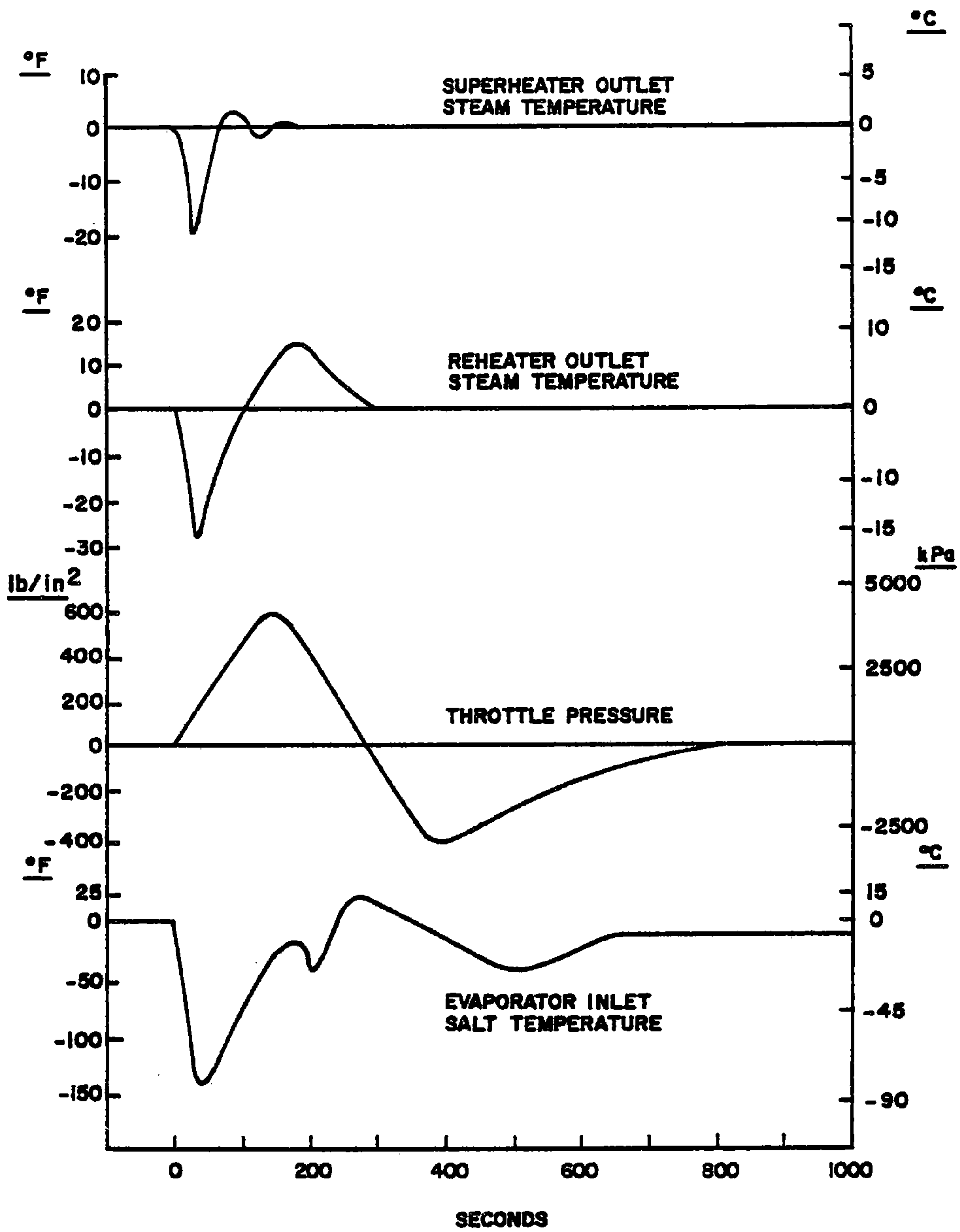

Figure 5.33 Transient Response for Emergency Shutdown

coupling effect of reheater, superheater, and bypass salt valves, the response of the SGS was also obtained for the uncoupled case. The decoupling is obtained by providing an intermediate tank to which the reheater, superheater, and bypass salt flows (Figure 5.34). The salt is then pumped to the evaporator. In effect, the intermediate tank achieves a constant evaporator inlet salt temperature. The SGS responds to the same emergency shutdown conditions shown in Figure 5.35. The control system reacts fast enough to prevent the safety valves from blowing during the emergency transients. However, the intermediate tank, the evaporator, the salt pump, and piping add cost to the SGS.

In conclusion, the dynamic analysis shows:

- In addition to proportional and integral controls, feed-forward and derivative controls are required for quick, stable response.
- Through the bypass line, 1-percent salt flow is required at 100-percent load for pressure control.
- An evaporator and preheater salt bypass line under pressure control is required to sustain pressure surges during emergency conditions.

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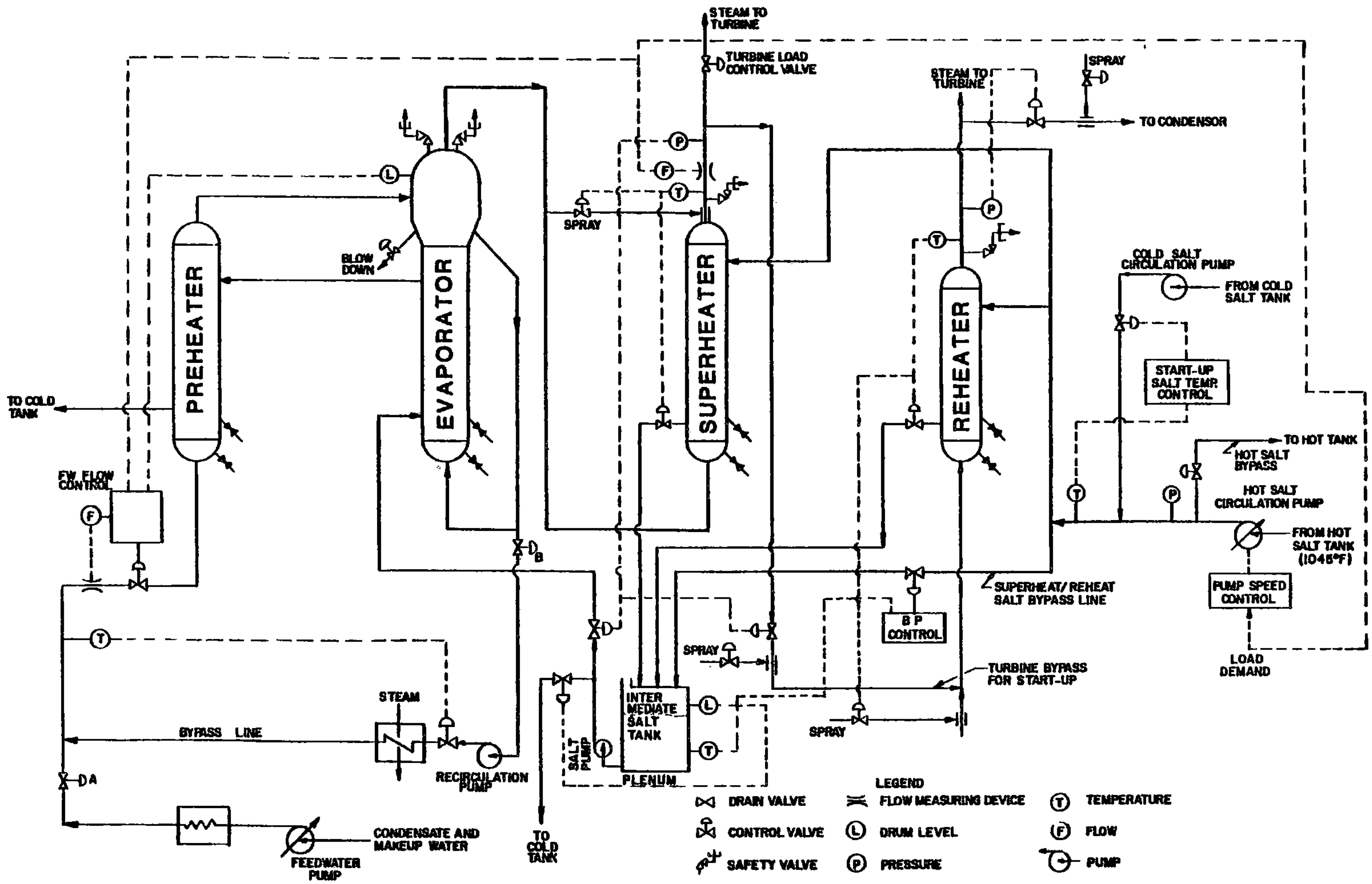


Figure 5.34 Alternate Control Diagram--100-MWe Solar Stand-Alone SGS

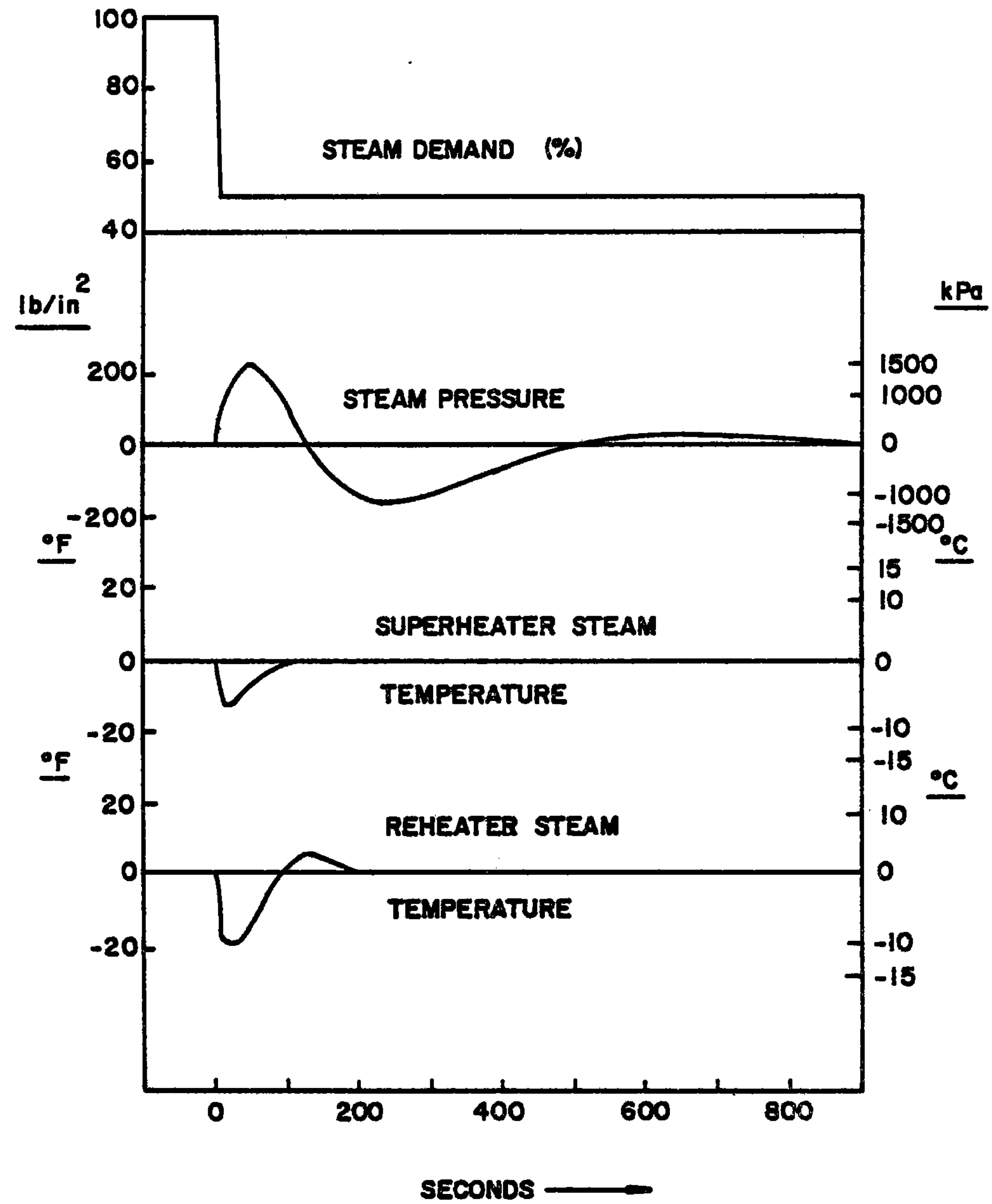


Figure 5.35 Transient Response for Emergency Shutdown--Alternate Control Scheme

5.7 AUXILIARY EQUIPMENT

5.7.1 Plot Plan

The plot plan for the 100-MWe solar stand-alone SGS, shown in Figure 5.36, is similar to the Sierra Pacific plot plan, which was used as the basis for this project. As with the Sierra Pacific plan, the solar SGS is located between the molten salt storage tanks and the power plant turbine/generator. The molten salt drain-tank system is located adjacent to the solar SGS, near the reheater and just behind the salt system pipe rack. Molten salt from the sump is pumped to the cold-salt return line using the molten salt sump pump. The start-up heater is located about 15.2 m (50 ft) to the right of the solar SGS, near the pipe rack. This location was chosen to minimize the recirculating water and fuel lines and the system controls.

The 50-MWe hybrid SGS differs only slightly from the stand-alone. The 50-MWe hybrid does not require a start-up heater and has smaller salt and steam lines (Figure 5.37).

A molten salt pressure-relief system will be required to protect the lower-pressure molten salt side of the heat exchangers against leaks from the higher-pressure boiler feedwater and steam generation side. This relief system will be located adjacent to the solar SGS.

5.7.2 Equipment Arrangement

This section describes the general arrangements of the four molten salt-to-steam/water heat exchangers, the two recirculation water pumps, associated control valves, and piping for both the 100-MWe solar stand-alone SGS and the

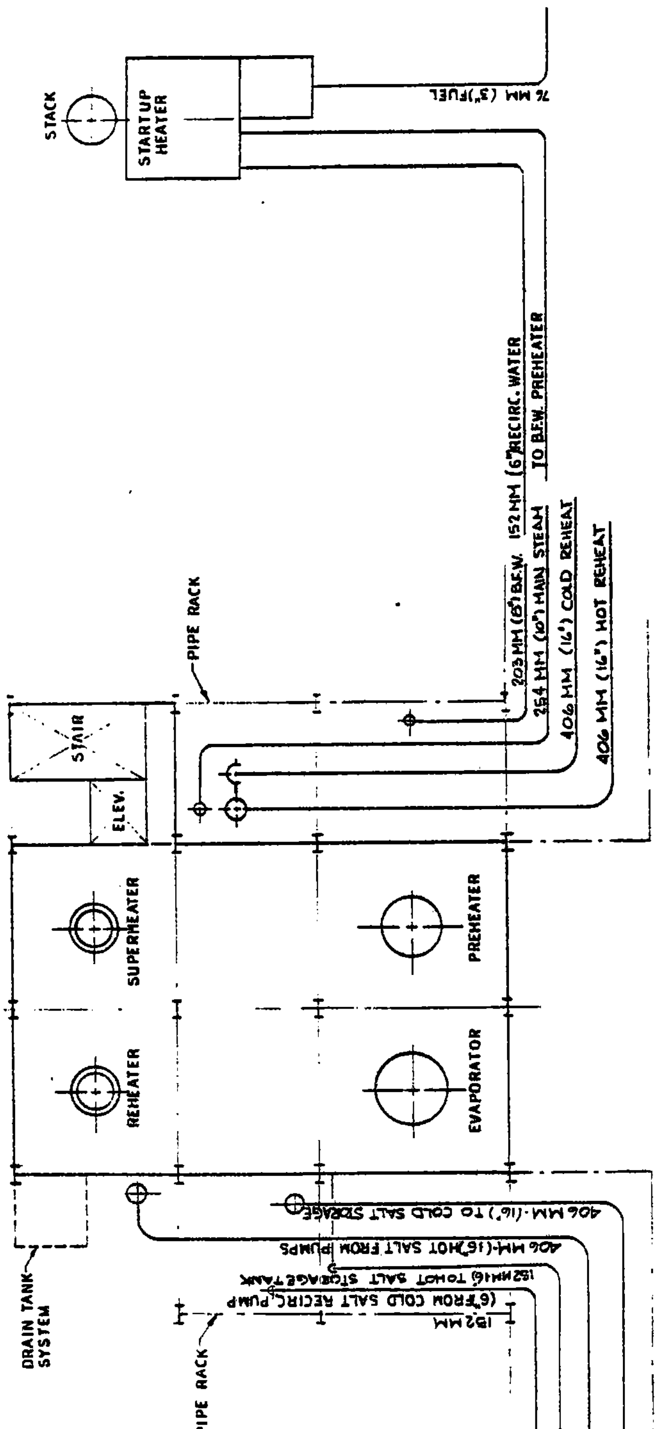


Figure 5.36 Plot Plan--100-MWe Solar Stand-Alone SGS

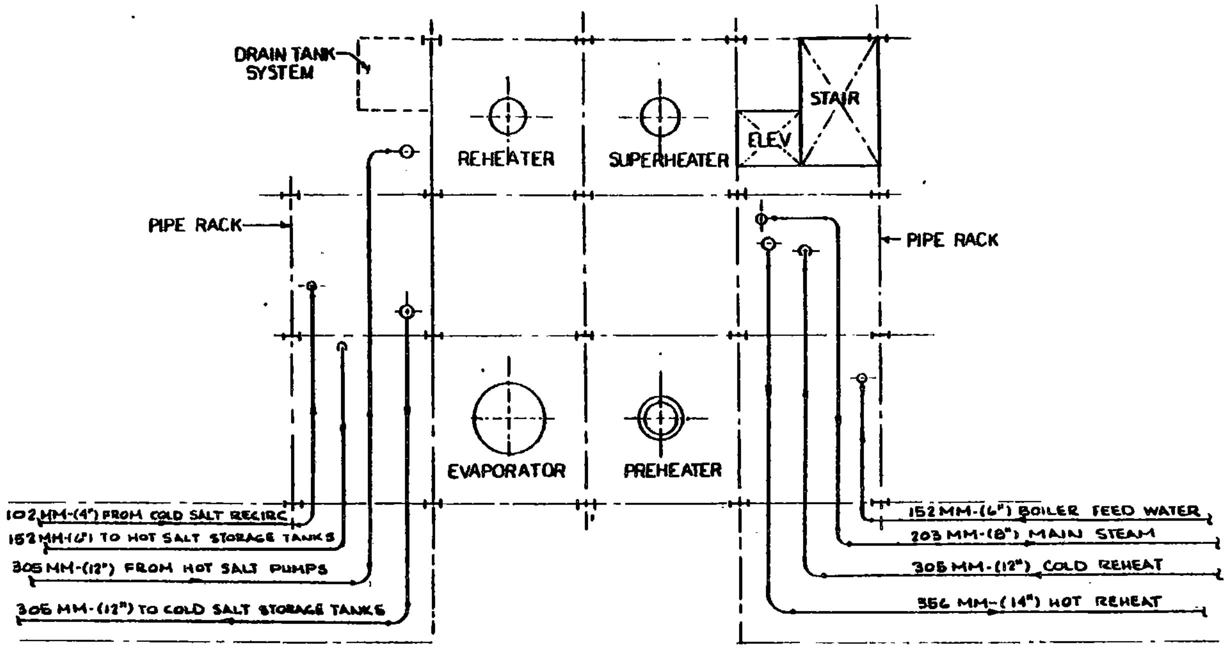


Figure 5.37 Plot Plan--50-MWe Hybrid SGS

50-MWe hybrid SGS. The 100-MWe SGS has an oil-fired start-up heater; the 50 MWe does not. Hot boiler feedwater for the 50-MWe unit is available from the fossil-fuel-fired boiler system.

The plot plans (Figures 5.36 and 5.37) show the relative location of the equipment and the general arrangement of the exchangers, supporting equipment, pipe racks, salt drain system, elevator, and stairwell. The two plot plans are identical except for a slight variation in size.

Figures 5.38 through 5.40 show the vertical and plan arrangements for the 100-MWe solar stand-alone SGS; Figures 5.41 through 5.43 show the vertical and plan arrangements for the 50-MWe hybrid SGS. These figures show the vertical and horizontal locations of the four heat exchangers and the circulation water pumps, as well as the locations of the pipelines and control valve stations. For reliability of operation and to comply with power plant requirements, full control stations are provided. Most of these are strategically located on platforms for ease of maintenance and accessibility in the service aisle between the exchangers. Others, also easily accessible, are located on the pipe rack just outside the solar SGS structure. The two pipe racks are located on the first level [Elevation 6.1 m (20 ft)], which is also the main operating floor--site of the major control stations.

The vertical-recirculation water pumps are beneath the evaporator, mounted on spring supports to allow for the vertical growth of the evaporator. The strategic location of these pumps minimizes piping and provides maximum static head so that any probability of flashing or cavitation in the pump suction lines and the pumps themselves is eliminated.

