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CONCEPTUAL DESIGN OF ADVANCED CENTRAL RECEIVER POWER SYSTEMS

Volume 1, Part 2. Final Technical Report

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Volume I, Part 2

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Final Technical Report

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GENERAL ELECTRIC

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Section 2

INTRODUCTION

2.1 OBJECTIVE

The economy of the United States relies on the availability of electric energy at a reasonable cost. The growing scarcity and increasing cost of fuels for power generation, along with the fact that electrical generation accounts for more than 25 percent of domestic fuel consumption, has led to the realization that new sources for electric energy must be developed in the future.

The sun is an inexhaustable source of additional energy for electric power generation, and the central receiver concept for converting solar thermal energy into electric energy is a promising technology for electrical utility applications. First generation central receiver power plants have been designed, and a pilot plant project has been initiated to demonstrate the concept.

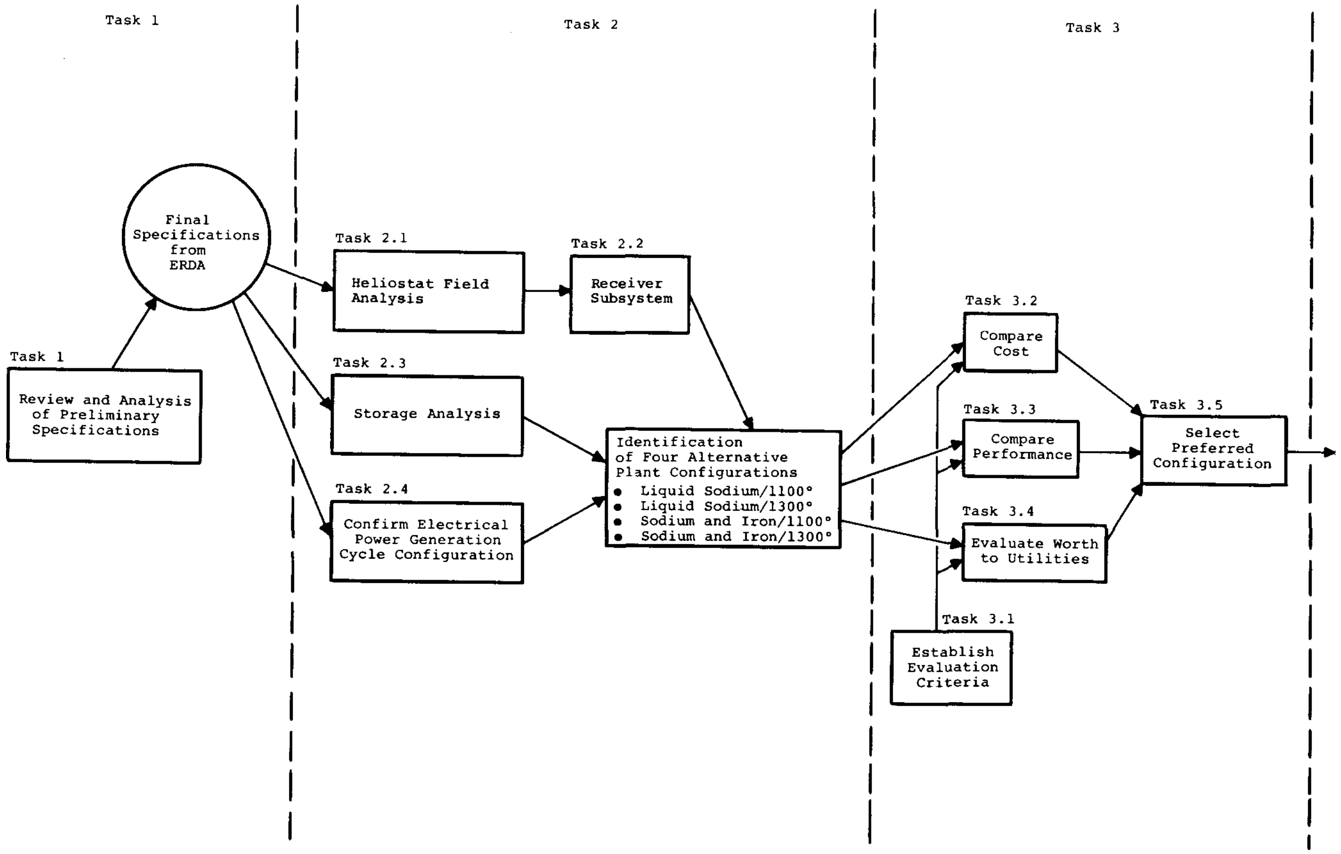
The objective of the Conceptual Design of Advanced Central Receiver Power Systems study is to investigate new technology which will improve the cost effectiveness of solar thermal electric energy. This report presents the results of a DoE-funded General Electric study which evaluated an approach with high potential for meeting the objective of lower cost. This approach utilizes the very favorable high temperature thermal properties of liquid sodium to achieve improved plant performance. A plant conceptual design and analyses to quantify potential performance and cost improvements were conducted as a part of this evaluation. A development plan defining the steps needed to achieve commercialization of this technology was also established.

The results of the study include a conceptual design and cost estimate for a 100 MW_e commercial power plant, the concept and cost estimate for a pilot plant scaled to model the commercial plant, a description of subsystem research experiments required to support the design, and a development plan for a program that will lead to the detailed design of the commercial plant.

2.2 TECHNICAL APPROACH

The work flow diagram for the General Electric Conceptual Design of Advanced Central Receiver Power Systems study is shown in Figure 2-1. Following a review of the preliminary system specifications and ground rules, a parametric analysis designed to identify the preferred commercial plant components and configuration was initiated. The results of the parametric analysis were evaluated, and a commercial plant configuration was selected. Conceptual design of the selected 100 MW_e commercial plant concept was then performed along with an estimate of the plant performance. The capital cost of the commercial plant was estimated, and opportunities for improvements to the plant design were identified. A safety analysis and review of implementation factors for electric utility service were conducted to identify potential impediments to the commercialization of the technology.

Having completed the commercial plant conceptual design, a development plan was established providing a road map for a program which would lead to the commercial plant detailed design from which construction could begin by the late 1980s.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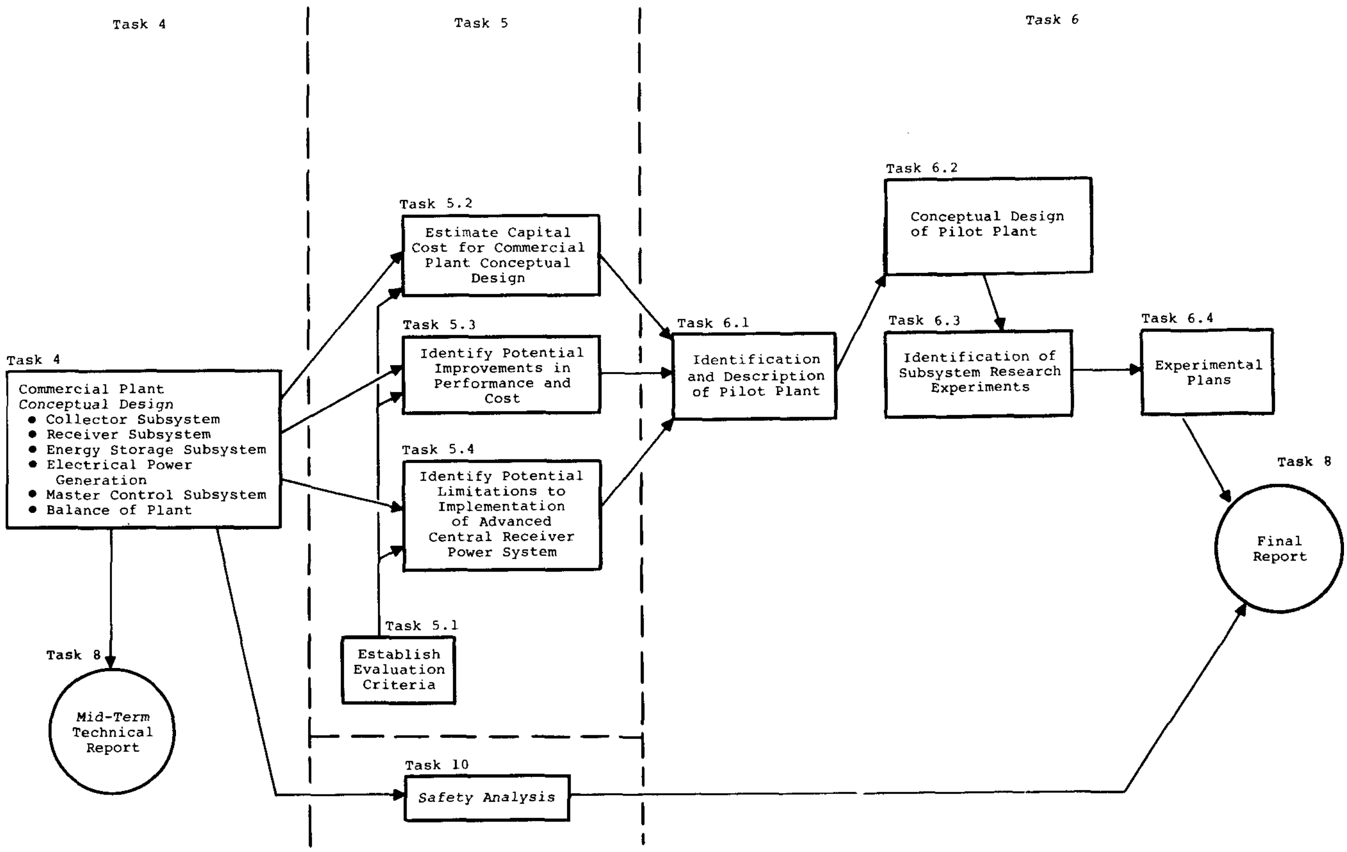


Figure 2-1. Work Flow Diagram

2.2.1 PLANT CONCEPT

The power plant design concept is based upon a steam cycle which receives thermal energy from a liquid-sodium-cooled receiver. Liquid sodium was selected as the receiver coolant and heat transfer medium based upon its very favorable high temperature heat transfer characteristics. The high overall heat transfer coefficients associated with the use of liquid sodium as the coolant, compared to those that result when water/steam or a gas is used as the receiver coolant, permit higher solar fluxes at the receiver surface. Consequently the size of the receiver may be reduced, decreasing receiver thermal losses and capital cost.

The use of liquid sodium as the heat transfer medium also permitted the design of a more efficient reheat steam cycle because the steam generators are located at the base of the tower. A reheat cycle is not practical in a system employing a water/steam-cooled receiver because steam would have to be piped from the high pressure turbine exhaust to the receiver to be reheated.

The storage subsystem, which is designed to contain sufficient thermal energy to permit operation of the steam turbine at full rated output for a period of three hours, employs liquid sodium as the storage medium. With this storage concept, steam can be generated at full throttle inlet conditions during the entire storage discharge cycle. Consequently, the steam turbine need not be derated during operation from storage as is the case with systems employing a water/steam-cooled receiver, but can produce full power during this mode of operation.

The reference advanced central receiver power system which formed the basis of the initial plant design concept is shown schematically in Figure 2-2. The collector subsystem provides a high flux of solar energy averaging up to two megawatts/square meter (north field) to the receiver absorber panels. The receiver is constructed of a number of individual absorber panels to permit factory fabrication of the individual panels and to expedite installation and any required maintenance operations. The flow of sodium through the absorber panels is controlled by a separate electromagnetic (EM) pump associated with each panel to ensure accurate panel cooling flow and control of the sodium outlet temperature.

An intermediate heat exchanger located at the base of the tower was proposed to isolate the high hydrostatic pressure of the primary sodium loop from the balance of the liquid sodium circuit. This would permit components of the secondary sodium loop to be designed for low pressure. The secondary loop contains the steam generator and thermal energy storage subsystem. During operation of the plant directly from the receiver, hot sodium passes into the superheater and reheater and/or the thermal energy storage subsystem as appropriate. Sodium exiting from the superheater and reheater then passes into the evaporator section of the steam generator and leaves the evaporator at a temperature of approximately 630 F, from which it returns to the intermediate heat exchanger. On the steam side of the system, steam from the superheater expands through the high pressure stages of the steam turbine and is then reheated prior to being expanded down to exhaust conditions.

2.2.2 PARAMETRIC ANALYSIS

The parametric analysis of the reference plant concept included a large number of subsystem or component options which are summarized in Table 2-1. Two

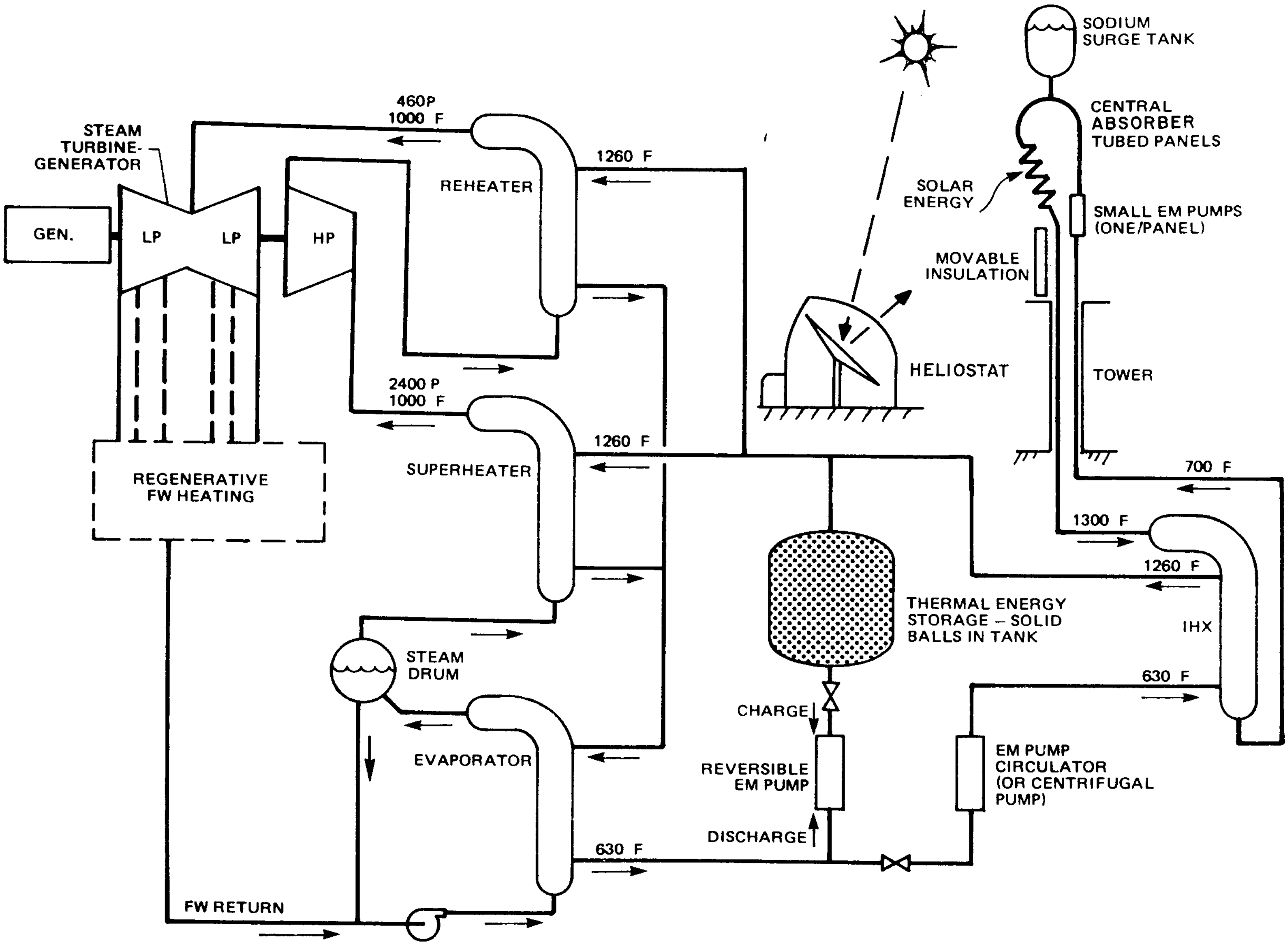


Figure 2-2. Reference Advanced Central Receiver Power System

Table 2-1

SUMMARY OF PARAMETRIC CASES

Subsystem or Component	Options
Heliostat	GE-Enclosed Heliostat Glass Heliostat
Heliostat Field	North Field 360° Field
Receiver	1100 F Peak Sodium Temperature 1300 F Peak Sodium Temperature Geometry
Sodium Loop	Two Loops Flow Throttling
Thermal Storage	Hot Sodium Hot Sodium Plus Iron Hot Molten Salt 1100 F Peak Temperature 1300 F Peak Temperature Factory-Assembled Tanks Field-Assembled Tanks Vessel Configuration
Steam Cycle	Throttle Pressure and Temperature Feedwater Temperature

advanced heliostat designs--an enclosed heliostat, and a glass heliostat--were considered. These heliostats were optimally arranged in both 360° and north field configurations. Cylindrical and flat receiver geometries corresponding to the respective collector subsystem geometries were investigated, and a trade-off study comparing an external versus a cavity receiver configuration was conducted. In addition, peak sodium temperature levels of 1100 F and 1300 F were considered.

A sodium loop configuration (consisting of two loops separated by an intermediate heat exchanger) and a single loop configuration (using flow throttling to reduce the pressure in the sodium circuit at the base of the tower) were investigated.

A number of thermal storage subsystem configurations were also investigated as part of the parametric analysis. These configurations included thermal energy storage using hot sodium, sodium plus iron (to improve the volumetric thermal storage capacity), and hot molten salt. Two storage temperatures corresponding to the 1100 F and 1300 F peak receiver temperatures were considered. The thermal storage vessel fabrication technique was also considered to be an important factor in controlling the cost of this system, and both factory and field assembly of these vessels were investigated. In addition, a large number of storage vessel configurations and control strategies were considered.

The parameters studied within the electric power generation subsystem included steam turbine throttle pressure and temperature as well as feedwater heating alternatives.

The parametric analysis was conducted on a subsystem level. The analysis was begun by considering the collector and receiver subsystem together. Inputs to the analysis included component specifications such as heliostat cost; optical and mechanical characteristics; a cost model for the receiver, tower, and riser/downcomer; a thermal loss model for the receiver (to specify receiver losses as a function of size and temperature); and a description of the auxiliary loads imposed upon the collector and receiver subsystems for functions such as sun-tracking and pumping of liquid sodium. This information was used to design four optimized heliostat fields:

- Enclosed heliostat, 360° field
- Enclosed heliostat, north field
- Glass unenclosed heliostat, 360° field
- Glass unenclosed heliostat, north field

A figure of merit, defined as the cost per unit of thermal energy delivered at the tower base, was computed for each of the four collector/receiver subsystem configurations. This figure of merit was the primary discriminator used in the selection of the preferred collector/receiver subsystem configuration.

In the parametric analysis of the storage subsystem, sodium temperature, storage medium, storage vessel fabrication technique, and sodium circuit configuration were the design factors considered. Storage temperature was a parameter of interest because storage at the highest possible temperature would result in minimum storage volume but would require more expensive construction materials.

Three storage media were considered in the analysis: sodium, sodium in a vessel containing iron spheres, and molten salt which has a lower unit cost than sodium but poorer heat transfer characteristics and higher viscosity.

A study of storage vessel fabrication costs was made to determine whether field-assembled storage vessels would be more cost effective than smaller factory assembled vessels whose maximum size would be determined by shipping constraints.

In the parametric analysis of the electric power generation subsystem, steam throttle conditions and the number of feedwater heaters were varied to determine the effect of steam cycle efficiency on the capital cost of the plant. The items of plant capital cost which would vary as a function of steam cycle efficiency were estimated for a number of steam conditions and feedwater heating strategies. Beginning with a reference inlet steam condition of 1800 psi/1000 F with a 1000 F reheat, the incremental capital cost of the plant at other steam conditions was estimated. It should be noted that all steam conditions considered in this parametric analysis were consistent with the requirement for daily startup and shutdown of the steam turbine.

2.2.3 SELECTION OF COMMERCIAL PLANT CONFIGURATION

A set of criteria for evaluating the results of the parametric analysis and selecting the configuration for the commercial plant design were established and are shown in Table 2-2. These criteria include those factors considered to be of major importance, namely acceptability and introduction of the technology into the electrical utility industry within the time period established as a study ground rule.

The selection process for the commercial plant configuration was conducted at three distinct levels of review. The Technical Team met to integrate the results of the subsystem and component parametric analyses and to select a commercial plant configuration by application of the evaluation and selection criteria to the parametric results. These conclusions were presented to a Technical Review Panel consisting of equipment suppliers and users in the electrical utility industry. The Technical Review Panel reviewed the selection process and offered commentary that was factored into the conceptual design. The results were then presented to General Electric management for review and concurrence.

2.2.4 COMMERCIAL PLANT CONCEPTUAL DESIGN

Following the selection of the preferred plant configuration, a commercial plant conceptual design was conducted. The conceptual design provides descriptions for the major components and balance-of-plant as well as plant and component drawings to a level of detail required for estimating the performance, cost, major operational characteristics and development requirements of the liquid-sodium-cooled central receiver power plant.

The organization of the conceptual design effort is shown in Figure 2-3. A detailed steam turbine heat balance was first established to provide the information needed to design the steam generators and set the thermal power required at the tower base. Having thus established this required solar power input, the collector subsystem was designed. A field optimization study established the tower height, receiver dimensions, and receiver flux plot. These factors formed the basis for designing the receiver subsystem.

Table 2-2

ADVANCED CENTRAL RECEIVER SELECTION CRITERIA

A.	Capital Cost
B.	O&M Requirements
C.	Control Characteristics
D.	Forced Outage Rate
E.	Startup Power Requirements
F.	Potential for Improvements in Cost and Performance
G.	Environmental Intrusion
H.	Land Requirements
I.	Hardware Materials Availability
J.	R&D Required
K.	Industrial Capability of Manufacture
L.	Plant Safety

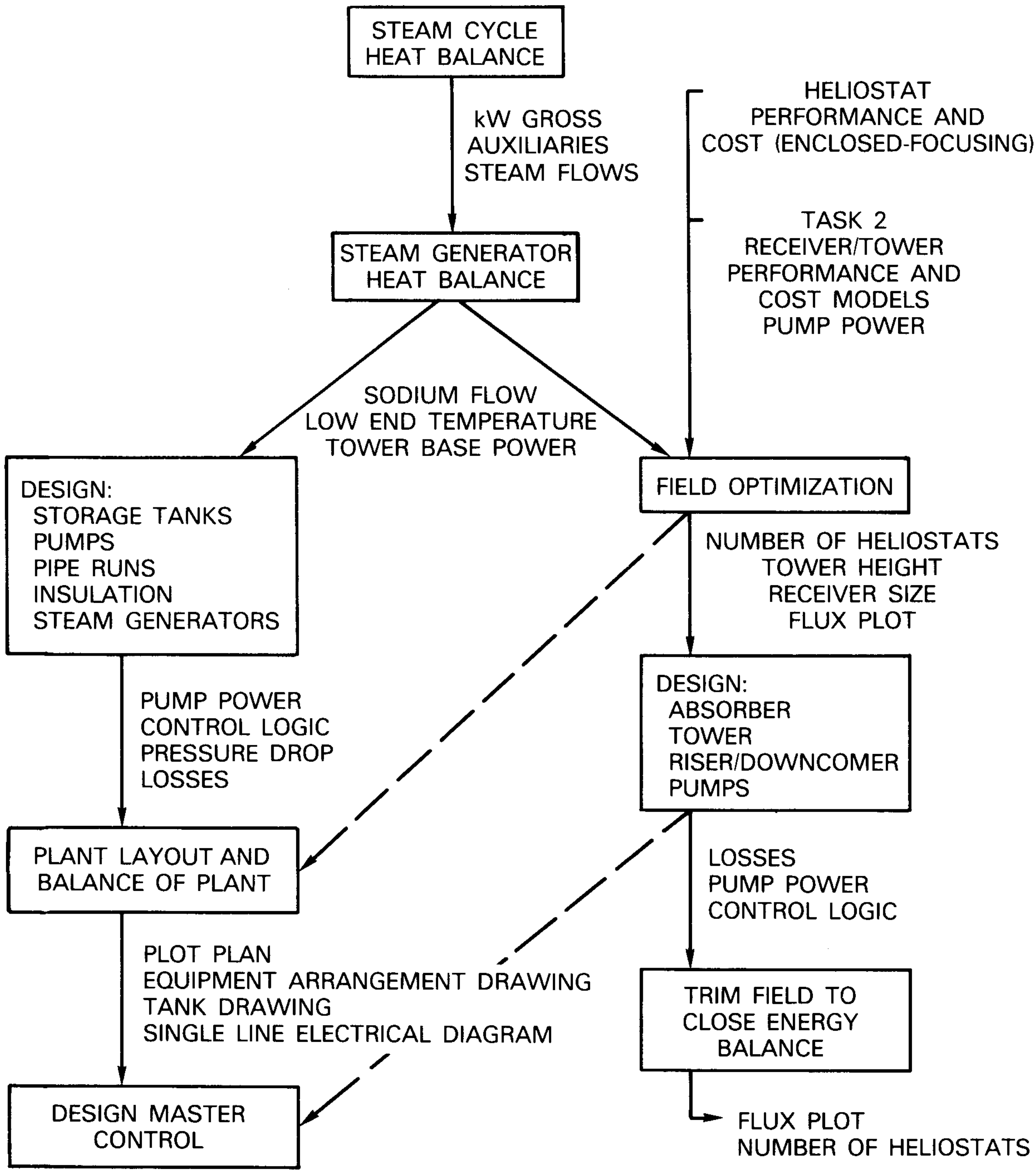


Figure 2-3. Commercial Plant Conceptual Design - Work Flow

The thermal storage subsystem design was carried out in parallel with the field optimization study. The steam generator analysis and specified period of operation from storage provided the data required to establish the volume of tankage and pump sizes.

Following completion of the heliostat field optimization and major component design work, the plant layout and balance-of-plant design was started. This work involved equipment arrangement considerations and specifications for conventional plant equipment such as feedwater heaters and cooling towers, interconnective piping, power conditioning equipment, and structures and buildings. Finally, the master control subsystem was designed to integrate the other subsystem control strategies. This involved some iteration on subsystem control concepts, especially in the area of receiver flow control.

After completing the conceptual design, the performance and cost of the commercial plant were estimated. Plant performance was estimated at the design point (noon of the summer solstice), during the design point day, and at several other times during the year. The analysis of plant performance included audits of the power flow at key locations within the plant at the design points and audits of the net electric energy available throughout the design point day as well as an estimate of the plant efficiency during the design point day.

The capital cost estimate for this plant was based on 1978 dollars. Estimates for both the first commercial plant and a plant representative of the mature technology were made. The capital cost estimates were itemized by the accounting categories shown in Table 2-3. Within each of these categories, costs are reported for major components, field labor, and field material. The estimate covers all hardware, engineering, construction, and management costs associated with the plant construction and startup. The distributables account was developed with the assumption that an architect-engineering firm would design the plant, procure the major components, and manage a constructor who would procure the field materials and provide the construction labor force. An assessment of the potential for utility market penetration by this commercial plant concept was made. A list of power plant evaluation criteria reflecting factors generally considered by a utility in evaluating a plant procurement decision was prepared (Table 2-4). The plant design was evaluated with respect to these criteria and compared to competing power generation technologies.

2.2.5 DEVELOPMENT PLAN

A plan outlining the stages of development leading to a detailed design for construction of the first commercial plant was identified. This development plan trades off risk vs. development time and cost and provides a program in which the commercial plant detailed design would be completed in the late 1980s.

The plan identifies a series of subsystem research experiments that will support the plant design effort by providing the data base required for a detailed design and establishing the required level of confidence in the design and operational characteristics of the unique or advanced components.

Table 2-3

ACCOUNTING COST CATEGORIES

Site, Structures, and Miscellaneous Equipment
Turbine Plant Equipment
Electric Plant Equipment
Collector Equipment
Receiver Equipment
Thermal Storage Equipment
Distributables

Table 2-4

POWER PLANT EVALUATION CRITERIA

Economic Viability
Capital Cost
Operation and Maintenance Requirements
Plant Performance
Reliability
Safety
Power Plant Siting
Availability of Plant Components

The plan also calls for a pilot plant which will model the critical areas of the commercial plant design that could not be addressed in a subsystem research experiment, and will provide a vehicle for analyzing and demonstrating the integration and control characteristics of the design.

The development plan identifies stages where the commercial plant design may be updated with respect to new technical information as it becomes available from this program and others.

2.3 TECHNICAL TEAM

Table 2-5 identifies the General Electric team and the respective roles of each member organization. The team includes Foster Wheeler Development Corporation, Kaiser Engineers, and the University of Houston Energy Laboratory. Within the General Electric Company, Corporate Research and Development had responsibility for the conduct of the program, including program and technical management and all system integration functions. The Energy Systems Programs Department had responsibility for the integration of the collector and receiver subsystems and for the heliostat specifications and field optimization work conducted at the University of Houston. The Advanced Reactor Systems Department had responsibility for the receiver and storage subsystems. Steam turbine design, performance, and cost estimates were developed by the Medium Steam Turbine Department. The Electric Utility Systems Engineering Department was responsible for the master control subsystem and an assessment of the commercial plant concept in an electric utility situation.

A Technical Review Panel consisting of eight senior technical managers reviewed the technical progress of the program. The panel was chaired by C.H. Holley, General Manager of General Electric's Electric Utility Systems Engineering Department. L.T. Papay, Director of Research of the Southern California Edison Company served as a member of this panel and provided an evaluation from a utility point of view.

Table 2-5

PROGRAM ORGANIZATION

Contributor	Activity
General Electric Company	
Corporate Research and Development	Program and Technical Management Systems and Plant Integration Electrical Energy Conversion Subsystem
Energy Systems Programs Department	Heliostat Design Collector/Receiver Subsystem Integration
Electric Utility Systems Engineering Department	Master Control Subsystem Electric Utility Implementation
Advanced Reactor Systems Department	Receiver Subsystem Storage Subsystem
Medium Steam Turbine Department	Steam Turbine-Generator
Energy Laboratory, University of Houston	Heliostat Field Design Solar Flux Plot at Receiver
Foster Wheeler Development Corporation	Heat Exchanger Design Storage Vessel Design
Kaiser Engineers, Inc.	Tower Design Storage Vessel Design Balance of Plant Design Plant Layout Safety Analysis Environmental Impact Analysis

Section 3

PARAMETRIC ANALYSIS

3.1 INTRODUCTION

Figure 3.1-1 describes the reference system configuration which seemed to offer the greatest promise for reduced costs when the parametric analysis was begun. The major features of this configuration are high peak temperature (1300 F), enclosed plastic heliostats, and thermocline type energy storage in iron balls.

Since a large number of technical options were to be considered, it was not possible to construct a complete system performance and cost estimate for each option. Instead the options were assessed by analyzing only those subsystems which would be most strongly affected when the plant configuration is changed from the baseline to the new design. For instance, changing heliostats from the enclosed design to a nonenclosed glass design would require significant design changes in the collector and receiver subsystems, but the impact of this change on the storage and power generation subsystems would be small. In a similar way, changes in the receiver configuration would affect the field design but would not have an impact on the storage subsystem or the electric power generation subsystem (EPGS). This argument in summarized in Table 3.1-1 which shows that the technical options associated with receiver configuration and heliostat type were assessed by analyzing only the collector and receiver subsystems.

The selection of peak working fluid temperature affects the storage subsystem as well as the receiver and collector subsystems, so this option was addressed as part of the storage concept trade-off as shown in Table 3.1-2.

The steam cycle selection affects all of the other subsystems and can only be made on the basis of the complete plant cost. The steam cycle options which were investigated are listed in Table 3.1-3.

Thus the parametric analysis was divided into three sets of cases:

- Receiver/collector
- Storage
- EPGS

The analysis and data for these cases are described in Sections 3.2 through 3.6 below, and the assessment of these results and selection of the preferred configuration are discussed in Section 4.