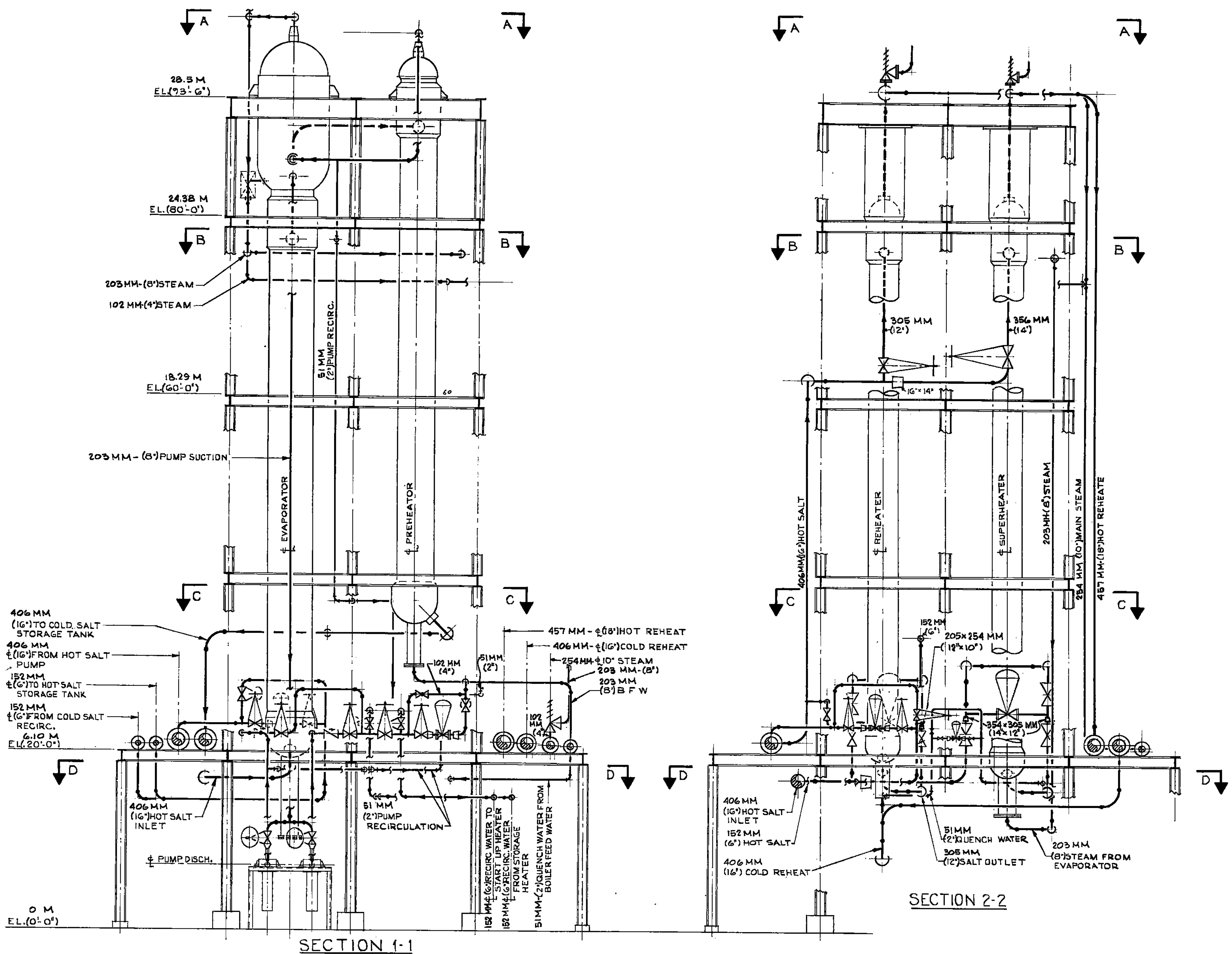


Figure 5.38 Equipment Layout (Side Elevation)--100-MWe Solar Stand-Alone SGS

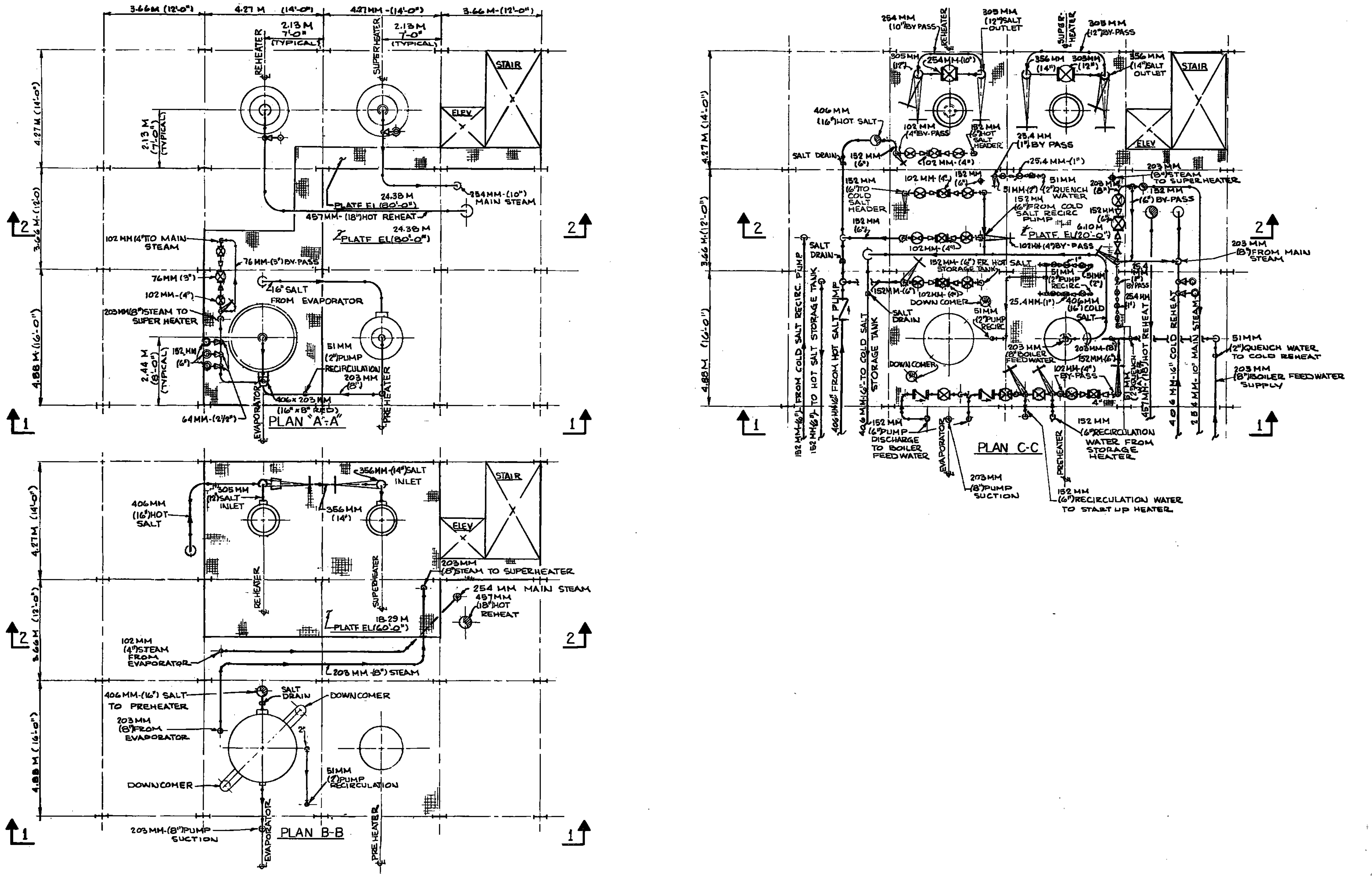
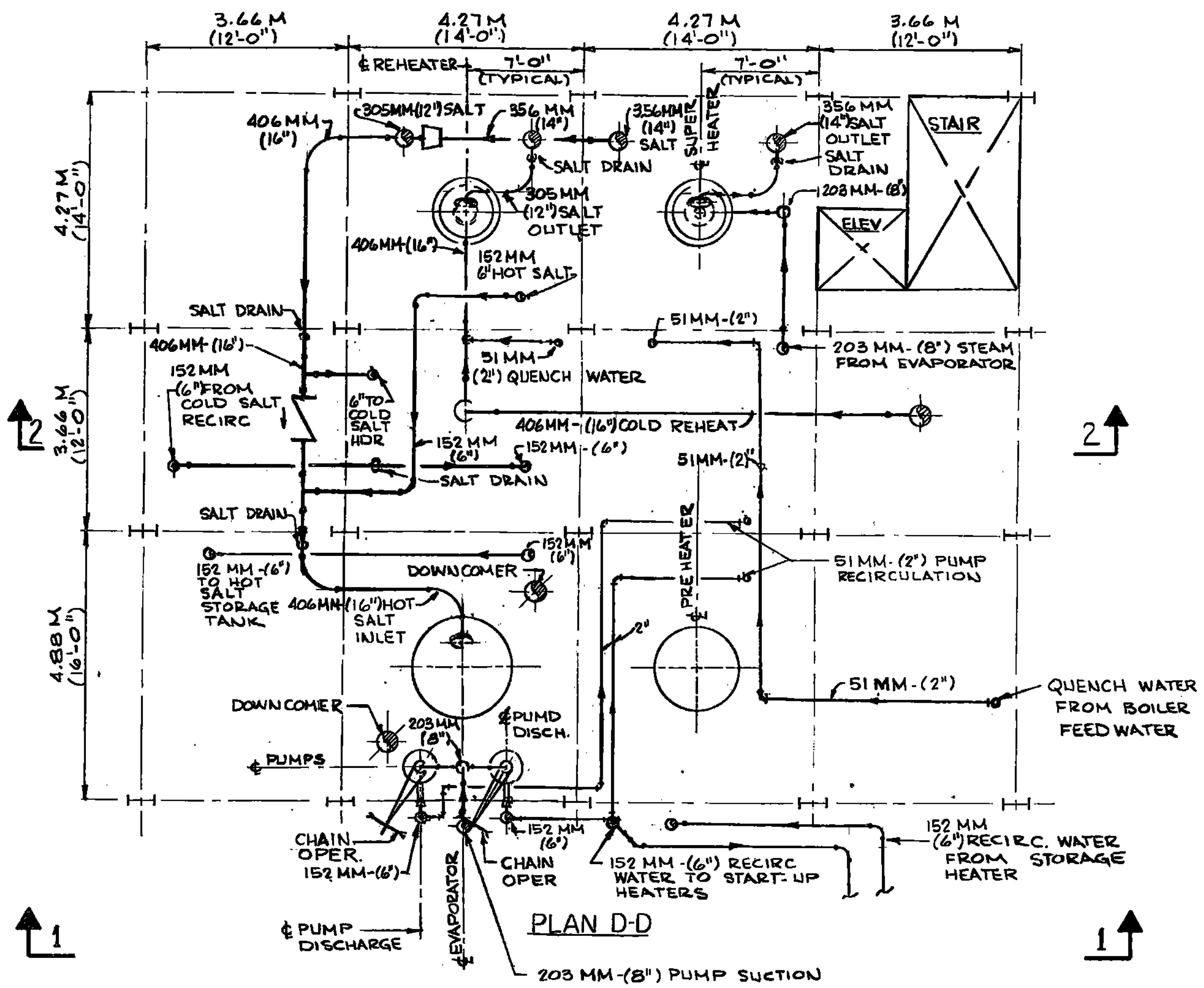
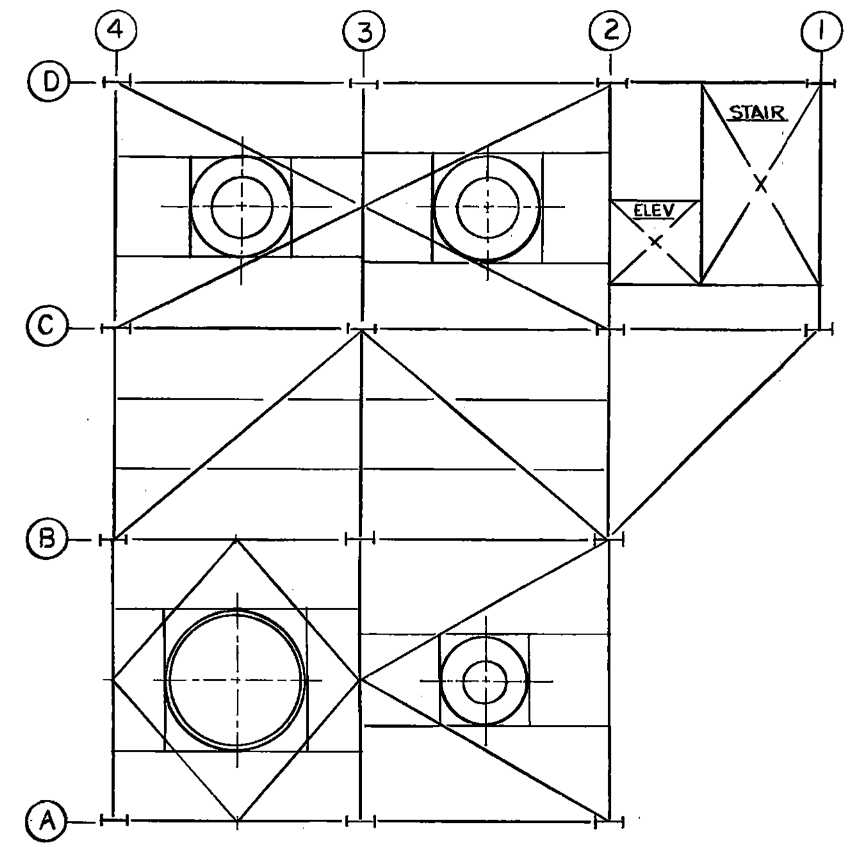


Figure 5.39 Equipment Layout (Plan View)--
100-MWe Solar Stand-Alone SGS





STRUCTURAL BRACING

NOTES:

(1) ALL HORIZ BRACING ⅃LS 3½ x 3 x ⅜ LG LEGS B. TO B. UNLESS NOTED

(2) BRACING SHOWN TO BE INSTALLED AT ELEV'S 6.10 M-(20'-0"), 12.19M-(40'-0") 18.29 M-(60'-0"), 24.38 M-(80'-0"), 28.5 M-(93'-6")

Figure 5.40 Equipment Layout and Structural Bracing (Plan View)--100-MWe Solar Stand-Alone SGS

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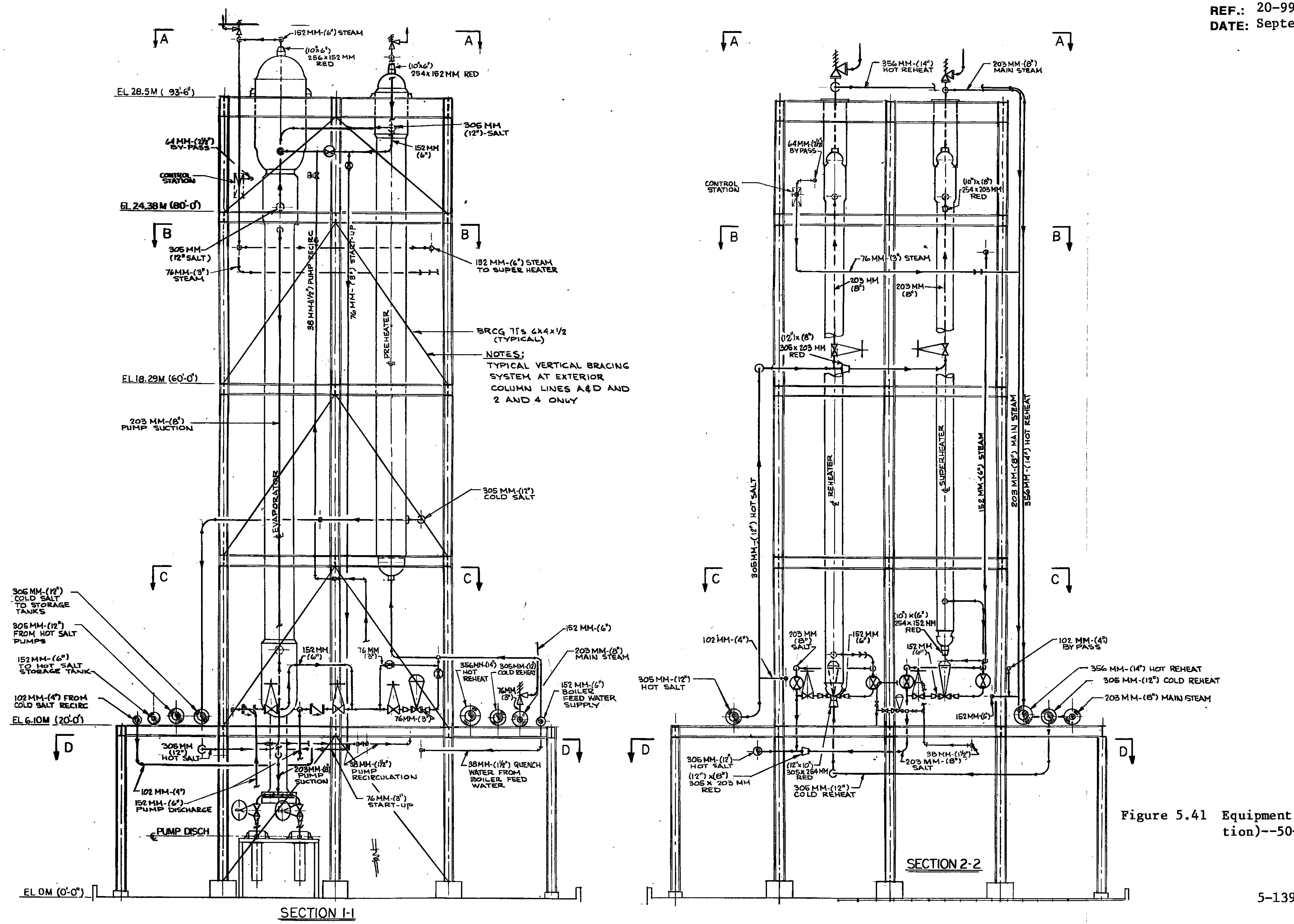
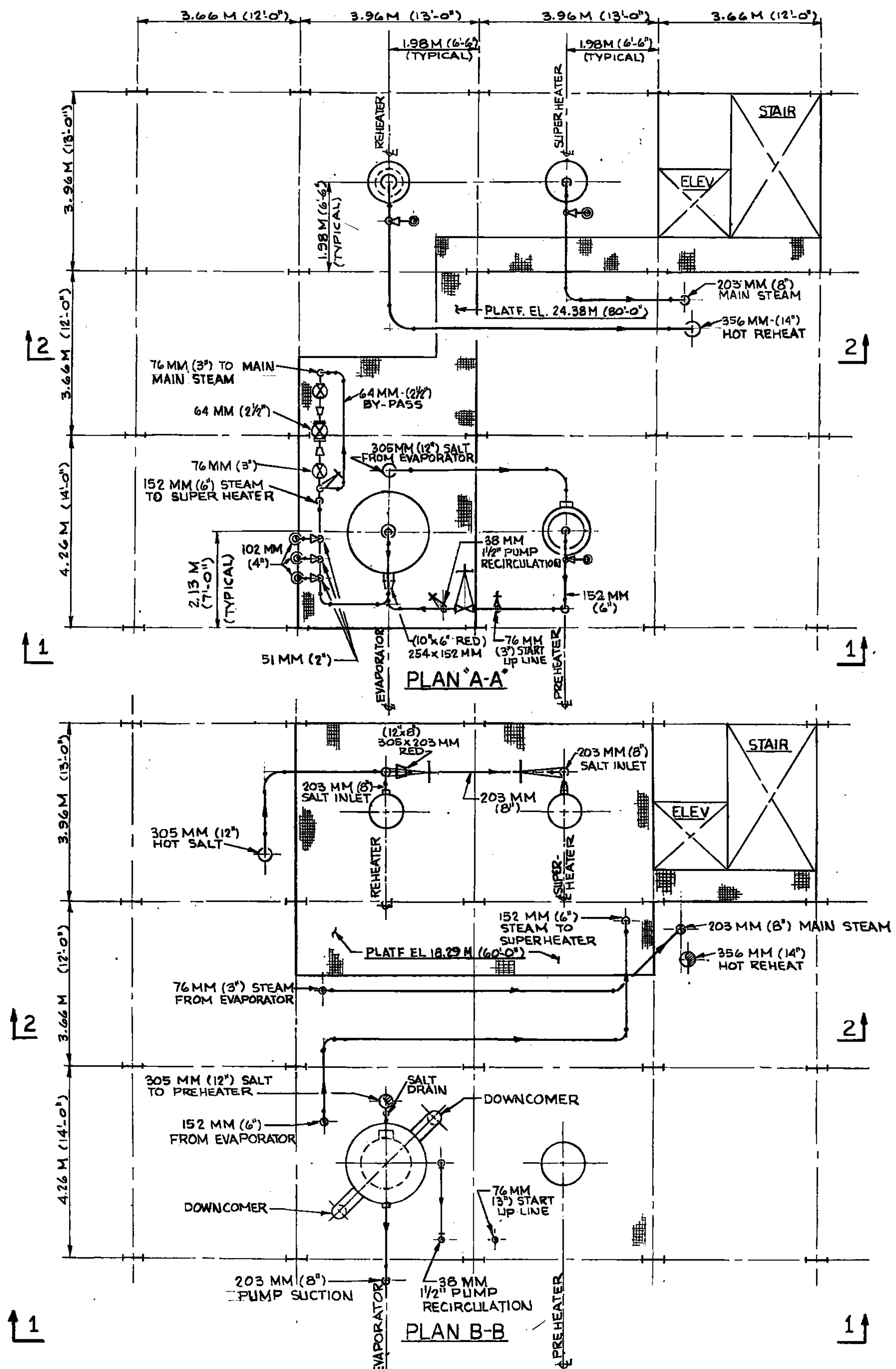


Figure 5.41 Equipment Layout (Side Elevation)--50-MWe Hybrid SGS



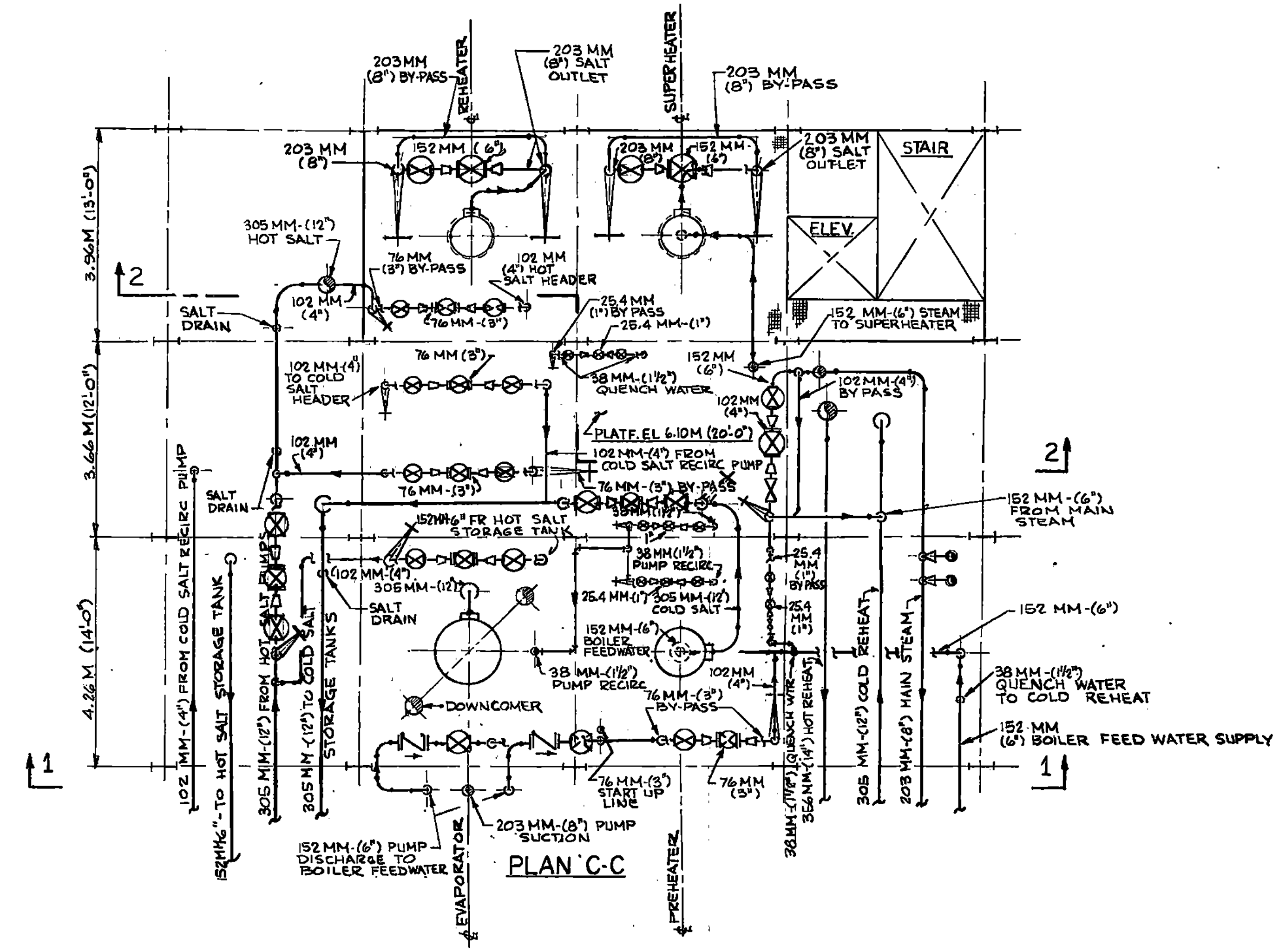
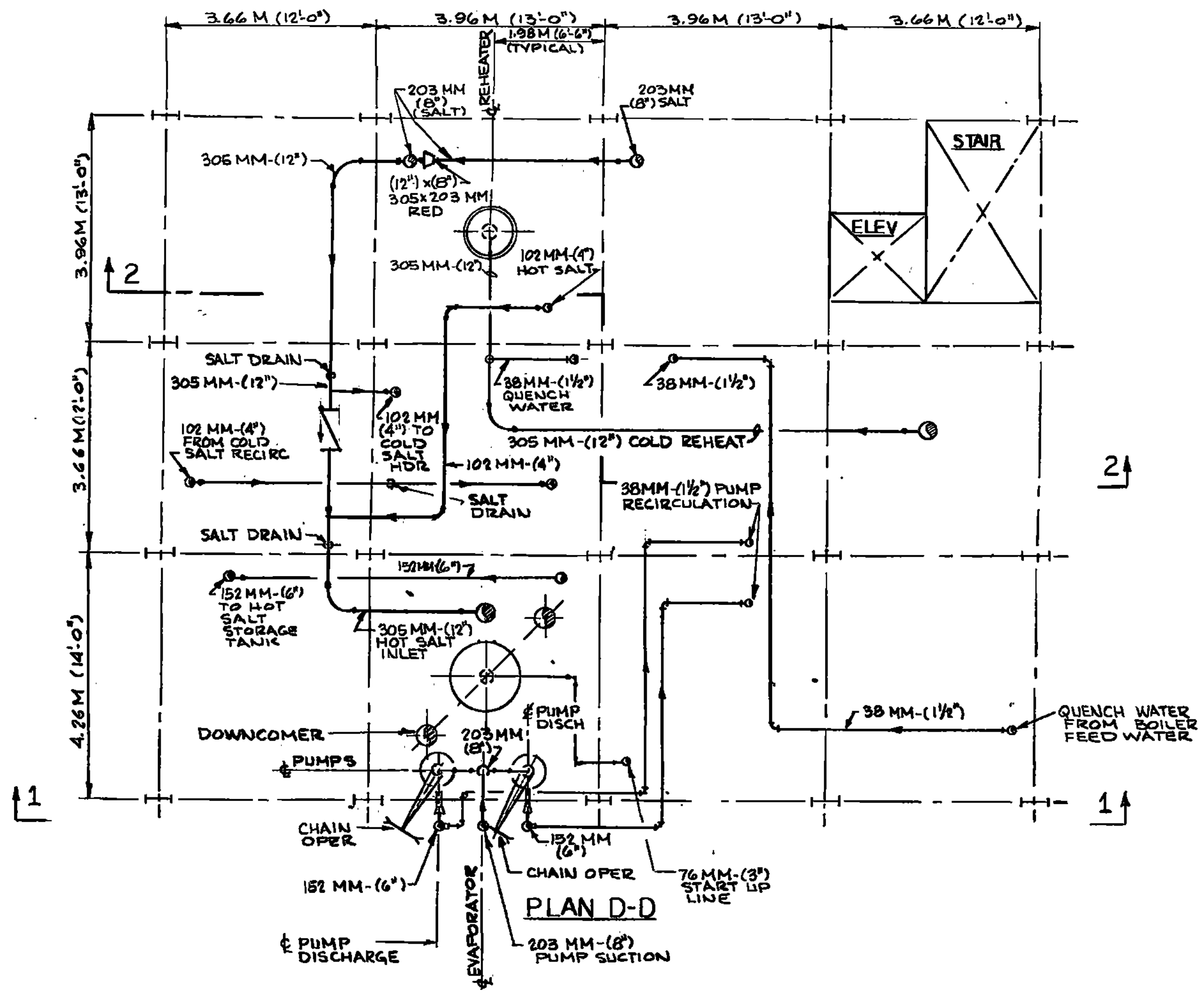
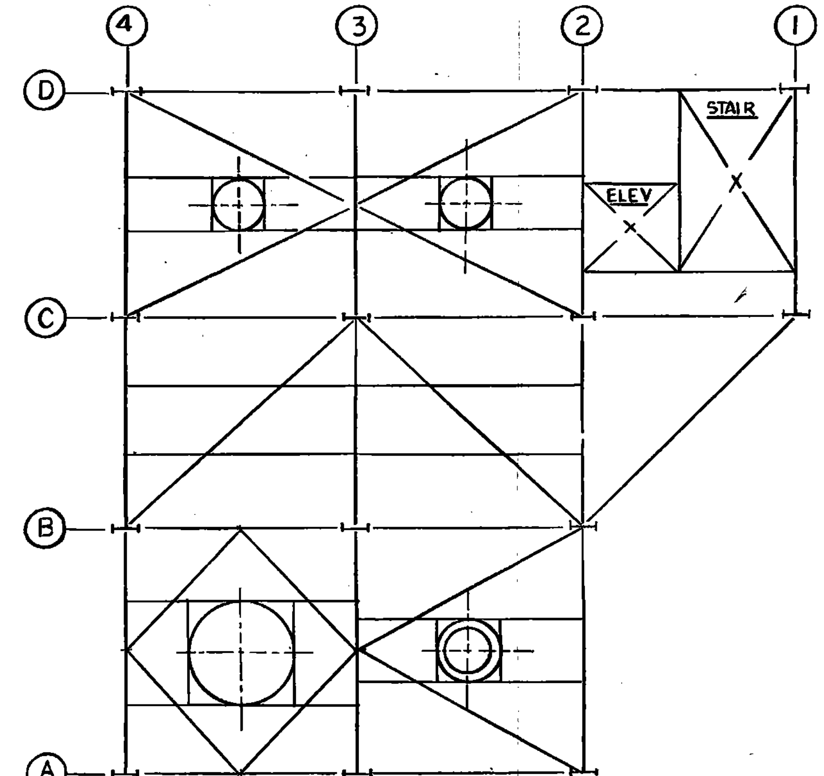


Figure 5.42 Equipment Layout (Plan View)--
50-MWe Hybrid SGS





STRUCTURAL BRACING

NOTE:

(1) ALL HORIZ BRACING ⅃L's 3½ x 3 x ⅜ LG. LEGS B.TO B. UNLESS NOTED

(2) BRACING SHOWN TO BE INSTALLED AT ELEV'S 6.10 M-(20'-0"), 12.19 M-(40'-0") 18.29 M-(60'-0"), 24.38 M-(80'-0"), 28.5 M-(93'-6")

Figure 5.43 Equipment Layout and Structural Bracing (Plan View)--50-MWe Hybrid SGS

The start-up heater for the 100-MWe solar stand-alone SGS is tied into the evaporator/preheater recirculation water pump discharge lines. This heater is to be used for start-up only, when the recirculation water system is operating at pressures lower than the design pressure of the heater. Bypass and shutoff valves isolate the heater from the solar SGS steam system during normal operation. The control system is designed so that these shutoff valves cannot be opened during normal operation. The pumps are accessible from grade for operation and maintenance.

The safety-relief valves are located above the upper platform level [28.5 m (93 ft-6 in.)]. The only exception is the relief valve on the reheater inlet pipeline, which is located on the pipe rack and vented to the atmosphere in such a manner that there is no danger to the personnel in the solar SGS structure.

The elevator and stairwell terminate at the upper platform level. This platform elevation and all other elevations where there are walkways and equipment to be serviced are protected by safety railings.

Molten salt pumped from the hot-salt storage tank enters the solar SGS, flows in parallel through the shell side of the superheater and reheater, combines, and passes in series through the evaporator and preheater, returning afterward to the cold-salt storage tank. Feedwater is pumped through the tube side of the preheater and brought to the flash point in the evaporator drum. Dry steam is then passed through the superheater for superheating, while saturated water is recirculated through the evaporator to generate steam by natural circulation and through the preheater by the two vertical-recirculation water pumps for temperature control of the incoming feedwater when required.

5.7.3 Molten Salt Drain System

A drainage system removes molten salt from equipment and pipelines during periodic shutdown of the plant for maintenance and in an emergency. The components that constitute the drain system are the piping, sump pump, and sump tank. Lines to and from the heat exchangers are sloped toward the sump tank to allow them to drain by gravity. The drain system is designed to accommodate a scheduled removal of the total volume of solar SGS salt in 2 hours. The salt flows to the sump tank and is pumped into the cold-salt storage tank. The entire system is insulated and heat traced to prevent salt from freezing (see Sections 5.7.8 and 5.7.9).

The 100-Mwe solar stand-alone SGS and the 50-MWe hybrid SGS molten salt drain system arrangements are shown isometrically in Figures 5.44 and 5.45. Piping characteristics are listed in Table 5.16.

The drain system sump tank characteristics are presented in Table 5.17. The cylindrical sump tank is sized to contain 5 minutes of salt flow based upon a 2-hour draining period. A level-actuated control valve on the tank inlet line prevents overflow. The tank is located below grade and is covered to prevent rain and dust from entering it. The sump pit is concrete and large enough in diameter to create a 0.15-m (0.5-ft) annulus between the pit wall and the tank. Ceramic pipe in the bottom of the sump pit allows the free passage of water vapor from beneath the tank.

The specified pump capacity (listed in Table 5.18) reflects the determined flow rate and a 20-percent contingency. The design head represents a cold-salt tank height of 10 m (30 ft) and a relatively short piping distance [4150 m (495 ft)].

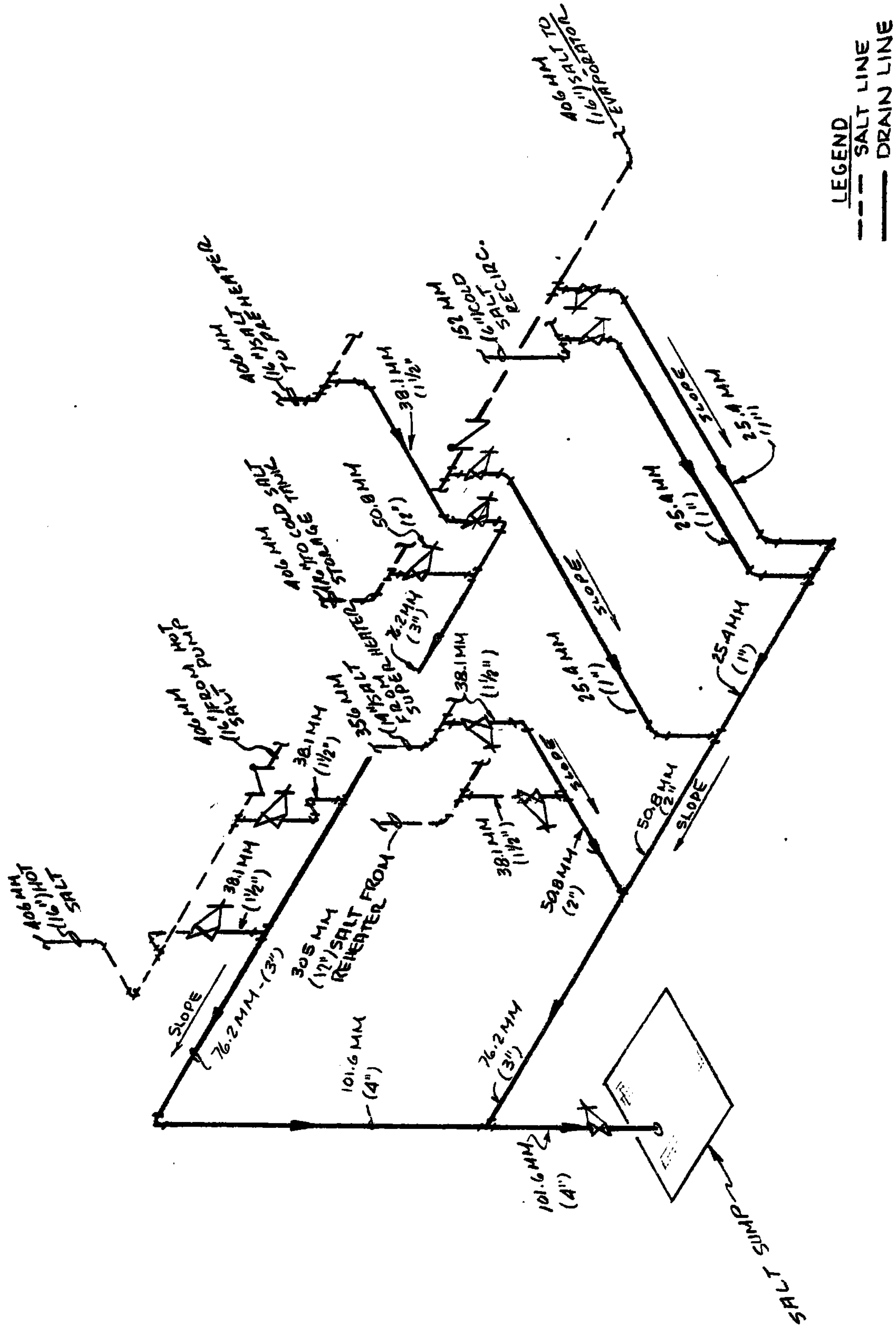


Figure 5.44 Drain System—100-MWe Solar Stand-Alone SGS

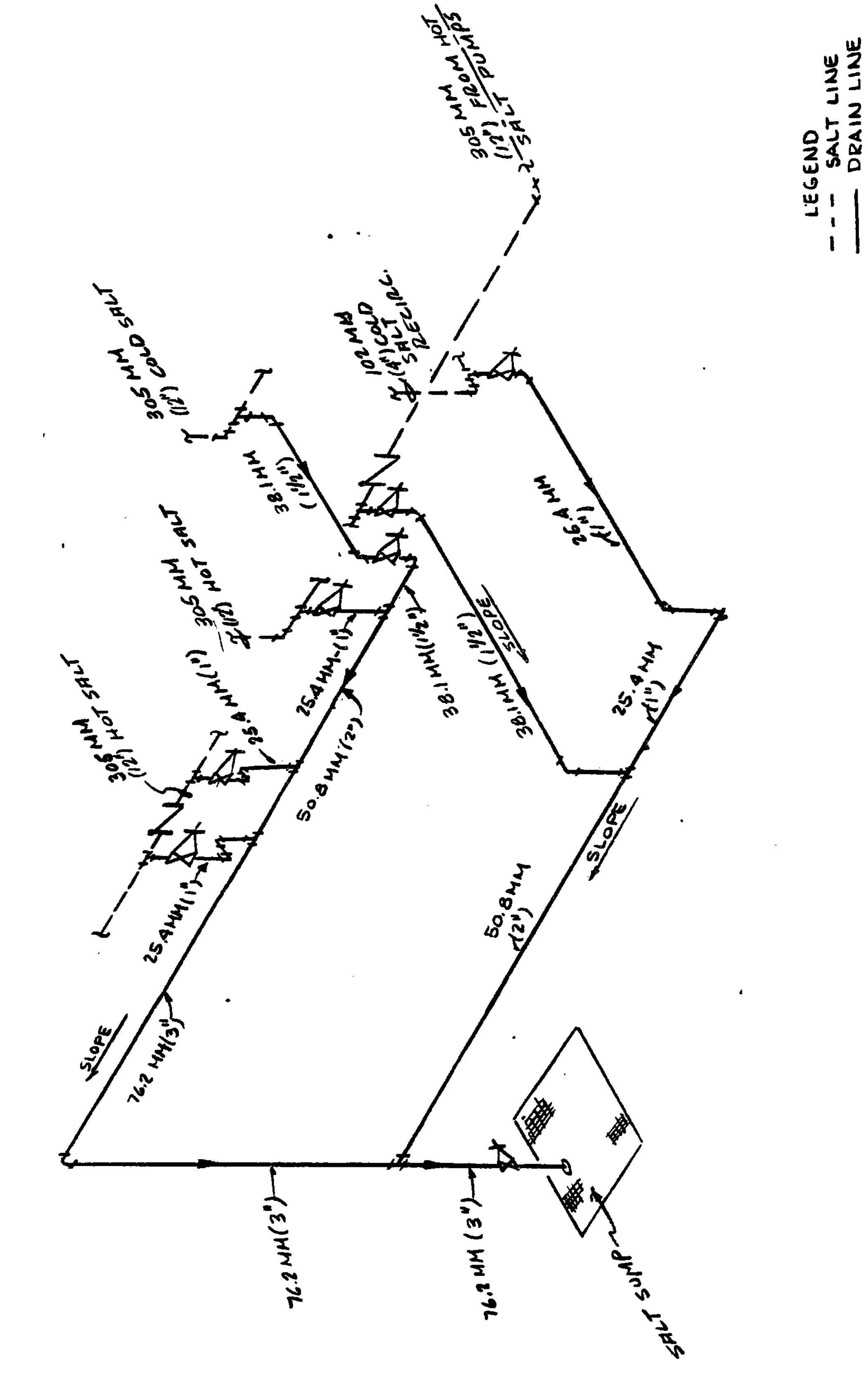


Figure 5.45 Drain System--50-MWe Hybrid SGS

Table 5.16 Molten Salt Drain System Piping Characteristics

	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Design Pressure, kPa gage (lb/in^2g)	1480 (200)	1480 (200)
Design Temperature, °C (°F)	593 (1100)	593 (1100)
Pipe Material	ASTM A312 Stainless Steel	ASTM A312 Stainless Steel
Code	ANSI B31.1	ANSI B31.1
Pipe Sizes (Nominal-Sch. 10S), mm (in.)	254 (1) 381 (1-1/2) 508 (2) 762 (3) 1016 (4)	254 (1) 381 (1-1/2) 508 (2)

Table 5.17 Molten Salt Drain System Sump Tank Characteristics

	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Design Pressure, kPa gage (lb/in^2g)	100 (0)	100 (0)
Design Temperature, °C (°F)	482 (900)	482 (900)
Tank Material	ASTM A312 Stainless Steel	ASTM A312 Stainless Steel
Height, m (ft)	1.5 (5)	1.2 (4)
Diameter, m (ft)	1.5 (5)	1.2 (4)
Wall Thickness, mm (in.)	48 (3/16)	48 (3/16)

Table 5.18 Salt Sump Pump Specifications

	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Number of Pumps	1	1
Design Flow, L/s (gal/min)	9.2 (145)	4.7 (75)
Differential Pressure, kPa (lb/in^2)	220 (32)	220 (32)
Design Head, kPa absolute (lb/in^2a)	320 (47)	320 (47)
Design Temperature, °C (°F)	482 (900)	482 (900)
Specific Gravity	1.77	1.77
Motor Rating, kW (hp)	5.6 (7.5)	3.2 (5.0)
Design Pump kW (hp)	4.7 (6.3)	2.2 (3.0)
Design NPSH, m (ft)	3 (10)	3 (10)

5.7.4 Boiler-Feedwater Recirculation Pumps

Boiler-feedwater recirculation pumps are required to circulate hot water from the steam drum and blend it with the incoming boiler feedwater to maintain the required inlet temperature of 238°C (460°F) to the preheater.

During periods of part-load operation, start-up, and shutdown and at variable operating load factors, the temperature of the feedwater from the turbine-generator boiler feedwater system to the solar SGS preheater may be below the required minimum of 238°C (460°F). To avoid a molten salt freeze-up in the preheater when feedwater temperature is low, hot recirculation water from the evaporator drum at 336°C (636°F) is blended under temperature control with the feedwater to maintain the specified 238°C (460°F) temperature to the preheater.