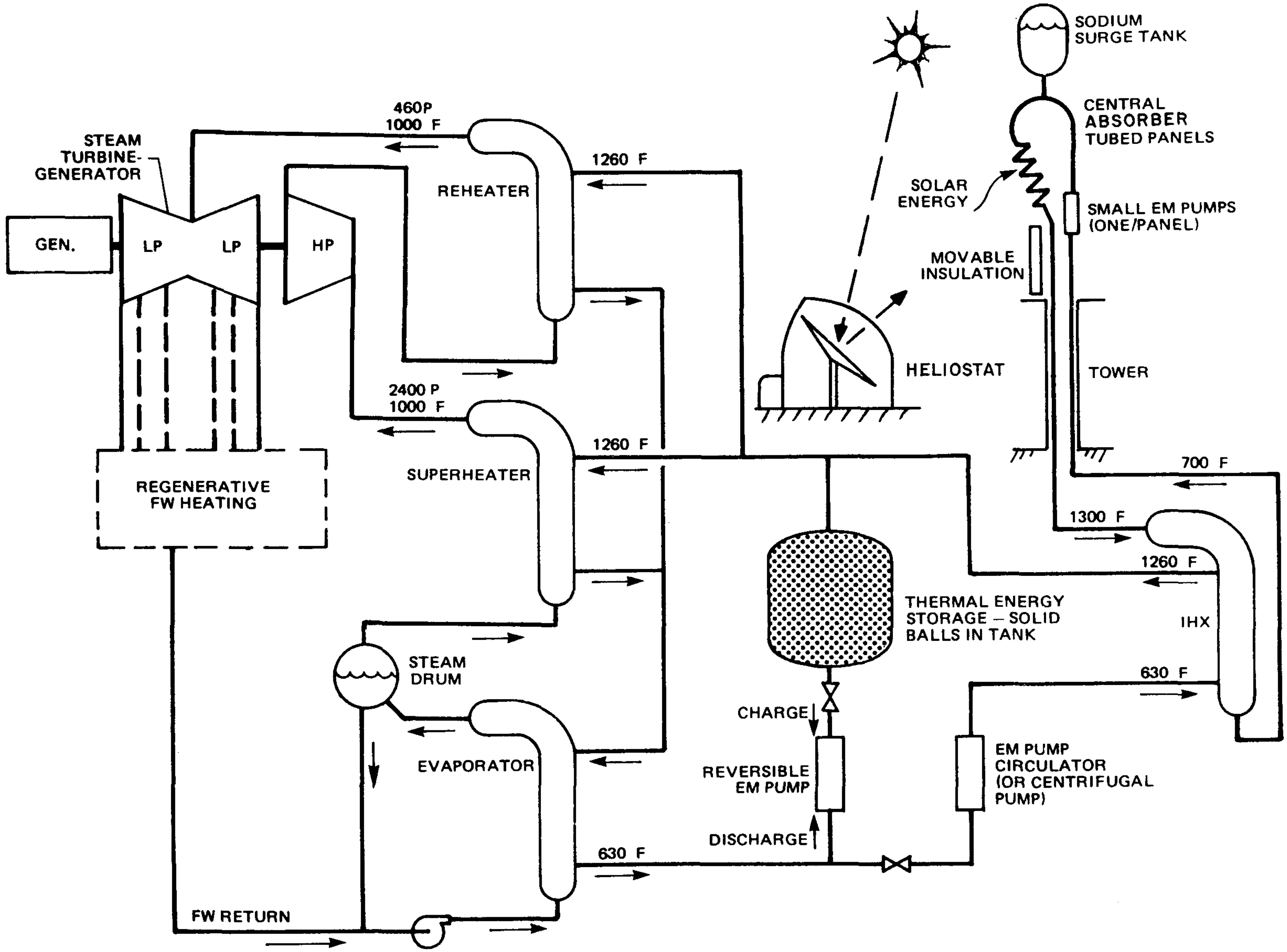


Figure 3.1-1. Advanced Central Receiver Reference System Configuration

Table 3.1-1

PARAMETRIC ANALYSIS LOGIC - RECEIVER/COLLECTOR CASES

Systems Analyzed	Options Considered
Collector Subsystem	
Heliostats	• Enclosed Plastic Heliostat
	• Glass Heliostat
Wiring & Land	• North Field
	• 360° Field
Receiver Subsystem	
Absorber, Riser, Downcomer, Tower and Pumps	• External Absorber Flat Cylindrical
	• Cavity Absorber

Table 3.1-2

PARAMETRIC ANALYSIS LOGIC - STORAGE CASES

Systems Analyzed	Options Considered
Collector Subsystem	• Glass Heliostat
	• 360° Field
Receiver Subsystem	• External Cylindrical Receiver
	• 1100 F Peak Temperature
	• 1300 F Peak Temperature
	• EM Pumps in Absorber
	• Control Valves in Absorber
Storage Subsystem	
Storage Tanks	• Field Assembled Tanks
	• Factor Assembled Tanks
Storage Medium	• 1100 F Peak Temperature
	• 1300 F Peak Temperature
	• Hot Sodium
	• Hot Sodium + Iron
	• Hot Molten Salt
Valves, Heat Exchangers, Pumps, and Piping	• Intermediate Heat Exchanger
	• Throttling Valve
	• Two Pressure Storage
	• Jet Pump

Table 3.1-3

PARAMETRIC ANALYSIS LOGIC - EPGS CASES

Systems Analyzed	Options Considered
Collector Subsystem	• Glass Heliostat • 360° Field
Receiver Subsystem	• External Cylindrical Absorber • Control Valves in Absorber • 1100 F Peak Temperature
Storage Subsystem	• Field Assembled Tanks • 1100 F Peak Temperature • Hot Sodium • Throttling Valve
Electric Power Generation Subsystem	
Steam Turbine, Condenser, Feedwater Heaters, and Cooling Towers	• 1450 P/1000 F/1000 F • 1800 P/1000 F/1000 F • 2400 P/1000 F/1000 F • 2400 P/1000 F/1000 F HARP* • 2400 P/1050 F/1050 F HARP • 2400 P/1050 F/1050 F/1050 F

*HARP = Feedwater Heater Above Reheat Point

3.2 COLLECTOR SUBSYSTEM

In Task 2 of this study, the four field configurations listed in Table 3.1-1 were designed by the Energy Laboratory of the University of Houston.

The input information required for the University of Houston heliostat field optimization program consists of heliostat specifications and cost and performance models for the receiver and tower. Heliostat input data were supplied by General Electric and Sandia Livermore Laboratories (glass heliostat) and were carefully reviewed by both organizations to insure consistency. Receiver and tower cost and performance models were derived from data generated by Foster Wheeler, GE Advanced Reactor Systems Department, and Kaiser Engineers. The input information is summarized in Section 3.2.1.

The heliostat field design methodology involves minimizing the figure of merit, defined as total capital cost for collector and receiver subsystems divided by net annual thermal energy derived at the base of the tower. The objective of the optimization was to minimize the figure of merit for the field layouts with respect to receiver size, tower height, and field arrangements. The results of the field analysis are presented in Section 3.2.2.

3.2.1 FIELD DESIGN INPUT DATA

Heliostat Specifications

Heliostat data required as input to the field design program includes the dimensions, costs, and performance of the heliostat.

Dimensions. These are the dimensions of the reflector, support, enclosure (for enclosed heliostat), and the minimum spacing distance required between the centerlines of two neighboring heliostats.

Costs. The costs fall into two categories: unit cost and land cost. The unit cost includes material, installation labor, overhead, interconnecting cabling, central computer for the heliostat system, AE fees, site survey, etc. The land cost includes acquisition, drainage, and clearing of obstacles such as trees.

Performance. The performance data include the outage rate estimates, minimum solar elevation for tracking, optical performance of reflector and enclosure (aging and fouling effects included), and guidance error which includes errors due to misalignment, mechanical drive tolerance, computer precision, reflector surface aberration, and wind effects. The guidance error is expressed as the Gaussian standard deviation of the angular optical errors.

The enclosed and unenclosed heliostat data initially proposed by GE for the parametric analysis are given in Table 3.2-1. The enclosed heliostat data were taken from the General Electric prototype heliostat study proposal (RFP No. EG-77-R-03-1468), and correspond to first commercial units. The glass heliostat data were extracted from the 10 MW pilot plant preliminary design performed by McDonnell Douglas Astronautics (Ref. 3.2-1).

Following a review of the proposed specifications by Sandia Laboratories, it was decided that a less conservative projection of advanced heliostat technology would be more appropriate. Consequently the DoE Prototype Heliostat Program goals were adopted as the guidelines. These goals correspond to a mature technology available in 1990 and high production levels.

Preliminary field optimization results indicated that the enclosed heliostat produced a more diffuse image and required a larger absorber than the glass heliostat. The enclosed heliostat specification was therefore changed to include a focused reflector to achieve image quality similar to that of the glass heliostat. As discussed below, this increased the peak flux, but did not have a significant effect on receiver size. Focusing the enclosed heliostat is relatively simple and an incremental cost of $0.50/square meter was allocated for focusing equipment.

The heliostat design specifications that were selected as a result of this review process are itemized in Table 3.2-2. These specifications were used in the study.

Receiver Cost and Performance Models

The models summarized in Table 3.2-3 for estimating the performance and costs of the receiver were derived from data provided by Foster Wheeler, GE Advanced Reactor Systems Department, and Kaiser Engineers.

3.2.2 FIELD ANALYSIS

As mentioned previously, optimized fields for the four concepts considered were designed by the University of Houston Energy Laboratory. The constraint was to design a field/receiver which would deliver 376 MW_t net power at equinox noon. The figures of merit (FOM) for the four optimized fields were then compared. The FOM is the total capital cost for the receiver and collector subsystems divided by the net annual energy:

$$FOM = \frac{C_R + C_c + C_o}{\int_{annual} (\dot{Q} - \Delta\dot{Q})dt} \qquad \frac{\$}{MW_t h}$$

Where C_R = receiver subsystem cost (including receiver, tower, pumps, and piping)

C_c = collector subsystem cost (including land, wiring, and heliostats)

C_o = fixed cost to open site (including costs for permits, environmental impact statements, road and rail connections)

$\dot{Q}$ = thermal power at tower base

$\Delta\dot{Q}$ = thermal power required to operate field and receiver auxiliaries

t = time

The integration in the denominator is over the period of one year.

Table 3.2-1

HELIOSTAT SPECIFICATIONS (CORRESPONDING TO FIRST COMMERCIAL PLANT)

		Enclosed (Plastic)	Exposed (Glass)
1.0	**Dimensions**		
1.1	Reflector Area	40 M^2	37.5 M^2
1.2	Support Height	3.81 M	2.74 M
1.3	Minimum Spacing Distance Between ¢ or Adjacent Heliostats	8.83 M	9.57 M
1.4	Enclosure Dimensions	8.2 M(D) x 7.46 M(H)	None
2.0	**Costs**		
2.1	Unit Cost	$47.50/$M^2$	$100/$M^2$
2.2	Land Cost	$1.14/$M^2$	$1.08/$M^2$
2.3	Cost Basis (Year)	1978	1977
2.4	Escalation Rates for Conversion to '78 $'s	None	6%
3.0	**Performance**		
3.1	Outage Rate Estimates	1/4 hr/yr/Heliostat	1/2 hr/yr/Heliostat
3.2	Minimum Sun Tracking Angle	10^{o}	10^{o}
3.3	Effective Relectivity	0.65*	0.91
3.4	Focusing	No	No
3.5	Guidance Error (1 σ estimate)	3.22 MRAD	Azi. 5.0 MRAD Ele. 3.6 MRAD

* Combined performance of : enclosure transmissivity in and out, each 0.86; reflectivity 0.88.

Table 3.2-2

HELIOSTAT SPECIFICATIONS USED IN TASK 2 STUDY
(CORRESPONDING TO MATURE TECHNOLOGY AND MASS PRODUCTION)

	Enclosed (Plastic)	Exposed (Glass)
1.0 Dimensions		
1.1 Reflector Area	55 M^2	49 M^2
1.2 Support Height	4.31 M	4.09 M
1.3 Minimum Spacing Distance Between ø of adjacent Heliostats	10.2 M	10.5 M
1.4 Enclosure Dimensions	9.60 M (D) x 9.09 M (H)	None
2.0 Costs (1978 Dollars)		
2.1 Unit Cost	\$25.28/M^2	\$65.0/M^2
2.2 Land Cost	\$1.14/M^2	\$1.14/M^2
3.0 Performance		
3.1 Method to account for Outage	Design field 1.8% oversize	Design field 1.8% oversize
3.2 Minimum Sun Tracking Angle	10^o	10^o
3.3 Effective Reflectivity	0.574*	0.90
3.4 Focusing	Yes	No
3.5 Guidance Error (1 σ estimate)	2.8 MRAD	2.5 MRAD
3.6 Power Required - Tracking	35 watts/hr	90 watts/hr
- Slew	55 watts/hr	-
- Stowed	15 watts/hr	0

*Combined performance of: enclosure transmissivity in and out, each 0.86; reflectivity 0.88; enclosure blockage 0.99; and surface degradation 0.89.

Table 3.2-3

RECEIVER PERFORMANCE AND COST MODELS
FOR PARAMETRIC ANALYSIS

		Cylindrical Receiver	Flat Receiver
Losses	Convective Loss (Mw_t)	$3.37[4.194E\text{-}3(D \times H)+1.557E\text{-}4\ Qn]$	$2.74[3.017E\text{-}3(W \times H)+3.405E\text{-}4Qn]$
	Radiative Loss (Mw_t)	$15.80[3.839E\text{-}3(D \times H)+3.324E\text{-}4\ Qn]$	$8.68[2.462E\text{-}3(W \times H)+6.748E\text{-}4\ Qn]$
	Pumping Loss* (Mw_e)	$[1.2306+0.02557(T+H)]\left(\frac{Qn}{376}\right)$	Same
Costs (M$)	Panels	$3.0448+.1164(H-15)+.2029(D-15)$	$1.1375+.0374(H-17)+.0669(W-17)$
	Pumps*	$3.6+[2.507+.0341(T+H)]\left(\frac{Qn}{376}\right)$	$3.39+[2.507+.0341(T+H)]\left(\frac{Qn}{376}\right)$
	Tower, R/D & Sodium	$1.613\exp[.00772T]$	Same
	Support Structure & Plumbing	$0.04464\sqrt{D^2+H^2}$	$0.0378\sqrt{W^2+H^2}$

Where H = Receiver Height (M)
D = Receiver Diameter (M)
W = Receiver Width (M)
T = Tower Height (M), measured from ground level to base of receiver
Qn = Absorbed Thermal Power (MW)

*Correspond to system with throttle valve, large EM pump at tower base, small EM pumps for trim on panels, 1100 F peak sodium temperature

Field sketches for the four concepts are shown in Figure 3.2-1. Overall sizes are shown as well as the number of the square computational cells into which the fields are divided in the calculations.

Table 3.2-4 indicates that the GE enclosed heliostat in a 360° field configuration is likely to be the most cost-effective approach. Therefore, the GE enclosed heliostat in a 360° field configuration is chosen for the Task 4 conceptual design.

Note that the peak fluxes with the enclosed heliostats are higher than for the glass heliostats; this is a result of focusing the enclosed heliostats to the slant range. However, focusing did not increase the average flux very much; both enclosed heliostat cases have lower average fluxes and larger receivers than the glass heliostat cases.

Preliminary estimates of performance and cost for cavity receivers had indicated that cavities did not offer any cost advantage for the sodium-cooled receiver concept. This question was reviewed in Task 2, based on the results of the flat receiver with glass heliostats. Figure 3.2-2 shows how a hypothetical cavity was formed from the flat geometry by moving the vertical surface back from its original position and adding four surfaces to form a box.

As the original surface moves back, the effective absorptivity and emissivity of the cavity increase. If the radiation is assumed to be diffuse, the apparent absorptivity is given by

$$\alpha_E = [1 - \frac{a}{A}(1 - \frac{1}{\alpha})]^{-1}$$

where a = area of aperture

A = area of tubed surface

α = absorptivity of tubed surface

The relationship for effective emissivity is identical with ε substituted for α. Thus as the cavity gets deeper the amount of sunlight captured increases. The reradiated power actually goes down despite the increase in ε because the average flux at the tubed surface goes down and this reduces the surface temperature and emitted black body flux levels.

Thus both the absorption and emission characteristics of the receiver improve as the cavity becomes deeper. However, convection losses increase rapidly as the tubed surface area increases, and these losses eventually negate the gains due to absorption and emission as is shown in the receiver efficiency curve, Figure 3.2-3. This conclusion is based on the assumption that the air side convection coefficient does not change as the cavity deepens. The structure shown in Figure 3.2-2 is quite open to the environment, and it is unlikely that the added sides of the cavity would shield the tubed surfaces from wind or free convection currents.

The performance calculations summarized in Figure 3.2-3 are detailed in Appendix A.

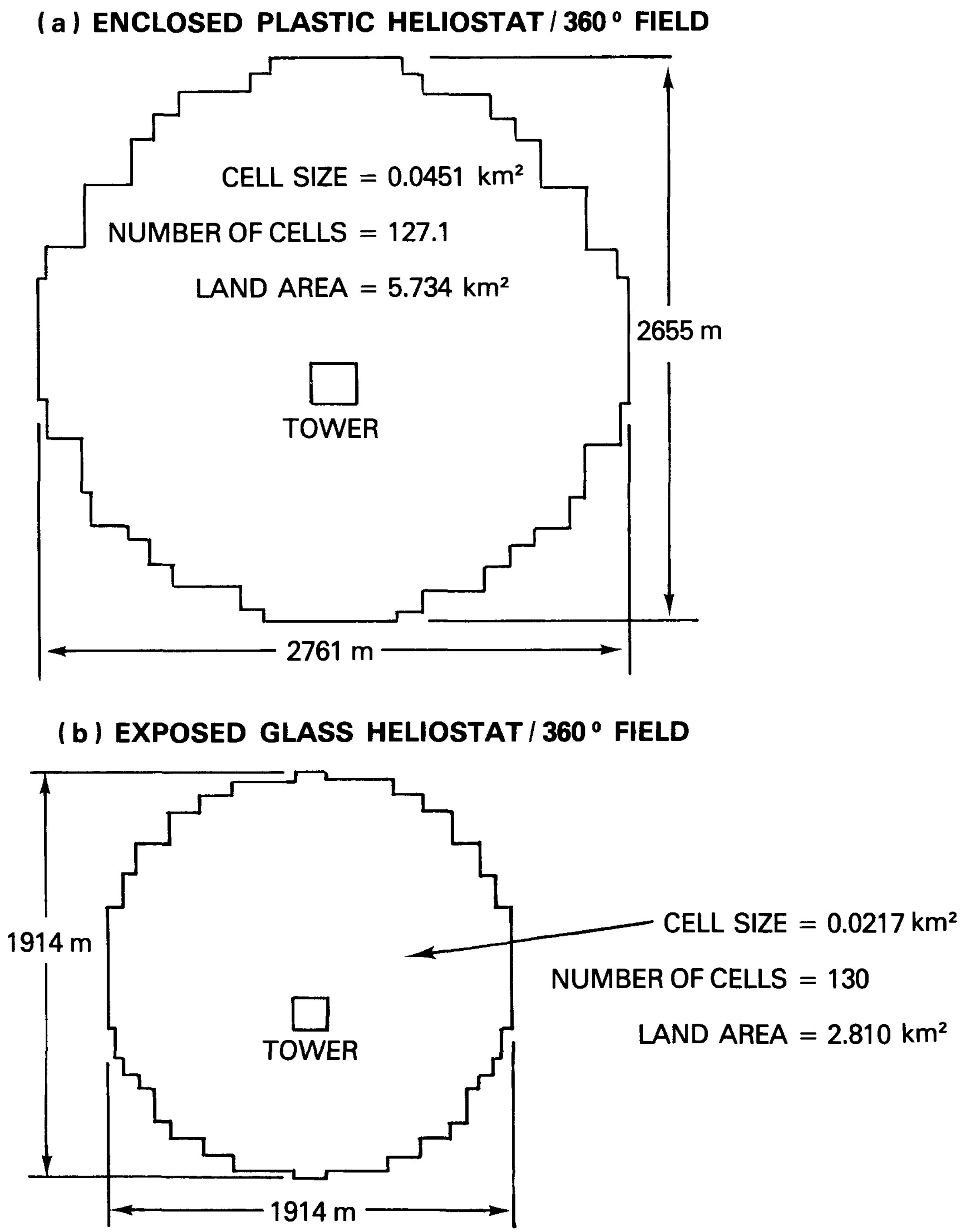
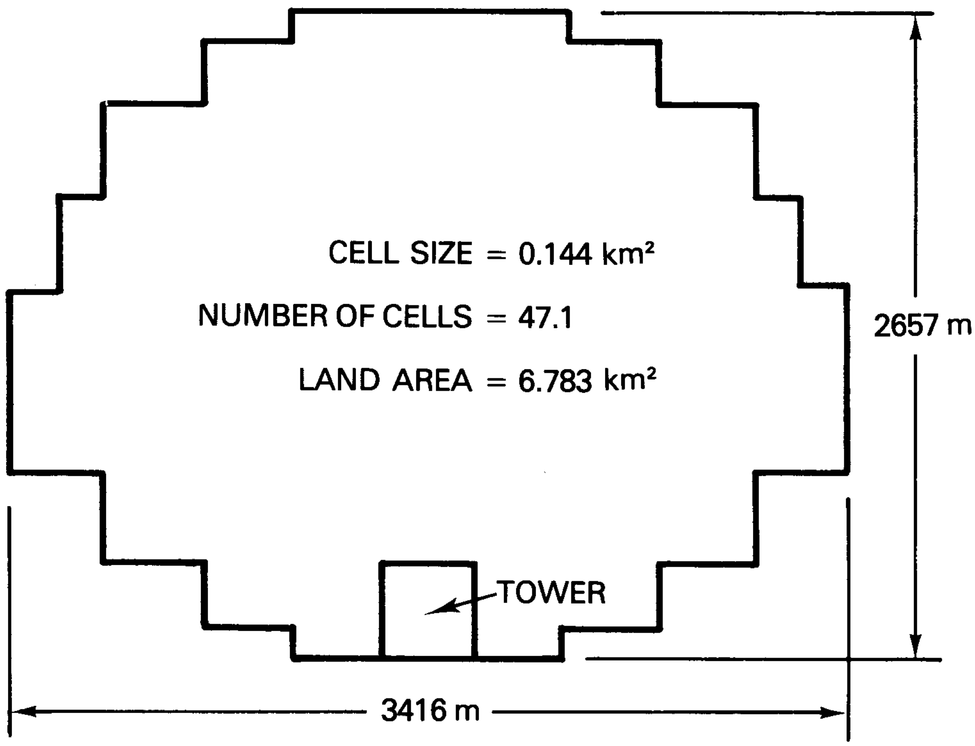


Figure 3.2-1. Field Layout Sketches

(c) ENCLOSED PLASTIC HELIOSTAT / NORTH FIELD



(d) EXPOSED GLASS HELIOSTAT / NORTH FIELD

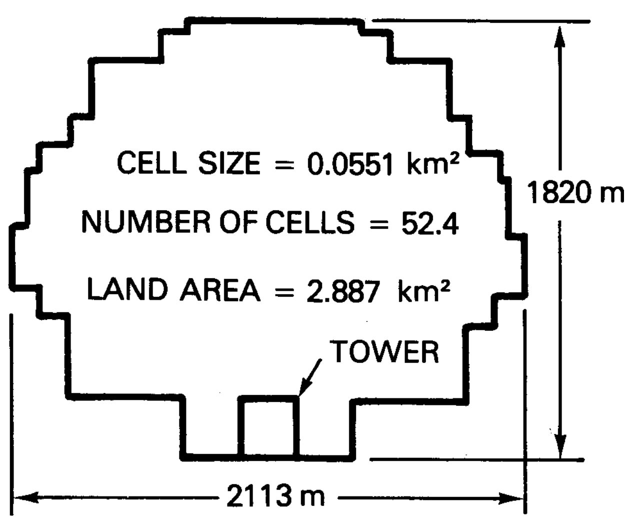


Figure 3.2-1. (Cont'd)

Table 3.2.4

SUMMARY OF TASK 2 HELIOSTAT FIELD DESIGN RESULTS

HELIOSTAT		GLASS/UNENCLOSED		GE/ENCLOSED	
RECEIVER		FLAT	CYLINDER	FLAT	CYLINDER
Receiver Elevation**	M	210	170	240	190
Receiver Size*	M x M	20.81x20.81	12.5x12.5	23x23	16x16
Heliostat Size	M^2	49	49	55	55
No. of Heliostats		12,216	13,908	21,017	20,936
Land Area	KM^2	2.887	2.810	6.783	5.734
Receiver Peak Flux	MW/M^2	3.732	1.665	5.1293	1.9073
Annual Power	TW_TH	0.846	0.886	0.835	0.863
Fixed Cost	$M	4.500	4.500	4.500	4.500
Tower Cost	$M	7.773	5.894	9.731	6.796
Receiver Cost	$M	1.535	2.247	1.763	3.364
Structure Cost	$M	1.112	0.789	1.230	1.010
Pump Cost	$M	13.279	12.046	14.297	12.765
Land Cost	$M	3.291	3.203	7.733	6.537
Wiring Cost	$M	1.661	1.713	2.427	2.030
Heliostat Cost	$M	37.247	42.584	26.795	27.079
Total Cost	$M	70.398	72.976	68.476	64.081
Figure of Merit	$\$/MW_TH$	83	82	82	74

* Width X Height For Flat Receiver, Diameter X Height For Cylinder Receiver

** Measured from top of heliostat pedestal to middle of receiver

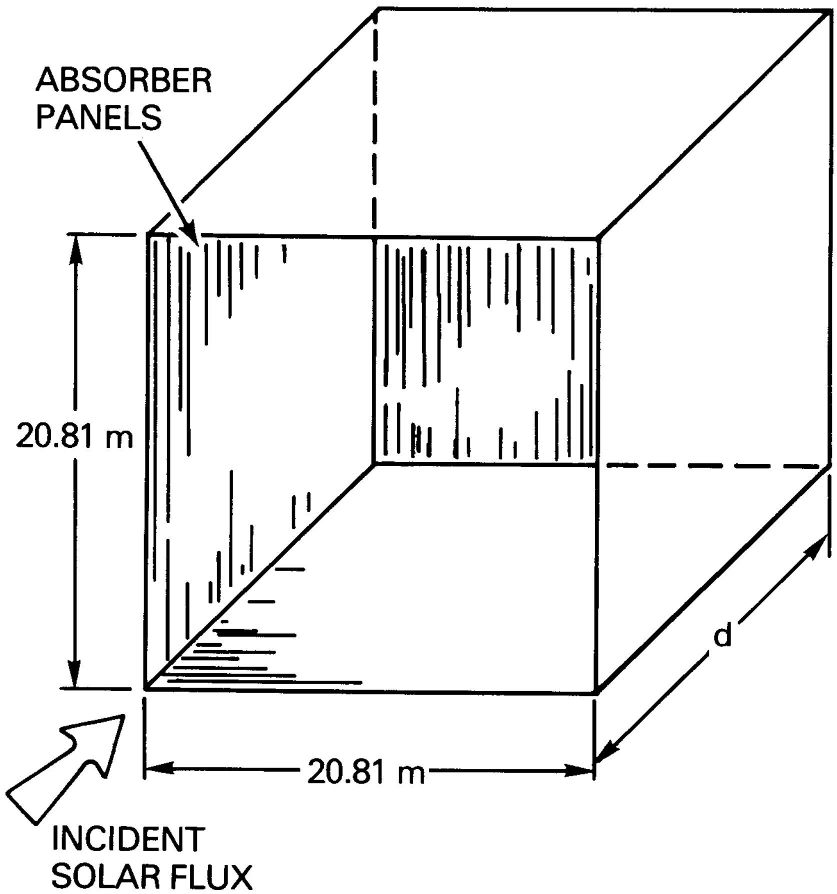


Figure 3.2-2. Cavity Receiver Model

Cost estimates were made for the cavities in Figure 3.2-3, based on the total cost for the receiver/collector subsystems shown in Table 3.2-4 and the scaling equations in Table 3.2-3. These costs are plotted in Figure 3.2-3, along with an estimate of annual energy delivered to the tower base (assumed proportional to efficiency). The quotient of these two numbers yields the figure of merit plotted in Figure 3.2-3. These cost estimates are also detailed in Appendix A.

The figure of merit for the cavity increases with depth. That is, the least expensive cavity is none at all (flat plate) which confirms the preliminary conclusion. Therefore a cavity receiver was not chosen for the Task 4 conceptual design.

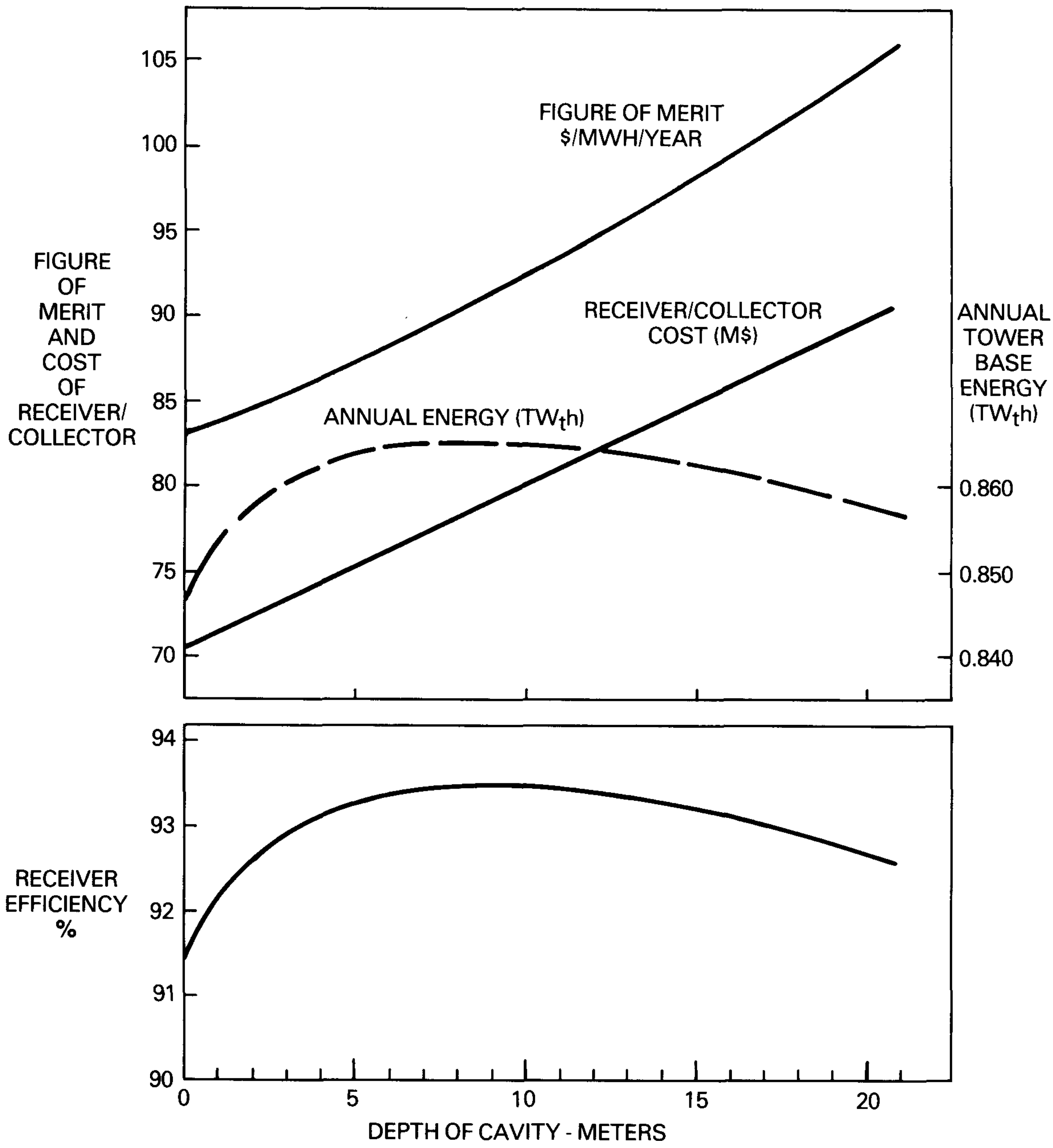


Figure 3.2-3. Cavity Receiver Performance and Cost

REFERENCES FOR SECTION 3.2

3.2-1. Central Receiver Solar Thermal Power System, Phase I, Pilot Plant Preliminary Design Report, Vol. III, MDC G-6776, Oct. 1977.

3.2-2. Solar Central Receiver Prototype Heliostat Phase I, Final Technical Report, SAN-1468-77-2, Sept. 1978.

3.3 RECEIVER SUBSYSTEM

3.3.1 RECEIVER CONCEPTS

Parametric Cases Studied

Two basic receiver types were evaluated as part of the parametric study, namely, a flat solar absorber with a north heliostat field and a cylindrical solar absorber with a surrounding heliostat field (360°). Each of these receiver configurations was designed to operate at two different temperature levels, 1100 F and 1300 F maximum sodium temperature. The purpose was to determine which of the four designs was most cost effective.