Two identical pumps, each sized for 100-percent capacity, are specified for both the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS to ensure continuous operation if the pump in service fails and to be consistent with conventional power plant practices. Two 100-percent-capacity pumps, as opposed to three at 50-percent capacity or any other combination, are recommended because of the lower overall capital cost and the reduced cost for piping, valves, controls, and installation.

Pump Specifications. The pumps selected for the service are two Hayward Tyler single-discharge, glandless, DVS-type-motor pumps capable of meeting the design requirements for both the stand-alone and hybrid configurations. They have a record of proven experience in utility power plant boilers operating at the same relative temperatures and pressures as the SGS. These glandless-motor pump units are totally enclosed and self-lubricating in the fluid handled. They are leak-free and run for long periods without servicing.

The pump case can be fitted into the system by its suction and discharge branches. These branches are not subsequently disturbed, since the motor contains all the moving parts and can be detached completely from the pump case.

The pump specifications for the two alternative services are shown in Table 5.19.

Table 5.19 Boiler-Feedwater Recirculation Pump Specifications

	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Number of Pumps	2	2
Design Flow, L/s (gal/min)	18.2 (289)	18.0 (285)
Differential Pressure, kPa (lb/in^2)	241 (35)	241 (35)
Pumping Temperature, °C (°F)	336 (636)	330 (636)
Specific Gravity at Temperature	0.625	0.625
Design NPSH, m (ft)	3.35 (11)	3.29 (10.8)
Design Head, m (ft)	39.6 (130)	39.6 (130)
Design Pump, kW (bhp)	6.4 (8.6)	6.3 (8.5)
Motor Rating, kW (hp)	14.9 (20)	14.9 (20)
Motor Speed, rpm	3500	3500

The pump is mounted directly in the pipeline with the motor below it. It is bottom-supported on a spring-loaded system that allows for thermal expansion of the evaporator system. This pump has been specifically designed for use in forced-circulation boilers.

Recirculation Rate. The hot water circulation rate from the drum to the preheater is a function of both the inlet feedwater temperature and its flow rate. Both variables increase or decrease as the main solar SGS flow rate varies.

The temperatures for the inlet feedwater, the resultant feedwater to the preheater, and the drum recirculation water for the 100-MWe solar stand-alone SGS are shown in Figure 5.46 as a function of SGS load. For example, at a 25-percent main steam flow rate, the inlet feedwater temperature is about 96°C (205°F), and it increases with increasing load factor up to 238°C (460°F) at 100 percent. Recirculation water will thus be required over the entire load range to maintain a constant 238°C (460°F) into the preheater.

The required drum-water recirculation rate as a function of the operating load is shown in Figure 5.47. The combination of variables results in a parabolic pump capacity profile. For the stand-alone configuration, the maximum required pump capacity of 11.30 kg/s (89,715 lb/h), 18.2 L/s (289 gal/min), occurs at 54.3 percent of maximum main steam flow.

The recirculation pump design requirements for the hybrid system are practically the same as for the stand-alone (see Table 5.19). The 50-MWe hybrid SGS differs from the 100-MWe solar stand-alone SGS in that the inlet feedwater temperature is a function of the turbine load, with steam being generated in both the fossil-fuel-fired boiler and the solar units. The turbine can be operating at 100-percent load at every combination of solar SGS and fossil steam rate, or at the solar SGS steam rate only. Thus the inlet feedwater temperature may vary as much as 238°C (460°F) from that shown in Figure 5.46 for all hybrid SGS steam flow rates.

Figures 5.48 and 5.49 illustrate the wide range of recirculation pump capacities that will be required for the hybrid option as a function of total main steam flow rate to the turbine.

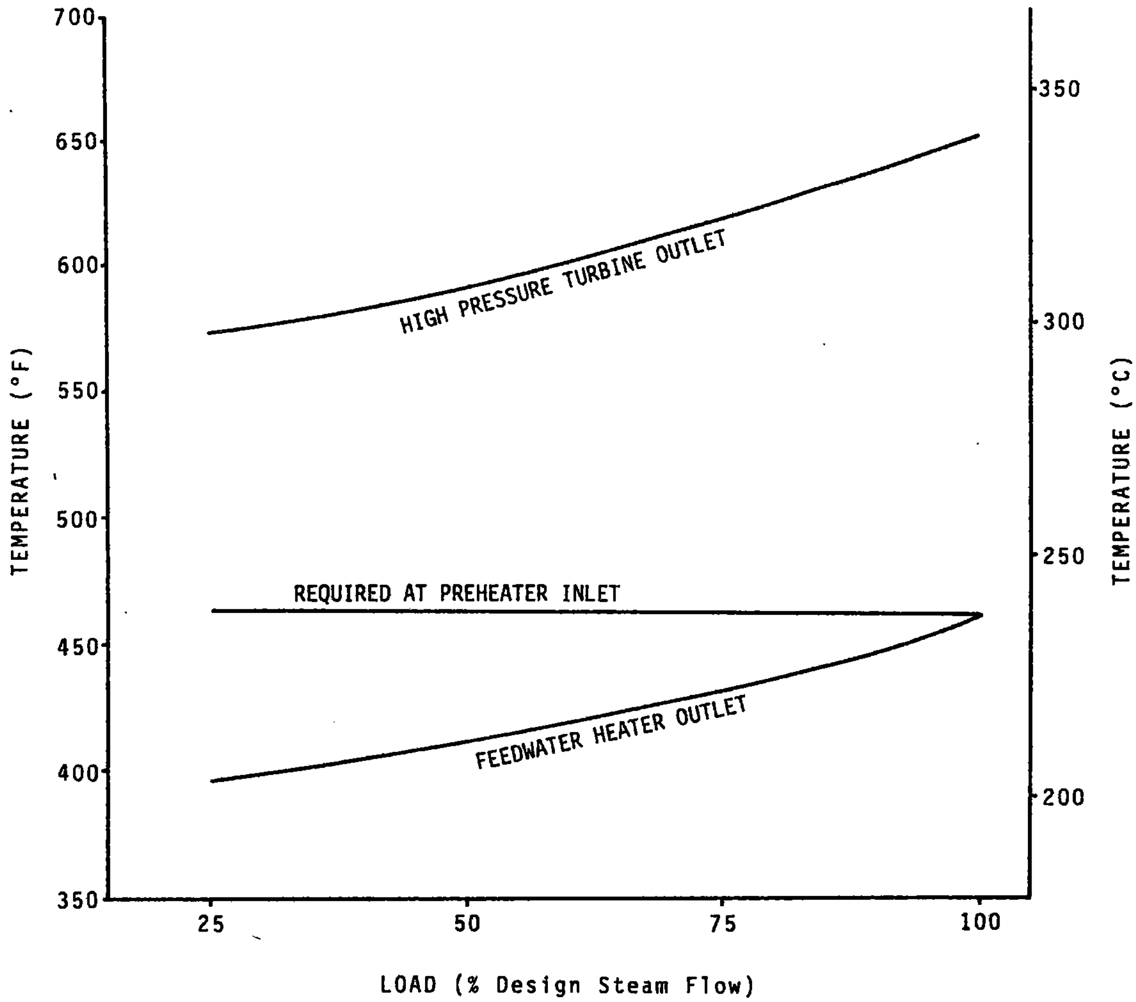


Figure 5.46 High-Pressure Turbine Steam Outlet and Feedwater Temperature Variation--100-MWe Solar Stand-Alone SGS

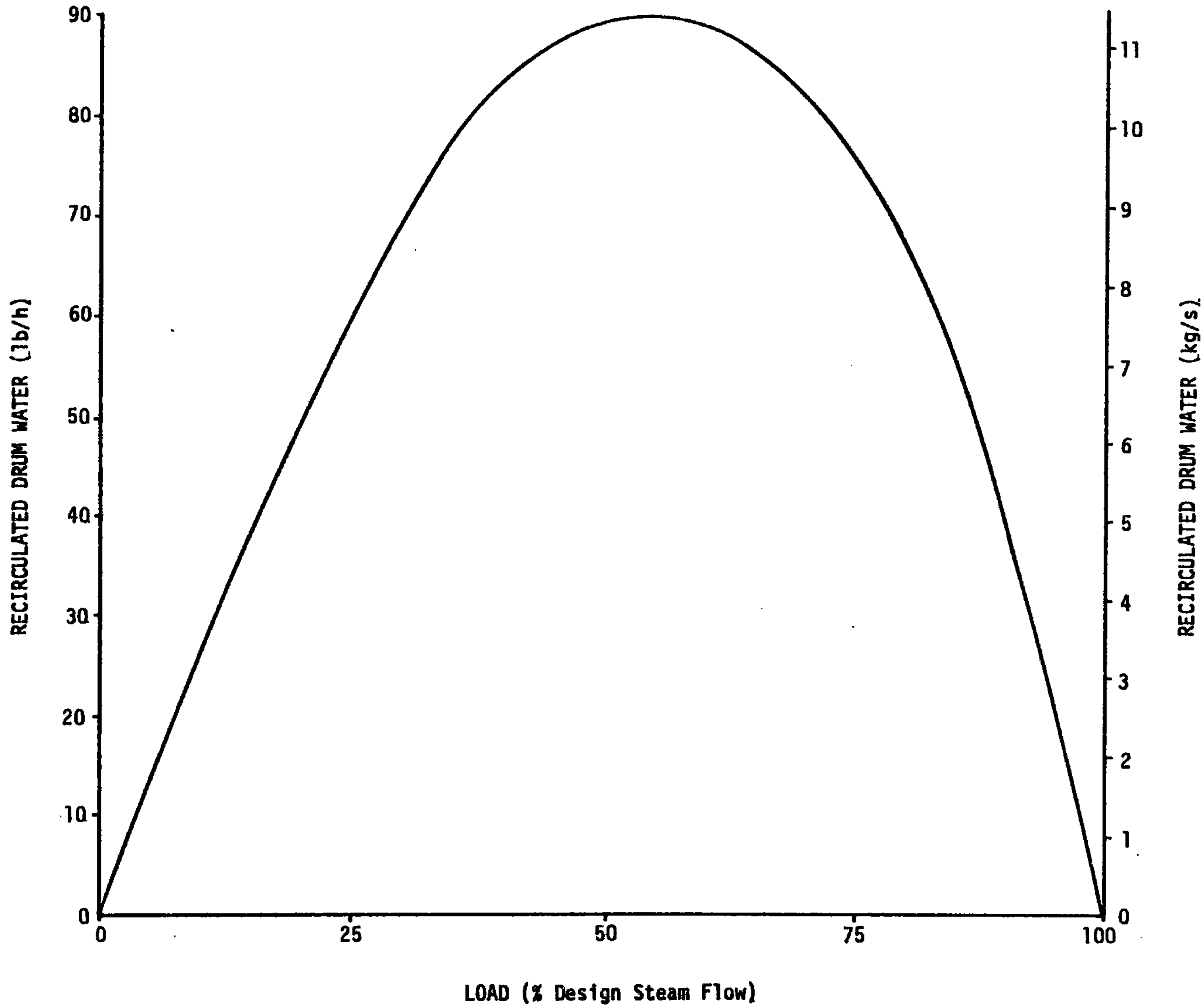


Figure 5.47 Recirculated Drum-Water Requirements--100-MWe Solar Stand-Alone SGS

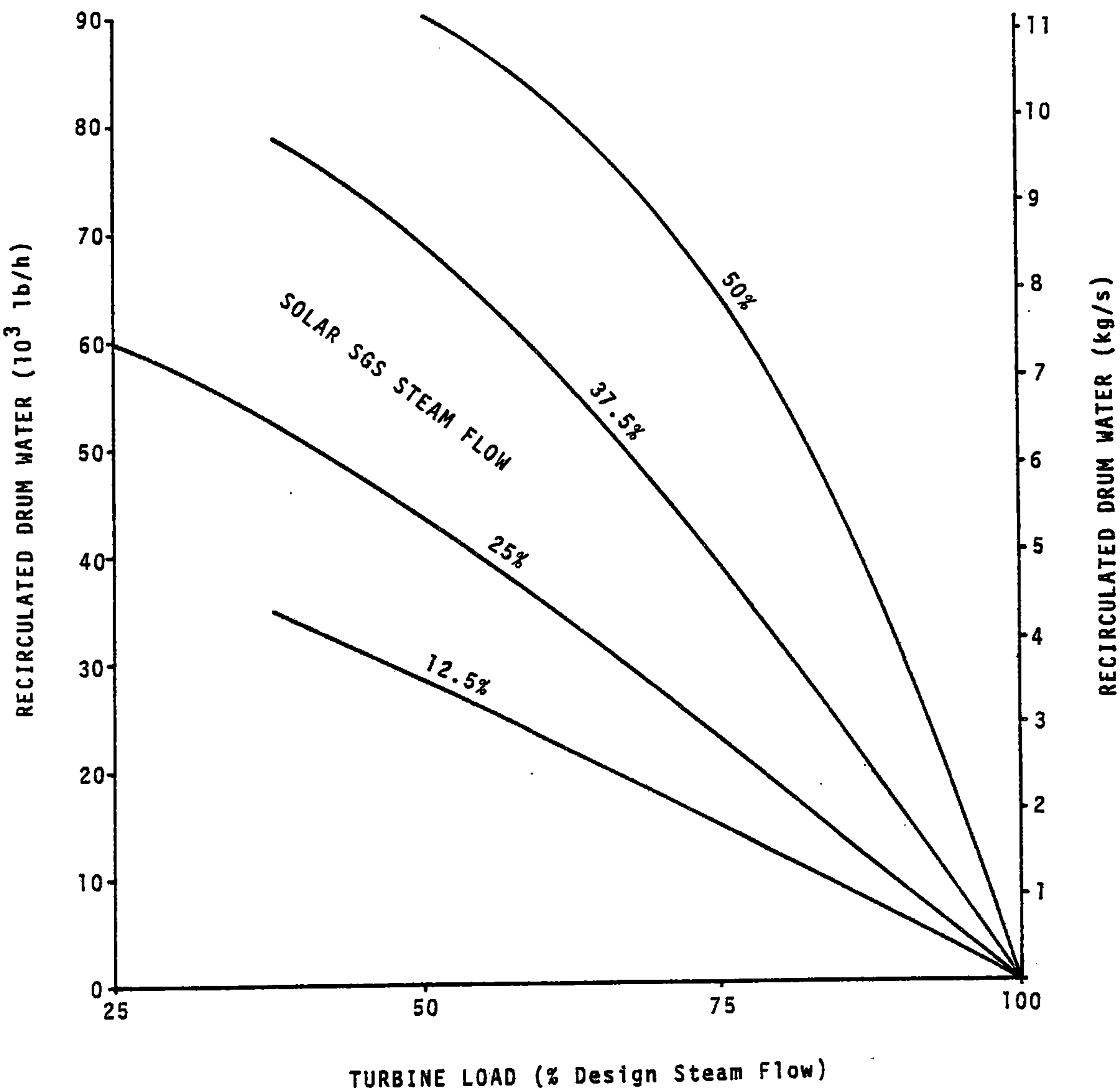


Figure 5.48 Recirculated Drum-Water Requirements (Constant Solar SGS Steam Flow)--50-MWe Hybrid SGS

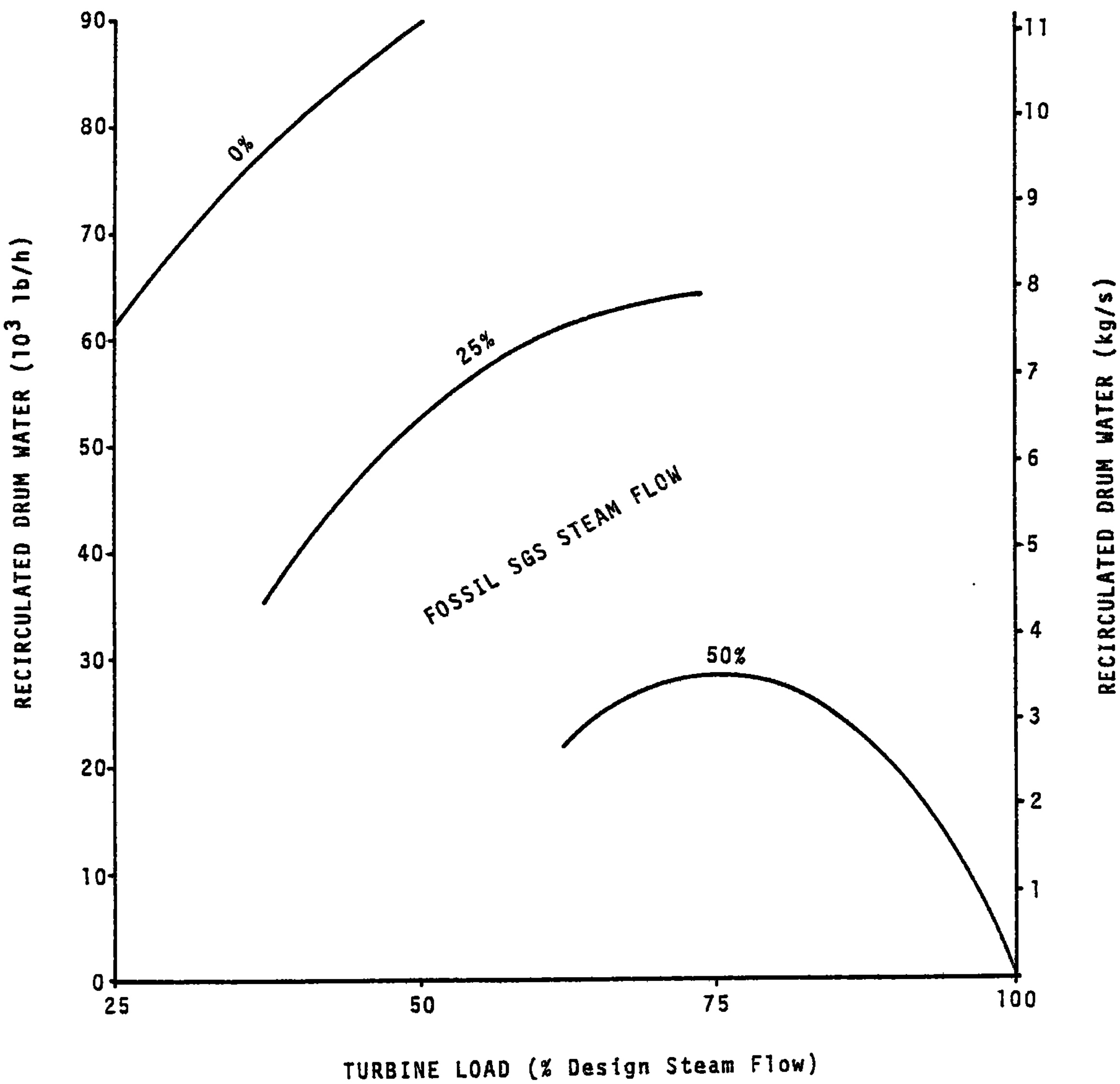


Figure 5.49 Recirculated Drum-Water Requirements (Constant Fossil SGS Steam Flow)--50-MWe Hybrid SGS

Figure 5.48 shows the drum-water recirculation rate as a function of turbine load for four different solar SGS load conditions. For example, at 50-percent solar SGS load, the drum-water recirculation rate varies from zero at full turbine load to 11.14 kg/s (88,450 lb/h), 18.0 L/s (285 gal/min), at 50-percent turbine load. Values for 50-, 37.5-, 25-, and 12.5-percent load for the solar SGS are all plotted on Figure 5.48.

Figure 5.49 shows the drum-water recirculation rate as a percentage of turbine load for three different fossil-fuel-fired boiler load conditions.

5.7.5 Salt Recirculation Pump

A cold-salt recirculation pump is required to control the inlet temperature of the molten salt during start-up. During cold start-up, the temperature of the salt admitted to the steam generating system must be slowly increased from 343 to 563°C (650 to 1045°F). The gradual rise in salt temperature is necessary to avoid thermal shock in the heat exchangers and to meet the steam requirements of prewarming and rolling the turbine. To achieve the desired temperature of the incoming salt, salt from the cold-salt tank [288°C (550°F)] is blended with hot salt [563°C (1045°F)] at the SGS inlet.

To ensure continuous operation and to be consistent with power plant codes, two 100-percent-capacity pumps are specified for the aforementioned duty. Two 100-percent-capacity pumps were selected instead of three at 50 percent (or any other combination) because of the lower capital cost and associated costs of piping, valves, controls, and installation for the two-pump arrangement. The specifications for the cold-salt recirculation pumps for stand-alone and hybrid services are shown in Table 5.20.

Table 5.20 Cold-Salt Recirculation Pump Specifications

	100-MWe Stand-Alone SGS	50-MWe Hybrid SGS
Number of Pumps	2	2
Design Flow L/s (gal/min)	50.5 (800)	25.2 (400)
Differential Pressure kPa (lb/in^2)	1597 (231.6)	1391 (201.8)
Design Head kPa (lb/in^2)	1697 (231.6)	1491 (201.8)
Design Temperature, °C (°F)	287.8 (550)	287.8 (550)
Specific Gravity	1.89	1.89
Motor Rating, kW (hp)	121.3 (161)	56.8 (76)
Design Pump, kW (hp)	113.2 (161)	53.5 (72)
Design NPSH, m (ft)	7.6 (25)	7.6 (25)

The determination of the design flow rates [50.5 L/s (800 gal/min) for the stand-alone and 25.2 L/s (400 gal/min) for the hybrid] were based upon estimated start-up requirements with a margin added to account for variations in the estimated requirements.

5.7.6 Auxiliary Feedwater Heater

An auxiliary feedwater heater brings the boiler feedwater temperature up to the required 238°C (460°F) during 100-MWe stand-alone SGS start-up, while simultaneously heating the SGS by heating the recirculation water when there

there is no molten salt flow through the system. For the 50-MWe hybrid SGS, the auxiliary feedwater heater is not needed because feedwater at the required conditions will be available from the fossil-fuel-fired boiler system.

During cold start-up, 343°C (650°F) salt is admitted to the preheater and evaporator after the preheater and evaporator have been heated to 232°C (450°F) with water to prevent possible thermal shock and to ensure that the molten salt will not freeze. An auxiliary heater, along with the drum-water recirculation pump, performs this function because the boiler feedwater heaters cannot operate during this phase of start-up in the absence of steam generation. The duty of the auxiliary heater is a function of the weight of the preheater, evaporator, piping, total water volume, and the following assumptions:

- 27°C (80°F) initial system temperature
- 38°C/h (100°F/h) allowable rate of temperature change of the heat exchanger
- 75-percent unit efficiency
- Drum-water recirculation pump operation at 100-percent capacity.

Based upon these assumptions, an auxiliary heater capable of delivering 3220 kW (11 x 10^6 Btu/h) is required for the stand-alone configuration. Tentative auxiliary heater specifications are shown in Table 5.21.

Table 5.21 Auxiliary Heater Specifications

Manufacturer	Cleaver Brooks
Model Number	Delta 26 (modified for high pressure)
Flow, L/s (gal/min)	18.2
Minimum Inlet Temperature, °C (°F)	27 (80)
Maximum Outlet Temperature, °C (°F)	232 (450)
Operating Pressure, kPa gage (lb/in^2g)	2758 (400)
Design Pressure, kPa gage (lb/in^2g)	3447 (500)
Net Heat Output, kW (10^6 Btu/h)	3220 (11.0)
Fuel	Oil

5.7.7 Piping

The plant piping system consists of the molten salt piping, the steam and boiler feedwater piping, and the molten salt drain piping. The characteristics of the molten salt piping for the 100-MWe solar stand-alone SGS are shown in Table 5.22 and for the 50-MWe hybrid SGS, in Table 5.23. Pipe sizing is in accordance with standard guidelines governing costs and fluid velocities (with respect to corrosion/erosion and frictional pressure loss). All salt bypass lines adhere to the same design points and material, code, and schedule selections as the primary salt piping. The material of construction for the hot-salt piping is ASTM A312 stainless steel and for the cold-salt piping, ASTM A106B carbon steel.

Table 5.22 Salt Piping Characteristics--100-MWe Solar Stand-Alone SGS

Location	Design Pressure kPa absolute (lb/in²a)	Design Temperature °C (°F)	Material	Code	Schedule	Nominal O.D. mm (in.)
Hot Salt Pump Outlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	406 (16)
Reheater						
Inlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	305 (12)
Outlet	2070 (300)	510 (950)	ASTM A312 Stainless Steel	ANSI B31.1	20	305 (12)
Superheater						
Inlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	356 (14)
Outlet	2070 (300)	454 (850)	ASTM A312 Stainless Steel	ANSI B31.1	20	356 (14)
Evaporator						
Inlet	2070 (300)	468 (875)	ASTM A312 Stainless Steel	ANSI B31.1	20	406 (16)
Outlet	2070 (300)	357 (675)	ASTM A106B Carbon Steel	ANSI B31.1	20	406 (16)
Preheater Outlet	2070 (300)	302 (575)	ASTM A106B Carbon Steel	ANSI B31.1	20	406 (16)

Table 5.23 Salt Piping Characteristics--50-MWe Hybrid SGS

Location	Design Pressure kPa absolute (lb/in²a)	Design Temperature °C (°F)	Material	Code	Schedule	Nominal O.D. mm (in.)
Hot Salt Pump Outlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	305 (12)
Reheater						
Inlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	203 (8)
Outlet	2070 (300)	510 (950)	ASTM A312 Stainless Steel	ANSI B31.1	20	203 (8)
Superheater						
Inlet	2070 (300)	593 (1100)	ASTM A312 Stainless Steel	ANSI B31.1	20	203 (8)
Outlet	2070 (300)	454 (850)	ASTM A312 Stainless Steel	ANSI B31.1	20	203 (8)
Evaporator						
Inlet	2070 (300)	468 (875)	ASTM A312 Stainless Steel	ANSI B31.1	20	305 (12)
Outlet	2070 (300)	357 (675)	ASTM A106B Carbon Steel	ANSI B31.1	20	305 (12)
Preheater Outlet	2070 (300)	302 (575)	ASTM A106B Carbon Steel	ANSI B31.1	20	305 (12)

The characteristics of the steam and boiler feedwater piping system for the 100-MWe solar stand-alone SGS are shown in Table 5.24 and for the 50-MWe hybrid SGS configuration, in Table 5.25. Materials of construction and pipe sizes were selected in accordance with the appropriate codes and standards. The molten salt piping system is discussed in Section 5.7.3.

5.7.8 Insulation and Lagging

The molten salt, feedwater, and steam pipelines are insulated to reduce heat loss and to prevent molten salt pipeline freeze-up. Calcium silicate insulation with aluminum jacketing for weather protection was chosen for this application. A calcium silicate system was selected over a fiberglass or mineral-wool system for strength and durability, thus minimizing future maintenance costs.

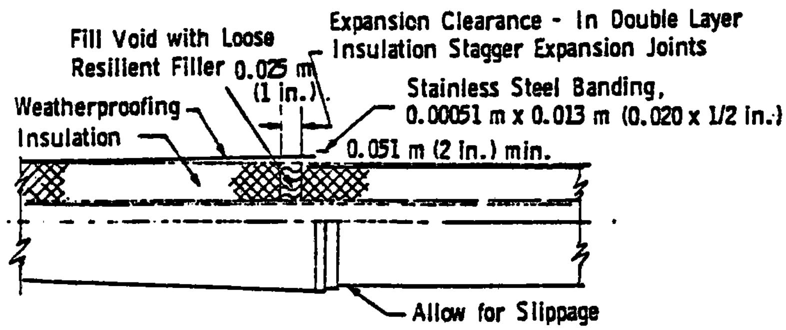
The insulation will accommodate a significant amount of longitudinal growth resulting from the high operating temperatures of the pipelines. The process industries commonly use expansion joint details similar to those shown in Figure 5.50. For longitudinal growth, the high-temperature resilient fill can either be fiberglass or mineral wool, compressed to about 50 percent so that, when the gap widens as the insulation block moves with the pipe during thermal growth, the resilient fill expands and plugs the gap. The expansion joint will be staggered with double- and triple-layer insulation. A similar arrangement for circumferential expansion will be required solely on the largest lines in the plant.

Table 5.24 Steam/Water Piping Characteristics--100-MWe Solar Stand-Alone SGS

Location	Design Pressure MPa absolute (lb/in²a)	Design Temperature °C (°F)	Material	Code	Schedule (or Wall Thickness)	Nominal O.D. mm (in.)
Boiler Feedwater	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	140	203 (8)
Drum Inlet	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	140	203 (8)
Recirculation Pump Suction	15.4 (2225)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	203 (8)
Recirculation Pump Outlet	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	152 (6)
Evaporator Downcomers	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	140	406 (16)
Superheater Inlet	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	120	203 (8)
Main Steam	15.4 (2225)	566 (150)	ASTM A335 P22	ANSI B31.1	41.3 mm (1.63 in.)	254 (10)
Turbine Bypass	15.4 (2225)	566 (1050)	ASTM A335 P22	ANSI B31.1	33.2mm (1.30 in.)	203 (8)
Turbine Bypass	15.4 (2225)	566 (1050)	ASTM A335 P22	ANSI B31.1	41.3 mm (1.62 in.)	254 (10)
Cold Reheat	4.1 (575)	371 (700)	ASTM A335 P22	ANSI B31.1	STD	406 (16)
Hot Reheat	4.1 (575)	566 (1050)	ASTM A335 P22	ANSI B31.1	100	457 (18)
Superheater Bypass	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	120	102 (4)

Table 5.25 Steam/Water Piping Characteristics--50-MWe Hybrid SGS

Location	Design Pressure MPa absolute (lb/in²a)	Design Temperature °C (°F)	Material	Code	Schedule (or Wall Thickness)	Nominal O.D. mm (in.)
Boiler Feedwater	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	152 (6)
Drum Inlet	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	152 (6)
Recirculation Pump Suction	15.4 (2225)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	203 (8)
Recirculation Pump Outlet	15.6 (2250)	371 (700)	ASTM A106B Carbon Steel	ANSI B31.1	120	152 (6)
Evaporator Downcomers	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	140	254 (10)
Superheater Inlet	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	120	152 (6)
Main Steam	15.4 (2225)	566 (1050)	ASTM A335 P22	ANSI B31.1	33.2 mm (1.30 in.)	203 (8)
Turbine Bypass	15.4 (2225)	566 (1050)	ASTM A335 P22	ANSI B31.1	25.4 mm (1.30 in.)	152 (6)
Turbine Bypass	15.4 (2225)	566 (1050)	ASTM A335 P22	ANSI B31.1	33.2 mm (1.30 in.)	203 (8)
Cold Reheat	4.1 (575)	371 (700)	ASTM A335 P22	ANSI B31.1	STD	305 (12)
Hot Reheat	4.1 (575)	566 (1050)	ASTM A335 P22	ANSI B31.1	80	356 (14)
Superheater Bypass	15.4 (2225)	371 (700)	ASTM A335 P22	ANSI B31.1	120	76 (3)



Expansion Joint Spacing - Longitudinal and Circumferential

Temp of Line K	Temp of Line °F	Approximate Spacing Required, m (ft) Carbon Steel	Stainless Steel	Aluminum
422	300	15.2 (50)	10.7 (35)	7.6 (25)
491	425	10.7 (35)	6.1 (20)	5.2 (17)
533	500	6.1 (20)	5.2 (17)	3.4 (11)
630	675	5.2 (17)	3.7 (12)	— —
700	800	4.6 (15)	3.4 (11)	— —
811	1000	3.4 (11)	2.7 (9)	— —
922	1200	2.4 (8)	1.8 (6)	— —
1033	1400	2.1 (7)	1.5 (5)	— —

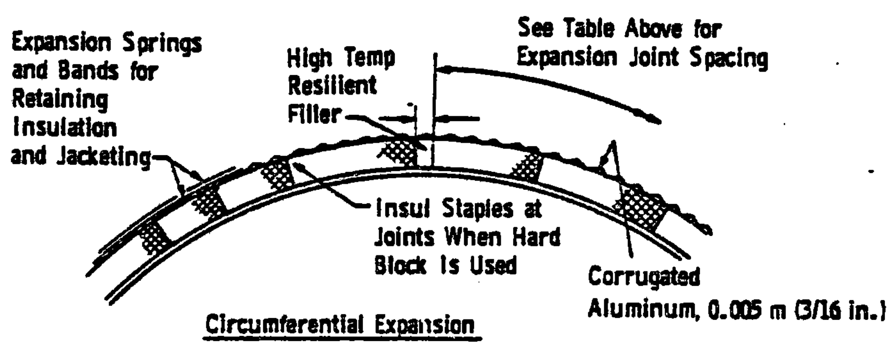


Figure 5.50 Insulation Expansion Joint for Piping

A trade-off between the incremental cost of insulation over the minimum thickness required for personnel protection and the cost of heliostats that would be needed to replace the heat lost will determine the optimum insulation thickness. This method is valid as long as the rate of heat loss is a small fraction of the total plant heat rate.

The insulation thickness specified is based on standard power plant practices. Heat losses are calculated based on these thicknesses and assuming the inside fluid film and pipe-wall resistances to be negligible. The outside air film resistance, while small, is not negligible and has been assumed as 11.35 W/m²·°C (2 Btu/h·ft²·°F) at an ambient temperature of 16°C (60°F). Specifically, we chose molded semicylindrical or block insulation composed of asbestos-free, hydrous calcium silicate, such as Johns-Manville Thermo-12 or equal, with the following physical properties:

Maximum Service Temperature, °C (°F)	815.5 (1500)
Density (Average Dry), kg/m² (lb/ft³)	208.3
Thermal Conductivity-- 371°C (700°F) Mean Temperature, W/m²·cm/°C (Btu/h·ft²·°F/in.)	9.37
Specific Heat, $\frac{W \cdot h}{kg \cdot {}^\circ C}$ (Btu/lb·°F)	0.25 (0.22)

Tables 5.26 through 5.29 show the design temperature, insulation thickness, and heat loss per unit length for the lines and equipment for the 100-MWe solar stand-alone SGS. Tables 5.30 through 5.33 show similar information on insulation and heat loss for the 50-MWe hybrid SGS.

Table 5.26 Steam/Water Line Insulation--100-MWe Solar Stand-Alone SGS

Location	Pipe Size* mm (in.)	Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h°ft)
Superheater Inlet	203 (8)	336 (636)	89 (3.5)	214 (223)
Superheater Bypass	102 (4)	541 (1005)	102 (4)	250 (260)
Main Steam	254 (10)	541 (1005)	127 (5)	389 (405)
Turbine Bypass	203 (8)	541 (1005)	127 (5)	332 (346)
Turbine Bypass	254 (10)	541 (1005)	127 (5)	389 (405)
Cold Reheat	406 (16)	342 (648)	89 (3.5)	355 (370)
Hot Reheat	457 (18)	541 (1005)	127 (5)	576 (600)
Evaporator Downcomers	406 (16)	336 (636)	89 (3.5)	349 (363)
Recirculating Pump Suction	203 (8)	336 (636)	89 (3.5)	214 (223)
Recirculating Pump Outlet	152 (6)	336 (636)	76 (3)	196 (204)
Boiler Feedwater	203 (8)	238 (460)	64 (2.5)	182 (189)
Drum Inlet	203 (8)	366 (636)	89 (3.5)	214 (223)
Recirculating Pump Recirculation Lines	51 (2)	336 (636)	64 (2.5)	111 (116)
Drum Bypass	102 (4)	336 (636)	76 (3)	182 (190)
Quench	51 (2)	238 (460)	51 (2)	108 (113)

*Nominal diameter.

Table 5.27 Salt Line Insulation--100-MWe Solar Stand-Alone SGS

Location	Pipe Size* mm (in.)	Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
Hot Salt Pump Outlet	406 (16)	563 (1045)	127 (5)	342 (356)
Reheater Inlet	305 (12)	563 (1045)	127 (5)	280 (292)
Superheater Inlet	356 (14)	563 (1045)	126 (5)	311 (324)
Superheater, Reheater Bypass	152 (6)	563 (1045)	152 (6)	272 (283)
Reheater Outlet	305 (12)	490 (914)	114 (4.5)	205 (213)
Superheater Outlet	356 (14)	445 (833)	114 (4.5)	240 (250)
Evaporator Inlet	406 (16)	457 (854)	114 (4.5)	244 (254)
Evaporator Outlet	406 (16)	342 (648)	89 (3.5)	296 (308)
Evaporator Preheater Bypass	152 (6)	457 (854)	89 (3.5)	151 (157)
Preheater Outlet	406 (16)	293 (560)	89 (3.5)	296 (308)
From Cold Salt Pump	152 (6)	288 (550)	64 (2.5)	188 (196)
To Hot Salt Tank	152 (6)	563 (1045)	152 (6)	272 (283)

*Nominal diameter.

Table 5.28 Drain Line Insulation--100-MWe Solar Stand-Alone SGS

Drain Line:

Pipe Size* mm (in.)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
25 (1)	51 (2)	82 (85)
38 (1.5)	51 (2)	100 (104)
51 (2)	51 (2)	114 (119)
76 (3)	51 (2)	148 (154)
102 (4)	64 (2.5)	152 (158)

Drain Tank:

Diameter m (ft)	Height m (ft)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
1.5 (5)	1.5 (5)	38 (1.5)	4350 (4532)

*Nominal diameter.

Table 5.29 Equipment Insulation--100-MWe Solar Stand-Alone SGS

Equipment	Cylindrical Size m (ft - in.)	Shell Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
Reheater	0.89 (2 - 11-1/16)	563 (1045)	127 (5)	1054 (1097)
Superheater	0.95 (3 - 1-1/2)	563 (1045)	127 (5)	1118 (1164)
Evaporator				
Body	1.50 (4 - 11)	446 (835)	114 (4.5)	1318 (1372)
Drum	2.45 (8 - 1/2)	336 (636)	89 (3.5)	1790 (1863)
Preheater	1.34 (4 - 4-3/4)	343 (649)	89 (3.5)	1031 (1073)

Table 5.30 Steam/Water-Line Insulation--50-MWe Hybrid SGS

Location	Pipe Size* mm (in.)	Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
Superheater Inlet	152 (6)	336 (636)	76 (3)	196 (204)
Superheater Bypass	76 (3)	541 (1005)	64 (2.5)	269 (280)
Main Steam	203 (8)	541 (1005)	127 (5)	332 (346)
Turbine Bypass	152 (6)	541 (1005)	152 (6)	244 (254)
Turbine Bypass	203 (8)	541 (1005)	127 (5)	332 (346)
Cold Reheat	305 (12)	342 (648)	89 (3.5)	356 (371)
Hot Reheat	356 (14)	541 (1005)	127 (5)	461 (480)
Evaporator Downcomers	254 (10)	336 (636)	89 (3.5)	305 (317)
Recirculating Pump Suction	203 (8)	336 (636)	89 (3.5)	214 (223)
Recirculating Pump Outlet	152 (6)	336 (636)	76 (3)	196 (204)
Boiler Feedwater	152 (6)	238 (460)	64 (2.5)	192 (200)
Drum Inlet	152 (6)	336 (636)	76 (3)	196 (204)
Recirculating Pump Recirculation Lines	38 (1.5)	336 (636)	64 (2.5)	119 (124)
Drum Bypass	76 (3)	336 (636)	64 (2.5)	170 (177)
Quench	38 (1.5)	238 (460)	51 (2)	95 (99)

*Nominal diameter.

Table 5.31 Salt-Line Insulation--50-MWe Hybrid SGS

Location	Pipe Size* mm (in.)	Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h·ft)
Hot Salt Pump Outlet	305 (12)	563 (1045)	127 (5)	426 (443)
Reheater Inlet	203 (8)	563 (1045)	127 (5)	325 (338)
Superheater Inlet	203 (8)	563 (1045)	127 (5)	325 (338)
Superheater, Reheater Bypass	102 (4)	563 (1045)	102 (4)	243 (253)
Reheater Outlet	203 (8)	490 (914)	102 (4)	298 (310)
Superheater Outlet	203 (8)	445 (833)	89 (3.5)	279 (290)
Evaporator Inlet	305 (12)	457 (854)	114 (4.5)	338 (352)
Evaporator Outlet	305 (12)	342 (648)	89 (3.5)	289 (301)
Evaporator Preheater Bypass	102 (4)	457 (854)	76 (3)	213 (222)
Preheater Outlet	305 (12)	293 (560)	89 (3.5)	171 (178)
From Cold Salt Pump	102 (4)	288 (550)	64 (2.5)	134 (139)
To Hot Salt Tank	102 (4)	563 (1045)	102 (4)	243 (253)

*Nominal diameter.

Table 5.32 Drain-Line Insulation--50-MWe Hybrid SGS

Drain Line:

Pipe Size* mm (in.)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
25 (1)	51 (2)	82 (85)
38 (1.5)	51 (2)	100 (104)
51 (2)	51 (2)	114 (119)
76 (3)	51 (2)	148 (154)

Drain Tank:

Diameter m (ft)	Height m (ft)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
1.2 (4)	1.2 (4)	38 (1.5)	4350 (4532)

*Nominal diameter.

Table 5.33 Equipment Insulation--50-MWe Hybrid SGS

Equipment	Cylindrical Size m (ft - in.)	Shell Temperature °C (°F)	Insulation Thickness mm (in.)	Heat Loss W/m (Btu/h•ft)
Reheater	0.66 (2 - 2)	518 (965)	127 (5)	669 (696)
Superheater	0.61 (2)	494 (921)	127 (5)	587 (611)
Evaporator				
Body	1.07 (3 - 6)	394 (742)	89 (3.5)	928 (966)
Drum	1.98 (6 - 6)	343 (649)	114 (4.5)	1132 (1178)
Preheater	0.76 (2 - 6)	318 (605)	89 (3.5)	528 (550)

5.7.9 Electric Trace Heating

Electric trace heaters are required on all molten salt equipment and pipelines to prevent freeze-up (solidification) of the liquid molten salt. The trace heaters are not on during periods of normal operation. They are used primarily for start-up, shutdown, and standby operations, when there may be little or no salt flow for extended periods, and for emergency operation. If the system is designed with dead-end pockets or salt lines where there is no net salt flow (such as in drain lines on the upstream side of the valves, vent lines, safety valves, or rupture disks), trace heaters may need to be used constantly to prevent local salt freeze-up.

Electric trace heaters on the heat exchanger shells, all interconnecting salt piping, and the molten salt drain system are sized to preheat and maintain the salt piping and exchanger shells at 287.8°C (550°F). The heating elements on the salt piping are positioned along the pipe axis at approximately six locations around the pipe surface. Heating elements on the heat exchanger shells are positioned parallel to the exchanger axis in single loops. The heat exchangers and all interconnecting piping are insulated with calcium silicate and covered by aluminum lagging. For preheat, the trace heaters require a 240-V power supply; for protection from freezing, a 120-V power supply is required.

Two layers of insulation surround the trace heaters on the high-temperature salt lines. The flexible stainless steel cable is insulated with magnesium oxide and covered with a solid Type 304L seamless stainless steel sheath (Thermon SSK or equivalent) with the following specifications:

Maximum Temperature Rating, °C (°F)	537.8 (1000)
Output, W/m (W/ft)	328 (100)

Table 5.34 summarizes the overall heat-tracing requirements for both the 100-MWe solar stand-alone and 50-MWe hybrid SGS units. The electric trace heating requirements for these units are shown in Tables 5.35 and 5.36. The tables show the individual heating requirements for each of the four heat exchangers, the individual molten salt lines, the drain system piping, and the drain tank.

Table 5.34 Heat-Tracing Power Requirements (W)

Equipment	100-MWe Solar Stand-Alone SGS	50-MWe Hybrid SGS
Heat Exchangers	62,600	51,908
Salt Piping	37,770	35,372
Drain System Piping	10,184	9,405
Drain Tank	6,630	5,304
Total	117,184 (117.2 kW)	101,989 (102.0 kW)

5.7.10 Safety Valves

The steam generating systems of the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS arrangements are equipped with safety valves in accordance with the ASME Boiler and Pressure Vessel Code, Section I. The valves, shown in Tables 5.37 and 5.38 are reaction-type, designed for high pressure and high capacity. They are sized according to the following design criteria.