North Heliostat Field with Flat Absorber

The north field receiver is shown in Figure 3.3-1. The flat absorber surface is assembled from a number of smaller rectangular absorber panels stacked edge to edge in the vertical orientation. The riser line brings sodium up the tower from the cold leg of the IHX and terminates at a horizontal header mounted behind the absorber panels. Feeder lines from the header carry the individual panel flows to small EM pumps which regulate the flow to the panels and provide tower loop circulation of the sodium. The pumps discharge into the individual panels at mid-length where the flow splits and flows in two paths upward and downward through the panels. At the ends of the panels the hot sodium flow is returned through panel outlet feeder lines to an outlet header mounted in back of the panels on the steel support structure. The downcomer line is connected to the outlet header and returns the sodium to the hot leg of the IHX at the base of the tower. A sodium surge tank is mounted at the highest point in the system to accomodate thermal expansion of the sodium and to provide a place to vent gas bubbles from the sodium when the system is filled. The surge tank cover gas pressure (argon) is adjusted to give near atmospheric operating pressures in the receiver components.

The backs and sides of the absorber panels are insulated to reduce parasitic heat loss while operating. When shut down, a moveable insulation curtain is provided which rolls up across the front face of the panels to keep the heat loss to the atmosphere to a minimum. In this way the receiver can be placed on a hot standby mode overnight or during cloud cover without heavy heat loss and danger of freezing the sodium in the tubes of the absorber panels.

Convection damper curtains were placed across the top and sides of the absorber with the purpose of partially blocking wind and natural convection currents from direct impingement on the absorber tubes.

Based on a preliminary field design, the flat absorber was sized at 17 meters by 17 meters. Because the flux gradients were high across the width, a relatively large number of panels were used to limit the hot to cold tube performance differential. Therefore the designs for 1100 F and 1300 F both used 17 panels which were 1 meter wide and 17 meters long.

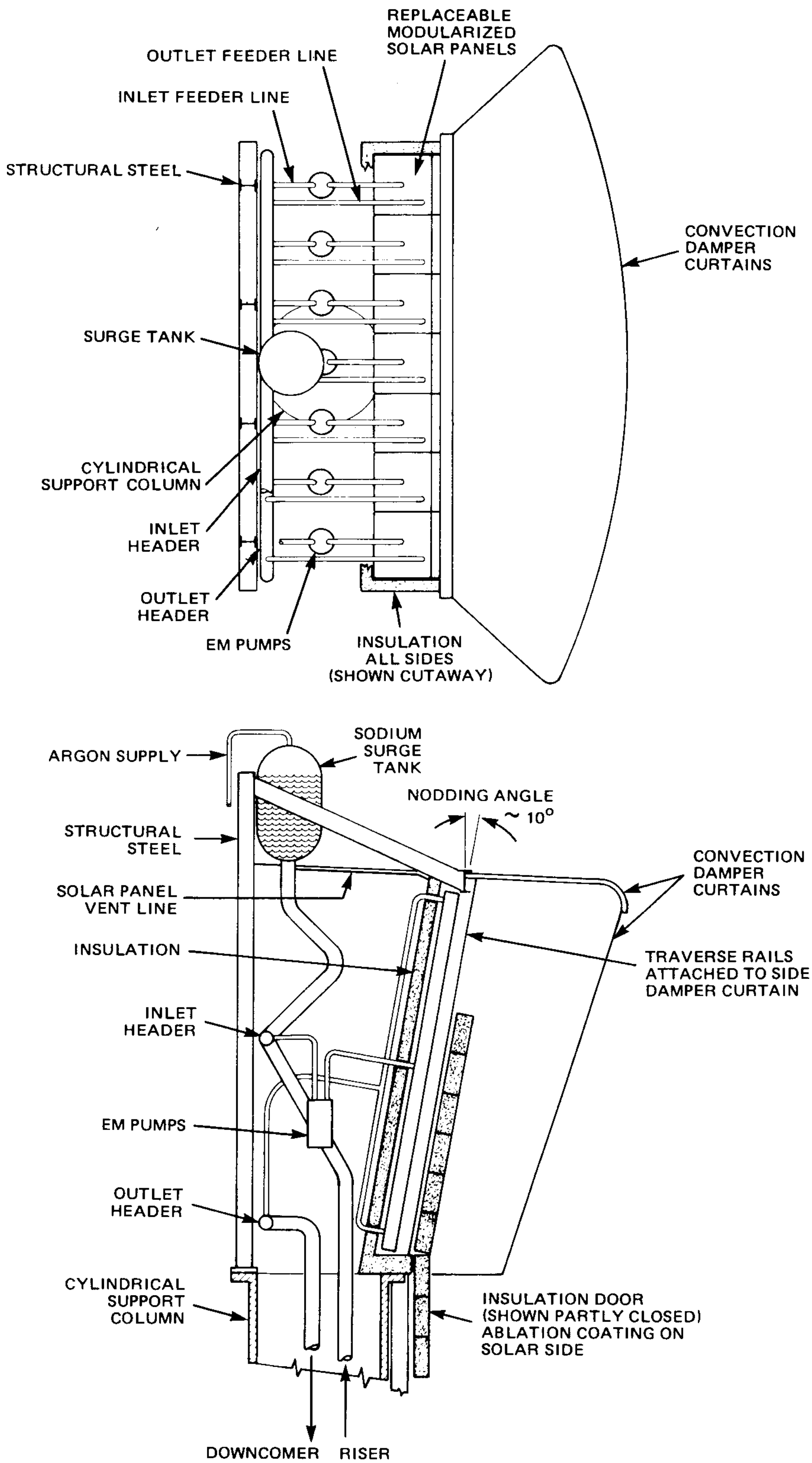


Figure 3.3-1. Flat Receiver Concept

Surrounding Field (360°) with Cylindrical Solar Receiver

The 360° field receiver is shown in Figure 3.3-2. The cylindrical surface of the absorber is composed of many flat rectangular absorber panels stacked edge to edge to form a cylinder. These panels are almost identical to the panels used in the flat receiver except that the edges of the panel must be wedged together at one angle to fit into the pie sections resulting from the cylindrical shape.

The riser brings cold sodium up into two headers, the preheat header and the main header. The preheat header is fed with sodium pressurized in an EM pump sized to handle the total flow to several absorber panels. The main header feeds flow through small EM pumps which pressurize the sodium into the panel inlets located at mid-length. The flow splits into up-flow and down-flow through the two halves of the panel. The panel outlet flows are gathered together in an outlet header and removed to the IHX via the downcomer line.

Early in the study, it was thought that economic advantages might result from preheating the sodium on the colder south side of the cylindrical absorber. It was found, however, that running the flow in series through two panels increases the pressure head. Also the inlet temperature to the hot panels is raised, causing cooling problems. Therefore, the preheat concept was dropped and the preheat panels were reconnected into the main inlet header in a fashion identical to the hot north side panels.

As in the case of the flat north field absorber, the cylindrical absorber is provided with insulation on the back and moveable front face insulation for periods of zero solar flux.

A surge tank is provided to allow for sodium expansion, venting during fill, and a means for regulating system pressures. Convection curtains were added initially and then removed from the design since the benefits did not justify the cost.

For the parametric study, the cylindrical receiver was sized at 15 meters in diameter by 15 meters in height. A total of 24 panels for both the 1100 F and 1300 F systems were used. The panels were 1.963 meters wide by 15 meters high.

Materials Selection

Table 3.3-1 shows a listing of materials used in the design of the sodium components of the 1100 F and 1300 F advanced central receiver systems for the Task 2 Parametric Analyses.

The alloy 2-1/4 Cr - 1 Mo was selected for the evaporator section of the steam generator because this material has been proven to have superior resistance to stress chloride cracking in water and resistance to decarburization in sodium. This material was also specified for the cold leg piping and cold storage tanks to avoid the necessity of making alloy transition welds between these components.

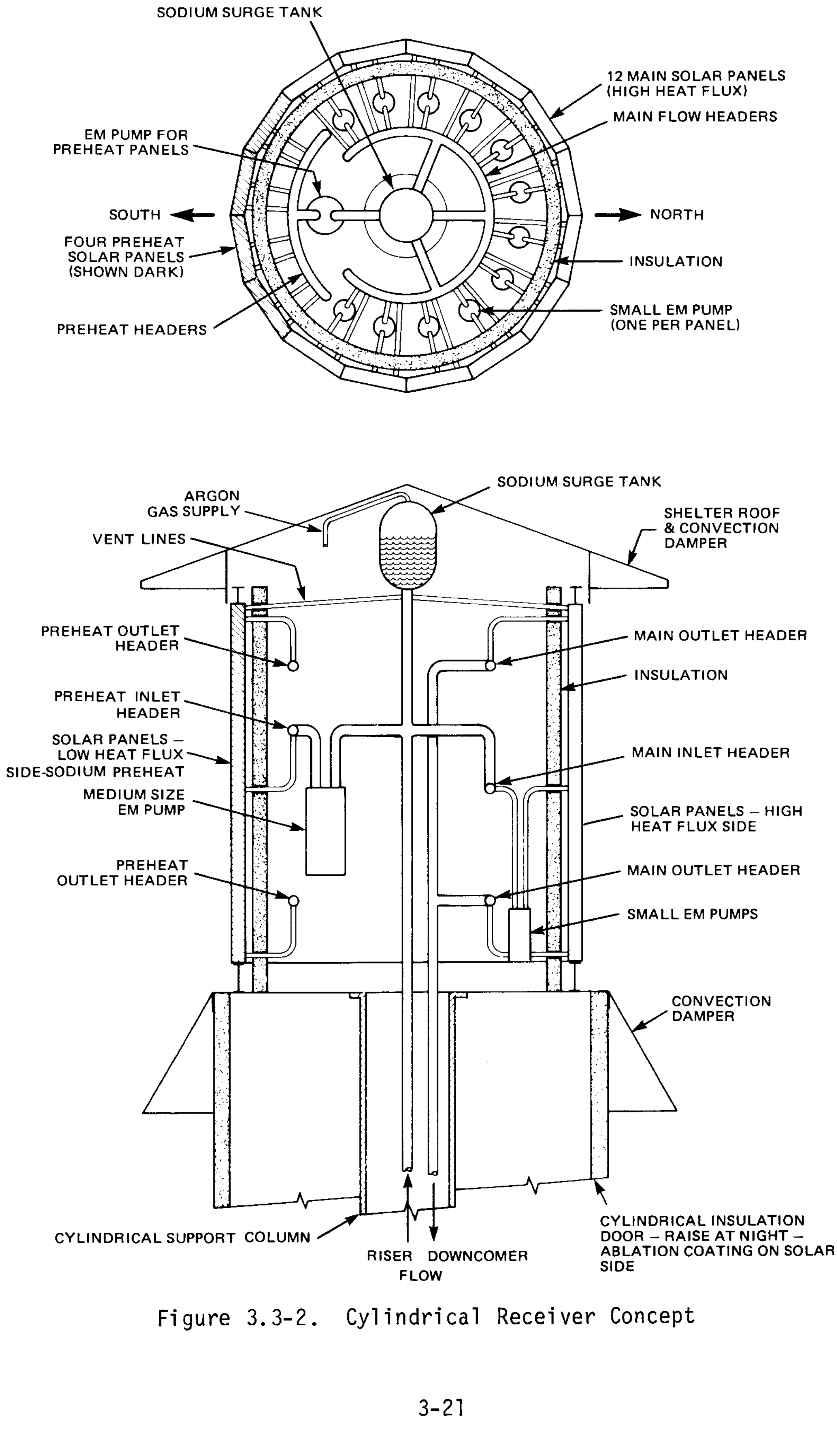


Figure 3.3-2. Cylindrical Receiver Concept

Table 3.3-1

MATERIALS SELECTION

	1100 F	1300 F
Cold Leg Piping	2-1/4 Cr - 1 Mo	2-1/4 Cr - 1 Mo
Hot Leg Piping	316SS	Inconel 625
Absorber Panels	316SS	Inconel 625
Storage Tanks (Hot)	316SS	Inconel 625
Storage Tanks (Cold)	2-1/4 Cr - 1 Mo	2-1/4 Cr - 1 Mo
IHX	316SS	Inconel 625
Evaporator	2-1/4 Cr - 1 Mo	2-1/4 Cr - 1 Mo
Superheater	Incoloy 800	Inconel 625
Reheater	Incoloy 800	Inconel 625

It was thought that the solar power plant size and cost could probably be reduced by using high peak sodium temperatures. However, the upper service temperature for 2-1/4 CR - 1 Mo is 950 F because it has unacceptably high decarburization rates in sodium and low creep strength at higher temperatures. Therefore, other alloys were considered for the hot leg components and the peak sodium temperature was treated as a parametric variable.

The peak temperature of 1100 F was selected in conjunction with type 316 stainless steal because this is the temperature at which the alloy begins to lose creep strength. All of the hot leg components except the steam generators were specified to be made of 316 SS. However, like all 300 series stainless steels, type 316 is subject to severe stress chloride cracking in aqueous environments, so Incoloy 800 has been specified for the steam superheater and reheater modules because of its good resistance to the type of corrosion. Both 316 SS and Incoloy 800 exhibits good sodium corrosion resistance at 1100 F.

The peak sodium temperature of 1300 F was selected on the basis of an initial look at sodium corrosion data for stainless steels and nickel base alloys. Corrosion rates for 316 SS are acceptable at 1300 F, but increase rapidly with increasing temperature. Although 316 SS has sufficient strength to be used in the absorber panels, the IHX, and hot leg piping, it was decided to investigate the possibility of using Inconel 625 because its superior strength might reduce the cost of the storage tanks. The sodium corrosion behavior of Inconel 625 was an unknown to be evaluated through a more detailed literature survey during the parametric analysis. In the superheater and reheater modules, Inconel 625 was specified to replace Incoloy 800 because Inconel was expected to have good stress chloride cracking resistance and Incoloy 73 known to have poor sodium corrosion resistance at 1300 F.

3.3.2 ABSORBER ANALYSIS

Absorber Panel Flux Plots

Preliminary receiver sizes were selected for the flat and cylindrical geometries based on collector/receiver optimizations performed by the University of Houston Energy Laboratory in previous work. Flux plots were also estimated on the basis of these earlier studies to make a preliminary estimate of receiver performance.

The 17-meter square flat receiver selected for analysis purposes was divided into 289 nodes as shown in Figure 3.3-3. The flux plot at noon (design point) is symmetrical about the vertical centerline of this square, and is almost symmetrical about the horizontal centerline as well. To reduce the complexity of the loss analysis, it was decided to assume the plot to be exactly symmetrical about both centerlines. Under this assumption it is possible to characterize the plot with only one quadrant; this quadrant is presented in Figure 3.3-4. The fluxes shown are heat-to-sodium, i.e., incident less reflection (5%), radiation, and convection. At the base of each column of numbers, the panel power is listed; all of the panels together provide 376.46 MW_t of thermal power to the sodium.

The cylindrical receiver was selected to be 15 meters tall by 15 meters in diameter, and it was divided into 408 nodes as shown in Figure 3.3-5. The flux plot at noon is symmetrical about the north/south centerlines and nearly so about the "belt." As with the flat receiver, a single quadrant of the flux plot was used to characterize the whole receiver (see Figure 3.3-6). The 24 panels in this receiver provide 397.77 MW_t heat to the sodium coolant.

Absorber Losses

The approach taken to estimate absorber losses for the parametric analysis is described below; these equations were programmed for the Hewlett Packard - 65 calculator. The detailed results of these calculations are presented in Appendix B.

The sodium side convection coefficient was obtained from Lyon's correlation (Ref. 3.2-1):

$$N_u = \frac{d_i}{12K_n} h_i = 7.0 + 0.025\ Pe^{0.8} \qquad (3.3\text{-}1)$$

where:

$$Pe = \frac{48\ W\ C_p}{\pi d_i\ K_n\ N_t} = \text{Peclet No.}$$

h_i = sodium side convection coefficient (Btu/hr-ft^2-°F)

d_i = tube ID (inches)

K_n = sodium conductivity (Btu/hr-ft-°F)

W = sodium flowrate (lb/hr) per panel

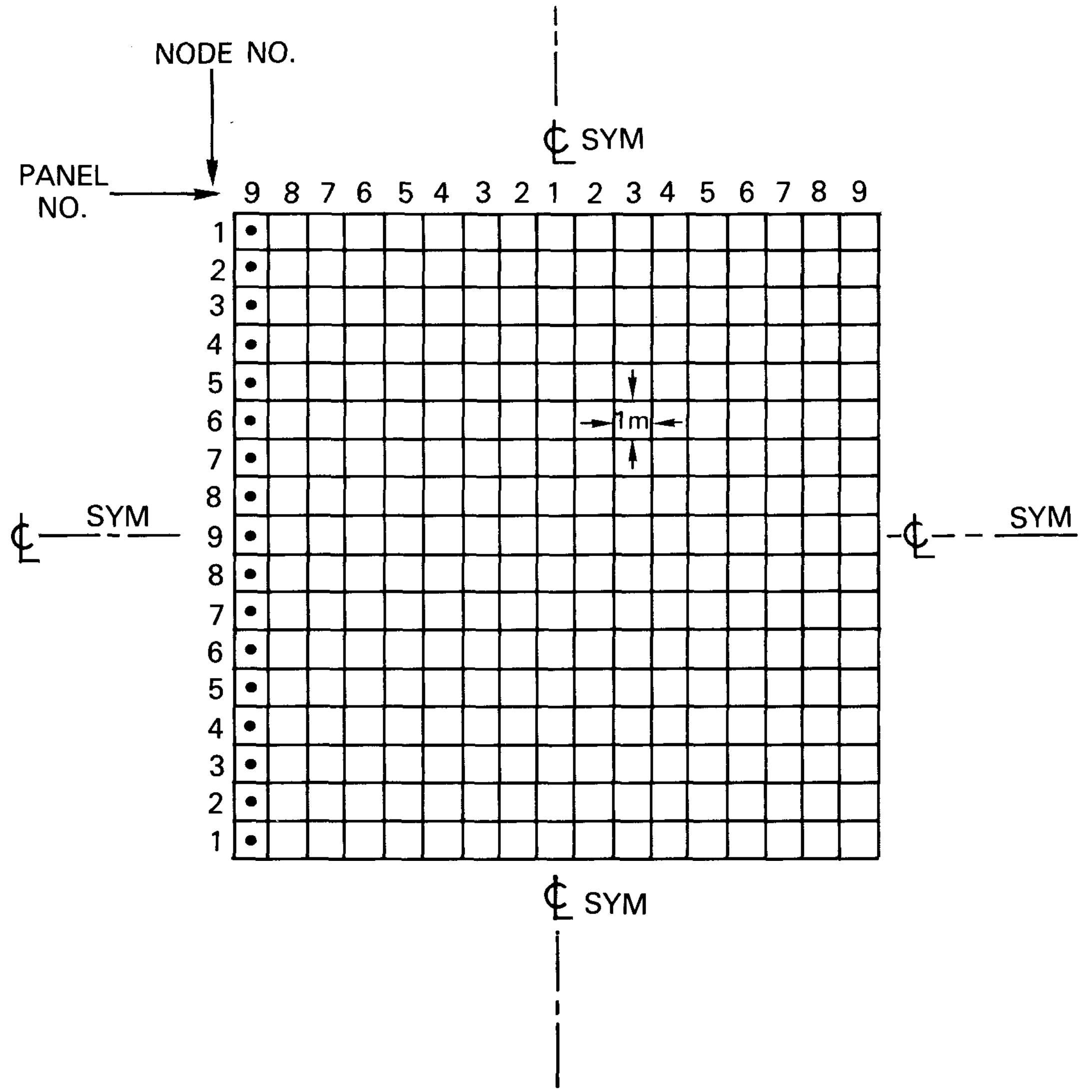


Figure 3.3-3. Key to Flat Receiver Loss Calculations

FLUX INTO SODIUM (MW/m^2)

$\Delta x = \Delta y = 1$ m

NODE NO. \ PANEL NO.	9	8	7	6	5	4	3	2	1
1	0.204	0.246	0.289	0.329	0.362	0.386	0.404	0.413	0.416
2	0.273	0.337	0.406	0.474	0.536	0.585	0.619	0.638	0.645
3	0.354	0.447	0.552	0.663	0.772	0.867	0.937	0.976	0.995
4	0.443	0.572	0.726	0.897	1.077	1.244	1.378	1.462	1.490
5	0.536	0.706	0.917	1.167	1.438	1.703	1.922	2.063	2.110
6	0.621	0.834	1.108	1.443	1.820	2.194	2.503	2.702	2.769
7	0.691	0.940	1.271	1.687	2.160	2.631	3.017	3.259	3.339
8	0.737	1.011	1.381	1.852	2.395	2.931	3.363	3.627	3.714
℄ 9	0.753	1.035	1.418	1.909	2.475	3.033	3.481	3.753	3.841
PANEL POWER MW_t	8.471	11.221	14.718	18.933	23.595	28.115	31.767	34.013	34.797

Figure 3.3-4. Flat Absorber - Flux Plot

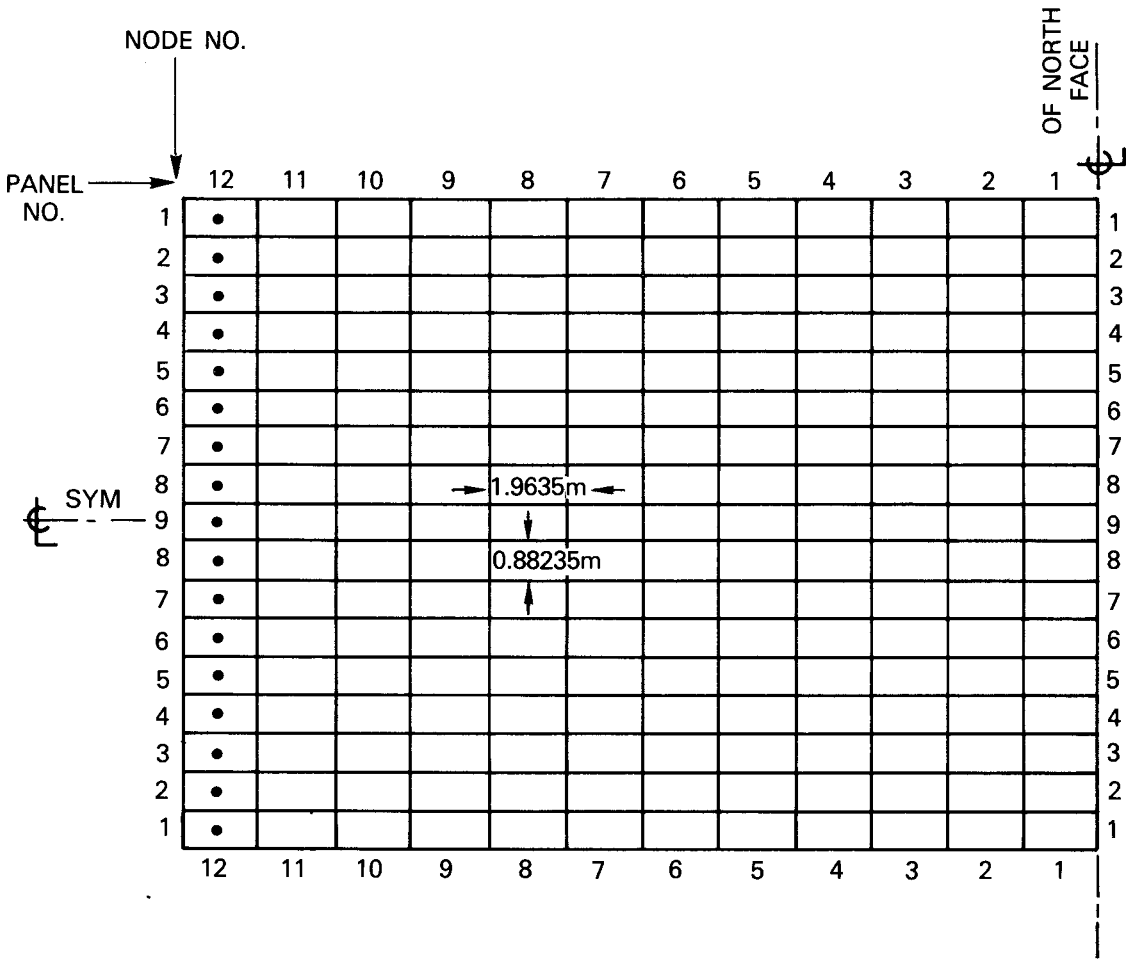


Figure 3.3-5. Key to Cylindrical Receiver Loss Calculations

FLUX INTO SODIUM (MW/m^2)

$\Delta x = 1.9635$ m

$\Delta y = 0.88235$ m

NODE NO. / PANEL NO.	12	11	10	9	8	7	6	5	4	3	2	1
1	0.00592	0.00825	0.0121	0.0229	0.0353	0.0641	0.0864	0.108	0.135	0.158	0.179	0.190
2	0.0193	0.0238	0.0368	0.0517	0.0766	0.125	0.163	0.197	0.239	0.271	0.304	0.319
3	0.0514	0.0591	0.0691	0.108	0.148	0.225	0.285	0.338	0.397	0.439	0.485	0.505
4	0.116	0.125	0.145	0.194	0.260	0.369	0.453	0.530	0.607	0.662	0.719	0.745
5	0.209	0.227	0.248	0.316	0.405	0.703	0.657	0.749	0.841	0.920	0.983	1.020
6	0.326	0.346	0.366	0.447	0.562	0.724	0.856	0.983	1.090	1.170	1.240	1.270
7	0.440	0.452	0.475	0.563	0.699	0.872	1.030	1.170	1.270	1.360	1.440	1.470
8	0.518	0.529	0.546	0.637	0.791	0.960	1.130	1.270	1.380	1.470	1.550	1.590
℄ 9	0.534	0.541	0.564	0.652	0.801	0.975	1.150	1.280	1.390	1.480	1.560	1.600
PANEL POWER MW_t	6.765	7.070	7.554	9.236	11.703	15.695	18.141	20.738	23.056	24.913	26.611	27.405

℄ SOUTH SIDE — ℄ NORTH SIDE

Figure 3.3-6. Cylindrical Absorber - Flux Plot

C_p = sodium specific heat (Btu/lb-°F)

N_r = number of tubes/panel

The overall tube thermal conductance referenced to the tube OD is given by

$$\frac{1}{U_o} = \frac{1}{h_i} \; \frac{1}{1-2\delta/d_o} + \frac{d_o}{24K_w} \; \ell n \left[\frac{1}{1-2\delta/d_o} \right] \qquad (3.3\text{-}2)$$

where δ = tube wall thickness (inches)

d_o = tube OD (inches)

K_w = tube conductivity (Btu/hr-°F-ft)

The inside and outside tube temperatures were then evaluated by

by

$$T_{wi} = T_{Na} + q/h_i \qquad (3.3\text{-}3)$$

$$T_{wo} = T_{Na} + q/U_o \qquad (3.3\text{-}4)$$

where

T_{Na} = sodium temperature (°R)

q = heat flux into sodium (Btu/hr-ft^2) from flux plot

U_o = overall tube conductance

As an approximation to avoid the necessity of balancing heat and mass flow along the tube length, the temperatures T_{wo} were evaluated at the inlet and outlet ends with only a linear interpolation between them. The value of T_{wo} vs. tube length was obtained in this manner and used to calculate the convective and radiation heat losses from each panel in turn using the following loss equations:

$$MW_C = h_c \; (t_{wo} - T_A) \; a \qquad (3.3\text{-}5)$$

$$MW_R = \sigma\varepsilon_w \; (T_{wo}^4 - T_A^4)a \qquad (3.3\text{-}6)$$

where:

h_c = air side convection coefficient (Btu/ft^2-hr-°F)

T_{wo} = outside tube wall temperature (°F)

t_{wo} = outside tube wall temperature (°R)

t_a = air temperature (°F) = 70

T_a = air temperature (°R)

ε_W = emissivity of Pyromark coating = 0.9

a = node area = $\Delta x \cdot \Delta y$

A major uncertainty in these calculations is associated with the estimate of the air side convective heat transfer coefficient. The uncertainty is due to the difficulty of accounting for the rough nature of the absorber surface and the combined effects of free and forced convection.

In the case of the cylindrical receiver, some limited data exists on the effect of roughness. The correlations used in these Task 2 calculations are based on the experiments by Achenbach (Ref. 3.3-2) with six-inch diameter roughened cylinders in cross flow.

Using the advanced central receiver specification for design wind speeds the velocity at the top of the 300 meter high tower was found to be 13.325 mph. This gives a Reynolds number of 6 x 10^6. The roughness of the receiver based on one-inch diameter tubes and 15 meter diameter is 0.00085. Achenbach's data indicates that heat transfer under these conditions is enhanced by about a factor of two over the smooth cylinder case.

Assuming a wind out of the north, the local convection coefficient, $N_u/\sqrt{R_e}$, varies around the circumference as given by Figure 2 in Auchenbach's paper. Local heat transfer values for each panel are listed in Table 3.3-2. These are the values of h_i used with Equations B-1 through B-6 of Appendix B to compute losses for the cylindrical receiver.

The north field absorber is a flat plate with the half tubes projecting from the surface. A brief review of the literature did not turn up any information on the effect of roughness on heat transfer from a flat plate. Therefore, to make the losses comparable for a valid parametric comparison, a smooth flat plate heat transfer coefficient was computed and then h_i was set equal to twice this value. The flat plate heat transfer correlation used was taken from the General Electric Heat Transfer Manual Section G503.5, p. 7, eq. 5-14.

$$h_x = \left[.0296 \left((c_p)_b \rho_b V \right) \left(\frac{c_\rho \mu}{K} \right)_b^{-0.4} \left(\frac{\rho_b V x}{\mu_b} \right)^{-0.20} \left(\frac{T_{wo}}{T_A} \right)^{-0.4} \right] F_r \qquad (3.3\text{-}7)$$

where: T_{wo} = outside tube temperature (°R)

T_A = air temperature (°R)

V = wind speed (ft/hr) 13.33 mph

ρ_b = 0.075 lb/ft^3

K = 0.014 $\frac{Btu}{ft\ hr\text{-}°F}$

Table 3.3-2

CYLINDRICAL RECEIVER AIR SIDE CONVECTION COEFFICIENTS
(Auchenbach)

Panel No.	$N_u/\sqrt{R_e}$	h_c (Btu/hr-ft^2-°F)
1	1.25	0.83
2	2.90	1.96
3	3.65	2.46
4	3.90	2.63
5	3.90	2.63
6	3.60	2.43
7	2.0	1.35
8	1.2	0.81
9	1.25	0.84
10	1.35	0.91
11	1.70	1.15
12	2.0	1.35

$$\mu = 0.44 \frac{lb}{ft\ hr}$$

Fr = roughness factor, based on cylinder correlation = 2

Table 3.3-3 lists 17 panel values of h_x for the 1100 F and 1300 F surface temperatures. In evaluating the panel losses in Appendix B, the average coefficient for the 1100 F case was used for both 1100 F and 1300 F.

Free convection on both flat and cylindrical absorber designs is vertical and parallel to the tubes (no roughness), whereas forced convection is across the tubes (with roughness). Combining the two effects is an area of heat transfer for which there is currently little guidance. Kreith (Ref. 3.3-3) notes that when the ratio of Grashoff's number to Reynold's number squared is the order of one or greater, then free convection effects are likely to be an important component in the total convection coefficient. For the design point conditions, this ratio is

$$\frac{Gr}{(Re)^2} \cong 5$$

for both receivers. Using the General Electric Heat Transfer Manual, the following values for natural convection were calculated for the flat plate receiver:

$$h_N = 1.43 \text{ Btu/hr/ft}^2\text{°F for 1100 F system}$$

$$h_N = 1.52 \text{ Btu/hr/ft}^2\text{°F for 1300 F system}$$

Thus free convection may be important in enhancing the convection losses from both the cylindrical and flat receivers. However, due to the uncertainty in roughness effects and the lack of a combining rule for free and forced convection, only the forced convection coefficients described above were used. Further work was done on this problem in Task 4 and is reported in Section 5.3.2.

Summary of Absorber Heat Losses

A summary tabulation of all the resulting convection and radiation heat losses for the four receiver designs is presented in Table 3.3-4. The details of these calculations are given in Appendix B.

Figure 3.3-7 shows a plot of temperature vs. length along the panel which was obtained using a more sophisticated computer program for the 1100 F system, north field on the hot panel. Also shown is the corresponding hand calculation using the approximate techniques outlined above. This shows a reasonable check between the outside wall temperature on the tube for both methods. The hand calculation can be said to give slightly conservative heat losses since, in general, it gives higher surface temperatures.