Superheater Outlet and Evaporator Safety Valves.

- The superheater outlet valve capacity is at least 20 percent of total design steam flow.
- The set pressure of the superheater outlet valve is less than the set pressure of the drum valve (less pressure differential)
- The set pressures of additional drum valves are less than 103 percent of the lowest set pressure.

Table 5.35 Heat-Tracing Demand--100-MWe Solar Stand-Alone SGS

Line or Equipment	Tracing Requirement W/m (W/ft)	Length or Height m (ft)	Demand/ Component, W
Drain Tank	4350 (1326)	1.5 (5)	6,630
25 mm (1 in.) Drain Line	81 (25)	34.7 (114)	2,827
38 mm (1.5 in.) Drain Line	100 (31)	32.6 (107)	3,264
51 mm (2 in.) Drain Line	114 (35)	14.0 (46)	1,605
76 mm (3 in.) Drain Line	148 (45)	13.7 (45)	2,025
102 mm (4 in.) Drain Line	152 (46)	3.0 (10)	463
Preheater	962 (294)	19.2 (63)	18,527
Superheater	655 (200)	19.8 (65)	12,986
Evaporator	856 (262)	21.9 (72)	18,830
Reheater	618 (188)	19.8 (65)	12,257
Hot-Salt Pump Outlet	372 (113)	10.7 (35)	3,963
Reheater Inlet	305 (94)	9.1 (30)	2,809
Superheater Inlet	339 (103)	6.7 (22)	2,275
Superheater/Reheater Bypass	296 (90)	16.8 (55)	4,970
Reheater Outlet	223 (69)	3.0 (10)	686
Superheater Outlet	261 (79)	6.1 (20)	1,588
Evaporator Inlet	266 (82)	25.9 (85)	6,940
Evaporator Outlet	322 (99)	7.3 (24)	2,378
Evaporator/Preheater Bypass	254 (84)	16.8 (55)	4,610
Preheater Outlet	322 (99)	17.1 (56)	5,548
From Cold-Salt Pump	164 (50)	12.2 (40)	2,003
To Hot-Salt Tank	296 (90)	*	*
Total			117,184

*Included in superheater/reheater bypass.

Table 5.36 Heat-Tracing Demand--50-MWe Hybrid SGS

Line or Equipment	Tracing Requirement W/m (W/ft)	Length or Height m (ft)	Demand/ Component, W
Drain Tank	4350 (1326)	1.2 (4)	5,304
25 mm (1 in.) Drain Line	81 (25)	58.2 (191)	4,737
38 mm (1.5 in.) Drain Line	100 (31)	36.9 (121)	3,691
51 mm (2 in.) Drain Line	114 (35)	8.5 (28)	977
Preheater	528 (161)	17.1 (56)	9,016
Superheater	587 (179)	16.8 (55)	9,845
Evaporator	958 (292)	21.5 (70.5)	20,603
Reheater	669 (204)	18.6 (61)	12,444
Hot-Salt Pump Outlet	426 (130)	10.7 (35)	4,550
Reheater Inlet	325 (99)	9.1 (30)	2,970
Superheater Inlet	325 (99)	6.7 (22)	2,178
Superheater/Reheater Bypass	243 (74)	16.8 (55)	4,070
Reheater Outlet	298 (91)	3.0 (10)	910
Superheater Outlet	279 (85)	6.1 (20)	1,700
Evaporator Inlet	338 (103)	25.9 (85)	8,755
Evaporator Outlet	289 (88)	7.3 (24)	2,112
Evaporator/Preheater Bypass	213 (65)	16.8 (55)	3,575
Preheater Outlet	171 (52)	17.1 (56)	2,912
From Cold-Salt Pump	134 (41)	12.2 (40)	1,640
To Hot-Salt Tank	243 (74)	*	*
Total			101,989

*Included in superheater/reheater bypass.

Table 5.37 Safety Valves--100-MWe Solar Stand-Alone SGS

Location	Size	Crosby Style	Set Pressure kPa-gage (lb/in²g)	Capacity kg/s (lb/h)
Drum	2-1/2 x K x 6	HE86W	15,341 (2225)	24.0 (190,817)
Drum	2-1/2 x K x 6	HE86W	15,514 (2250)	24.3 (192,929)
Drum	2-1/2 x K x 6	HE86W	15,790 (2290)	24.7 (196,357)
Superheater	2-1/2 x KZ x 6	HCA88W	14,135 (2050)	24.2 (191,773)
Reheater Outlet	6 x R x 8	HCA38W	3,723 (540)	39.5 (313,417)
Reheater Inlet	4 x Q x 8	HC36W	3,965 (575)	33.6 (266,977)
Reheater Inlet	3 x M x 6	HC36W	4,068 (590)	11.2 (89,233)
Preheater Outlet	3 x M x 6	HE86W	15,514 (2250)	50.4 (399,950)
Preheater Outlet	3 x M x 6	HE86W	15,686 (2275)	51.1 (405,338)

Table 5.38 Safety Valves--50-MWe Hybrid SGS

Location	Size	Crosby Style	Set Pressure kPa-gage (lb/in²g)	Capacity kg/s (lb/h)
Drum	2-1/2 x K x 6	HE86W	15,341 (2225)	24.0 (190,817)
Drum	2-1/2 x K x 6	HE86W	15,514 (2250)	24.3 (192,929)
Superheater	2 x J x 6	HCA88W	14,135 (2050)	12.3 (97,544)
Reheater Outlet	4 x Q x B	HCA36W	3,723 (540)	27.3 (216,375)
Reheater Inlet	4 x P x 6	HC36W	3,965 (575)	19.6 (155,682)
Preheater Outlet	3 x M x 6	HE86W	15,514 (2250)	50.4 (399,950)

- The total valve capacity is greater than the design steam flow capacity.

Reheater Safety Valves.

- The outlet valve capacity is at least 15 percent of reheater design flow capacity.
- The set pressure of the outlet valve is less than the set pressure of the inlet valve (less pressure differential).
- The total valve capacity is greater than the design flow capacity.

Preheater Safety Valves.

- The set pressures of the additional valves are less than 103 percent of the lowest set pressure.
- The total valve capacity is greater than the design steam flow capacity.

For identification, Crosby-style designations are included in Tables 5.37 and 5.38.

5.7.11 Molten Salt Pressure-Relief System

The molten salt pressure-relief system protects the molten salt system and relieves any sudden pressure increase on the solar SGS molten salt system. The system depicted in Figure 5.51 is similar to those used in chemical facilities, where the danger of rapid expansion or explosions exists.

This system has been specified rather than designed for the solar SGS. As part of a detailed design, a very careful analysis must be made of the impact of leaking into the salt systems, including leak rate and the extent of pressure increase. Consideration should be given to locating the pressure-relief devices directly on the shell of each exchanger.

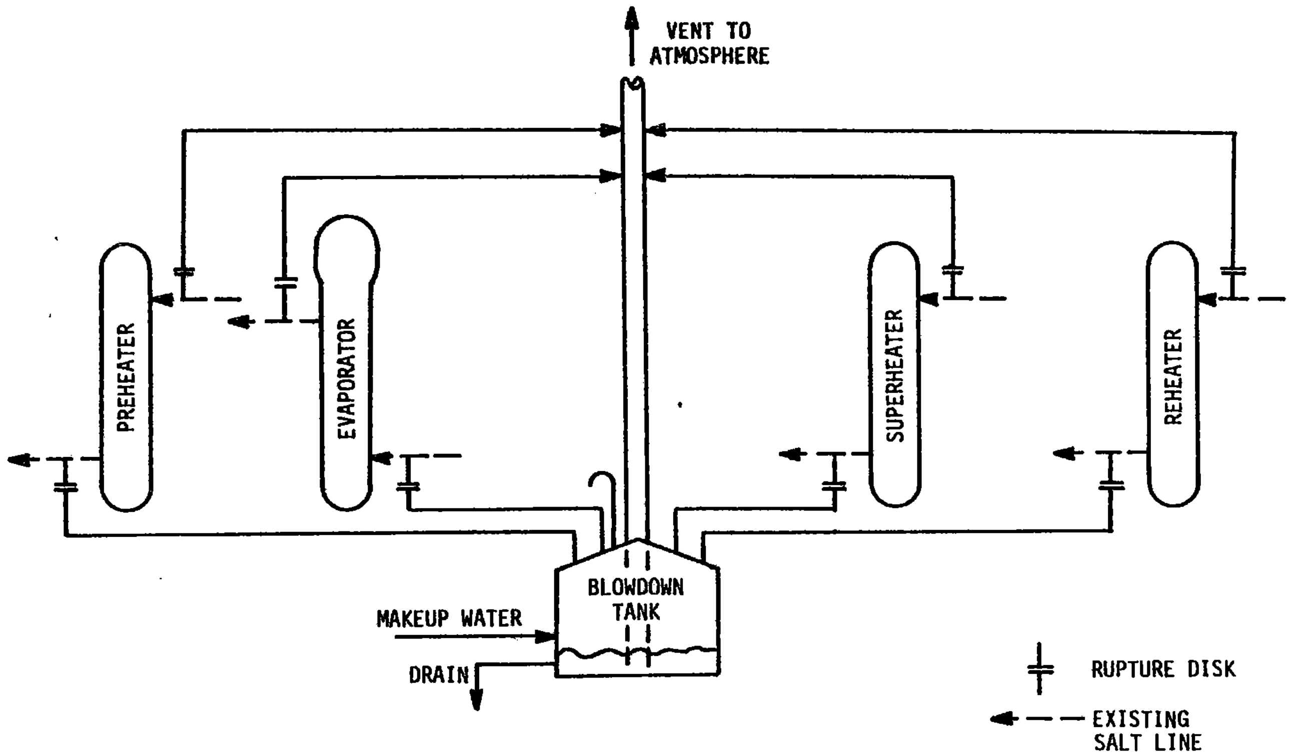


Figure 5.51 Salt Pressure-Relief System

Over its normal operating life, the possibility always exists that fluid will leak from the higher operating pressure side of a shell-and-tube heat exchanger to the lower pressure side, despite the care taken in its design and fabrication. The operating pressure on the steam generating side of the four heat exchangers is much higher than on the molten salt side. Therefore, any leak will be from the steam/water side into the molten salt side. The saturated boiler feedwater will vaporize and be superheated very rapidly up to the temperature of the salt. Without a pressure-relief system, there is a high probability that the inertia of the salt will rupture the shell (molten salt) side as well as send a shock wave throughout the entire solar SGS system and connecting molten salt piping.

Full line-size rupture disks are located on the inlet and outlet molten salt lines on each of the four heat exchangers. To minimize the probability of injury to personnel and facilities, the discharge from the rupture disks is sent to a blowdown tank. The upper four lines are directed into a standpipe, which is vented to the atmosphere to relieve the pressure. The bottom four lines are sent directly to the blowdown tank. The blowdown tank could be either an open-top tank or a tank adequately ventilated to prevent any pressure buildup from steam. A water level sufficient to cool and solidify the molten salt will be maintained in the tank.

The rupture disks, as specified, will be made of stainless steel or a higher-quality alloy and must be located, electrically heat traced, and insulated to prevent the salt from plating out or solidifying on the face of the rupture disk. Plating out or solidifying would tend to reduce the effectiveness of the disk and change its rupture pressure point.

The discharge lines can be of uninsulated carbon steel arranged so that there are no pockets to capture moisture or salt in them. All lines must slope so that they drain by gravity into the blowdown tank.

The hot-salt pumps will be instrumented to shut down automatically if there is a sudden pressure loss in the system.

5.7.12 Instruments and Controls

This section discusses instruments and controls included in the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS--instruments of measurement; actuators such as valves, switches, control loops (closed or open); instrumentation at the control desk such as displays and keyboards; and the recorders belonging to the solar SGS that are housed in the control room. The philosophy for system control and operation is described in Section 5.6.

The Instruments and Control Systems are divided into two structures--the infrastructure and the superstructure. Figure 5.52 shows, in an architectural approach, all control functions in a hierarchic arrangement of blocks surrounded by the data communications environment, which ensures the message exchange between control function blocks. A distributed control system of the third generation was considered featuring a complete peer-to-peer exchange of message between control function blocks.

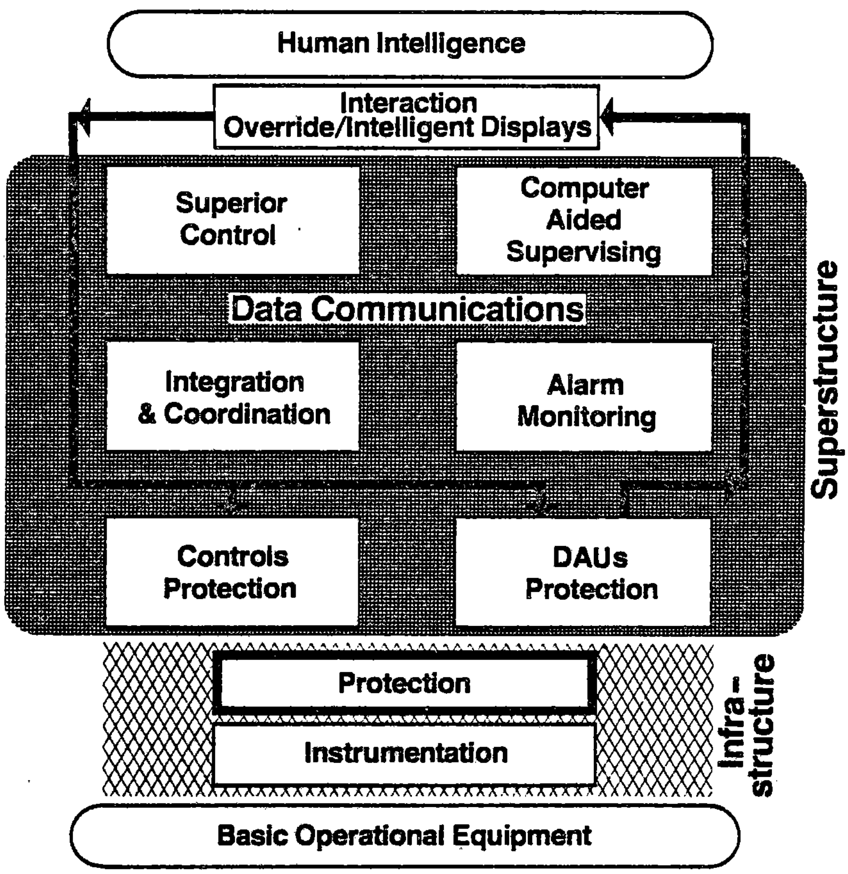


Figure 5.52 Gibbsway I&C Architectural Approach

The control system infrastructure consists of:

- Actuators. The actual working conditions considered are the chemical properties of the molten salt and the temperature and pressure of the salt and steam. The actuators for the 20 control-valve stations, as well as the spray, bypass, start-up, and emergency letdown stations and the molten salt drain system controls are included. Each control valve is framed by two isolation valves and a 100-percent bypass valve. All their actuators are motorized in accordance with their function--control or isolation. In total, there are 80 valves and 25 block valves with motorized remote control.

- Sensors for Discrete Inputs and Transducers for Electric Analog Inputs. There are a total of 50 analog transducers (differential pressure, level, pressure, temperature), 10 discrete sensors for the auxiliary electric system for the Data Acquisition System (DAS), and 2 distributed Data Acquisition Units (DAUs) included in the 100-MWe solar stand-alone SGS.

- Mechanical Protection Means. Safety valves, for example, are included within the protection block of Figure 5.52.

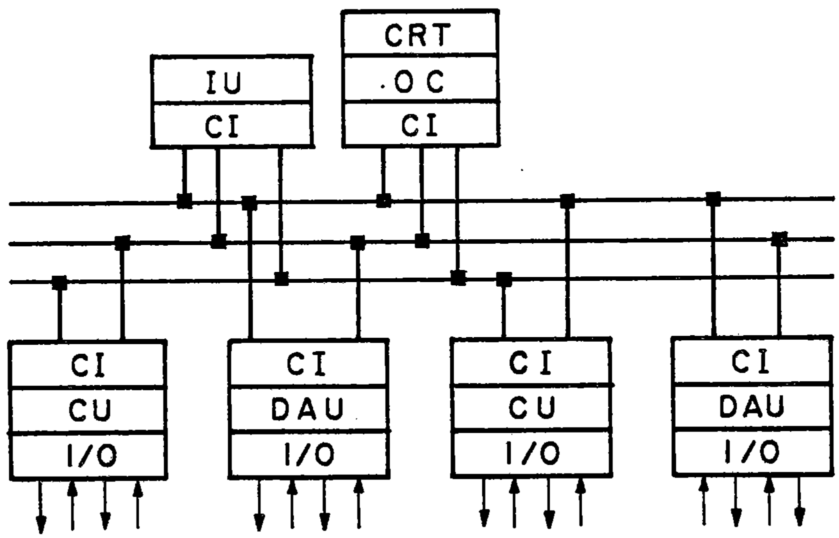
- Hardwired Interlocks and Protection. These are included as required for solar SGS safety (e.g., pressure switch linked directly to bypass stations).

- Signal Conditioning and Acquisition Logic. A distributed digital network, as explained in the Control System Superstructure subsection, is included. The signal conditioners convert electrical signals from sensors and transducers into electrical data pulses compatible with the input/output of digital control stations.

  Multiple transducers (1 out of 2) are used as a basis for the acquisition logic. All safety parameters, such as superheated steam pressure, are also acquired through multiple transducers (1 out of 3).

- Local Input/Output Wiring Between Instrumentation and Controls and Data Acquisition Units (DAUs). The wiring of input data from instrumentation apparatus or from the selector to the control units and DAUs is included.

A distributed digital control network is selected as the basis of the control superstructure. (The data communication layer of Figure 5.52 surrounds all distributed capacity.) This control network, as illustrated in Figures 5.52 and 5.53, is compatible with the third generation of distributed control systems



LEGEND:
CI - COMMUNICATIONS INTERFACE
CU - CONTROL UNIT
DAU- DATA ACQUISITION UNIT
IU - INTEGRATION UNIT
OC - OVERRIDE CONTROL
CRT- PARAMETER VIDEO DISPLAY
I/O - INPUT/OUTPUT

Figure 5.53 SGS Control Subnetwork

such as the Barly 90, Siemens Teleperm M, Forney, etc. The control functions of the superstructure consist of:

- Integrating the solar SGS control system with other plant subsystems--performed by the integrating unit (IU) and the common data buses (Figure 5.53).
- Coordinating of subgroups--IU.
- Integrated control with the turbine generator subsystem (no boiler or turbine following)--IU
- Accommodating the operating modes, regular and emergency--IU
- Semiautomatic start-up and shutdown--IU and Override Control OC
- Data Acquisition System and reporting system
- Alarm monitoring--through DAUs and data buses
- Operator control override unit--OC.

A subnetwork of the general plant network is considered for the solar SGS. The control stations perform the previously listed functions.

Figure 5.53 shows a station provided with a parameter video display (CRT) and a keyboard facility [override control (OC)]. These apparatus, placed on a control desk, will provide a local operator with the necessary information and also with the possiblity to override automatic controls. Local operator supervision is only required occasionally. All supervisory control is performed regularly by operators housed in the central control room.

Several analog recorders are included for the solar SGS but are located in the Central Control Room.

5.7.13 Auxiliary Electrical System

An electrical supply system is required for the 100-MWe solar stand-alone SGS. Figure 5.54 is a one-line electric diagram for the 100-MWe SGS; Figure 5.55, for the 50-MWe hybrid SGS. The electric power requirements for the SGS, as shown on the one-line diagrams, represent a relatively small part of the total power plant electrical system. As such, the transformers, the emergency diesel generators, and the high-power bus-bar lines are not included as part of the SGS auxiliary power system. The scope includes the two 460-V 600-A bus-bar lines, low-voltage distribution transformers, and distribution wiring.

The auxiliary electric systems for the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS are similar. The only difference is the larger power requirements and the start-up heater for the 100-MWe unit. The approximate installed power is 400 kVA for the 100-MWe unit and 300 kVA for the 50-MWe unit. The estimated simultaneity factor is 0.80 percent.

A 480-V, three-phase, 60 Hz distribution system is sufficient to meet all power needs. For redundancy, we elected a symmetrical configuration consisting of two motor-control centers (MCCs 1 and 2), shown in Figure 5.54, which feed from two different 4160/480 power centers. Each 4160/480 power center has an emergency diesel generator set which services the emergency requirements of the power plant. The two MCCs have a tie breaker (C), so that one power center can supply both 480-V MCCs. The tie breaker is interlocked with the main circuit breakers (A and B) a NAND logic operation.

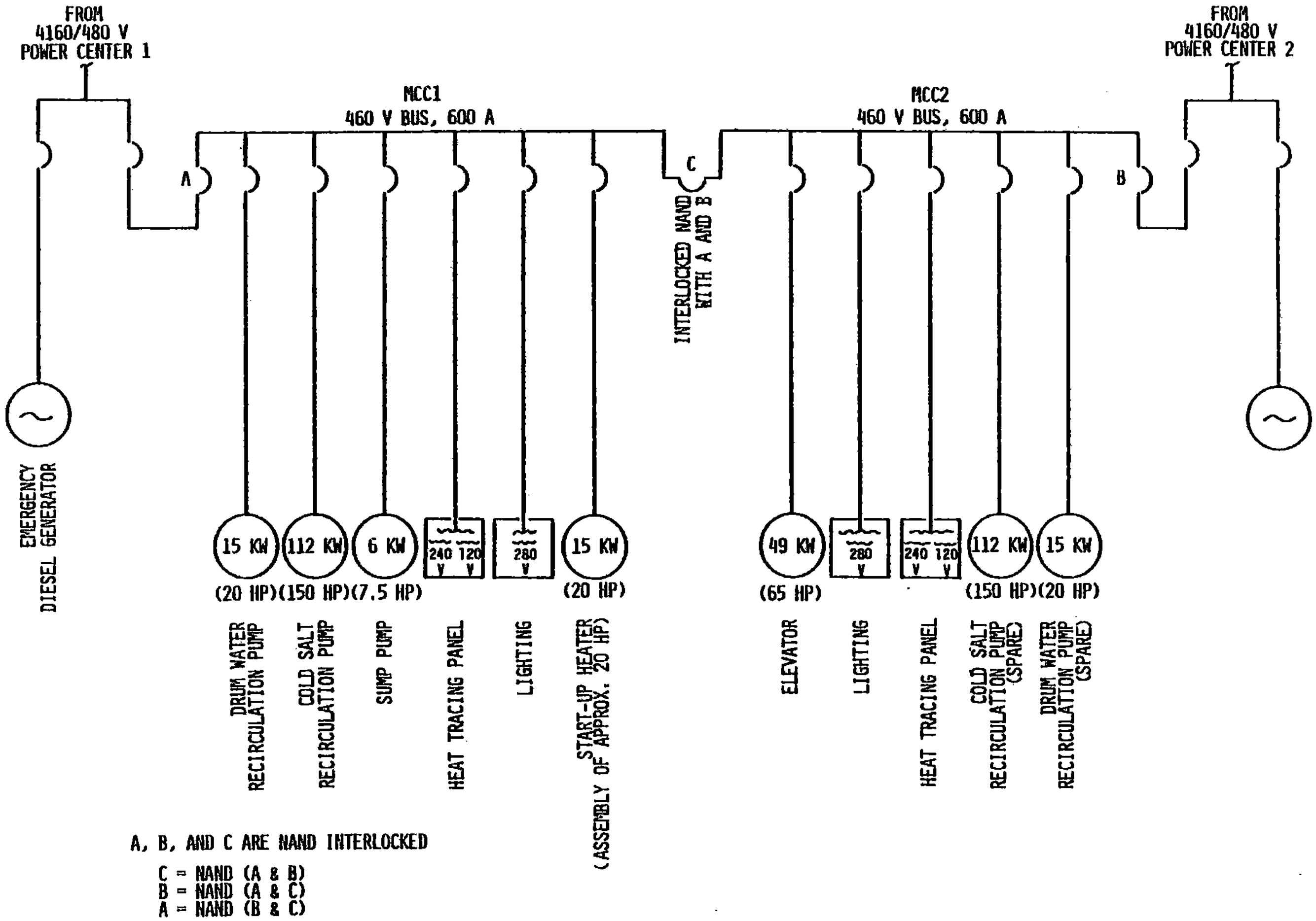


Figure 5.54 Electric One-Line Diagram--100-MWe Solar Stand-Alone SGS

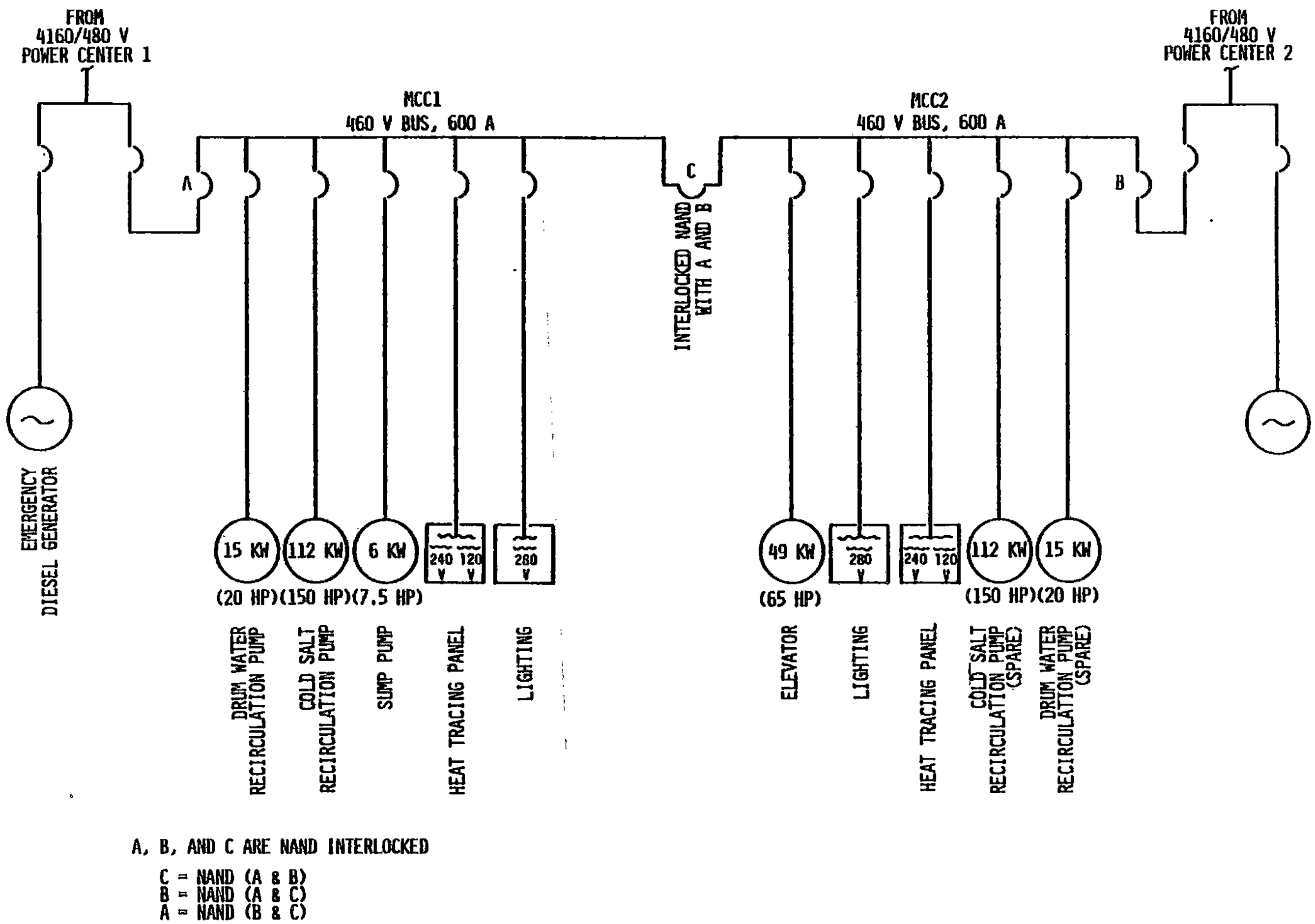


Figure 5.55 Electric One-Line Diagram--50-MWe Hybrid SGS

The overall unit lighting is estimated at 100 kVA for both units. Security lighting will be supplied by a special circuit from the EPGS subsystem. Heat tracing has a total power demand of 117,200 W (117.2 kW) with 240 V for preheat and 120 V for freeze protection for the 100-MWe unit and 102,000 W (102 kW) with 240 V for preheat and 120 V for freeze protection for the 50-MWe unit. The master control system will initiate the transfer between the two voltages. The start-up heater has several motors totalling 14,900 W (20 hp).

The MCCs are metal clad with dials on the door panels and access by front and rear doors. The circuit breakers are remotely and manually operated.

The main circuit breakers (A and B) have a 10,000 short-circuit interrupting power.

The superheating current for each motor and undervoltage of the two buses as well as the on/off position of circuit breakers are supervised by the local control panel and DAS of the main control room.

5.7.14 Structural Supports

The structural support system has two functions:

- To transmit the load of the heat exchangers, piping, and balance of plant equipment to the foundation through the proper placement of steel columns and beams that support the solar SGS components
- To allow access to the 100-MWe solar stand-alone SGS for operating and maintenance duties (a freight elevator, stairs, and floor grating at various elevations serve this purpose).

Each heat exchanger is supported by four columns; however, the superheater and reheater, as well as the evaporator and preheater, have two of their four columns in common. The size of each column is dictated by the total load it encounters. Tables 5.39 and 5.40 list each column and its respective load and size. Figures 5.56 and 5.57 show the locations of the columns.

Column spacing is dictated by the diameters of the individual heat exchangers. The spacing of the 100-MWe solar stand-alone SGS columns is greater than that of the 50-MWe hybrid SGS because of the larger diameters of the 100-MWe vessels. (See Figures 5.40 and 5.43 in the Equipment Arrangement section.)

Horizontal cross beams are situated between columns at 6.1-, 12.2-, 18.3-, 24.4-, and 28.5-m (20-, 40-, 60-, 80-, and 93.5-ft) elevations. At the topmost elevation [28.5 m (93.5 ft)], four beams surround the heat exchangers and tie them to the support structure. Thus the circular support plate upon which the heat exchanger rests is uniformly supported by the cross beams (see Figures 5.40 and 5.43). The evaporator and preheater are equipped with collars at the top of each vessel that sit on their respective support plates. The superheater and reheater are each hung from a support skirt which, in turn, sits on a support plate (see Figures 5.38 and 5.41).

Beams placed at intermediate levels of 6.1, 12.2, 18.3, and 24.4 m (20, 40, 60 and 80 ft) provide support for salt and steam piping and serve as floor grates and handrails. Floor grating creates working areas at all five levels (including the topmost elevation) that must be equipped with handrails to produce a safe working environment. A stairwell provides access to all elevations and is designed in accordance with all appropriate codes. A freight elevator is situated alongside the SGS to enable the lifting of heavy equipment to any level.

Table 5.39 Column Schedule and Loading--100-MWe Solar Stand-Alone SGS

Column Designation*	Column Size	Total Load 1000 kg (1000 lb)
A1	W8 x 20	3.2 (7)
A2	W8 x 40	51.7 (114)
A3	W12 x 60	104.3 (230)
A4	W8 x 48	60.3 (133)
A5	W8 x 20	3.2 (7)
B1	W8 x 20	5.9 (13)
B2	W8 x 48	66.2 (146)
B3	W12 x 72	133.8 (295)
B4	W12 x 53	74.8 (165)
B5	W8 x 20	5.9 (13)
C1	W8 x 24	15.0 (33)
C2	W8 x 40	46.7 (103)
C3	W12 x 53	74.4 (164)
C4	W8 x 31	38.6 (85)
C5	W8 x 20	2.7 (6)
D1	W8 x 24	11.8 (26)
D2	W8 x 31	37.6 (83)
D3	W8 x 40	49.9 (110)
D4	W8 x 28	23.6 (52)

*See Figure 5.56.

Table 5.40 Column Schedule and Loading--50-MWe Hybrid SGS

Column Designation*	Column Size	Total Load 1000 kg (1000 lb)
A1	W8 x 20	3.2 (7)
A2	W8 x 31	28.6 (63)
A3	W12 x 48	58.1 (128)
A4	W8 x 31	36.7 (81)
A5	W8 x 20	3.2 (7)
B1	W8 x 20	5.9 (13)
B2	W8 x 31	43.1 (95)
B3	W12 x 53	87.5 (193)
B4	W8 x 40	51.3 (113)
B5	W8 x 20	5.9 (13)
C1	W8 x 24	15.0 (33)
C2	W8 x 31	39.9 (88)
C3	W8 x 48	63.5 (140)
C4	W8 x 31	34.5 (76)
C5	W8 x 20	2.7 (6)
D1	W8 x 24	11.8 (26)
D2	W8 x 31	30.8 (68)
D3	W8 x 31	39.0 (86)
D4	W8 x 28	19.5 (43)

*See Figure 5.57.

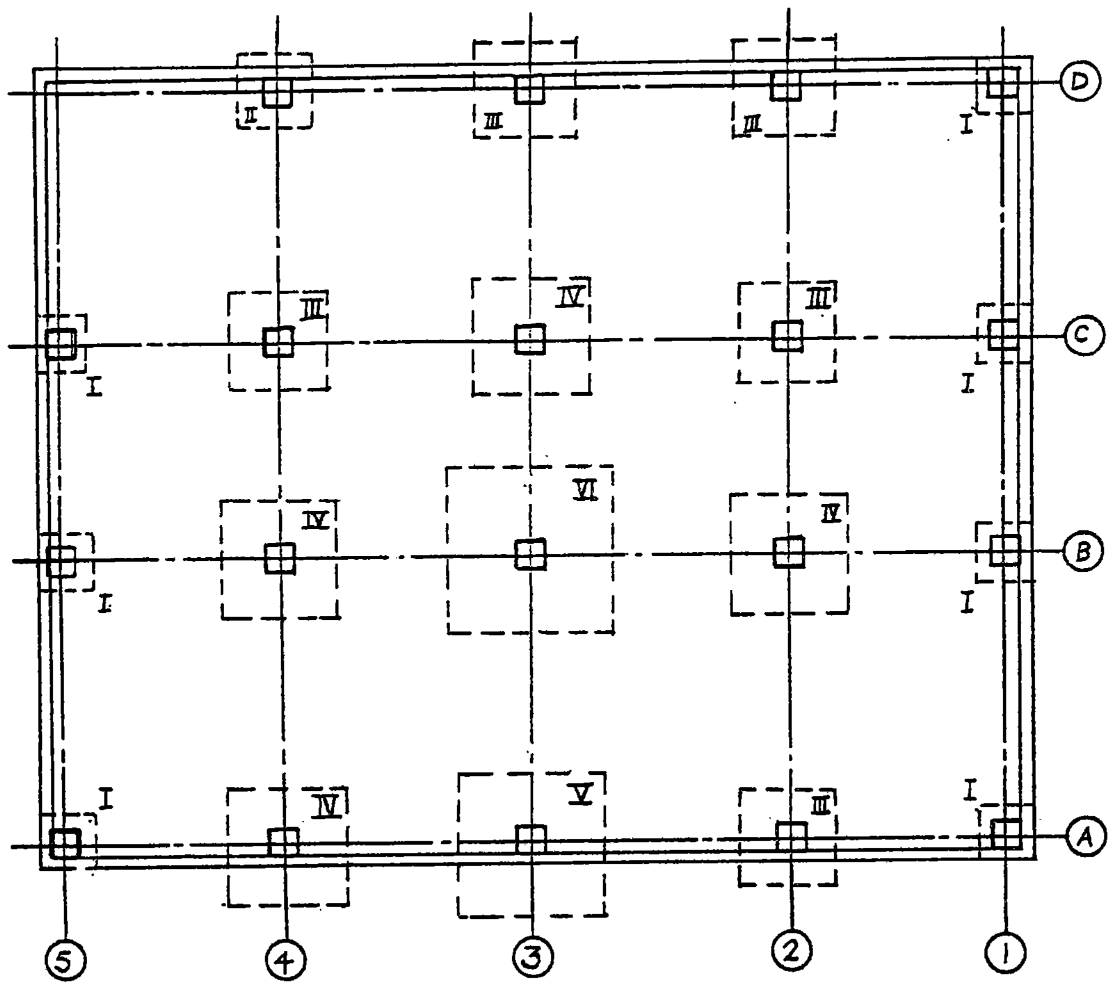


Figure 5.56 Column Schedule--100-MWe Solar Stand-Alone SGS

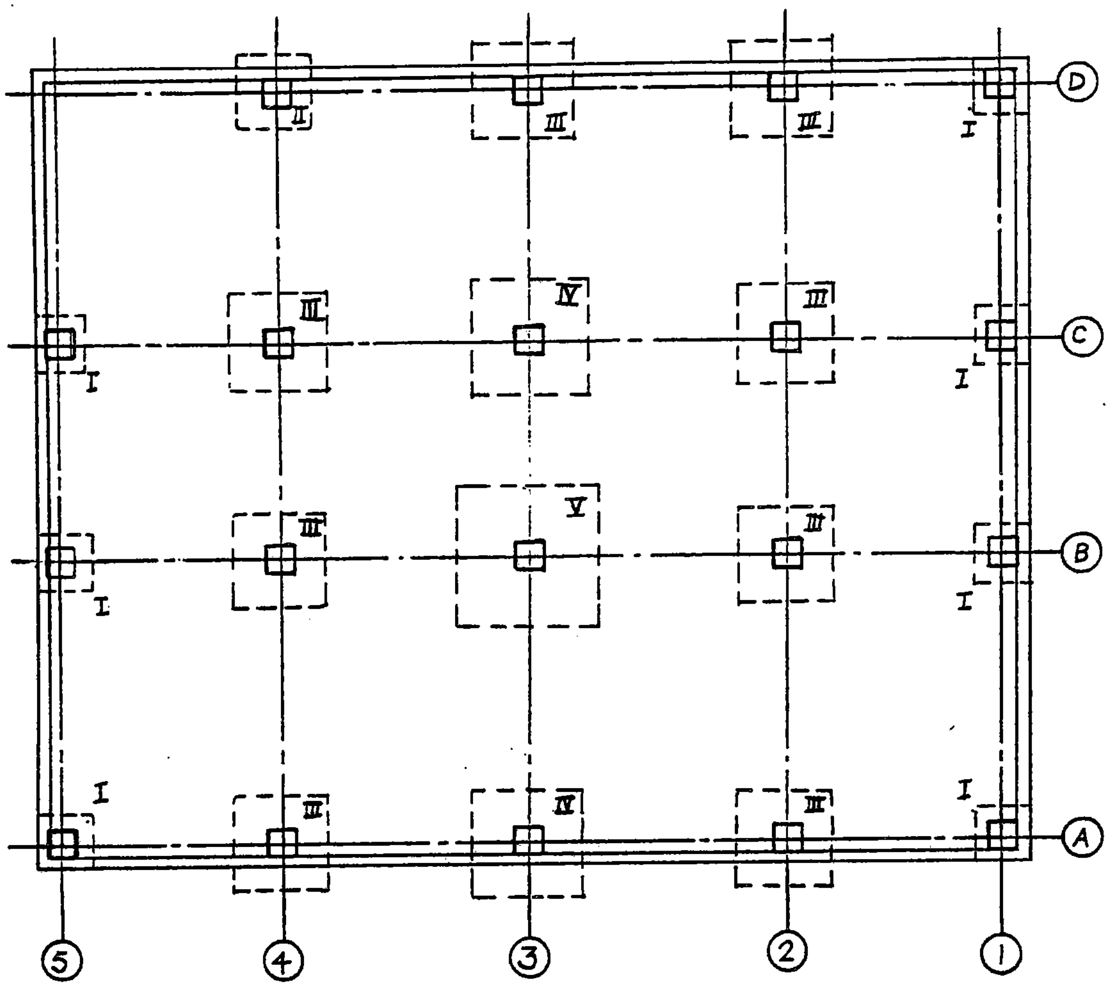


Figure 5.57 Column Schedule--50-MWe Hybrid SGS

Diagonal, horizontal, and vertical bracing (see Figures 5.40 and 5.43) supplies additional stability to the support structure. The bracing, beams, columns, and foundation arrangements (Section 5.7.15) comply with building codes applicable to a Zone 2 earthquake area.

5.7.15 Foundations and Dikes

The support foundation consists of footings for each column and a continuous slab with a dike to contain any salt spill.

The 100-MWe solar stand-alone SGS is a free-standing outdoor structure set on a 0.15-m (6-in.) concrete slab 12.8 m by 15.8 m (42 ft x 52 ft). The slab is surrounded by a 0.15-m (6-in.) curb to contain molten salt in the event of a leak. As shown in Table 5.41, 19 piers and footings are located under the slab to support the structural steel columns. The quantities of concrete and reinforcing bar which comprise the foundation components are listed in Table 5.42. The slab under the 50-MWe hybrid SGS is 11.9 m by 15.2 m (39 ft x 50 ft), 0.15 m (6 in.) thick with a 0.15-m (6-in.) curb. This slab is smaller than the slab for the 100-MWe unit because the 50-MWe heat exchangers are smaller in diameter, allowing a reduction in dimension for the horizontal steel structure. A smaller foundation requires less construction materials, as shown in Table 5.43. The foundation and pier schedule for the 50-MWe hybrid SGS is shown in Table 5.44.

Table 5.41 Footing and Pier Schedule--100-MWe Solar Stand-Alone SGS

Column Size	Pier Dimension m (in.)*	Pier Reinforcement Quantity/Size	Footing Dimension m (ft)†	Footing Reinforcement Quantity/Size	Anchor Bolts§ mm x m (in. x ft)
W8 x 20 or 8 x 24	0.30 x 0.30 (12 x 12)	4/No. 6	0.91 x 0.91 x 0.30 (3 x 3 x 1)	3/No. 6	2/25.4 x 1.07 (1 x 3.5)
W8 x 28	0.30 x 0.30 (12 x 12)	4/No. 6	1.22 x 1.22 x 0.30 (4 x 4 x 1)	6/No. 6	2/25.4 x 1.07 (1 x 3.5)
W8 x 31 or W8 x 40	0.30 x 0.30 (12 x 12)	4/No. 6	1.68 x 1.68 x 46 (5.5 x 5.5 x 1.5)	8/No. 6	4/25.4 x 1.07 (1 x 3.5)
W8 x 48 or W8 x 53	0.36 x 0.36 (14 x 14)	6/No. 8	1.98 x 1.98 x 0.46 (6.5 x 6.5 x 1.5)	10/No. 6	4/25.4 x 1.07 (1 x 3.5)
W12 x 60	0.41 x 0.41 (16 x 16)	8/No. 8	2.44 x 2.44 x 0.61 (8 x 8 x 2)	9/No. 8	4/25.4 x 1.07 (1 x 3.5)
W12 x 72	0.41 x 0.41 (16 x 16)	8/No. 8	2.82 x 2.82 x 0.61 (9.25 x 9.25 x 2)	12/No. 8	4/25.4 x 1.07 (1 x 3.5)

*Length x width; height = 1.4 m (4.5 ft) less footing height.
†Length x width x height.
§Diameter x length.