Mechanical Design of Absorber Panels

Maximum Allowable Solar Flux. In order to determine the maximum allowable peak flux on the receiver tubes, thermal/elastic stress analyses of the tubes were performed. The analytical model, the computer program used, parameters

Table 3.3-3

FLAT RECEIVER - AIR SIDE CONVECTION COEFFICIENTS

Panel No.	x ft	h_x(1100 F) Btu/hr-ft^2-°F	h_x(1300 F) Btu/hr-ft^2-°F
1	1.64	4.74	4.52
2	4.92	3.80	3.62
3	8.20	3.43	3.27
4	11.48	3.21	3.06
5	14.76	3.05	2.91
6	18.05	2.93	2.80
7	21.33	2.84	2.70
8	24.61	2.76	2.63
9	27.89	2.69	2.56
10	31.17	2.63	2.51
11	34.45	2.58	2.46
12	37.73	2.53	2.41
13	41.01	2.49	2.37
14	44.29	2.45	2.34
15	47.57	2.42	2.30
16	50.86	2.38	2.27
17	54.14	2.35	2.24
	(h_c)	AVE = 2.90*	AVE = 2.76

*Value used for both 1100 F and 1300 F

Table 3.3-4

RECEIVER LOSS SUMMARY

Field Type:	North	North	360^o	360^o
Maximum Sodium Temperature (°F):	1100	1300	1100	1300
Incident Power (MW_t)	408.29	410.77	438.88	444.21
Reflection Loss* (MW_t)	-20.42	-20.54	-21.94	-22.21
Convection Loss (MW_t)	-2.74	-2.96	-3.37	-3.66
Radiation Loss* (MW_t)	-8.68	-10.81	-15.80	-20.57
Power to Sodium (MW_t)	376.46	376.46	397.77	397.77
Receiver Efficiency (%)	92.2	91.6	90.6	89.5

*assumes Pyromark coating with $\alpha = 0.95$, $\varepsilon = 0.9$

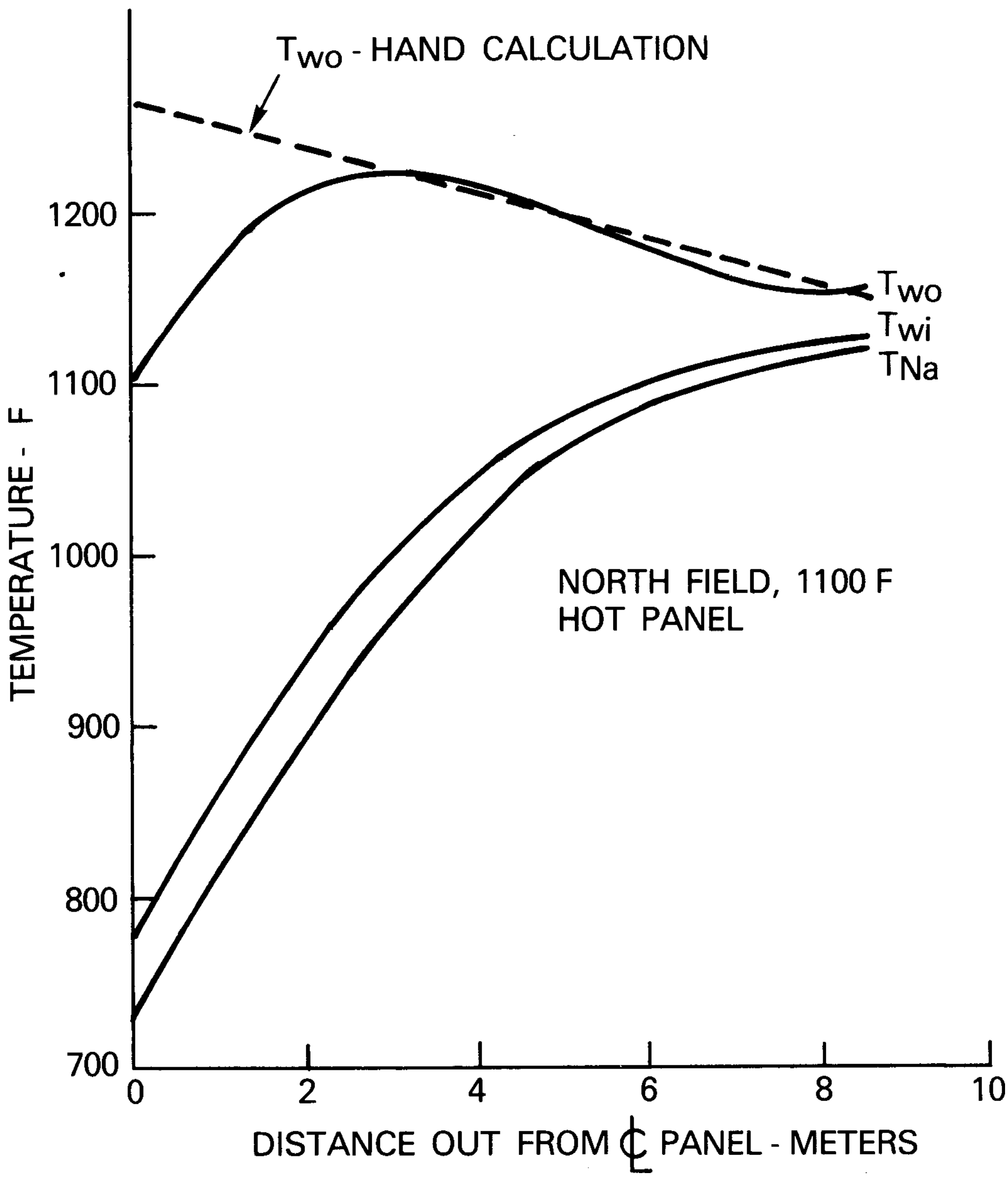


Figure 3.3-7. Accuracy Check for Approximate Panel Loss Calculation Method

considered and the criteria used in evaluating the allowable flux are described below:

Analytical Model. The receiver tube is assumed to be a long thick cylindrical shell. The ends of the tube are bent and welded to the headers. This support condition simulates that of an elastic spring of given rotational and elongational stiffness. However, in this preliminary analysis the tube is assumed to be free to expand axially, but is not allowed to rotate at the ends. This corresponds to zero elongational stiffness and infinite rotational stiffness. The heat flux is assumed to have a cosine variation, i.e., $q = q_0 \cos \theta$, on the sun side of the tube and the rear side is assumed to be insulated. Axial variation of the flux is ignored, i.e., the tube is assumed to be subjected to the peak flux over the length of the tube. Thus the problem is reduced to that of generalized plane strain. The above assumptions, in general, are conservative, but are suitable for a preliminary study. A more accurate analysis, possibly including an elastic-plastic-creep analysis, considering the actual elastic support conditions should be done as part of the commercial plant preliminary design.

Computer Program. A specialized in-house computer program called NONSYM (Refs. 3.3-4, 3.3-5) developed by Foster Wheeler Development Corporation (FWDC) was used in this analysis. At present this program has the capability to do thermal and stress analyses of tubes subjected to non-axisymmetric radiant heating. The tube material can be elastic, or elastic-plastic undergoing creep. Elastic spring supports are permissible. This program has been verified extensively with exact solutions as well as with finite element solutions. It has been shown that this program is very accurate and much less expensive than the general purpose finite element programs.

Evaluation Criteria. Since the sodium pressure inside the receiver tube is low (150 psi) the hoop stresses are very low and ductile rupture is not a predominant failure mode. Hence, the minimum tube thickness is determined from welding considerations rather than pressure stresses. The solar receiver is subjected to startup and shutdown cycles due to diurnal solar flux variations and cloud covers. Hence, fatigue and creep-fatigue interactions are important failure modes. One set of possible criteria for the creep-failure evaluation of the tubes is that of the ASME Code Case 1592 (Elevated Temperature Design of Class 1 Nuclear Components). These criteria are consistent with the reliability and integrity required in nuclear components. Consequently, these criteria may be too conservative for solar applications and may result in severe economic penalties. Hence, the following criteria which are a combination of Section I, Section VIII Division 2, and Code Case 1592, are used in this study (Ref. 3.2-6).

- Limit the primary stress P_m to the allowable stress S given in Section I (S is identical to S_0 in Code Case 1592).
- Primary and secondary stress ($P_L + P_B + Q \leq 3\ S_m$) is an extension of the shakedown limit given in Section VIII Division 2 to the elevated temperature region.
- For fatigue criteria, use the inelastic fatigue curves of Code Case 1592. The elastic fatigue curves are very conservative. The creep damage is ignored. This ignoring of creep damage is justified at least in regions where the stresses are predominantly compressive. In solar tubes, the stress state in those regions where the temperature is the highest is compressive.

In all cases dealt with in this study, the third criteria is the governing one. (It is assumed that the receiver will be subjected to 13,000 diurnal and cloud cover cycles during the 30-year lifetime.) For 316SS the fatigue curves of Code Case 1592 are used. These curves have a factor of safety of 20 on the cycles or 2 on the strain range, whichever is more conservative. For Inconel-625 no fatigue curves are given in Code Case 1592. Hence, the manufacturers fatigue curve (Ref. 3.2-7) with a factor of safety of 20 on the cycle is used in this study. This is consistent with the ASME Code approach.

Allowable Flux Values. The various parameters involved in this study and the allowable peak flux values are given in Tables 3.3-5 and 3.3-6. For the Inconel tubes in the 360° field design, the allowable fluxes are higher than those shown in Table 3.3-6, because the reduced tube diameter and increased film coefficient will result in a decrease in thermal stresses.

Receiver Assembly Weight Estimates

Table 3.3-7 lists the weight breakdown for the 360° field receivers operating at 1100 F and 1300 F. The structural steel weight calculation is detailed in Figure 3.3-8 and Table 3.3-8. The piping weights were obtained using the data generated on pipe sizes and lengths for the pressure loss calculations. The EM pump weights were obtained from Figure 3.3-9. The absorber panel weights were estimated from the Foster Wheeler panel design detail. Convection curtains and movable insulation were roughly sized and weights were obtained.

Table 3.3-9 lists the weight breakdown for the north field receivers operating at 1100 F and 1300 F. The structural steel weight calculation is detailed in Figure 3.3-10 and Table 3.3-10. Other than this, the weight calculation followed the same approach as that used to estimate weights for the receivers for the 360° field.

The pound/foot numbers for an increase in height or width on both designs were used to permit parametric weight scaling.

Manufacturing Plan for Absorber Panels

Table 3.3-11 summarizes the design data used in establishing a manufacturing cost estimate for the absorber panels. Sketches of the panels (Figures 3.3-11 and 3.3-12) were also developed to aid in making this estimate. The absorber manufacturing plan proposed by Foster Wheeler on the basis of this information is shown in Figure 3.3-13. The major steps of the manufacturing precess are:

1. Fabricate attachments -- lugs and plate
2. Wash and machine end, and make one automatic inert gas (AIG) weld
3. Bend tubes
4. Set up and brace panels
5. Make longitudinal tube to tube welds
6. Set up and weld attachments -- lugs and plate
7. Fabricate inlet header -- machine, roll, weld, drill openings

Table 3.3-5

316SS TUBES

Sodium Temperature - Inlet = 725 F

Sodium Temperature - Outlet = 1100 F

Design Pressure = 150 psi

Allowable Strain Range at 1100 F = 0.21×10^{-3} mm/mm

	North Field Tube ID = 0.465 in.			360° Field Tube ID = 0.265 in.	
Tube Thickness (in.)	0.035	0.050	0.080	0.05	0.08
Sodium Side Coefficient ($Btu/hr-ft^2-F$)	8000	8000	8000	10,000	10,000
Maximum Allowable Flux (MW/m^2)	2.575	1.953	1.312	2.01	1.33

Table 3.3-6
INCONEL 625 TUBES

Sodium Temperature - Inlet	= 680 F
Sodium Temperature - Outlet	= 1300 F
Design Pressure	= 150 psi
Allowable Stress Range (S_{alt}) for 13,000 Cycles	= 52 ksi
Tube ID	= 0.351

Tube Thickness (in.)	0.035	0.05	0.08
Sodium Side Coefficient (Btu/hr, ft^2, F)	9000	9000	9000
Maximum Allowable Flux (MW/m^2)	4.86	3.623	2.393

Table 3.3-7

SUMMARY OF WEIGHTS - SOLAR RECEIVER ASSEMBLY

360^o Field - 1100 F	Metal	Sodium	Insulation	lb/Diam	lb/Height
Structural Steel	356,558	-	-	2,855	2,745
Piping	84,424	114,378	66,406	3,200	328
EM Pumps	75,000	4,000	3,000	-	-
Moveable Insulation	25,000	-	83,000	2,117	2,077
Convection Curtains	20,855	-	-	469	-
Absorber Panels	334,000	82,000	128,000	10,750	8,026
	895,837	200,378	280,406	19,391	13,176

Weight Σ = 1,376,621 lb

360^o Field - 1300 F	Metal	Sodium	Insulation	lb/Diam	lb/Height
Structural Steel	356,558	-	-	2,855	2,745
Piping	75,239	87,919	57,745	2,848	219
EM Pumps	50,830	2,700	2,000	-	-
Moveable Insulation	25,000	-	83,000	2,117	2,077
Convection Curtains	20,855	-	-	469	-
Absorber Panels	321,000	63,000	128,000	10,120	7,917
	849,482	153,619	270,745	18,409	12,958

Weight Σ = 1,273,846 lb

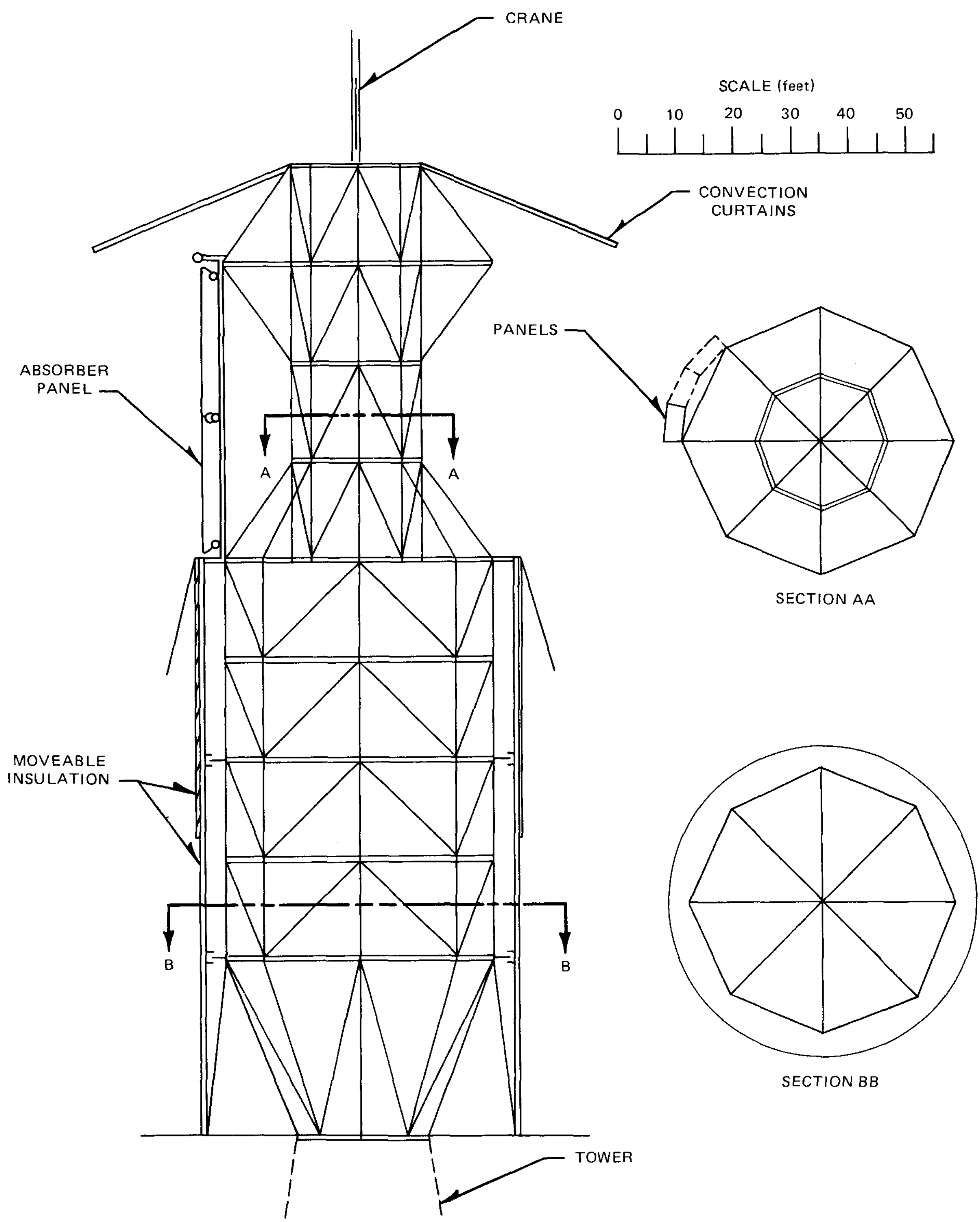


Figure 3.3-8. Structural Steel for 360° Field

Table 3.3-8

STRUCTURAL STEEL WEIGHT ESTIMATE FOR 360° FIELD

Member	Size	Length	lb/ft	lb
1	18 x 11-3/4	432	105	45,360
2	ST 61	360	25	9,000
3	12 x 3 Channel	270	30	7,920
4	8 x 2-1/4 Channel	800	18.75	15,000
5	18 x 11-3/4	512	105	53,750
6	18 x 8-3/4	560	85	47,600
7	8 x 6-1/2	880	28	24,640
8	12 x 8	720	50	36,000
9	12 x 6-1/2	920	31	28,520
10	8 x 6-1/2	288	24	6,912
11	8 x 6-1/2	480	24	11,520
12	12 x 3 channel	653	25	16,336
13	12 x 3 channel	800	25	20,000
Ladders		200	20	4,000
Platforms				30,000
				356,558

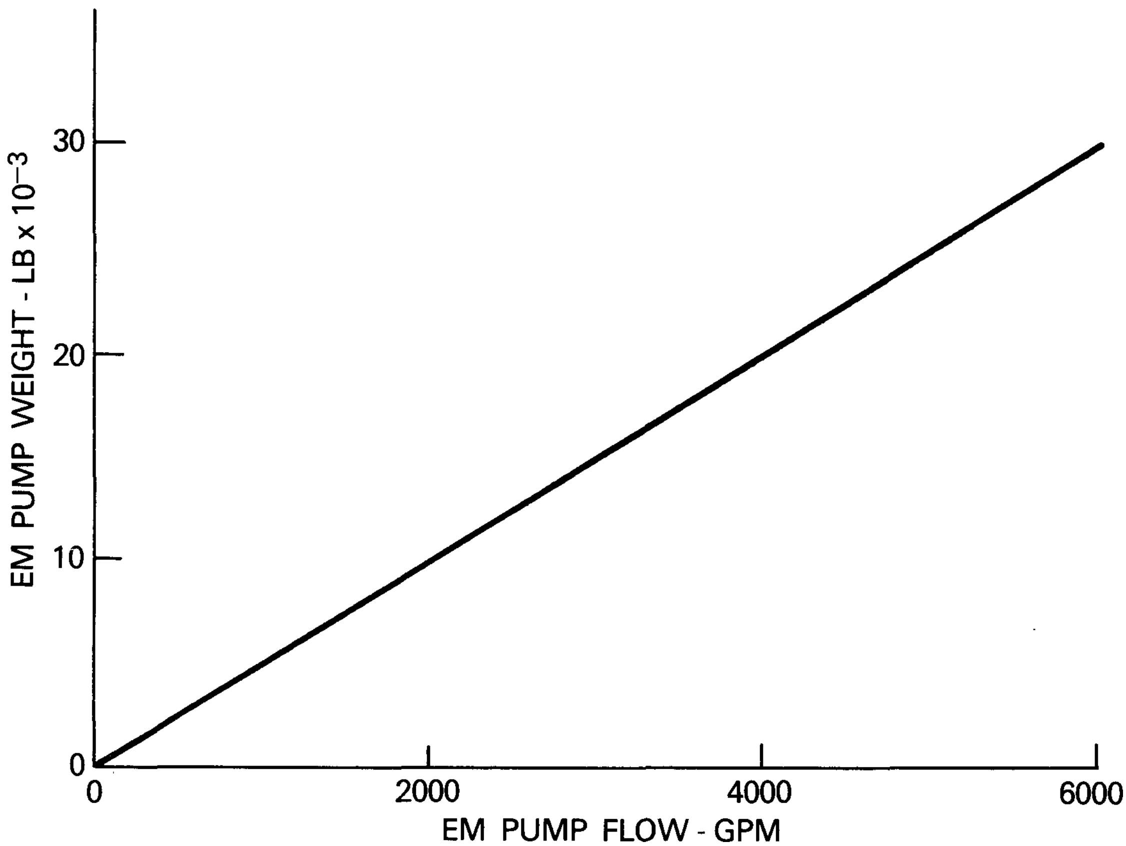


Figure 3.3-9. Small EM Pump Weights

Table 3.3-9

SUMMARY OF WEIGHTS - SOLAR RECEIVER ASSEMBLY

North Field - 1100 F	Metal	Sodium	Insulation	lb/ft lb/Width	lb/ft lb/Height
Structural Steel	258,560	-	-	2,628	705
Piping	139,710	135,652	95,603	671	3,030
EM Pumps	75,000	4,000	3,000	-	-
Moveable Insulation	10,000	-	36,000	767	767
Convection Curtains	18,000	-	-	1,000	2,000
Absorber Panels	120,161	29,566	46,000	1,697	2,887
Totals	621,431	169,218	180,603	6,763	9,389
	Weight $\Sigma = 9.71 \times 10^5$ lb				
North Field - 1300 F					
Structural Steel	258,560	-	-	2,628	705
Piping	109,681	83,714	88,281	409	2,650
EM Pumps	50,830	2,700	2,000	-	-
Moveable Insulation	10,000	-	36,000	767	767
Convection Curtains	18,000	-	-	1,000	2,000
Absorber Panels	115,300	22,940	46,000	1,506	2,848
Totals	562,400	109,354	172,281	6,310	8,970
	Weight $\Sigma = 8.44 \times 10^5$ lb				

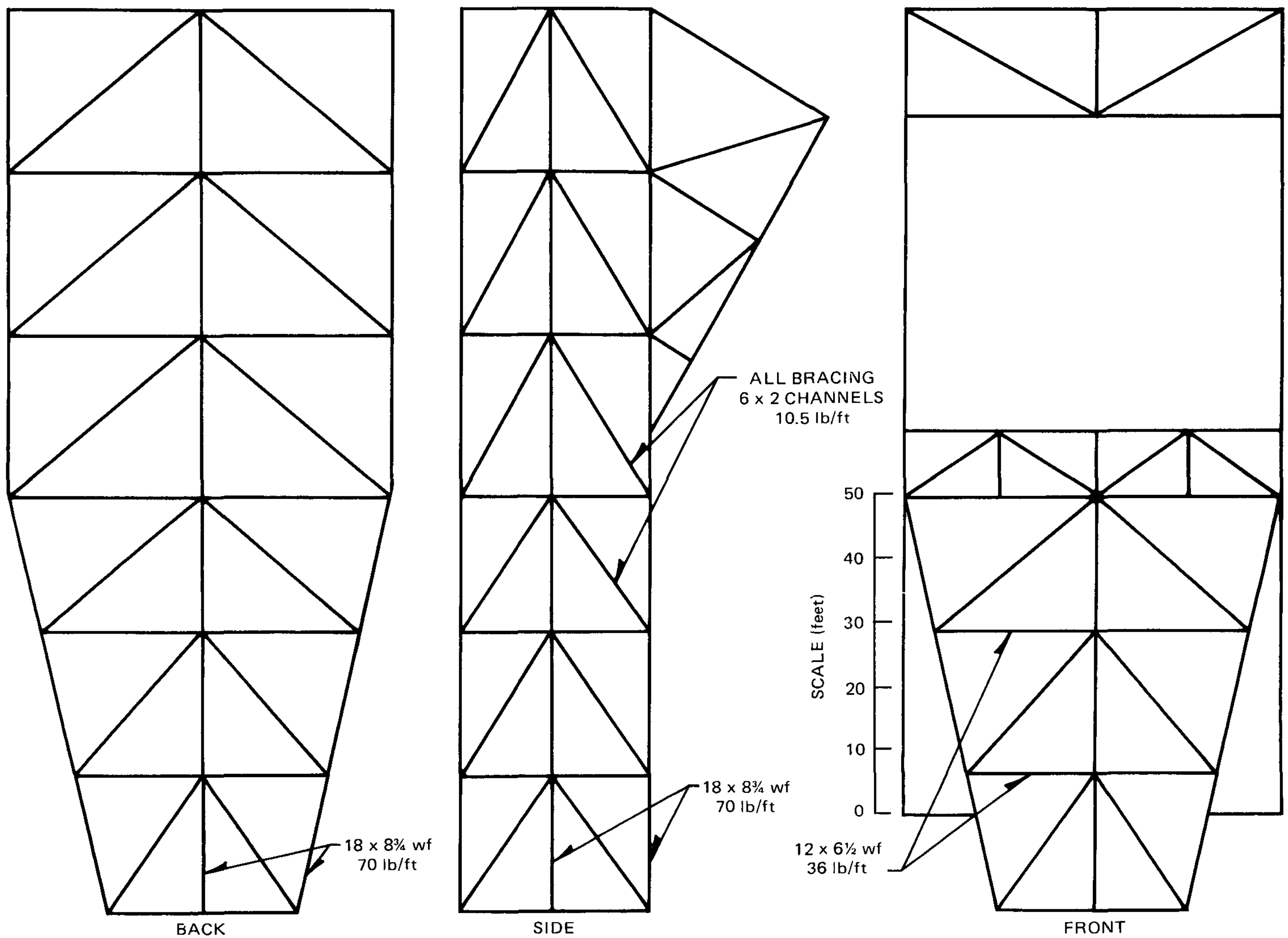


Figure 3.3-10. Structural Steel for North Field Receiver

Table 3.3-10

WEIGHT BREAKDOWN FOR STRUCTURAL MEMBERS
IN NORTH FIELD RECEIVERS

Vertical Members

4 Corners - 18 x 8-3/4 I-Beams 70 lb/ft		
560 x 70 ft	=	39,200
4 Intermediate - 18 x 8-3/4 I-Beams 70 lb/ft		
560 x 70 ft	=	39,200
Platforms - 7 Required		
12 x 6-1/2 I-Beams 36 lb/ft		
7 x (2 x 60' + 2 x 30') x 36 lb/ft Frame	=	45,360
7 x (3 x 60' + 3 x 30') x 36 lb/ft Joists	=	68,040
Platform and Supports - Guess	=	30,000
		143,400
Bracing - ∿34 Required - 40 ft Long		
6 x 2 Channel 10.5 lb/ft		
34 x 40 x 10.5 lb/ft	=	14,280
Ladders - 300 ft @ 20 lb/ft	=	6,000
Moveable Insulation Guide		
180 Ft 12 x 3 Channel 25 lb/ft	=	6,480
Pump Supports - Guess	=	10,000
		258,560 lb
Increase Height - 1 ft		
(39,200 + 39,200 + 14,280 + 6,000) 1 ft/140 ft = 705 lb	=	705 lb
Increase Width - 1 ft		
(143,400 + 14,280) 1 ft/60 ft = 2628 lb	=	2,628 lb

Table 3.3-11

ABSORBER PANELS - ADVANCED CENTRAL RECEIVER
PRELIMINARY DESIGN DATA FOR USE IN TASK 2

	North Field		360° Field	
	1100 F	1300 F	1100 F	1300 F
Tube Outside Diameter (in.)	1.0095	0.772	.5993	.4685
Tube Wall Thickness (in.)	0.035	0.035	.035	.035
Active Tube Length (ft)	55.8	55.8	49.22	49.22
Total Tube Length (ft)	70	70	64	64
Number of Tubes/Panels	39	51	129	165
Panel Width (ft)	3.281	3.281	6.442	6.442
Panel Length (ft)	57.8	57.8	51.22	51.22
Number of Panels	17	17	24	24
Inlet Header - OD (in.)	16	12	16	12
Inlet Header - Wall Thickness (in.)	0.312	0.250	0.312	0.250
Outlet Headers - OD (in.)	12	10	12	10
Outlet Headers - Wall Thickness (in.)	0.250	0.219	0.250	0.219
Sodium Temperature - Panel Inlet (F)	725	680	725	680
Sodium Temperature - Panel Outlet* (F)	1100	1300	1100	1300
Peak Heat Flux - (MW/m^2)	3.84	3.84	1.602	1.602
Average Heat Flux - (MW/m^2)	1.298	1.298	0.530	0.530
Panel Pressure Loss (psi)	10	10	10	10
Strongback - Channel Size	12x3 30 lb/ft	12x3 30 lb/ft	12x3 30 lb/ft	12x3 30 lb/ft
Design Pressure - psi	150	150	150	150
Flow in Peak Flux Panel (lb/hr)	1.047×10^6	$.635 \times 10^6$	0.783×10^6	0.474×10^6
Sodium Side Coefficient** Btu/hr-ft^2-F	8229	8894	10233	11757
Wall OD Coefficient Btu/hr-ft^2-F	4168	4579	4062	4432
Tube Temperature (OD) at Peak Flux	1176	1096	906	845
Tube Temperature (ID) at Peak Flux	884	831	781	731

* Adjust sodium flow to each panel to meet outlet temperature specified.
**Based on panel flow in peak heat flux panel (Referenced to tube ID).

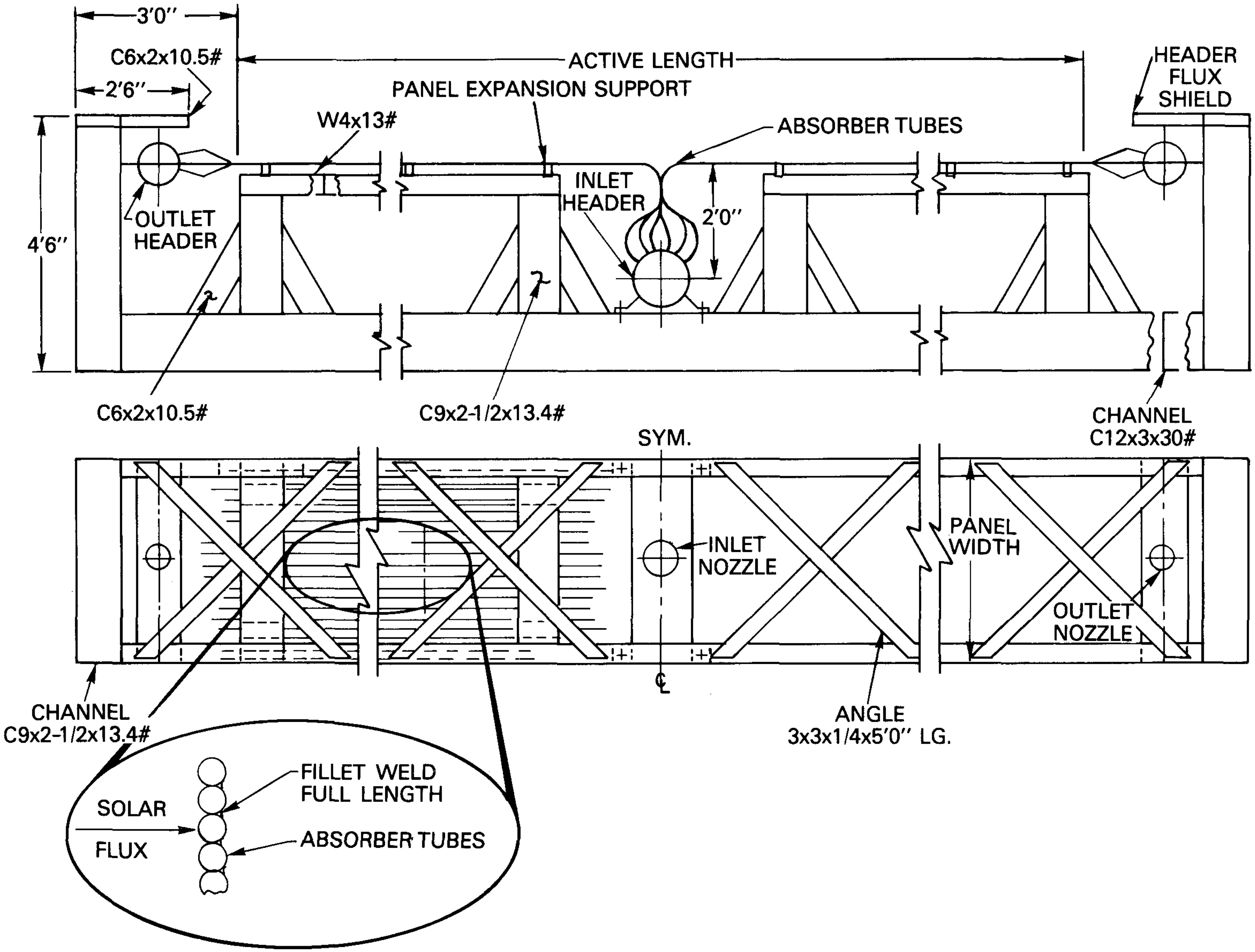
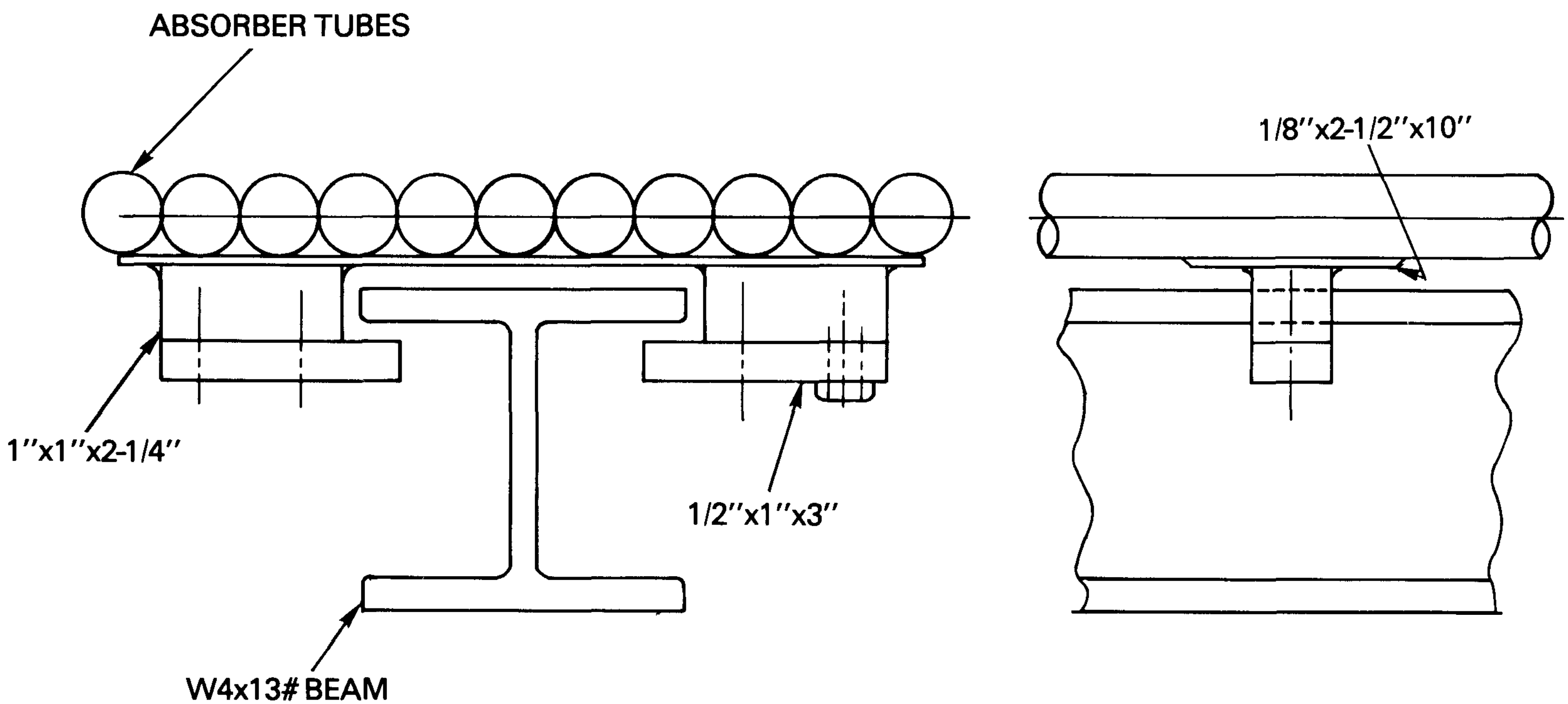


Figure 3.3-11. Absorber Panel - Advanced Central Receiver



NOTE: FOR ESTIMATING PURPOSES ONLY

Figure 3.3-12. Absorber Panel - Expansion Support

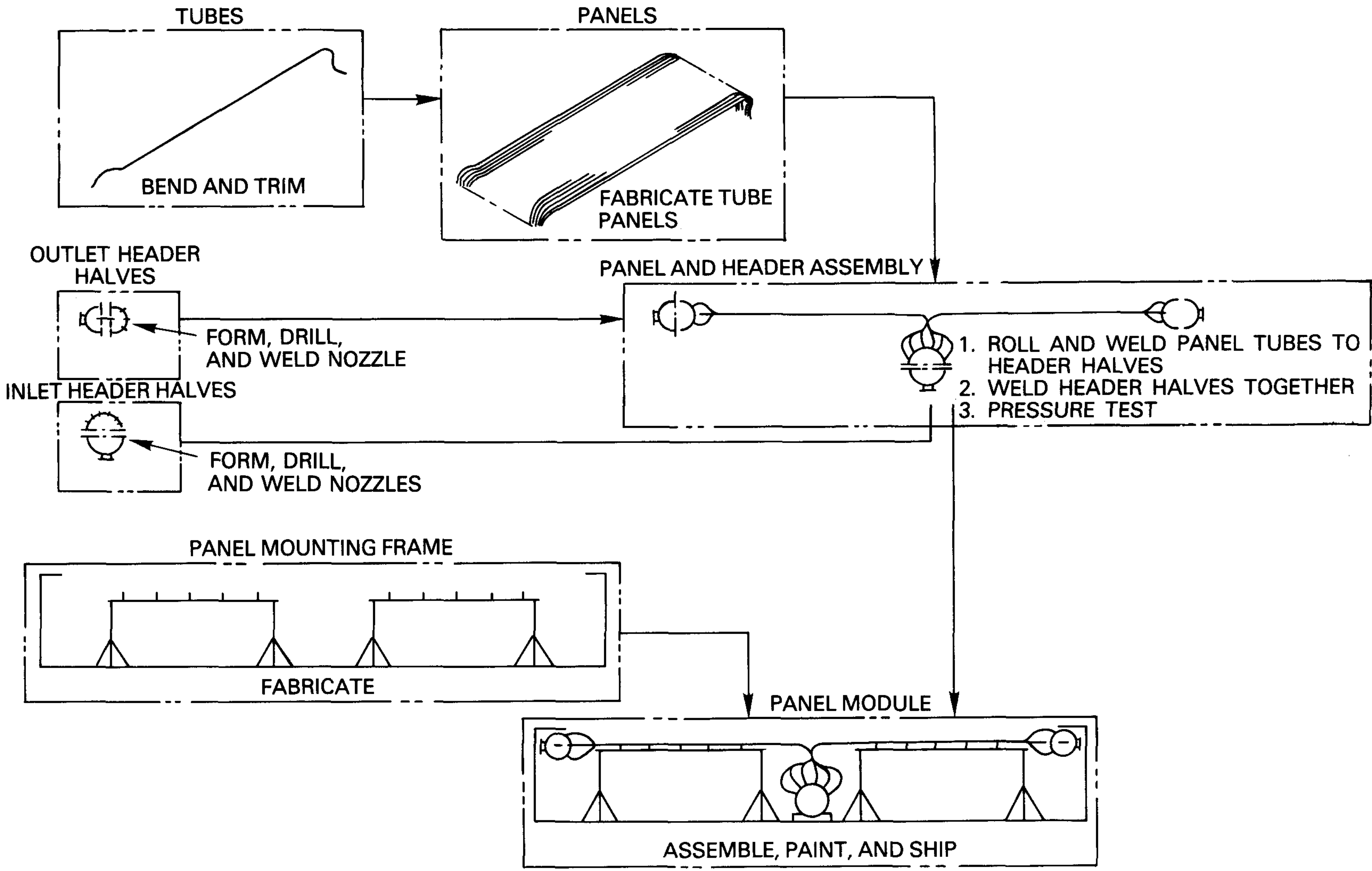


Figure 3.3-13. Basic Fabrication Sequence - Advanced Central Solar Receiver Absorber Panel Module

8. Fabricate outlet headers -- machine, roll, weld, drill openings
9. Insert tubes in headers
10. Weld tubes to headers
11. Stress relieve
12. Fabricate shipping frame
13. Apply paint
14. Prepare for shipment
15. Load on flatbed truck

Receiver Pumping Requirement

Small EM Pumps. Pumping power requirements were estimated for all combinations of receiver type, temperature level, and tower height considered in the parametric analysis. These calculations are described in Appendix B and the results are summarized in Table 3.3-12.

The results in Table 3.3-12 are for the baseline system including IHX. Pumping requirements for the throttle valve and two pressure storage cases were estimated by scaling these results.

Large EM Pumps. A rough estimate of the cost and power requirement of the EM pumps in the storage subsystem was made. The standard line sizes were all selected for a maximum flow velocity of 25 feet/second. An estimate was then made of the line lengths, elbows, tees, valves, expansions, contractions, etc., and pressure loss calculations were made from the total equivalent pipe runs. Static head effects were ignored as second order effects at full system flow. The line flow rates were determined from the system heat balance. The resulting parametric design data for all of the EM pumps are contained in Table 3.3-13 for storage concepts No. 1 through No. 4 for 1100 F and 1300 F respectively. All of the remaining parametric storage system pumps were obtained from extrapolations of this basic data. The new head requirements were obtained by applying the ratio of flows to the 1.8 power, $(W/W_{REF})^{1.8}$, and line lengths, to the first power (L/L_{REF}).

Table 3.3-12

RECEIVER PUMPING REQUIREMENTS
SMALL EM PUMPS IN ABSORBER

	Tower Height m	Power Required MW_e	Number of Pumps*	Capital Cost of Pumps M$
1100 F North Field	150	1.972	17	7.10
	225	2.219	17	7.64
	300	2.468	17	8.03
1300 F North Field	150	1.619	17	6.29
	225	1.854	17	6.67
	300	2.081	17	7.12
1100 F 360^{o} Field	150	1.962	24	8.47
	225	2.208	24	8.79
	300	2.456	24	9.21
1300 F 360^{o} Field	150	1.616	24	7.59
	225	1.842	24	7.93
	300	2.068	24	8.42

Note: These numbers correspond to the baseline system concept which includes an intermediate heat exchanger (IHX). All pumping is provided by small EM pumps located in receiver.

*One pump per panel.

Table 3.3-13

SUMMARY OF PARAMETRIC BASE CASES FOR LARGE EM PUMP DATA

No.	Storage Concept	Parametric System	Flow lb/hr	Flow gpm	Head psid	Head ft	Pump Eff - %	Electric Power-MW_e
1	Steel Balls-Loop Pump	1100 F	9.909×10^6	23,007	75	201.1	50	1.501
1	Steel Balls-Stor. Pump	1100 F	9.909×10^6	23,007	75	201.1	50	1.501
1	Steel Balls-Loop Pump	1300 F	6.235×10^6	14,386	75	199.9	50	0.939
1	Steel Balls-Stor. Pump	1300 F	6.235×10^6	14,386	75	199.9	50	0.939
2	Switch Tanks-Loop Pump	1100 F	9.909×10^6	23,007	125	282.6	50	2.110
2	Switch Tanks-Stor. Pump	1100 F	6.606×10^6	16,150	75	211.8	50	1.054
2	Switch Tanks-Loop Pump	1300 F	6.235×10^6	14,386	125	335.2	50	1.574
2	Switch Tanks-Stor. Pump	1300 F	4.157×10^6	10,620	75	221.3	50	0.693
3	Hot/Cold Tanks-IHX Pump	1100 F	9.909×10^6	23,007	50	134.1	50	1.001
3	Hot/Cold Tanks-S.G. Pump	1100 F	6.606×10^6	16,150	75	211.8	50	1.054
3	Hot/Cold Tanks-IHX Pump	1300 F	6.235×10^6	14,386	50	133.2	50	0.626
3	Hot/Cold Tanks-S.G. Pump	1300 F	4.157×10^6	10,620	75	221.3	50	0.629
4	IHX Pump	1100 F	9.909×10^6	23,007	50	134.1	50	1.001
4	S.G. Pump	1100 F	6.606×10^6	16,150	75	211.8	50	1.054
4	Storage Pump	1100 F	9.909×10^6	23,007	50	134.1	50	1.001
4	IHX Pump	1300 F	6.235×10^6	14.386	50	133.2	50	0.626
4	S.G. Pump	1300 F	4.157×10^6	10,620	75	221.3	50	0.629
4	Storage Pump	1300 F	6.235×10^6	14,386	50	133.2	50	0.626

3.3.3 TOWER AND RISER - DOWNCOMER CONCEPTS

Receiver Tower

Guyed vs. Non-Guyed Construction. The construction technique selected for the receiver tower is slip formed concrete, free standing. Guyed construction is not recommended for either the steel or contrete tower, although this system would probably require less construction material. Some disadvantages of a guyed system are

- its indeterminate interaction with the relatively stiff tower structure under the combined loadings of wind, heat, and icing of the cables
- potential interference of the numerous cables and their anchorages with the heliostat array and operations
- the possibility that failure of one cable could cause the collapse of the entire tower (an example is the Emley Moor Tower in Yorkshire, U.K., in March 1969)

Steel vs. Concrete Construction. In comparing the steel frame tower contruction costs of those of the slip formed reinforced concrete tower, it is considered that the latter is probably less costly. Slip formed concrete has the advantages of shorter design and construction time. Additionally, maintenance costs for the concrete tower are lower. Accordingly we have chosen the slip formed reinforced concrete tower concept. It should be noted that some dozen or more existing towers between 220 and 550 meters have been constructed in free standing concrete in the U.S. and overseas.

Tower Criteria. The following criteria were used in designing the towers for the parametric analysis.

Tower heights: 150,225,300 meters

Receiver size: 17m diameter x 17m tall cylinder

Receiver weight: 1000 kips

Central shaft, piping: 300 kips

Maximum allowable receiver displacement due to wind = 2 meters

Wind speed: 90 mph (30 ft above grade), Exposure C, ANSI - A58.1, speed increases as 0.15 power of height

Seismic loads: Per Uniform Building Code, Zone 3 with

Z = 0.75	C* = 0.046 (300M)
I = 1.0	0.047 (225M)
K = 2.0	0.066 (150M)
S = 1.0	

Allowable soil pressure: 10 ksf

*Reference 3.3-8.

Tower Design. A preliminary design of the receiver tower (Figures 3.3-14 through 3.3-16) has been prepared. These diagrams show the tower as a slip formed reinforced concrete tower with central elevator core and maintenance platforms at different levels. An access ladder extends the length of the elevator core.

The tower was designed to accept the total weights for the 360° field (1100 F and 1300 F) case which has greater weights than the north field case (1100 F and 1300 F). The tower design is not sensitive to the difference in weights between the 360° field case and the north field case (about 400 kips).

The procedure used to design the towers was as follows:

a) Set diameter of shell at 52'-6" at the point of connection to the receiver.

b) Select base shell diameter, shell thickness and foundation ring size by rough scaling of the new Emly Moor Tower.*

c) Compute the wind and seismic loads

d) Check for failure of the shell at grade.†

e) Check for overturning due to excessive force on the underside of the foundation.†

f) check for excessive deflection of receiver due to wind loading.

In all three cases (150, 225, and 300 meters), the initial selection of shell diameters and thickness was found to be slightly conservative for this application. However, the initial selection was judged to be near the optimum and was adopted without change.

Tower Cost Estimate. The design and cost estimate was done for three different heights. Results of the cost estimate are tabulated in Appendix D and in Table 3.3-14. Tower cost includes foundations (forms, rebar, concrete, embedments), tower (slip formed concrete), elevated slabs, steel (core, maintenance platforms), and service elevator. Costs are expressed in mid-1978 dollars.

Downcomer/Riser Design

Material Specification

1100 F System

Pipe diameter	24 inches (downcomer and riser)
Downcomer material	316SS
Riser material	2-1/4 CR - 1 Mo

1300 F System

Pipe diameter	18 inches (downcomer and riser)
Downcomer material	Inconel 625
Riser Material	2-1/4 CR - 1 Mo

*Original guyed steel structure was replaced by a slip formed concrete tower.

†Based on combinations of live and dead loads specified in ANSI A58.1

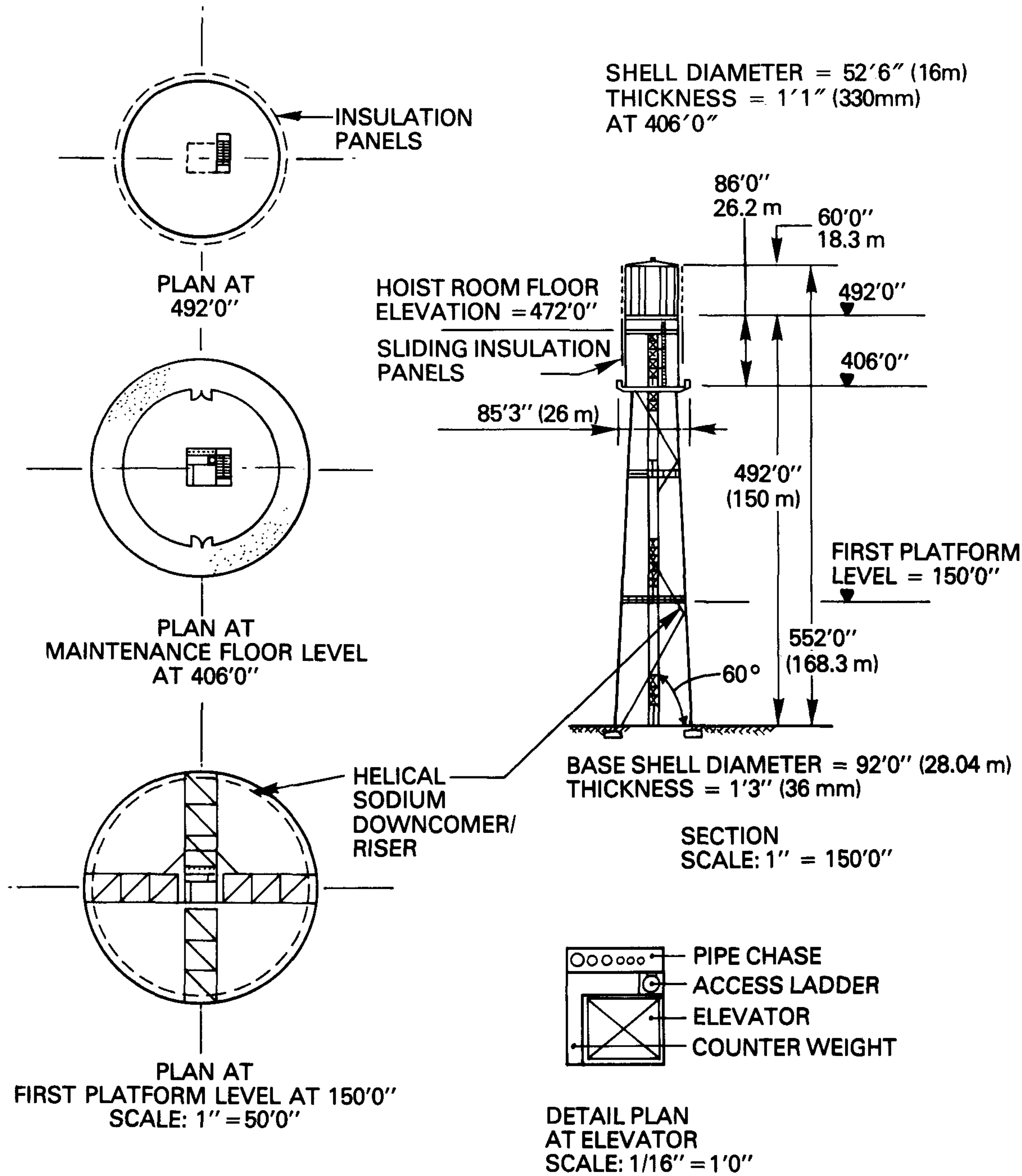


Figure 3.3-14. Tower - 150 Meters

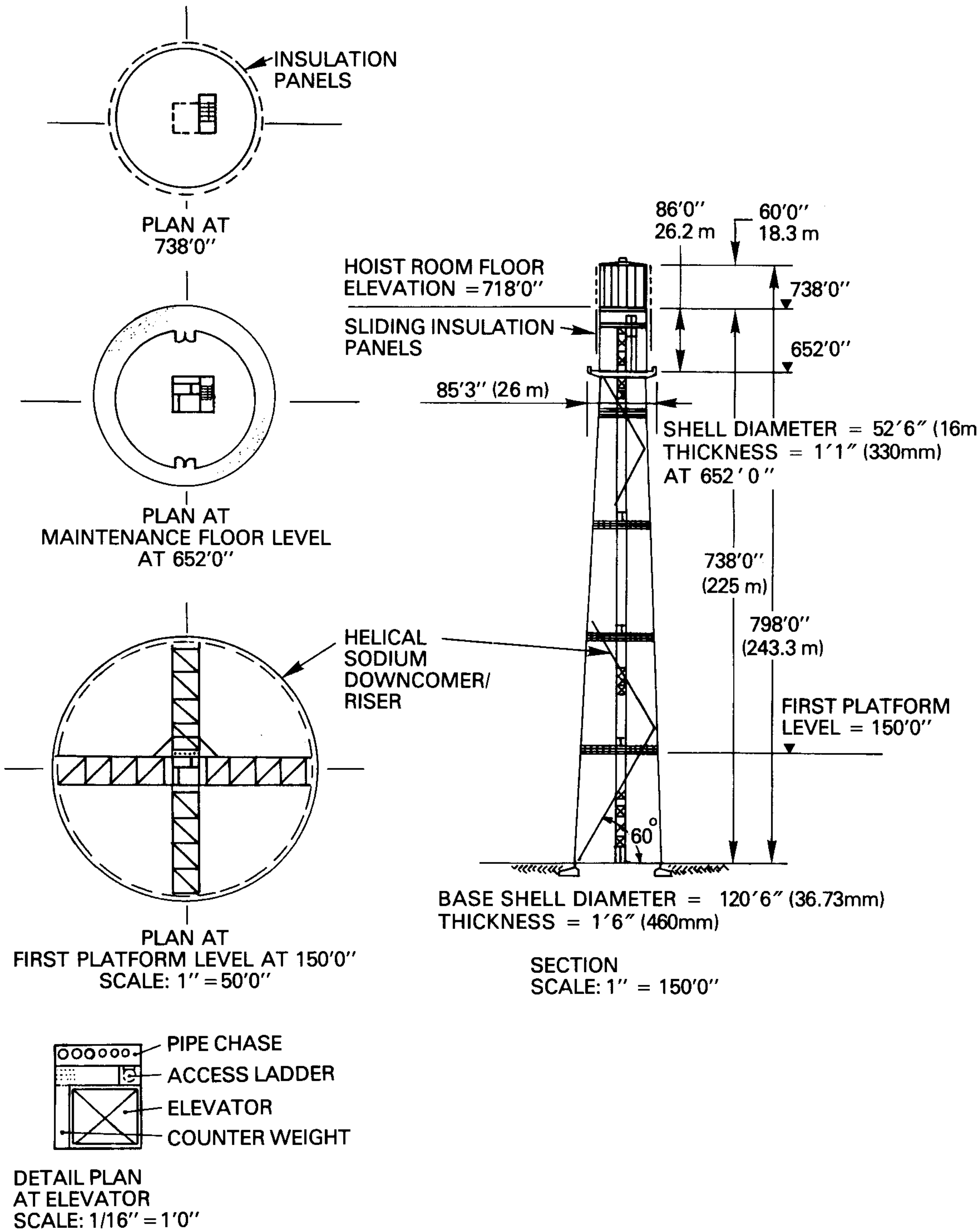


Figure 3.3-15. Tower - 225 Meters

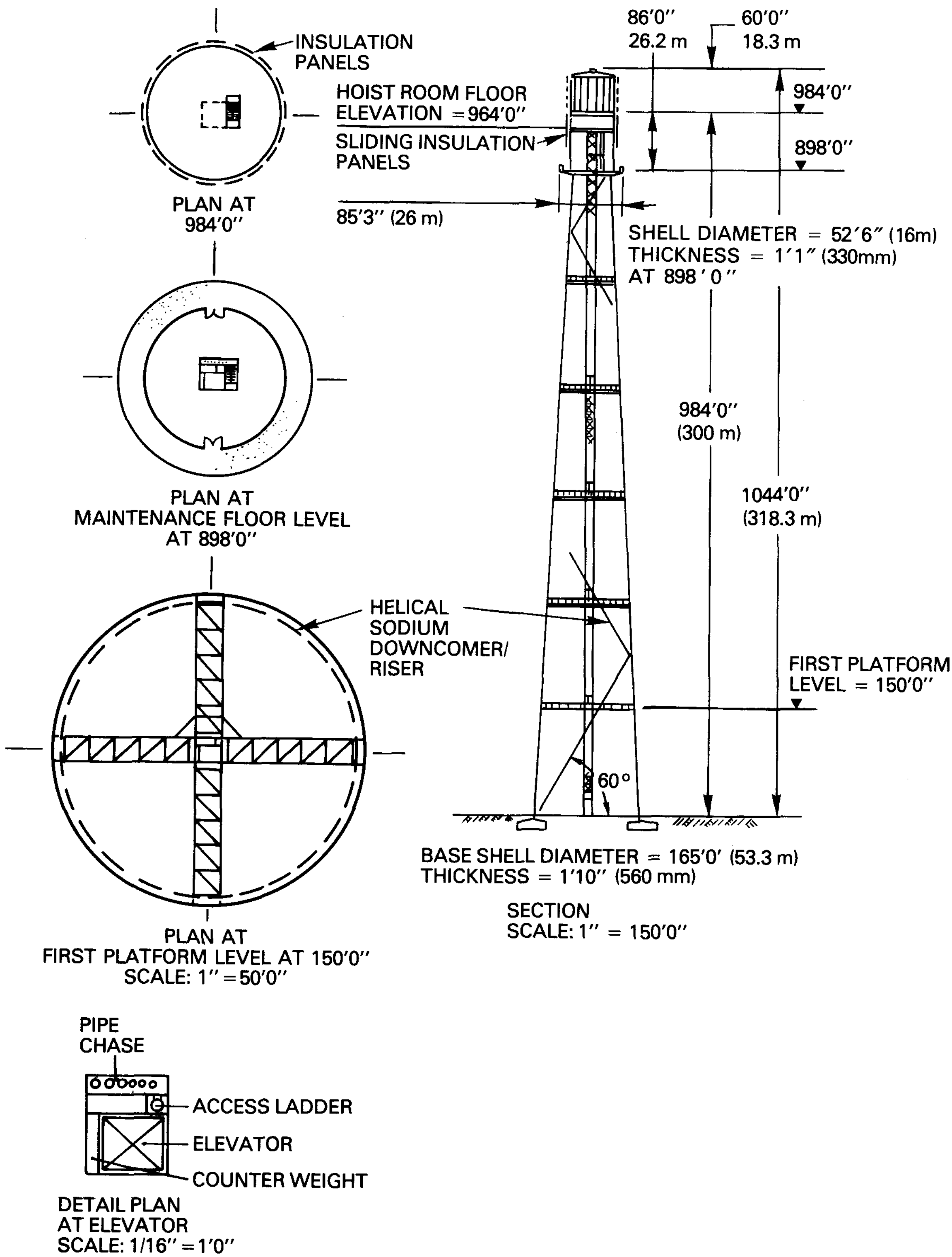


Figure 3.3-16. Tower - 300 Meters

Bellows vs. Helical Configuration. The use of bellows to accommodate thermal expansion in the downcomer and riser pipes was considered. Thermal expansion of 316 stainless steel from 70 F to 1300 F is approximately 15-1/2 inches per 100 feet. If standard bellows with axial expansion of 3 inches are adopted, approximately 100 bellows would be required for the 984 foot tower case for both downcomer and riser pipes. Cost for each bellows (nonflanged type) is assumed to be $4000. The total materials cost for bellows (including installation costs) would therefore be approximately $400,000. If flanged bellows are used, costs would be much higher.

A design for a helical downcomer/riser configuration was developed. An angle of 60° was chosen and allowable stresses checked. For this design approximately 15 percent extra material is required. For the 984 foot tower case, the cost of this extra material is approximately $100,000. Fabrication of the helix could be done by bending straight sections of pipe and the cost of pipe bending is included in the pipe cost estimate. The cost of welding of the incremental material would be approximately $15,000. The helical configuration is thus more economical than bellows for the assumptions made in this analysis.

It is considered, further, that bellows may present a problem with reliability. Since bellows failure could lead to a plant shutdown, the use of bellows is rejected. The helical configuration is recommended over use of expansion loops to minimize pressure drops which could lead to a lowering of plant efficiency.

Cost Estimate of Helical Configurations. Appendix E shows the downcomer riser cost estimate with the helical configuration. The cost estimate includes material, labor, hangers, insulation (Thermal 12 with aluminum jacket). Table 3.3-15 summarizes total downcomer/riser cost as a function of tower height.

3.3.4 RECEIVER COST ESTIMATES

Receiver subsystem cost estimates, using 1978 dollars, were developed for the parametric analysis (Task 2) for north field and 360° field collector configurations, maximum system temperatures of both 1300 F and 1100 F, and tower heights of 150, 225, and 300 meters. These estimates include costs for the following items:

Major Components

Absorber Panels

Trim EM Pumps and Peripherals

Tower

Riser/Downcomer

Other Items

Receiver Piping

Receiver Structural Steel

Dump Tank

Table 3.3-14

TOWER COSTS

Tower Height	Cost
150 meters	$ 4,300,000
225 meters	$ 7,700,000
300 meters	$14,700,000

Table 3.3-15

DOWNCOMER RISER COST ESTIMATE

Tower Height	
Case A - 1300 F	
150 meters	$ 500,000
225 meters	$ 800,000
300 meters	$1,200,000
Case B - 1100 F	
150 meters	$ 700,000
225 meters	$1,100,000
300 meters	$1,700,000

Table 3.3-16

RECEIVER COST SUMMARY

ITEM	NORTH FIELD (Cost=$\$10^6$)						360° FIELD (Cost=$\$10^6$)					
	150 m		225 m		300 m		150 m		225 m		300 m	
	1300 F	1100 F	1300 F	1100 F	1300 F	1100 F	1300 F	1100 F	1300 F	1100 F	1300 F	1100 F
1. Absorber Panel	1.304	1.119	1.304	1.119	1.304	1.119	3.965	2.998	3.965	2.998	3.965	2.998
2. EM Pumps*	6.290	7.100	6.670	7.640	7.120	8.030	7.590	8.470	7.930	8.790	8.420	9.210
3. Tower	4.232	4.232	7.719	7.719	14.652	14.652	4.232	4.232	7.719	7.719	14.652	14.652
4. Riser/Downcomer	0.463	0.701	0.763	1.108	1.114	1.637	0.463	0.701	0.763	1.108	1.144	1.637
Subtotal 1	12.289	13.152	16.456	17.586	24.220	25.438	16.250	16.401	20.377	20.615	28.181	28.497
1. Feeder Lines	0.714	0.702	0.714	0.702	0.714	0.702	0.583	0.455	0.583	0.455	0.583	0.455
2. Headers	0.057	0.056	0.057	0.056	0.057	0.056	0.157	0.154	0.157	0.154	0.157	0.154
3. Structural Steel	0.150	0.150	0.150	0.150	0.150	0.150	0.207	0.207	0.207	0.207	0.207	0.207
4. Dump Tank	0.155	0.200	0.155	0.220	0.180	0.220	0.155	0.180	0.155	0.200	0.180	0.220
5. Surge Tank	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100	0.100
6. Sodium	0.085	0.147	0.102	0.176	0.118	0.205	0.076	0.127	0.093	0.157	0.109	0.186
Subtotal 2	1.261	1.355	1.278	1.404	1.319	1.433	1.278	1.223	1.295	1.273	1.336	1.322
Total: Sub 1 + Sub 2	13.550	14.507	17.734	18.990	25.539	26.881	17.528	17.624	21.672	21.888	29.517	29.819

Note: Moveable Insulation Costs Are Not Included.
These costs apply to a system with an IHX.