Table 5.42 Foundation Materials--100-MWe Solar Stand-Alone SGS

Item	Concrete m^3	Concrete (yd^3)	Reinforcing Bar Type	Reinforcing Bar kg	Reinforcing Bar (lb)
Slab	30.9	(40.4)	4	1326	(2918)
Curb	1.3	(1.7)	4	114	(251)
Piers	1.9	(2.5)	6	195	(430)
			8	262	(578)
Footings	24.3	(31.8)	6	773	(1704)
			8	443	(977)
Ties	---	---	3	149	(328)
Total	58.5	(76.5)		3262	(7038)

Table 5.43 Foundation Materials--50-MWe Hybrid SGS

Item	Concrete m^3	Concrete (yd^3)	Reinforcing Bar Type	Reinforcing Bar kg	Reinforcing Bar (lb)
Slab	27.6	(36.1)	4	1182	(2605)
Curb	1.2	(1.6)	4	108	(238)
Piers	1.8	(2.4)	6	240	(529)
			8	174	(384)
Footings	19.7	(25.8)	6	775	(1709)
			8	132	(291)
Ties	---	---	3	149	(328)
Total	50.3	(65.9)		2760	(6084)

Table 5.44 Footing and Pier Schedule--50-MWe Hybrid SGS

Column Size	Pier Dimension m (in.)*	Pier Reinforcement Quantity/Size	Footing Dimension m (ft)†	Footing Reinforcement Quantity/Size	Anchor Bolts§ mm x m (in. x ft)
W8 x 20 or 8 x 24	0.30 x 0.30 (12 x 12)	4/No. 8	0.91 x 0.91 x 0.30 (3 x 3 x 1)	3/No. 6	2/25.4 x 1.07 (1 x 3.5)
W8 x 28	0.30 x 0.30 (12 x 12)	4/No. 6	1.22 x 1.22 x 0.30 (4 x 4 x 1)	6/No. 6	2/25.4 x 1.07 (1 x 3.5)
W8 x 31 or W8 x 40	0.30 x 0.30 (12 x 12)	4/No. 6	1.68 x 1.68 x 46 (5.5 x 5.5 x 1.5)	8/No. 6	4/25.4 x 1.07 (1 x 3.5)
W8 x 48 or W8 x 53	0.36 x 0.36 (14 x 14)	6/No. 8	1.98 x 1.98 x 0.46 (6.5 x 6.5 x 1.5)	10/No. 6	4/25.4 x 1.07 (1 x 3.5)
W12 x 60	0.41 x 0.41 (16 x 16)	8/No. 8	2.44 x 2.44 x 0.61 (8 x 8 x 2)	9/No. 8	4/25.4 x 1.07 (1 x 3.5)

*Length x width; height = 1.4 m (4.5 ft) less footing height.
†Length x width x height.
§Diameter x length.

5.8 FABRICATION/ERECTION REQUIREMENTS

5.8.1 Fabrication Requirements

The straight-tube heat exchanger design selected for molten salt steam generation is such that standard state-of-the art fabrication techniques can be used. These include the following:

- Drilling the tubesheet holes on multispindle Lahr drills
- Subarc welding of all shell courses
- Assembling the tube-support cage using tierods and spacers
- Inserting the tubes into the tube bundles from one side through the tubesheet
- Welding the tubes to the tubesheets using the fillet-type technique
- Kinetically expanding the tubes into the tubesheets.

These processes have been used by Foster Wheeler in the manufacture of feedwater heaters, nuclear steam generators, process heat exchangers, and intermediate heat exchangers for breeder reactor plants.

A step-by-step fabrication sequence, including all significant fabrication and inspection operations, is presented in the Fabrication/Erection Plans (Appendix D) for both the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS. The plans take into consideration the tools, jigs, fixtures, equipment, floor space, and special supplies required. Welding procedures, welding qualification requirements, and nondestructive examination and inspection requirements are identified. Also included are shop fabrication schedules showing fabrication of subassemblies (such as shells, tubesheets, and tube bundles) and final assembly of the heat exchanger components and schedules for tool design and fabrication, welding development, and mockups.

5.8.2 Erection Requirements

The field fabrication and erection requirements are described in Appendix D for both the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS. A schedule is included that identifies the interrelationships between and times required for preparing the site; pouring foundations; assembling structural supporting steel; installing the heat exchangers, piping, heat tracing, pumps, drum tanks, instrumentation and controls; and applying insulation.

5.9 MAINTENANCE REQUIREMENTS

The safe, reliable operation of equipment is of prime importance. To achieve this objective, a comprehensive maintenance program must be established and implemented. In general, two types of maintenance programs are required: daily unscheduled maintenance and scheduled, preventive maintenance.

5.9.1 Unscheduled Maintenance

Minor equipment failures must be eliminated so that operation is uninterrupted. This type of maintenance is usually limited to minor repair work, adjustments, and the elimination of leaks at various joints.

5.9.2 Preventive Maintenance

This program must be designed to include a policy for operating the equipment properly and within its range of capability. It should include proper instructions to maintain equipment in a clean and prime operating condition--verified by instrumentation and in-service observations. Preventive maintenance also includes regularly scheduled outages to permit those inspections that cannot be made during operation and to perform necessary repairs.

- Pumps
  - Bearing Lubrication. Operation of the unit without proper lubrication can result in bearing overheating and failure, pump seizure, and actual break-up of the equipment, exposing operating personnel to injury. To prevent this, the level of lubricating oil must be maintained within acceptable limits at all times. The oil in the bearing housing must be inspected periodically for deterioration, moisture, and sludge. The manufacturer's recommended practice should be followed for oil changes. The oil rings should be inspected periodically to ensure that they are riding free on the pump shaft or journal sleeve and are not hung up.
  - Shaft Packing. If used, the packing glands must be checked to prevent excessive process fluid leaks. A small amount of leaking is required to lubricate the packing. Its complete elimination from the packing will cause burned packing, a scored shaft sleeve, and possible rotor seizure.

- Alignment Check. The pump should be checked for excessive vibration, possibly caused by misalignment between the pump and the driver. If excessive vibration is found, the coupling alignment must be checked and corrected. If the cause of vibration is not traceable to coupling misalignment, the pump must be checked for damaged or worn bearings, cavitation, vapor lock from overheating, etc.

- Scheduled Outage. At the manufacturer's recommended intervals, the pump should be taken out of service and dismantled for a complete inspection and repair.

  At this time the bearings should be checked for wear, pitting, or other problems. The shaft should be checked for straightness and the keyways checked for wear. Follow manufacturer's recommended procedure if repair is required.

  The casing should be inspected for possible corrosion or signs of erosion. If a problem is found, the casing should be repaired by a competent pump shop.

  The impeller should also be inspected for corrosion or signs of erosion or rubbing. If a problem is found, repairs should be carried out by a competent pump shop.

- Heat Exchangers

  - Inspection of Unit. At regular intervals, as frequently as experience indicates, an examination should be made of the interior and exterior condition of the unit. Since the unit must be taken out of service for this inspection, the schedule should coincide with outage required for other equipment inspection.

  - Fouling.

    - Exchangers subject to fouling or scaling should be cleaned periodically.
    - A marked increase in pressure drop or reduction in performance usually indicates that cleaning is necessary.
    - The unit first should be checked for an air or vapor lock to confirm that this is not the cause for the reduction in performance.
    - Since cleaning becomes more difficult as the amount of deposit or scale increases, the intervals between cleanings should not be excessive.
    - Cleaning of heat exchanger surfaces may be done either by mechanical or chemical methods. Since the exchangers used in this installation are contacted on both sides by fluid only, we recommend the chemical method.

- External (Shell-Side) Cleaning. Salt deposits may be washed out by circulating hot fresh water at high velocity. If scaling is present, this treatment should be followed by cleaning with a compound formulated for scale removal (see internal cleaning).

- Internal (Tube-Side) Cleaning. In general, chemical cleaning is done in four steps:

  - The internal surfaces are washed with an acid solvent, containing a suitable inhibitor, to dissolve the deposits.
  - Clean water flushes out loose or loosened deposits, solvent adhering to the surfaces, soluble iron salts, and sludge. Any corrosive or otherwise dangerous gases that may have been formed during the solvent-wash phase are displaced.
  - The unit is neutralized (passivated). This treatment produces a passive surface, thus preventing an "after-rust" formation on the freshly cleaned surface.
  - A final flush of clean water removes any remaining loose deposits. There are two generally accepted methods utilized in chemical cleaning: the continuous circulation method and the soaking method. Since both methods are acceptable and result in good cleaning, selection should be based on local conditions.

  We recommend retaining a qualified organization that specializes in cleaning services. These organizations check the nature of deposits to be removed, furnish the proper chemicals and equipment, and provide experienced personnel to do a complete cleaning job.

- Gas-Fired Water Preheater. Maintenance requirements previously discussed for pumps and heat exchangers also apply for the gas-fired water preheater. Areas peculiar to this unit are:

  - Optimum burner operations
  - Prompt detection of flame failure
  - Detection of unburned combustibles in the flue gas. Whether monitored by instruments or visually observed, deviation in these areas must be corrected immediately to ensure safe, economical operation. Consult the manufacturer's manual for proper burner adjustment instructions.

- Valves. Three functional categories of valves are utilized in the installation: Safety, Control (manual or automatic), and Shut-Off.

  - Safety Valves. The safe operation of the plant depends on the proper operation of the relief valves. The valves should be kept clean and should be periodically tested and reconditioned to ensure proper functioning.

    Valve testing should be done on a calibrated testing bench by qualified technicians. If the need for reconditioning is indicated, the recommended manufacturer's procedures should be followed.

- Control Valves. Control valve components are subject to normal wear; they should be periodically inspected and replaced if necessary. The frequency of inspection and maintenance is solely dependent on the severity of service conditions.

  In general, all maintenance operations may be performed with the valve body in the line.

  To avoid injury to personnel or damage to equipment, the following points must be observed before any maintenance work is begun:

  -- Shut off the air pressure to the activator
  -- Isolate the valve from the process
  -- Release process pressure
  -- Vent the activator loading pressure.

  The following are the types of maintenance operations generally performed on control valves:

  -- Packing lubrication
  -- Packing replacement
  -- Trim replacement
  -- Seating surface lapping
  -- Bellows seal replacement
  -- Gasket replacement.

  For recommended parts and explicit instructions for maintenance, the manufacturer's manual should be followed.

- Shut-Off Valves. Shut-off valves, whether manual or remotely actuated, are subject to normal wear and should be periodically inspected. The frequency of inspection and maintenance depends solely on the severity of service conditions.

  Dependent on the type and make of the valve, most of the maintenance operations may be performed with the valve body in the line. If maintenance work is performed with the valve body in the line, the valve must be isolated from the process in a manner that ensures that no accidental pressurizing of the valve is possible. Once this is done and the process pressure is relieved from the valve, maintenance work may proceed.

  The following types of maintenance operations are generally performed on the valves:

  -- Packing lubrication
  -- Packing replacement
  -- Trim replacement
  -- Gasket replacement.

  For the proper maintenance procedures and replacement parts, the manufacturer's manual should be consulted.

SECTION 6

SGS COST AND SCHEDULE

Section 6

SGS COST AND SCHEDULE

6.1 COST

The installed capital costs based on first quarter 1982 dollars are summarized in Table 6.1 for the 100-MWe solar stand-alone SGS and the 50-MWe hybrid SGS. Interest during design, fabrication, and construction; State and local taxes; escalation; and owner's costs are not included. A more detailed breakdown of costs for heat exchangers, auxiliary systems and equipment, structure, and instrumentation and controls is included in Tables 6.2 and 6.3 for the 100-MWe solar stand-alone SGS and 50-MWe hybrid SGS respectively.

6.2 SCHEDULE

The time period from start of design to the end of preoperational testing is 32 months for the 100-MWe solar stand-alone SGS and 29 months for the 50-MWe hybrid SGS. The schedules for both units are included in Figure 6.1. More detailed schedules for shop fabrication and field fabrication and erection are included in Appendix D.

Table 6.1 Cost Summary

	100-MWe Solar Stand-Alone SGS ($)	50-MWe Hybrid SGS ($)
1. Heat Exchangers	3,614,100	2,480,900
2. Auxiliary Systems and Equipment	2,994,000	1,942,700
3. Structure	522,000	475,000
4. Instrumentation and Controls:	2,400,000	2,166,000
Subtotal	9,530,100	7,064,600
Contingency at 20% of Subtotal	1,906,000	1,412,900
5. Home Office Costs	1,715,400	1,271,600
6. Construction Management:	914,900	678,200
Total Cost	14,066,400	10,427,300
Fee at 8% of Total Cost	1,125,300	834,200
Sell Price	15,191,700	11,261,500

Table 6.2 Cost Breakdown--100-MWe Solar Stand-Alone SGS

		Labor	Material	Total ($)
1.	Heat Exchangers			3,614,100
1.1	Shop Material and Fabrication			2,214,200
1.1.1	Preheater	234,600	283,000	517,600
1.1.2	Evaporator/Drum	507,500	477,300	984,800
1.1.3	Superheater	158,200	237,500	395,700
1.1.4	Reheater	125,600	190,500	316,100
1.2	Other Shop Costs			388,500
1.2.1	Tooling			82,600
1.2.2	Manufacturing Development and Mock-Ups			3,800
1.2.3	Contract Reserve			204,000
1.2.4	Freight			97,000
1.2.5	Miscellaneous			1,100
1.3	Engineering, Management, and Miscellaneous Costs			578,900
1.3.1	Project Management			72,100
1.3.2	Contract Administration			15,000
1.3.3	Functional Engineering			99,000
1.3.4	Mechanical Design and Detailing			283,000
1.3.5	Weld Engineering			3,500
1.3.6	Manufacturing Engineering and Quality Assurance			52,900
1.3.7	Estimating			16,400
1.3.8	Reproduction			13,000
1.3.9	Travel and Living Expenses			14,000
1.3.10	Computer			10,000
	Subtotal			3,181,600
	Fee at 8% of Subtotal			254,500
1.4	Installation Cost			178,000

Table 6.2 Cost Breakdown--100-MWe Solar Stand-Alone SGS (Cont)

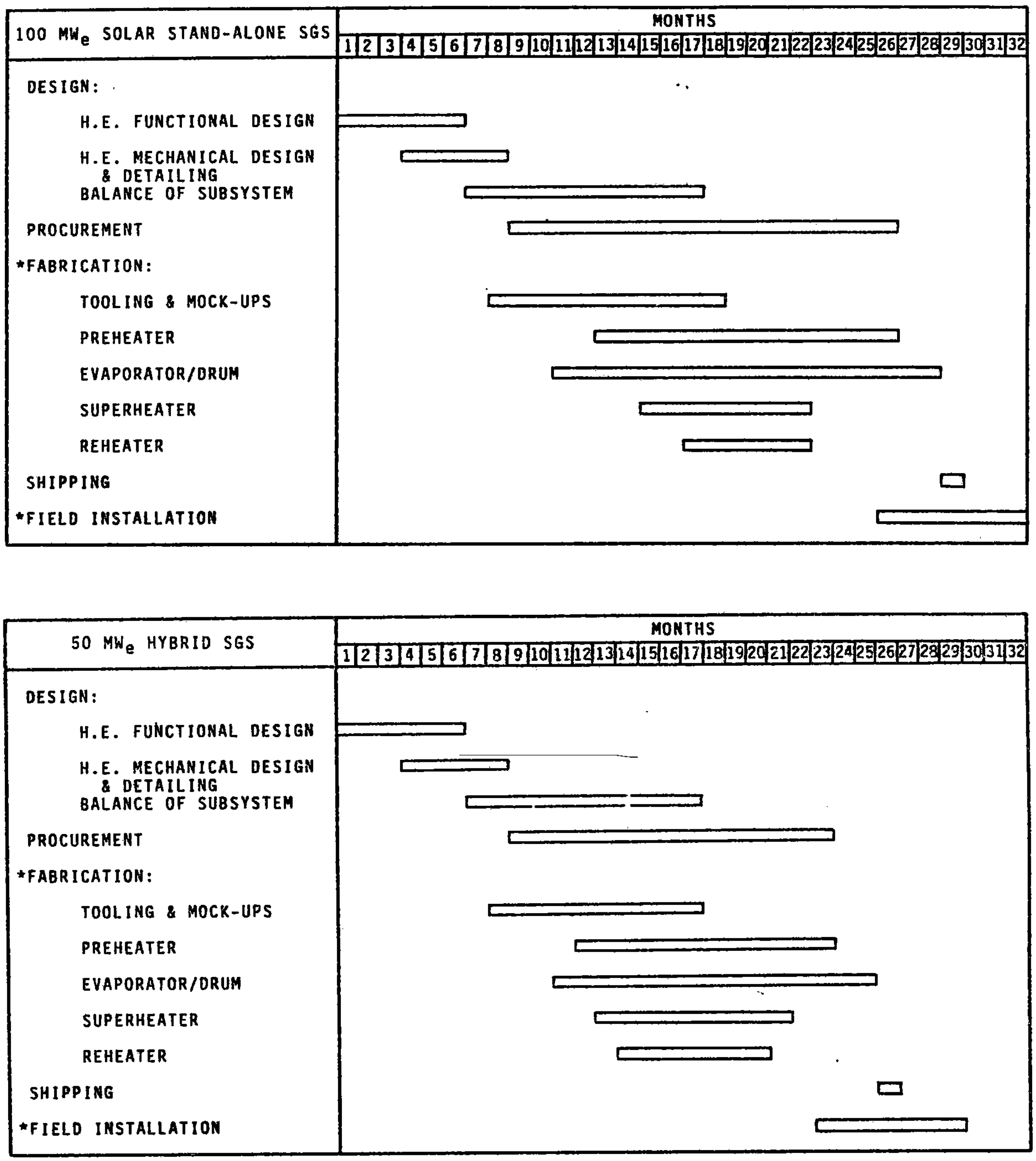
		Labor	Material	Sub-Contract	Total ($)
2.	Auxiliary Systems and Equipment				2,994,000
2.1	Drum-Water Recirculation Pumps	44,000	194,000	---	238,000
2.2	Cold-Salt Recirculation Pumps	8,000	131,000	---	139,000
2.3	Electric Trace Heating	73,000	116,000	---	189,000
2.4	Salt Drain System	30,400	45,400	---	75,800
2.5	Salt Pressure-Relief System	113,000	158,000	---	271,000
2.6	Safety Valves	19,400	38,000	---	57,400
2.7	Auxiliary Electric Equipment	110,000	78,000	---	188,000
2.8	Piping	844,600	602,200	---	1,446,800
2.9	Insulation and Lagging	---	---	188,000	188,000
2.10	Auxiliary Feedwater Heater	78,000	123,000	---	201,000
3.	Structure				522,000
3.1	Support Structure	219,000	203,000	---	422,000
3.2	Elevator	---	---	54,000	54,000
3.3	Foundation and Dike	34,000	12,000	---	46,000
4.	Instrumentation and Controls				2,400,000
4.1	Control Valve Stations	---	---	872,000	872,000
4.2	Block Valves	---	---	195,000	195,000
4.3	Transducers	---	---	117,000	117,000
4.4	Distributed Control System	---	---	1,055,000	1,055,000
4.5	Miscellaneous Wire, Conduit, and Connections	20,000	141,000	---	161,000

Table 6.3 Cost Breakdown--50-MWe Hybrid SGS

		Labor	Material	Total ($)
1.	Heat Exchangers			2,480,900
1.1	Shop Material and Fabrication			1,361,400
1.1.1	Preheater	132,400	90,200	222,600
1.1.2	Evaporator/Drum	418,400	260,000	678,400
1.1.3	Superheater	125,400	110,100	235,500
1.1.4	Reheater	118,200	106,700	224,900
1.2	Other Shop Costs			261,000
1.2.1	Tooling			66,900
1.2.2	Manufacturing Development and Mock-Ups			3,800
1.2.3	Contract Reserve			139,400
1.2.4	Freight			49,800
1.2.5	Miscellaneous			1,100
1.3	Engineering, Management, and Miscellaneous Costs			569,800
1.3.1	Project Management			65,300
1.3.2	Contract Administration			13,600
1.3.3	Functional Engineering			99,000
1.3.4	Mechanical Design and Detailing			283,000
1.3.5	Weld Engineering			2,600
1.3.6	Manufacturing Engineering and Quality Assurance			52,900
1.3.7	Estimating			16,400
1.3.8	Reproduction			13,000
1.3.9	Travel and Living Expenses			14,000
1.3.10	Computer			10,000
	Subtotal			2,192,200
	Fee at 8% of Subtotal			175,400
1.4	Installation Cost			113,300

Table 6.3 Cost Breakdown--50-MWe Hybrid SGS (Cont)

		Labor	Material	Sub-Contract	Total ($)
2.	Auxiliary Systems and Equipment				1,942,700
2.1	Drum-Water Recirculation Pumps	44,000	194,000	---	238,000
2.2	Cold-Salt Recirculation Pumps	3,500	96,000	---	99,500
2.3	Electric Trace Heating	67,000	104,000	---	171,000
2.4	Salt Drain System	24,100	37,300	---	61,400
2.5	Salt Pressure-Relief System	88,000	122,000	---	210,000
2.6	Safety Valves	13,700	23,500	---	37,200
2.7	Auxiliary Electric Equipment	86,000	52,000	---	138,000
2.8	Piping	565,400	289,200	---	854,600
2.9	Insulation and Lagging	---	---	133,000	133,000
2.10	Auxiliary Feedwater Heater	---	---	---	---
3.	Structure				475,000
3.1	Support Structure	197,000	183,000	---	380,000
3.2	Elevator	---	---	54,000	54,000
3.3	Foundation and Dike	30,000	11,000	---	41,000
4.	Instrumentation and Controls				2,166,000
4.1	Control Valve Stations	---	---	702,000	702,000
4.2	Block Valves	---	---	132,000	132,000
4.3	Transducers	---	---	117,000	117,000
4.4	Distributed Control System	---	---	1,055,000	1,055,000
4.5	Miscellaneous Wire, Conduit, and Connections	20,000	140,000	---	160,000



*See Appendix D for details.

Figure 6.1 SGS Design/Construction Schedules

SECTION 7

SUBSYSTEM RESEARCH EXPERIMENT

(SRE)

Section 7

SUBSYSTEM RESEARCH EXPERIMENT (SRE)

As part of Phase 1, a proposal was prepared for the design, construction, testing, and evaluation of an SRE of sufficient scale to ensure successful operation of the full-size subsystem designed in Phase 1. Abridged versions of the Phase 2 Proposal (Technical and Cost Volumes--Appendices E and F) describe the SGS SRE in detail.

SECTION 8

REFERENCES

SECTION 8

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