Surge Tank

Sodium (for receiver system only)

The results of these estimates are given in Table 3.3-16.

The bases of the costs for the above receiver subsystem items are given below.

Absorber Panels. Absorber panel costs were developed by FWDC for four preliminary panel designs as shown below. All panel designs included a center inlet header and a top and bottom outlet header as shown in Figure 3.3-11, and were assembled according to the manufacturing sequence described in Figure 3.3-13.

Panel Design	No. 1	No. 2	No. 3	No. 4
	North Field		360° Field	
	1300 F	1100 F	1300 F	1100 F
Tubing Material	I-625	I-625	I-625	SS316
Tubing OD (in.)	0.75	1.0	0.4685	0.5993
Tubing Wall Thickness (in.)	0.049	0.049	0.035	0.035
Tube Total Length (ft)	70	70	64	64
Number of Tubes per Panel	51	39	165	129
Number of Panels	17	17	24	24
Material Cost/Panel ($)	24,140	23,600	30,210	16,260
Labor Cost/Panel ($)	52,570	42,200	135,000	108,660
Total Cost/Panel ($)	76,710	65,800	165,210	124,920
Total Absorber Panel Cost ($M)	1.304	1.119	3.965	2.998

Trim EM Pumps. Cost and performance data for the EM trim pumps located in the receiver assembly (one per absorber panel) were developed from GE induction pump manufacturing experience. The total pump cost for each receiver configuration and system temperature are shown plotted vs. tower height in Figure 3.3-17. These pump costs include power supplies, capacitors, and cooling equipment.

Tower. Tower costs were developed by Kaiser Engineers Inc. for slip formed concrete towers 150, 225, and 300 meters high. Tower cost vs. tower height is shown in Figure 3.3-18. The elements of these cost estimates are shown below.

	Tower Height			
	150 m	225 m	300 m	
Material	1.815	3.272	6.118	
Labor	1.329	2.452	4.722	
Taxes and Contractors Indirect	1.088	1.995	3.812	
Total	$4.232	$7.719	$14.652	($Million)

Riser/Downcomers. Riser/downcomer estimated total costs vs. tower height are shown in Figure 3.3-19. Estimates are based on a 60^{0} helical piping configuration to accommodate differential thermal expansion between the pipe and tower. Pipe sizes and material used for these estimates are given below.

	Material: 1300 F System	Material: 1100 F System
Riser	2-1/4Cr - 1 Mo	2-1/4 Cr - 1 Mo
Downcomer	Inconel 625	316SS
Pipe Size	18 in.*	24 in.*

Cost Elements	($ in Millions) 1300 F			1100 F		
	150 m	225 m	300 m	150 m	225 m	300 m
Material	0.346	0.576	0.870	0.524	0.826	1.217
Labor	0.038	0.059	0.084	0.058	0.093	0.141
Taxes and Contractors Indirect	0.079	0.128	0.190	0.119	0.189	0.279
Total	0.463	0.763	1.144	0.701	1.108	1.637

Other Items. Receiver piping includes the interconnecting piping between the riser/downcomer and the absorber panels. Costs are installed costs including manifold branch pipe, fittings, hangers, traceheating, insulation, labor, and overhead. Structural steel costs are based on $0.58/pound, installed. The dump tank and surge tank are based on the use of carbon steel and 2-1/4Cr -1 Mo materials, respectively, designed and constructed in accordance with Section VIII of the ASME Boiler and Pressure Vessel Code. Costs are based on recent procurement of similar tanks. Sodium costs are based on $0.33/pound for large bulk quantities of commercial grade sodium.

*Wall thickness is specified in Tables E-1 through E-6 of Appendix E.

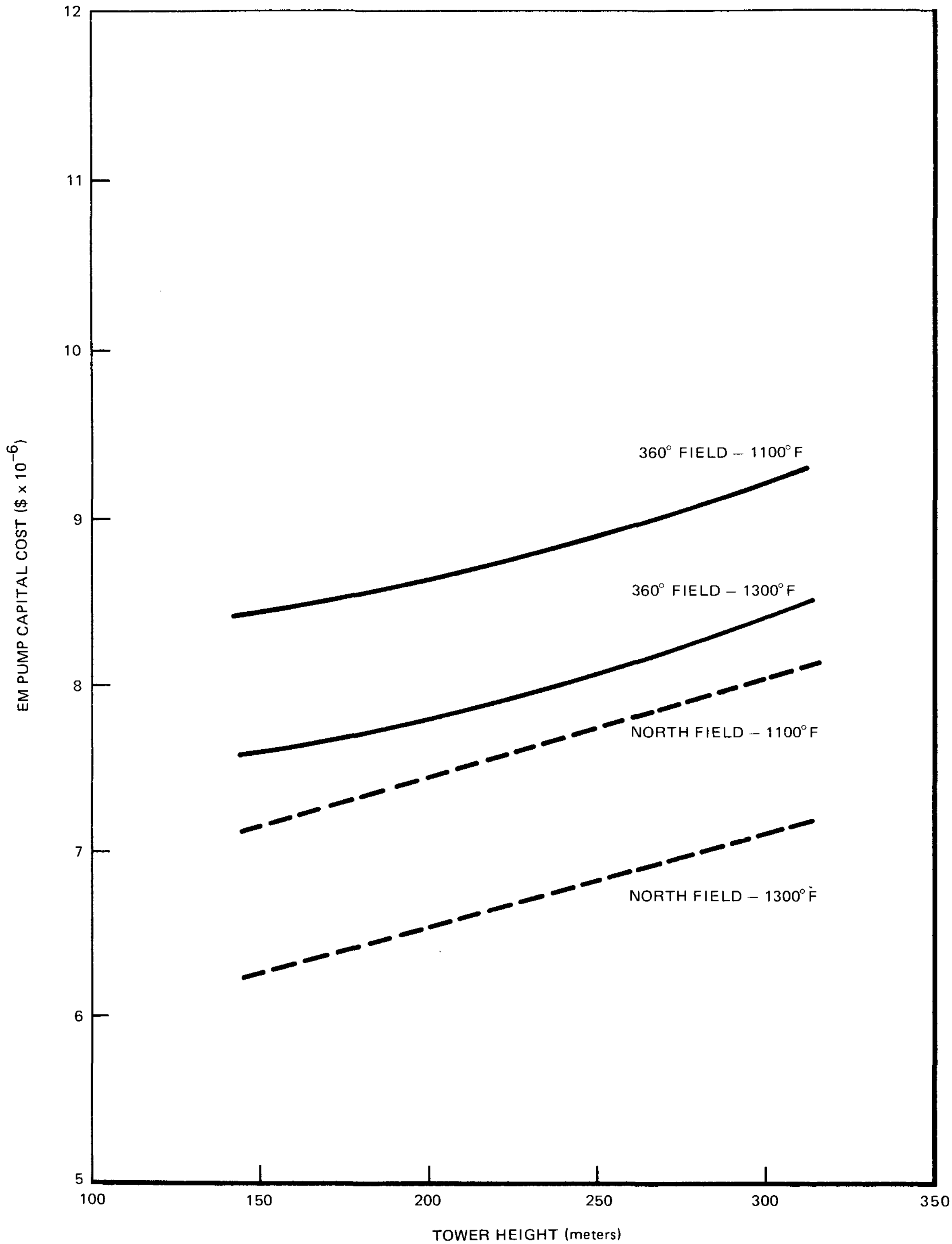


Figure 3.3-17. Capital Cost for Small EM Pumps for Absorber Panels

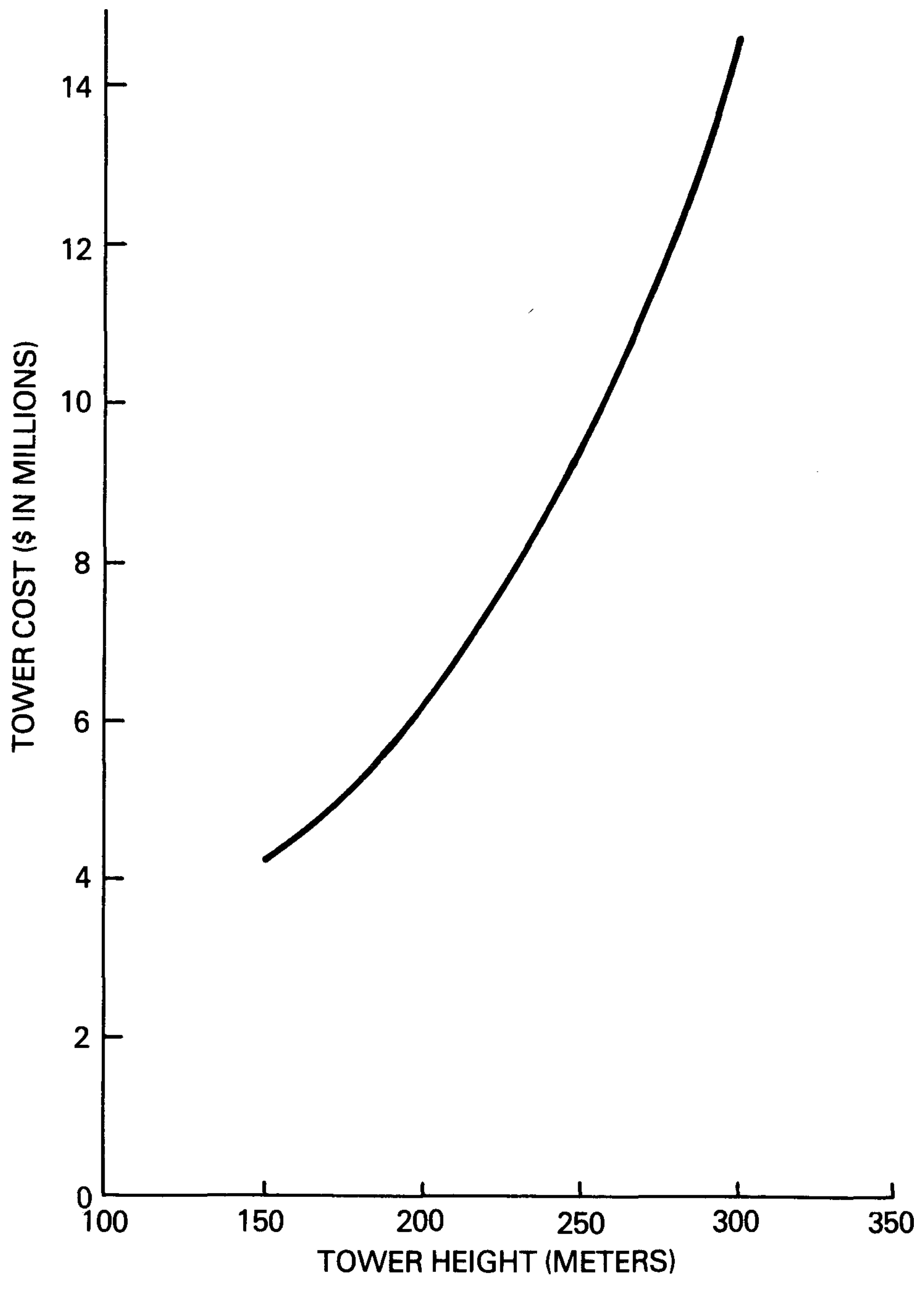
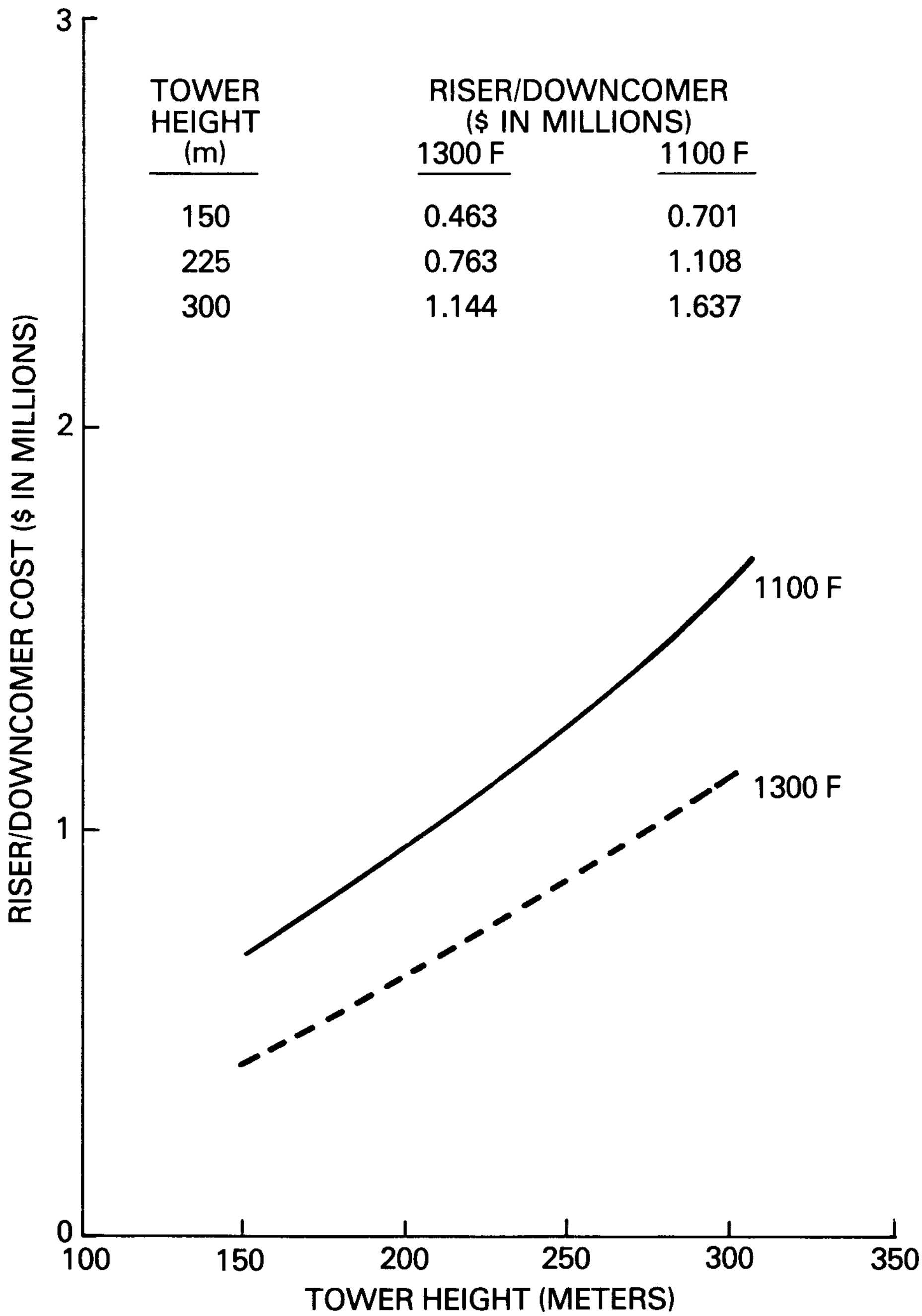


Figure 3.3-18. Plot of Tower Cost vs. Height



TOWER HEIGHT (m)	RISER/DOWNCOMER ($ IN MILLIONS) 1300 F	1100 F
150	0.463	0.701
225	0.763	1.108
300	1.144	1.637

Figure 3.3-19. Riser/Downcomer Costs vs. Tower Height

REFERENCES FOR SECTION 3.3

3.3-1. Richard N. Lyon, "Liquid Metal Heat Transfer Coefficients," Chemical Engineering Progress, Vol. 47, No. 2, February 1951.

3.3-2. "Evaluation of Measurements of the Local and Total Heat Transfer Iron Smooth and Rough Surface Cylinders In Cross Flow," Heat and Mass Transfer Source Book, Fifth All-Union Conference, Minsk, 1976, John Wiley and Sons.

3.3-3. F. Kreith, Principles of Heat Transfer, second edition, International Textbook Company, Scranton, Pennsylvania, 1965, p. 358.

3.3-4. G.D. Gupta, M.S.M. Rao, T.V. Narayanan, and A.C. Gangadharan, "Thermoelastic Analysis of Nonaxisymmetrically Heated Thick Cylindrical Shells," Journal of Pressure Vessel Technology, ASME, Vol. 100, No. 1, February 1978, pp. 107-111.

3.3-5. M.S.M. Rao, T.V. Narayanan, G.D. Gupta, "Inelastic Analysis of Non-axisymmetrically Heated Thick Cylindrical Shells" (to be published).

3.3-6. T.V. Narayanan, G.D. Gupta, M.S.M. Rao, "Structural Design of a Superheater for a Central Solar Receiver" (to be published in Journal of Pressure Vessel Technology, ASME).

3.3-7. INCONEL Alloy 625, Huntington Alloys, Inc., Huntington, West Virginia 25720, Report No. 15M 1-76-42, 1976.

3.3-8. ACI Manual of Concrete Practice, Part 2, 1973, pp. 307-308.

3.4 STORAGE SUBSYSTEM

The general design/operating/survival requirements considered in the parametric analysis of storage options are defined in Table 3.4-1

Table 3.4-1

STORAGE SUBSYSTEM REQUIREMENTS

		Baseline Design	Operating	Survival
• System Temperature	1300 F case	1260 F	630 - 1260 F	-
	1100 F case	1050 F	630 - 1050 F	-
• System Pressure	Na-Fe case	150 psi	35 - 150 psi	-
	Na case	50 psi	30 - 50 psi	-
• System Flow Rates				
1300 F	Charge	6.24×10^6 W	0 - 6.24×10^6 W	-
	Discharge	4.16×10^6 W	0 - 4.16×10^6 W	-
1100 F	Charge	9.91×10^6 W	0 - 9.91×10^6 W	-
	Discharge	6.61×10^6 W	0 - 6.61×10^6 W	-
• Design Life		30 years	30 years	-
• Minimum Storage Temperature (Trace heated, dump system)		350 F	350 F	
• Environment				
Outside environment		No sodium expulsion to atmosphere		Sodium drop and suppression ponds
Temperature		Wet bulb 74 F Dry bulb 82.6 F	-20 to 120 F	
Wind		8 mph	∿40 mph	90 mph
• Plant Availability		90%		
• Maintainability		Accessible for maintenance	Sodium removal from components	Sodium drip and suppression ponds
• Temperature Maintenance		<2% loss for 24 hour period at normal baseline conditions	<2% loss for 24 hour period at normal baseline conditions	
• Operating Modes		Standby Charge at solar-multiple of 0 to 1.5 Discharge to Steam Generators at plant design rating (100 MW_e)		Shutdown Drain

3.4.1 STORAGE CONCEPTS AND SIZING

Four basic storage concepts were identified at the start of the parametric analysis:

Concept 1 - sensible energy storage in liquid sodium and iron in one or more tanks using the increased volumetric heat capacity of iron with thermocline heat transfer for both charge and discharge (see Figure 3.4-1).

Concept 2 - sensible energy storage of sodium in multiple tanks with the equivalent of one tank always left empty for purposes of hot or cold sodium transfer (Figure 3.4-2).

Concept 3 - sensible energy storage of sodium in separate low pressure hot and cold tanks. The complex of hot or cold tanks is capable of accepting the total sodium inventory and independent operation of the steam generator and tower/receiver loop is provided. This configuration was also used to evaluate the possibility of using draw salt instead of sodium as the storage medium (Figure 3.4-3).

Concept 4 - hybrid sensible energy storage system which combines the sodium/iron feature of Concept 1 with the hot and cold independently operated tankage of Concept 3 (Figure 3.4-4).

During the analysis, these concepts were discovered to be quite expensive because the Intermediate Heat Exchanger (IHX) was included to isolate storage from high pressure at the base of the tower. Additional parametric cases were defined to investigate the following alternatives to the IHX:

1. Throttle valve at the base of the downcomer with low pressure storage vessels

Figure 3.4-5 describes two cases in which the receiver loop pumping is provided entirely by the EM pump allocated to each receiver panel. These pumps are located on the ground to provide adequate pump inlet pressure. Figure 3.4-6 is a variation of this configuration in which the major pumping input comes from a single ground level pump, while small EM pumps in the tower are used to trim the flow in each panel. Figure 3.4-7 shows the large ground level pump with control valves on each panel instead of EM trim pumps.

2. Jet pumps with low pressure storage vessels

Figure 3.4-8 shows two cases which use jet pumps to pump low pressure tank fluid into the high pressure steam generator loop. When hot fluid is being stored, it is throttled into the hot tank.

3. Two pressure storage with part of the storage at high pressure

Figure 3.4-9 shows a high pressure steam generator loop having two small high pressure tanks. These tanks hold just enough sodium to ride out short term transients (<15 minutes). Excess hot sodium is throttled and stored in large low pressure tanks; it is recovered by pumping back into the

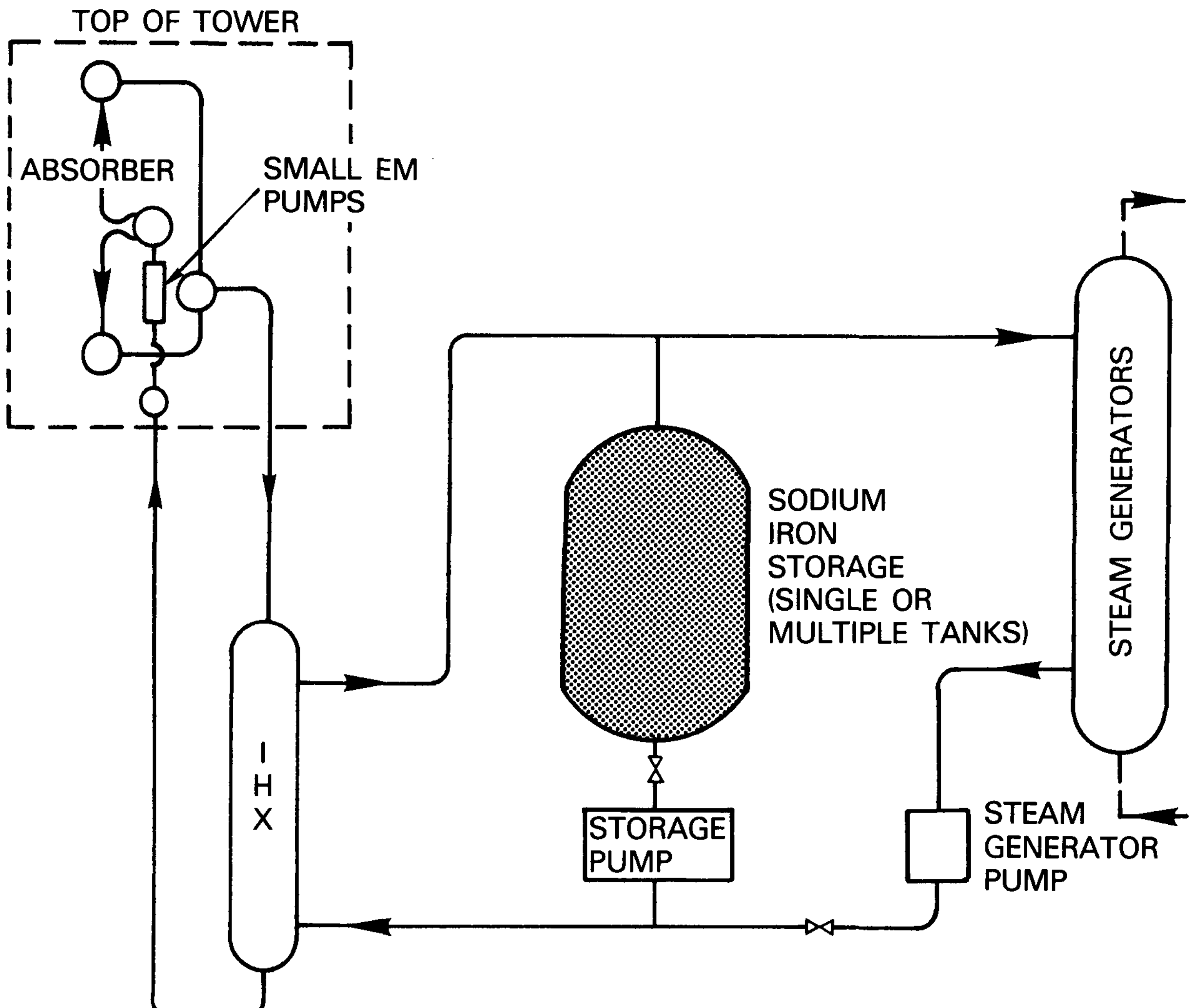


Figure 3.4-1. Concept 1-Sodium-Iron Storage (Cases S1 - 1100 F, S6 - 1300 F)

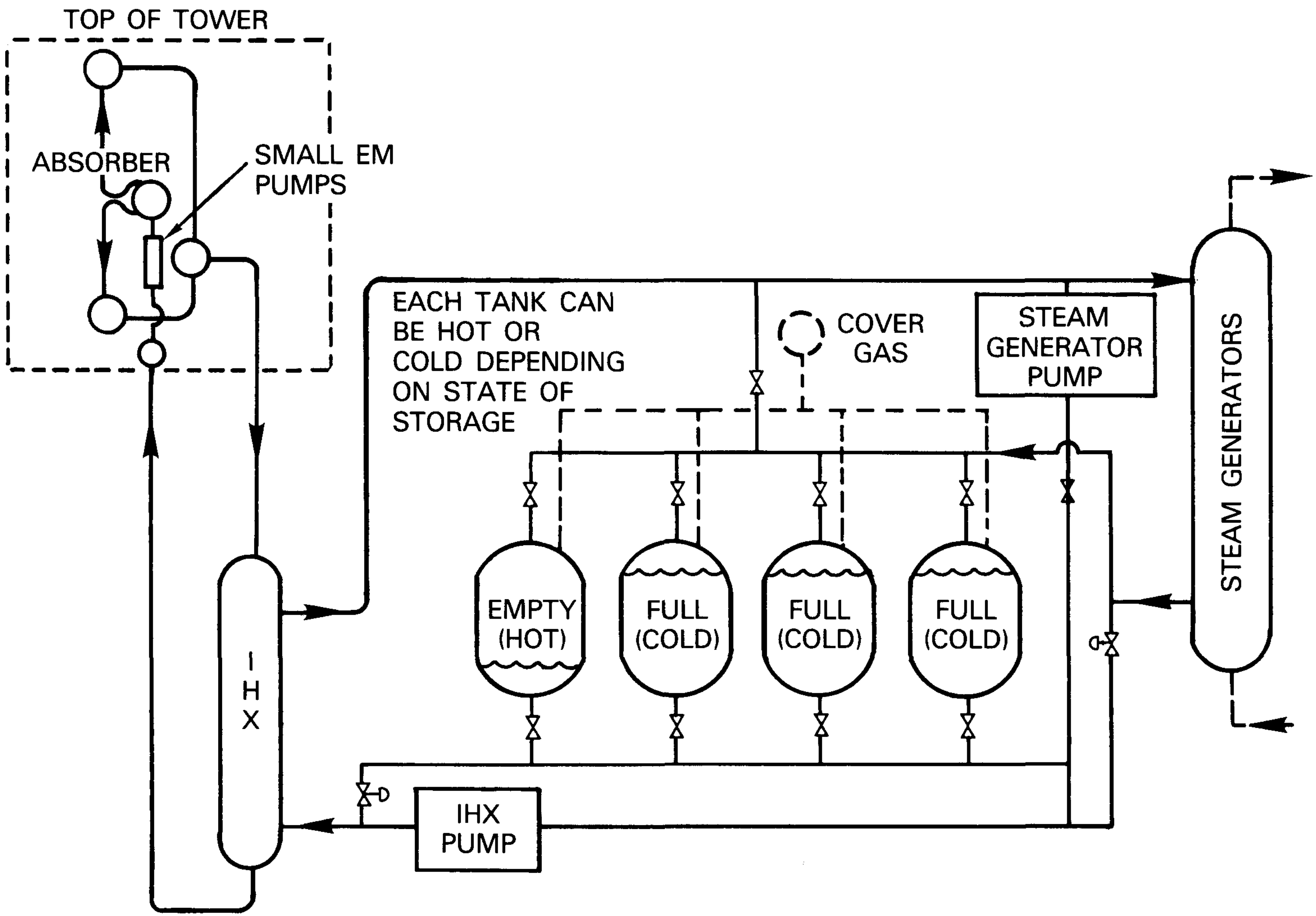


Figure 3.4-2. Concept 2 - Sodium - Empty Tank (Cases S2 - 1100 F, S7 - 1300 F)

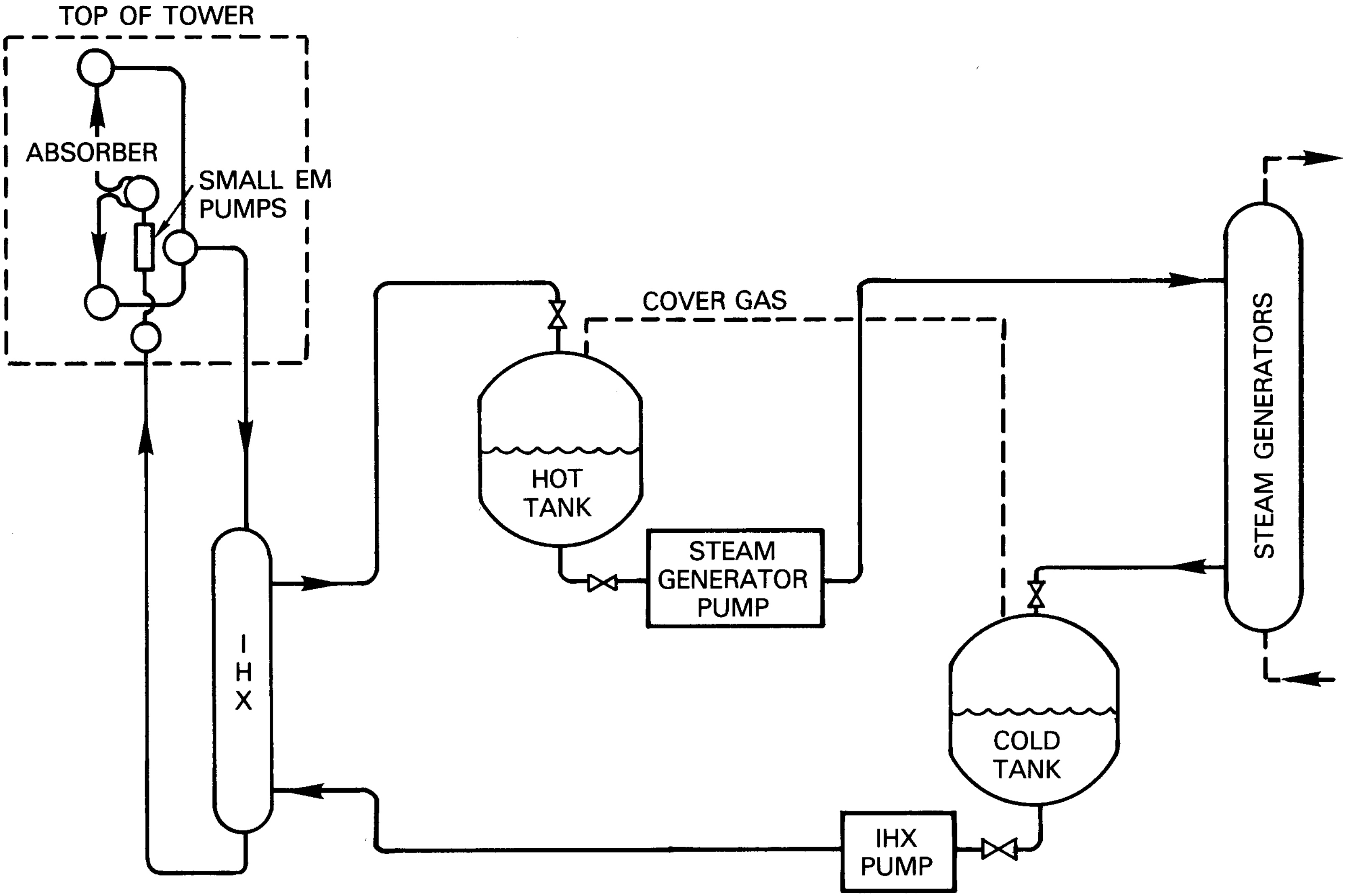


Figure 3.4-3. Concept 3 - Sodium - Hot and Cold Tanks (Cases S3 - 1100 F, S5 - 1100 F - Salt, S8 - 1300 F)

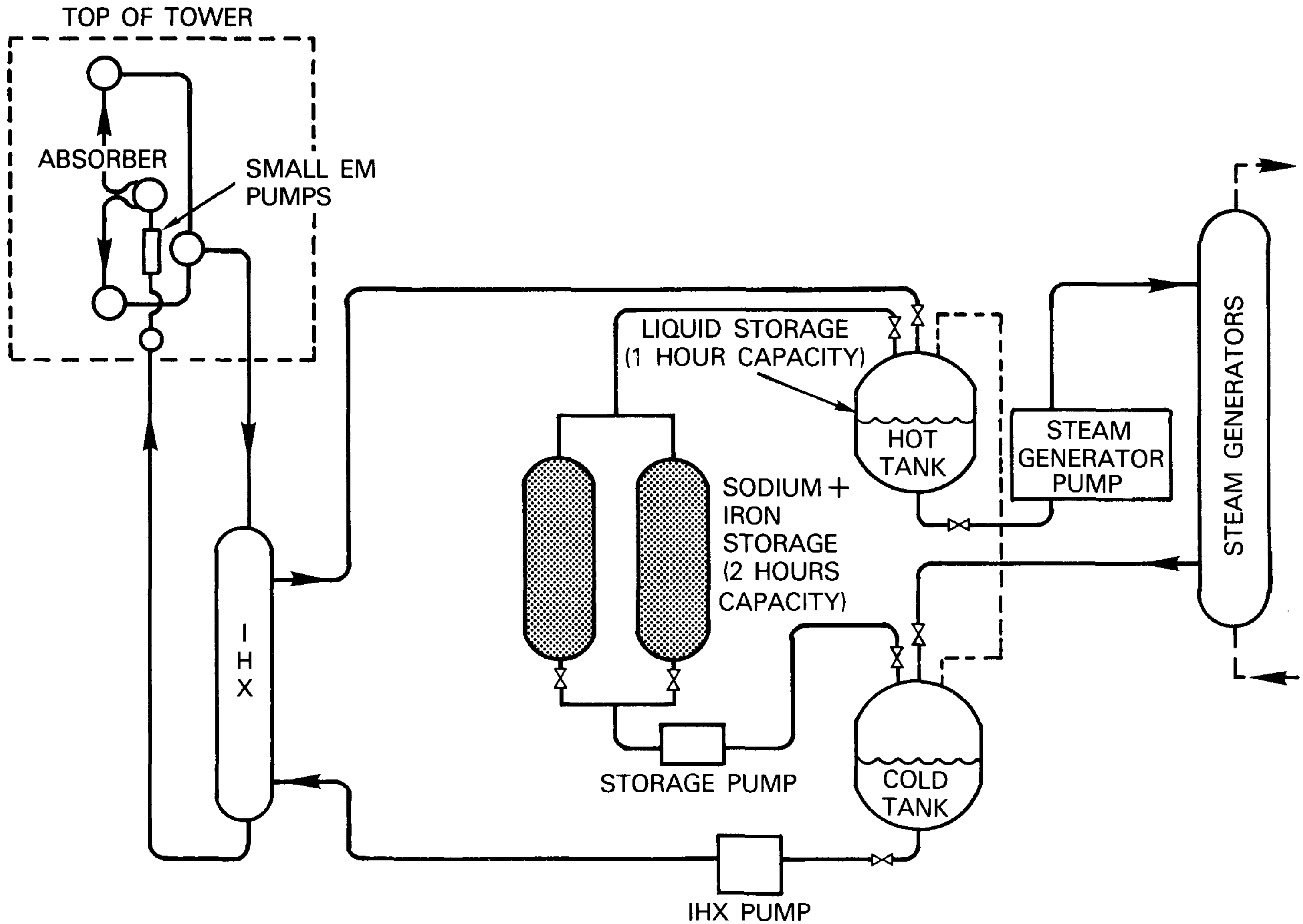


Figure 3.4-4. Concept 4-Hybrid of Concepts 1 and 3 (Cases S4 - 1100 F, S9 - 1300 F

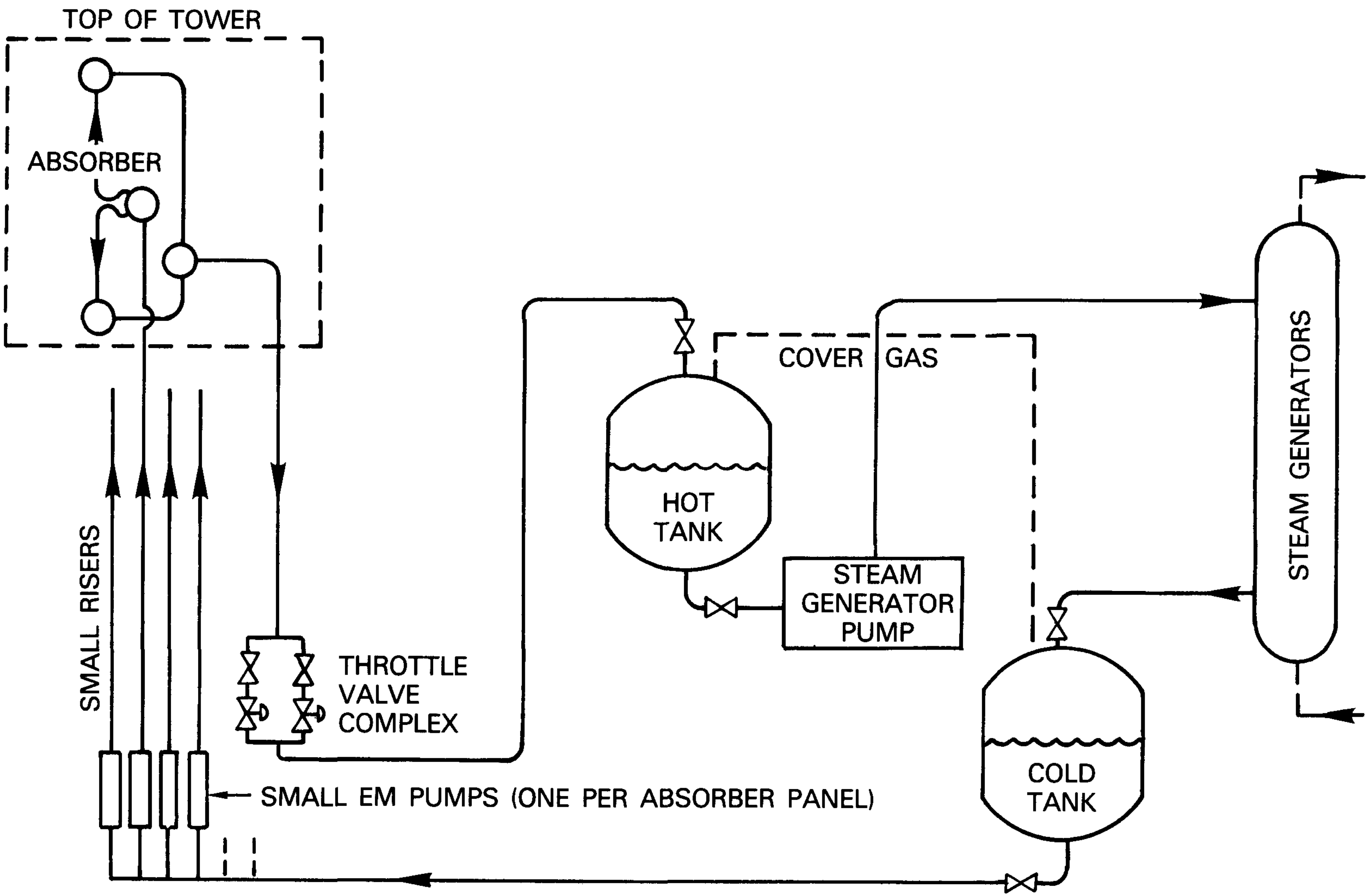


Figure 3.4-5. No IHX, Throttle Valve, Small EM Pumps on Ground with Multiple Risers (Case S10 - 1100 F)

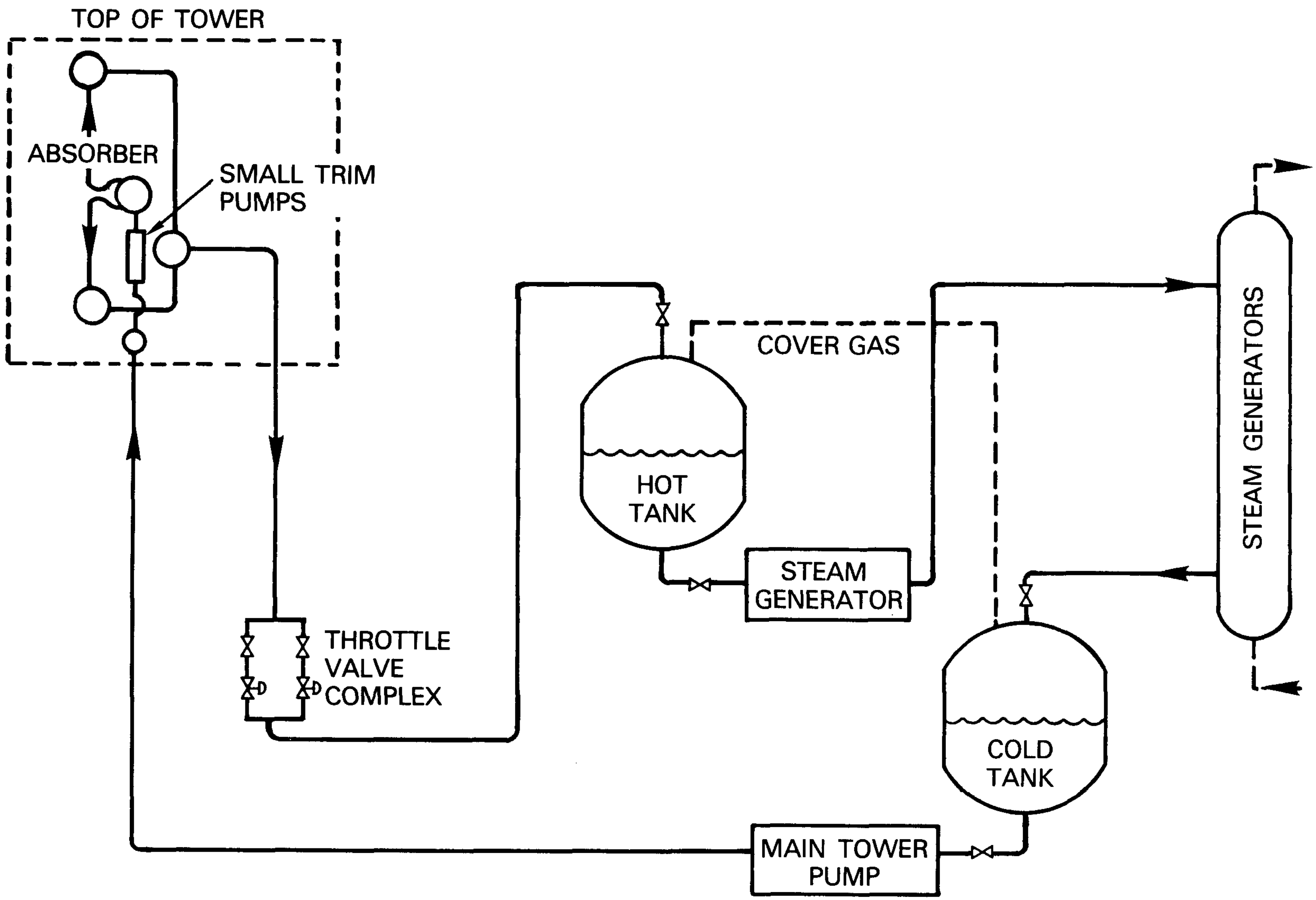


Figure 3.4-6. No IHX, Large Pump on Ground, Small EM Trim Pumps in Tower (Case S11 - 1100 F)

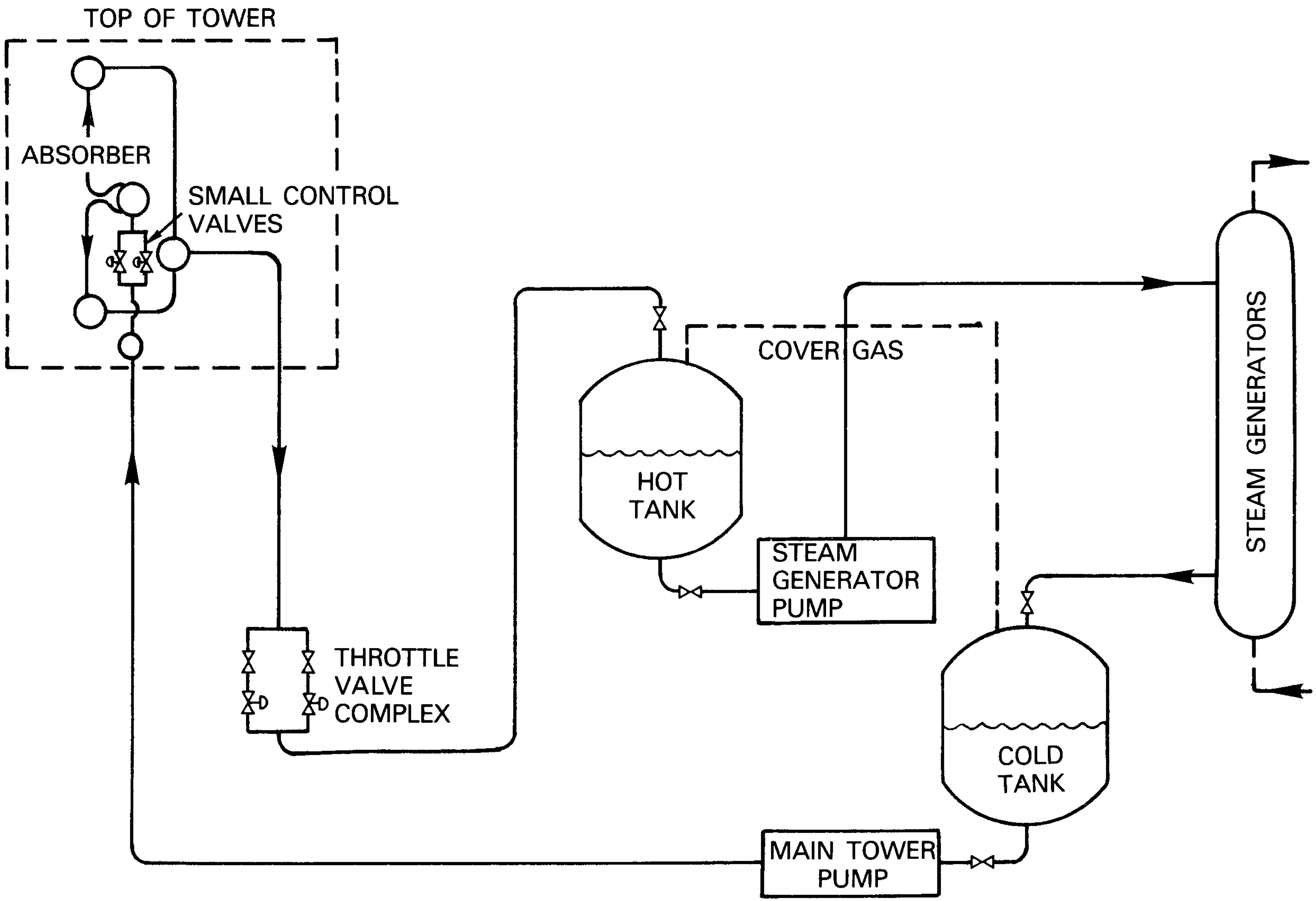


Figure 3.4-7. No IHX, Large Pump on Ground, Control Valves in Tower, 1100 F (Case S12)

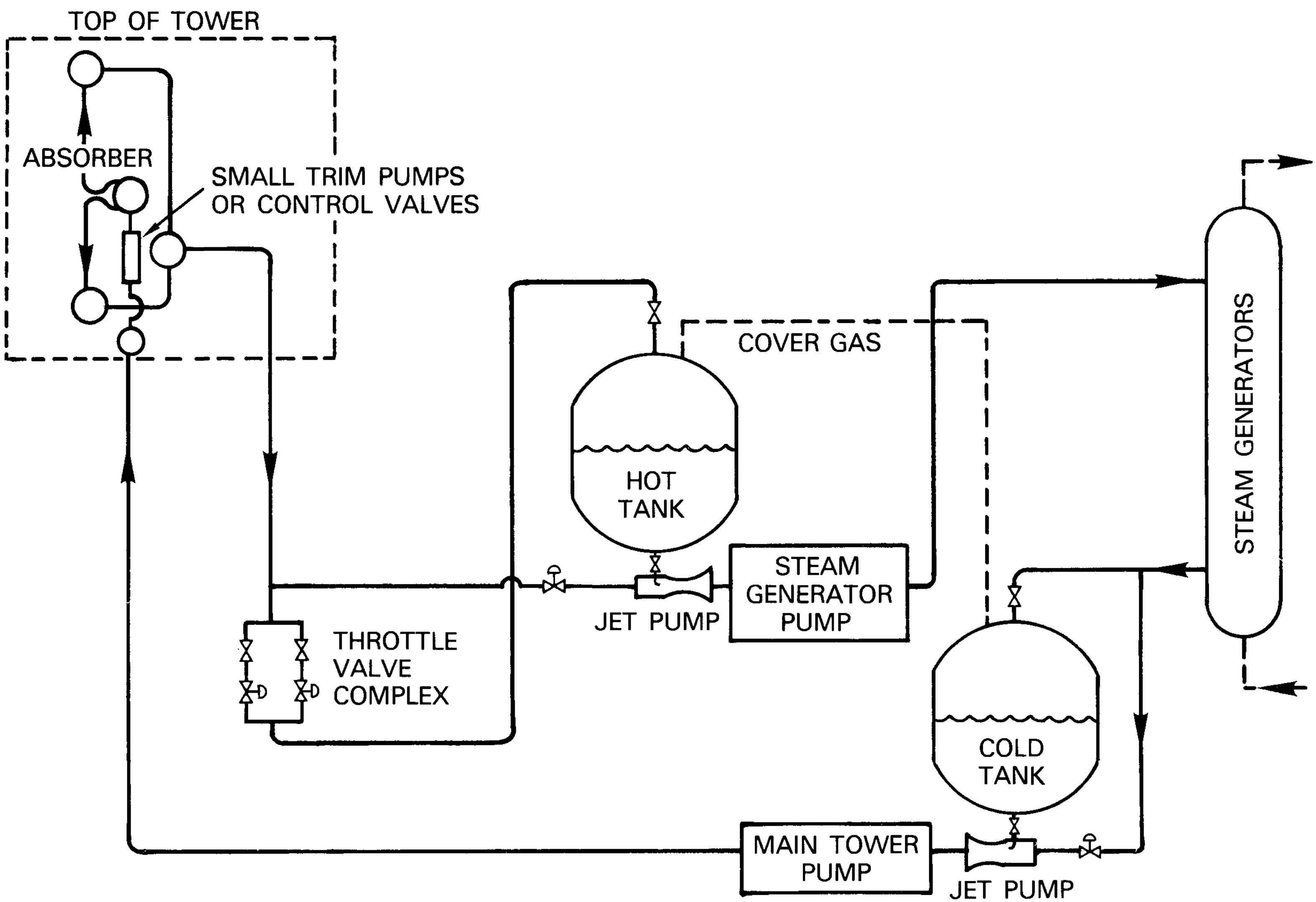


Figure 3.4-8. No IHX, Large Pumps and Jet Pumps on Ground (Cases S13 - Trip Pumps, S14 - Valves)

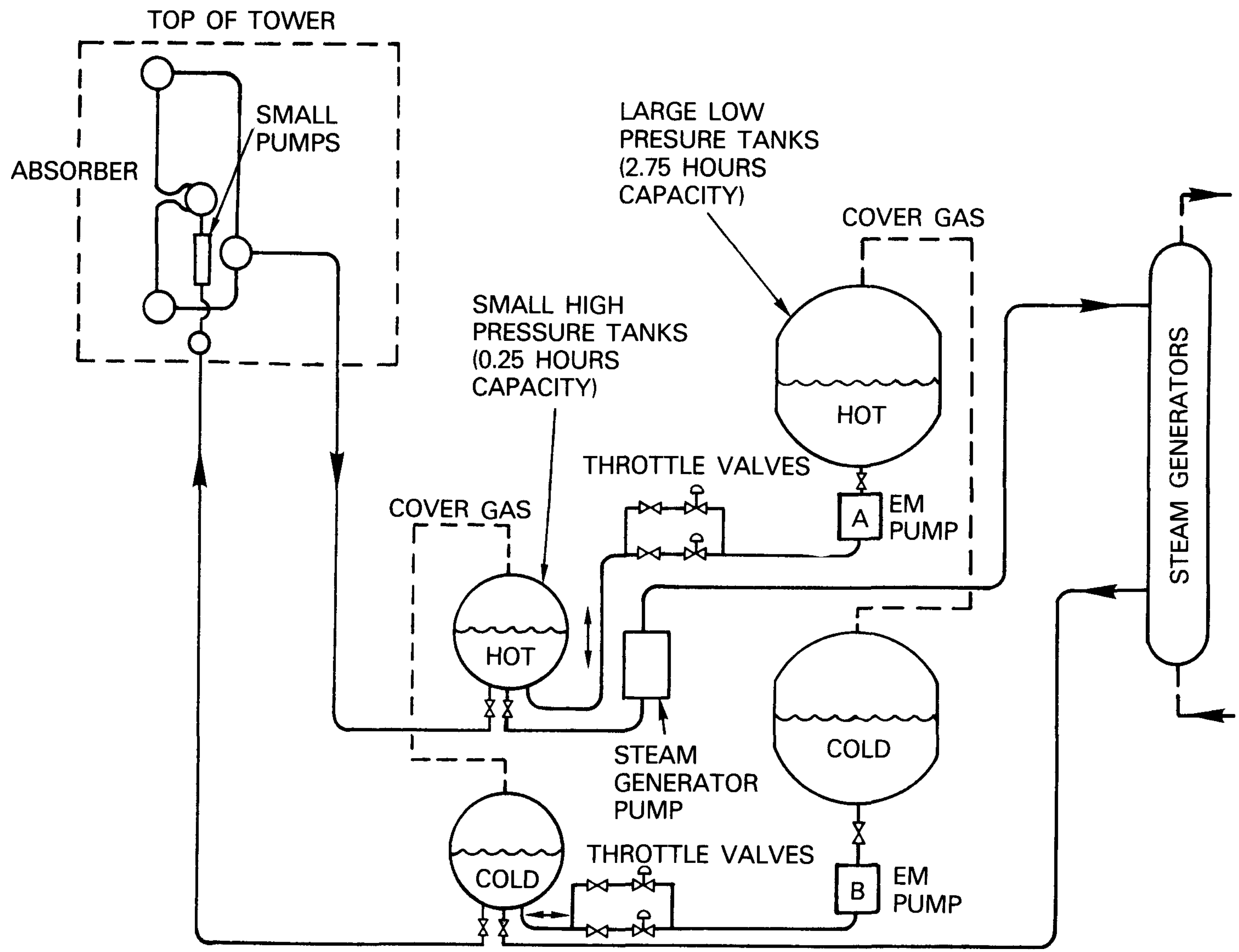


Figure 3.4-9. No IHX, Two Pressure Storage (Cases S15 - 1100F, S16 - 1300F)

small high pressure hot tank. Cold sodium is handled the same way. Figure 3.4-10 shows a variation on this concept which eliminates the smaller of the two pumps (Flow B/Flow A = 1.5 due to solar multiple). Figure 3.4-11 shows another variation which eliminates the larger pump and substitutes large high pressure cold tanks.

The complete matrix of storage parametric cases is summarized in Table 3.4-2. The major parametric variables are temperature level (1100 F vs. 1300 F), storage concept, pressure reducer, and tower flow control technique.

The storage subsystem for the advanced central receiver is required to provide a storage capactiy of 3 MW_e hr/MW_e. The steam cycle used as the basis for this parametric analysis (see Appendix H) requires approximately 250 MW_t at full output; thus the thermal capacity of storage must be 750 MW_th to meet the specification. Tank volumes and storage material requirements were determined for the four storage concepts so as to provide 750 MW_th of net recovered energy; losses were accounted for as follows:

	Sodium Storage (%)	Sodium + Iron Storage (%)
Losses*		
Insulation (24 hours)	1	1
Thermocline Loss	0	3
Turbine Startup	1	1
	2	5

*as a fraction of net capacity

In addition a 10 percent tank volume adder was applied to the sodium + iron cases to account for the volume occupied by flow distribution structures at the base of the tanks.

The insulation losses shown were estimated for a 24-hour duty cycle because it was assumed that the energy stored each day would be used completely during that day. Thermocline losses for the sodium + iron storage cases were estimated on the basis of a detailed transient thermal analysis. This is discussed in Appendix F and Section 3.4.5. Turbine startup energy was based on 2 percent steam flow during a one-half hour start.

It was assumed that the storage media had the following average properties:

	Density (lb/ft^3)	Specific Heat (Btu/lb-F)
Sodium	50	0.30
Iron	492	0.15
Sodium + Iron (75% Iron by Volume)*	381	0.15

*assumes hexagonal close packed spheres (one-half-inch diameter)

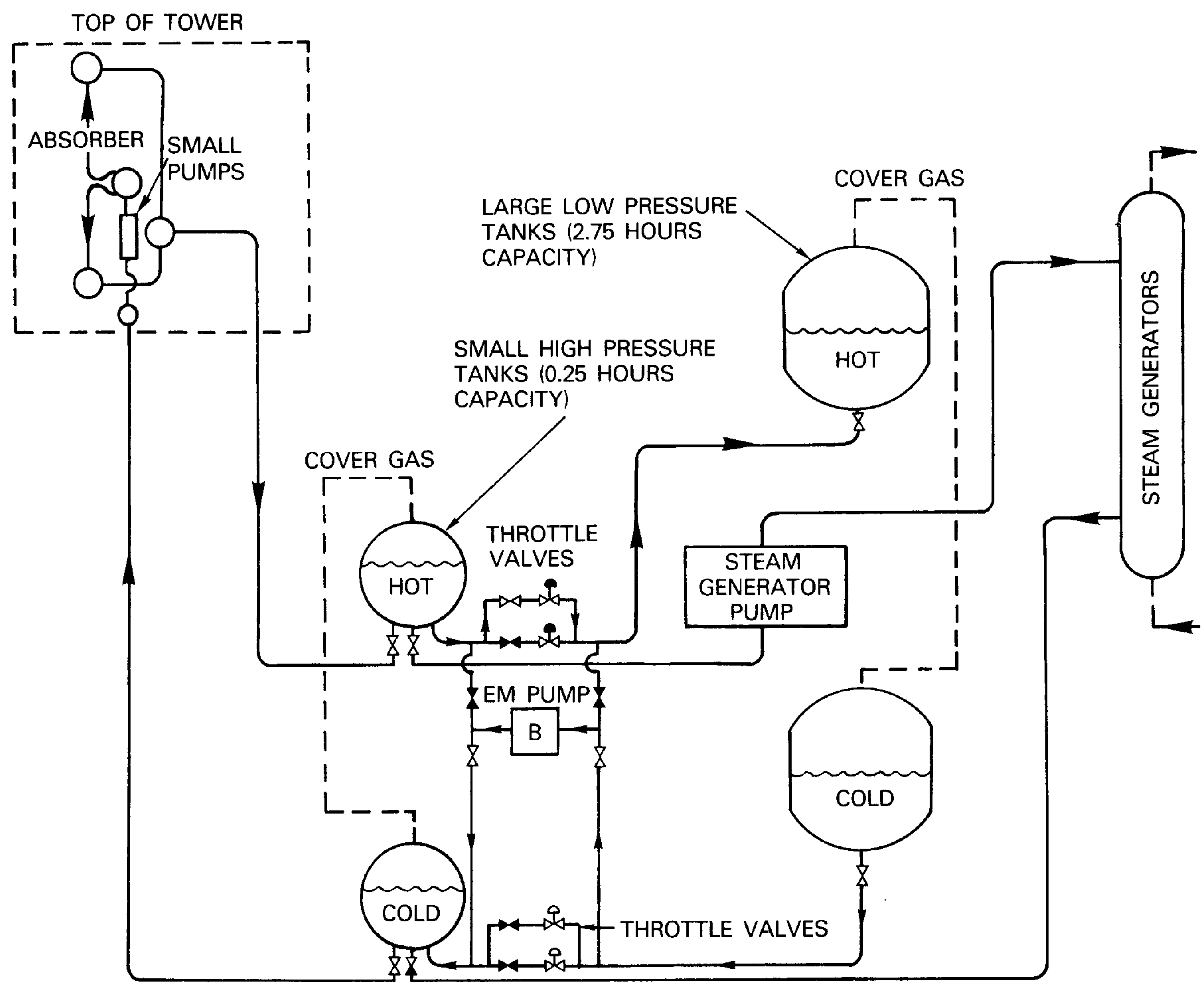


Figure 3.4-10. No IHX, Two Pressure Storage (Cases S19 - 1100 F, S20 - 1300 F)

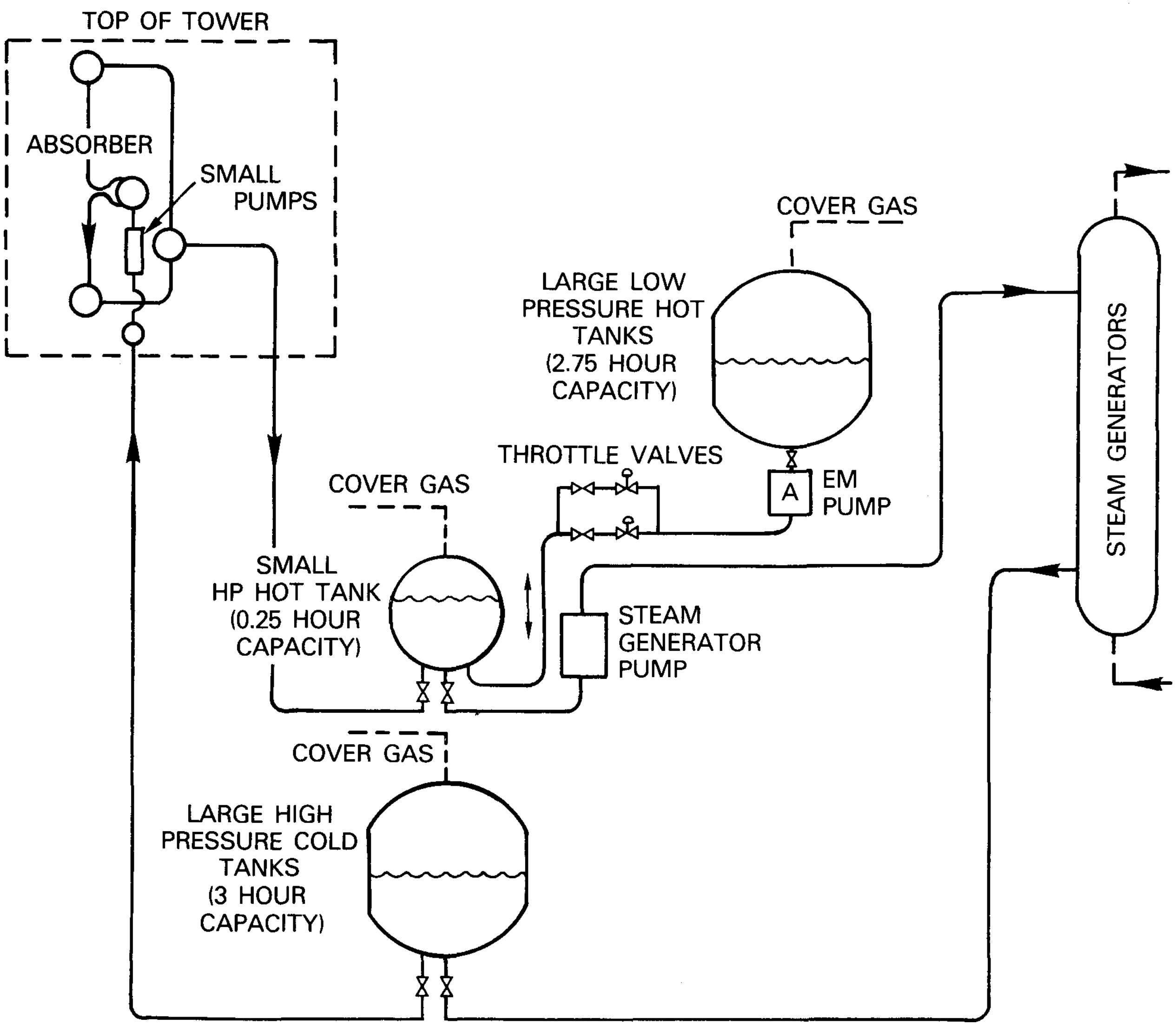


Figure 3.4-11. No IHX, Two-Pressure Storage (Cases S17 - 1100 F, S18 - 1300 F)

Table 3.4-2

THERMAL STORAGE SUBSYSTEM PARAMETRIC CASES S1 THROUGH S22

Case No.	Temp. (^{o}F)	Storage Concept	Pressure Reducer	Tank Assembly	Tower Flow Control
S1	1100	1	IHX	Factory	Small EM Pumps on Absorber Panels
S2	1100	2	IHX	Field	Small EM Pumps on Absorber Panels
S3	1100	3	IHX	Field	Small EM Pumps on Absorber Panels
S4	1100	4	IHX	Factory Field	Small EM Pumps on Absorber Panels
S5	1100	3/Salt	IHX	Field	Small EM Pumps on Absorber Panels
S6	1300	1	IHX	Factory	Small EM Pumps on Absorber Panels
S7	1300	2	IHX	Field	Small EM Pumps on Absorber Panels
S8	1300	3	IHX	Field	Small EM Pumps on Absorber Panels
S9	1300	4	IHX	Factory Field	Small EM Pumps on Absorber Panels
S10	1100	3	Valve	Field	Small EM Pumps at Tower Base, Multiple Risers
S11	1100	3	Valve	Field	Large Pump at Tower Base - Small Panel Pumps
S12	1100	3	Valve	Field	Large Pump at Tower Base - Panel Valves
S13	1100	3	Jet Pump	Field	Same as S11
S14	1100	3	Jet Pump	Field	Same as S12
S15	1100	Two Pressure Storage		Factory Field	Small EM Pumps on Absorber Panels
S16	1300	High/Low Hot and Cold Tanks		Factory Field	Small EM Pumps on Absorber Panels
S17	1100	Two Pressure Storage		Factory Field	Small EM Pumps on Absorber Panels
S18	1300	High/Low Hot and Cold Tanks		Factory Field	Small EM Pumps on Absorber Panels
S19	1100	Same as S15/S16		Factory Field	Small EM Pumps on Absorber Panels
S20	1300	Added Valves to Eliminate One Pump		Factory Field	Small EM Pumps on Absorber Panels

Table 3.4-3 gives a summary of the storage volumes which were calculated based on these data. These numbers have been rounded.

Preliminary calculations had indicated that the tankage cost would be a significant part of the total storage subsystem cost. For this reason a sub-optimization was performed to find the most cost-effective tank fabrication technique. The two major options investigated were field assembled tanks and factory assembled tanks; these were analyzed by Kaiser Engineers and Foster Wheeler Development Corporation respectively.

Figures 3.4-12 and 3.4-13 describe the way the storage vessels would be arranged and supported. The factory fabricated vessels would be mounted in groups of four to nine tanks to reduce insulation and valving requirements. The field fabricated tanks are large enough that this clustering is not necessary; each tank stands alone. Figure 3.4-14 shows three alternatives for arranging the iron in the thermocline tanks. This is not a trivial consideration since the arrangement affects the packing density of the iron and may also significantly increase the effective pressure on the tank walls if, for instance, iron balls are used. However, it was not possible to address this question in more detail in the parametric analysis.

The details of the various factory fabricated tank designs are described in Section 3.4.2; details of the field fabricated tanks are given in Section 3.4.3. A summary of the storage subsystem cost estimates is given in Section 3.4.7.

Table 3.4-3

STORAGE VOLUMES

	Operating Temperatures High (°F)	Operating Temperatures Low (°F)	Volume of Media ft^3	Nominal Tank Volume ft^3
Concept 1	1260	630	72,000	79,200[a]
	1050	630	108,000	118,800[a]
Concept 2	1260	630	276,300	345,375[b]
	1050	630	416,000	520,000[b]
Concept 3	1260	630	276,300	552,600[c]
	1050	630	416,000	832,000[c]
Concept 4	1260	630	140,100	237,000
	1050	630	210,700	356,600

[a] Accounts for flow spreaders

[b] Assumes a five tank array with one empty tank

[c] Volume is double that of sodium due to duplicate sets of tanks

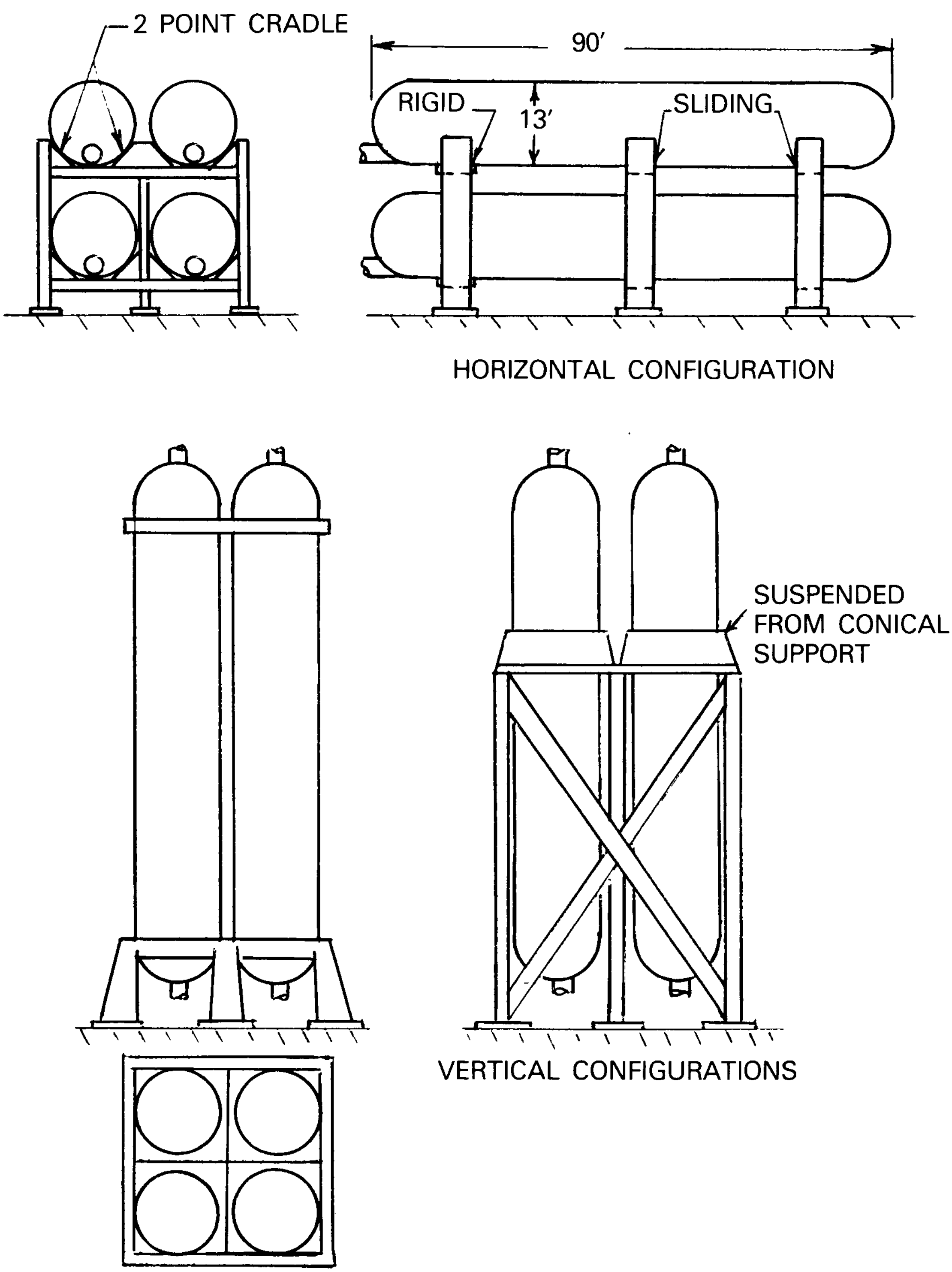


Figure 3.4-12. Typical Factory Fabricated Tank Clusters

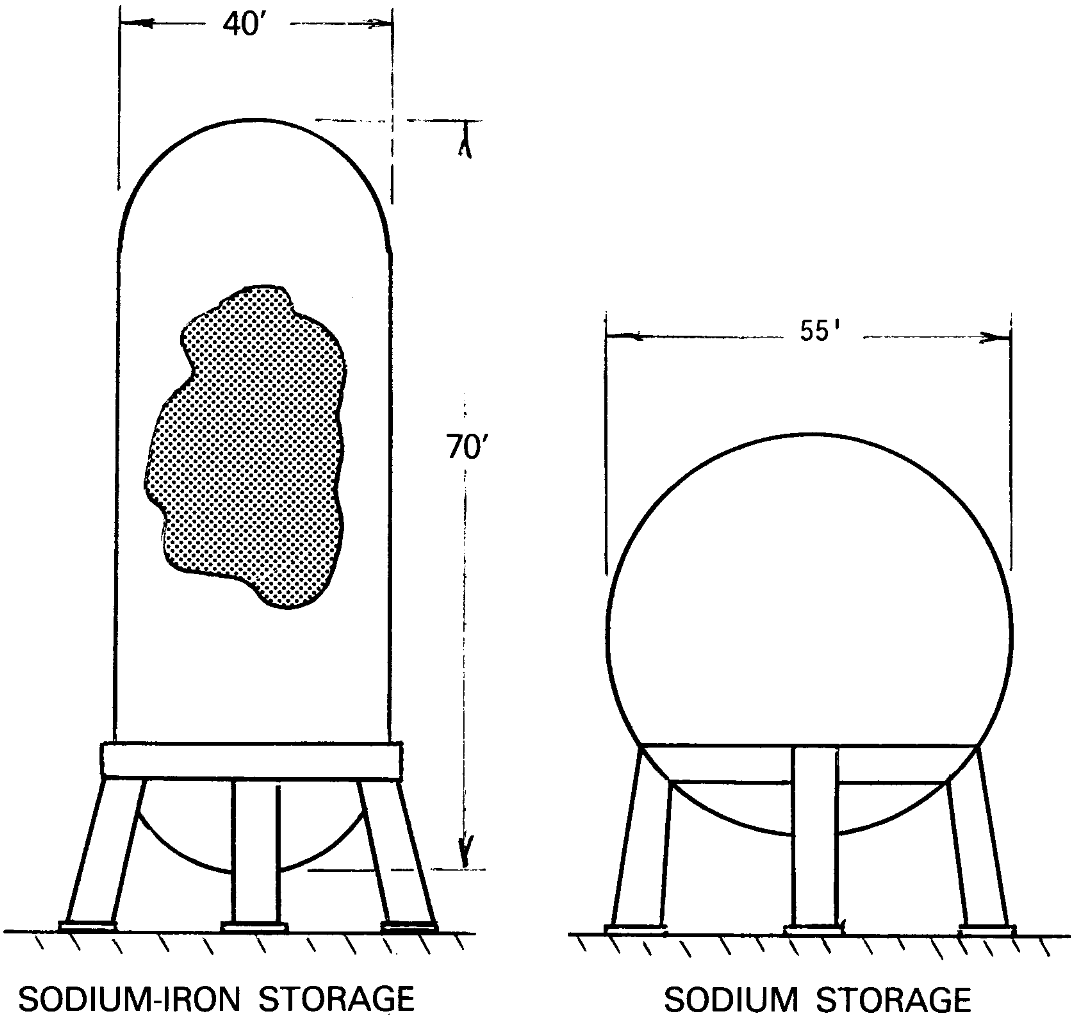


Figure 3.4-13. Typical Field Fabricated Tank Configuration

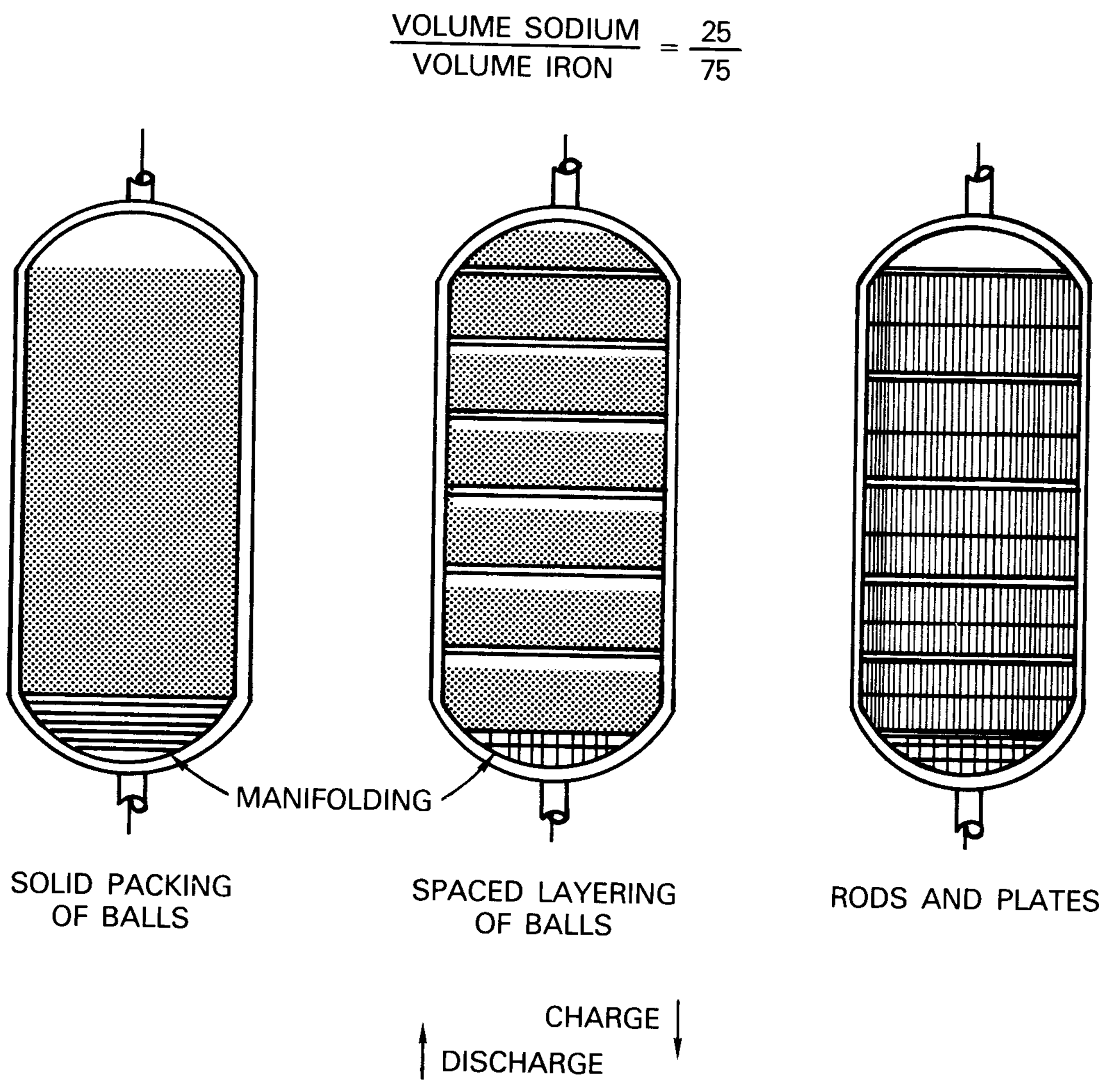


Figure 3.4-14. Sodium-Iron Tank Concept

3.4.2 FACTORY ASSEMBLED STORAGE VESSELS

The major difference between field and factory fabricated vessels is that factory assembled tanks are limited to sizes which are transportable while field tanks are not. Railroad limits are a maximum width of 13 feet with the maximum length variable from ∿60 feet out of New York City to ∿90 feet out of the Southeast. Shipment through the Panama Canal to coastal ports can accommodate lengths up to 120 feet. For purposes of this study, the factory fabricated tank size was selected to be 13 feet OD by less than 90 feet in length on the assumption that railroad routes through the Southwest will permit transportation of a 90 foot vessel.

Foster Wheeler analyzed several vessels of this type. Figure 3.4-15 shows a typical vertical tank for sodium + iron storage with a first guess at the arrangement of flow spreaders in the bottom of the tank. It was assumed that the iron was in the form of one-half inch balls. Figure 3.4-16 shows a typical horizontal configuration considered for containing liquid sodium.

Table 3.4-4 lists the results of the Foster Wheeler analysis. The first three tanks are mounted vertically to maintain thermal stratification in the thermocline storage system. At 1300 F the tank material is Inconel 625, while at 1100 F two materials were investigated -- Incoloy 800 and 316 stainless steel. Field installation costs were estimated by Kaiser Engineers for the clustering arrangements discussed in Section 3.4.1. Cases 3, 4a, and 4b are the liquid sodium tanks and were originally designed by Foster Wheeler for 150 psig service. Later analysis of the liquid metal system indicated that the actual service pressure would be much lower (50 psig) since the tanks operate on the inlet side of the pumps. The costs and weights originally provided by Foster Wheeler were scaled by the ratio of 50/150, and these are the results quoted in Table 3.4-4 for Cases 3, 4a, and 4b. The sodium + iron tank design pressure cannot be reduced in Concept 1 (Cases 1, 2a, 2b) because the tanks are between the steam generators and a large pump during the storage discharge cycle; however, the tanks in Concept 4 can be designed for lower pressure (Cases 1a, 2c).

In Concepts 3 and 4, some of the liquid storage tanks operate at 630 F and are constructed from 2-¼ Cr - 1 Mo. The price of this material (about $1.10/lb) is lower than for Inconel 625 or Incoloy 800. To estimate the cost of these tanks (Cases 3a, 4c) the material cost of Cases 3 and 4a were reduced by the appropriate ratio while the labor and freight costs were left unchanged.

Table 3.4-5 shows how these tank designs were combined to form storage systems Concepts 1 through 4. This table shows the type, number, and size of the tanks used, as well as the volumes and weights of the tanks, sodium, and iron. The cost for tanks, installation, and storage media is also listed. The breakdown of these costs is detailed in Table 3.4-6.

Field assembled tank costs are also given in Table 3.4-5; these costs are from the analysis by Kaiser Engineers, which is discussed in the next section. Based on these cost results, we concluded that factory assembled tanks are not cost effective for storing liquid sodium. This is principally because of the high field cost associated with installing a large number of separate tank units. For the sodium + iron cases, however, the number of tanks is smaller, and the factory assembled approach does seem to offer a saving.

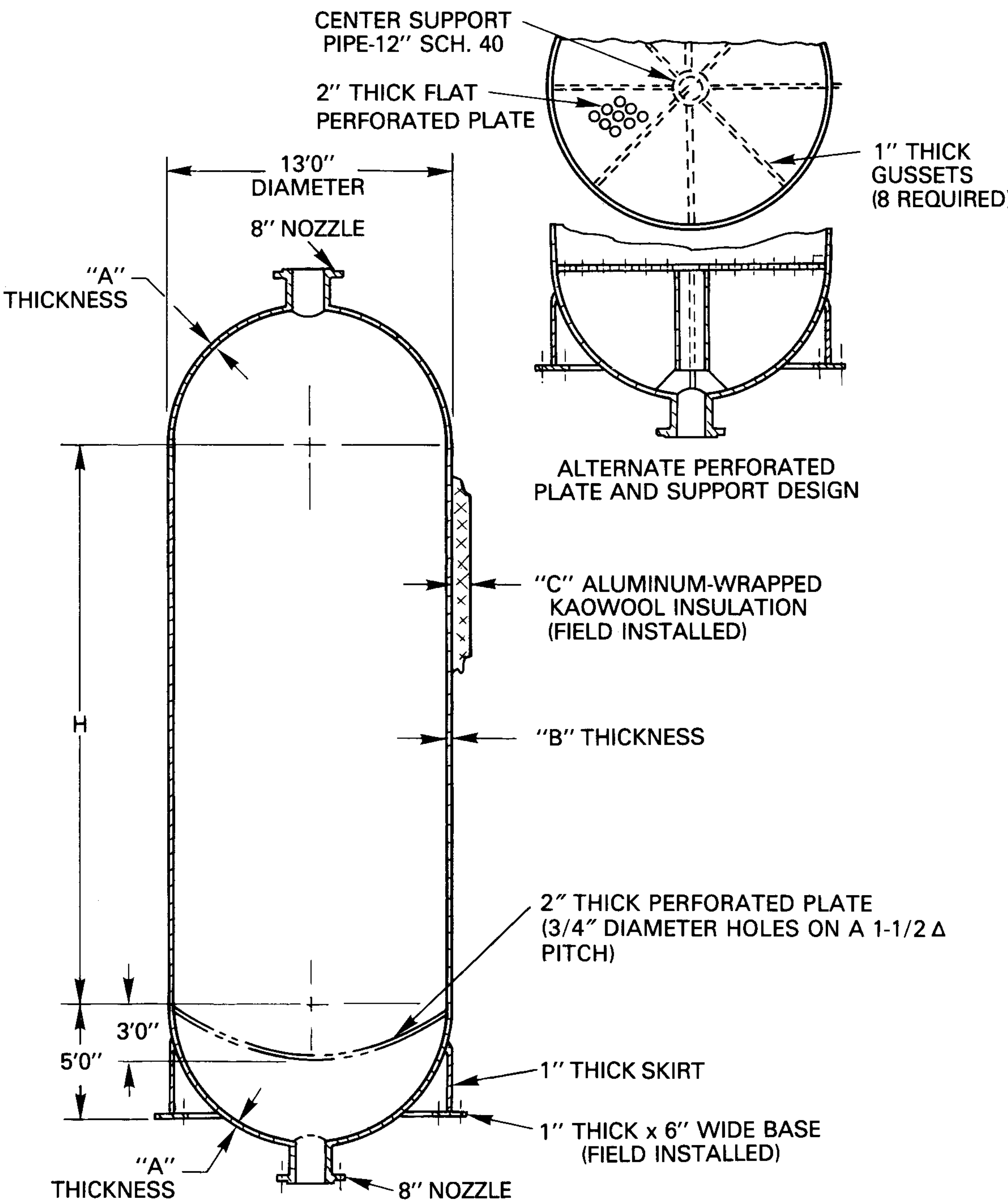


Figure 3.4-15. Typical Factory Assembled Vertical Storage Tank

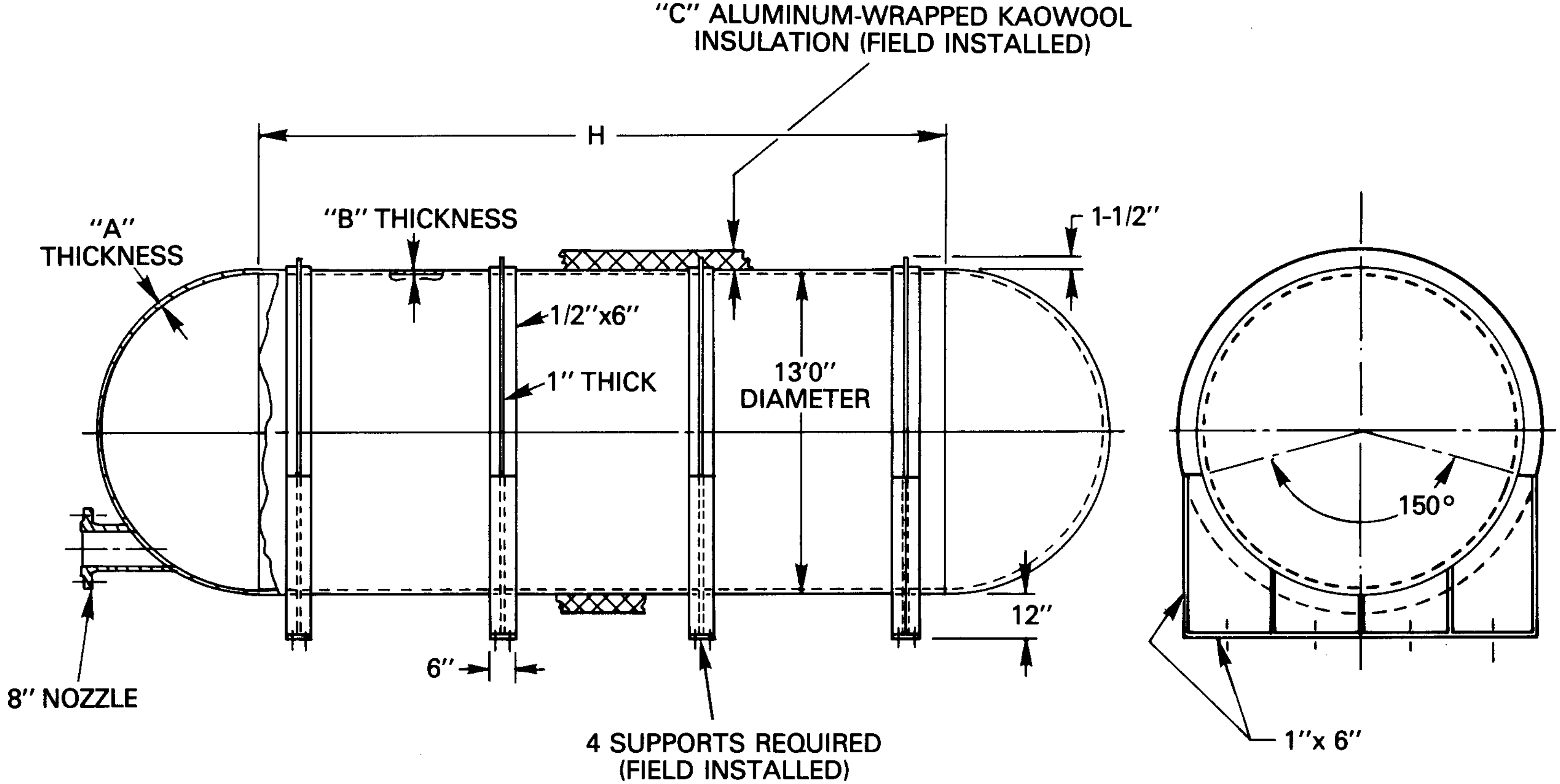


Figure 3.4-16. Typical Factory Assembled Horizontal Storage Tank

Table 3.4-4

FACTORY ASSEMBLED TANK COSTS

Case No.:	1	2a	2b	3	4a	4b	3a	4c	1a	2c
Storage Medium	Na+Fe	Na+Fe	Na+Fe	Na	Na	Na	Na	Na	Na+Fe	Na+Fe
Nominal Temperature (^{o}F)	1300	1100	1100	1300	1100	1100	630	630	1300	1100
Design Pressure (psig)	150	150	150	50	50	50	50	50	50	50
Volume of Tank (ft^3)	9902	9900	9900	9210	9905	9905	9210	9905	9902	9900
Orientation	Vertical	Vertical	Vertical	Horizontal	Horizontal	Horizontal	Horizontal	Horizontal	Vertical	Vertical
Material	Inconel 625	Incoloy 800	SS316	Inconel 625	Incoloy 800	SS316H	2-¼ Cr-1 Mo	2-¼ Cr-1 Mo	Inconel 625	Incoloy 800
Dimensions A (in.)	0.50	0.45	0.50	0.17	0.15	0.17	0.17	0.15	0.17	0.15
B (in.)	1.00	0.90	1.00	0.33	0.30	0.33	0.33	0.30	0.33	0.30
C (in.)	16.00	10.00	10.00	16.00	10.00	10.00	10.00	10.00	16.00	10.00
H (ft)	67.83	67.62	67.83	62.48	67.66	67.85	62.48	67.66	67.83	67.72
Weight Empty (lb)	144,072	132,672	144,072	46,344	45,599	49,390	46,344	45,599	48,024	44,224
Material Cost[a] ($)	803,503	330,352	217,417	254,566	111,470	69,367	51,006[e]	50,253[f]	267,834[g]	110,117[g]
Labor Cost[b] ($)	91,311	91,477	85,862	18,257	19,504	17,450	18,257	19,504	30,437[g]	30,492[g]
Freight Cost[c] ($)	1,441	1,327	1,440	463	456	494	463	456	480[g]	442[g]
Total Cost-FOB ($)	896,255	423,155	304,719	273,286	131,430	87,311	69,726	70,213	298,751	141,051
Field Installation[d] ($)	239,000	239,000	239,000	213,500	213,500	213,500	213,500	213,500	239,000	239,000

[a]Metal only, no insulation included.

[b]Factory labor only, no installation included.

[c]Northeast to Southwest United States by rail.

[d]Includes unloading from rail cars, foundations and supports, trace heating, insulation, and filling with iron balls; assumes tanks are mounted in groups of four as shown in Figure 3.4-15.

[e]Scaled from Case 3 by ratio of costs for 2-¼ Cr - 1 Mo vs. Inconel 625 (1.10/5.49).

[f]Scaled from Case 4a by ratio of costs for 2-¼ Cr - 1 Mo vs. Incoloy 800 (1.10/2.44).

[g]Scaled from Cases 1 and 2a by ratio of pressures (50/150).

Table 3.4-5

FIELD ASSEMBLED VS. FACTORY ASSEMBLED STORAGE VESSELS

Concept	Temperature (°F)	Volume of Storage Medium (ft³)	Storage Vessels Assembled in: Field[a]	Storage Vessels Assembled in: Factory[b]	Number of Tanks Required	Tank Description: Shape	Tank Description: Size[c] (ft)	Unit Tank Volume (ft³)	Total Volume of Tanks (ft³)	Tank Material	Weights (Tons): Tanks	Weights (Tons): Sodium	Weights (Tons): Iron	Cost[d] (M$)	Comments
Concept 1	1300	72,000	VIIa		1	Cyl.	40 D x 78 L	79,403	79,403	I-625	495	495	13,443	27.473	75% Volume Steel
			VIIb		2	Cyl.	32 D x 62 L	40,235	80,470	I-625	500	495	13,443	29.004	25% Volume Sodium
Sodium + Iron Thermocline				1	8	Cyl.	13 D x 81 L	9,902	79,216	I-625	576	495	13,443	21.507	All Tanks Mounted Vertically
	1100	108,000	IIa		1	Cyl.	45.5 D x 90.3 L	119,696	119.696	I-800	630	742	20,165	33.985	
Design Pressure: 150 psig			IIb		2	Cyl.	35.7 D x 71.5 L	58,313	116,626	I-800	624	742	20,165	32.385	
				2a	12	Cyl.	13 D x 80.6 L	9,900	118,800	I-800	796	742	20,165	26.584	
				2b	12	Cyl.	13 D x 80.8 L	9,900	118,800	SS316	864	742	20,165	25.163	
Concept 2	1300	276,300	Va		5	Sphere	51.2 D	69,075	345,375	I-625	600	6,907	--	19.204	One Tank Empty
Sodium in Hot/Cold Tanks				3	36	Cyl.	13 D x 75.5 L	9,210	331,560	I-625	834	6,907	--	22.033	Group of 6 Empty
	1100	416,000	IVa		5	Sphere	58.7 D	104,000	520,000	I-800	760	10,400	--	20.634	1 Tank Empty
Design				4a	48	Cyl.	13 D x 80.7 L	9,905	475,440	I-800	1094	10,400	--	23.421	Group of 6 Empty
Pressure: 50 psig				4b	48	Cyl.	13 D x 80.9 L	9,905	475,440	SS316	1186	10,400	--	21.303	Group of 6 Empty
Concept 3	1300	276,300	Va + Vc		4 Hot 4 Cold	Sphere	51.2 D	69,075	552,600	Hot I-625 Cold AS	1200	6,907	--	21.575	
Sodium in Separate Hot and Cold Tanks				3, 3a	30 Hot 30 Cold	Cyl.	13 D x 75.5 L	9,210	552,600	Hot I-625 Cold AS	1390	6,907	--	27.659	
Design Pressure: 50 psig	1100	416,000	IVa + IVc		4 Hot 4 Cold	Sphere	58.7 D	104,000	832,000	Hot I-800 Cold AS	1216	10,400	--	25.104	
				4a, 4c	42 Hot 42 Cold	Cyl.	13 D x 80.7 L	9,905	832,020	Hot I-800 Cold AS	1915	10,400	--	33.267	
Concept 4	1300	140,100	Vd + Ve		1 Hot 1 Cold	Sphere	56.4 D	92,100	243,612	Hot I-625 Cold AS	464	2,632	8,962	18.310	Sodium Tanks are Field Assembled, Sodium + Iron Tanks Are Factory Assembled
Hybrid of Concepts 1 and 3				1a	6	Cyl.	13 D x 81 L	9,902							
Design Pressure: 50 psig (Na) 150 psig (Na+Fe)	1100	210,700	IVd + IVe		1 Hot 1 Cold	Sphere	64.6 D	138,700	356,600	Hot I-800 Cold AS	581	3,961	13,443	23.487	Sodium Tanks are Field Assembled, Sodium + Iron Tanks Are Factory Assembled
				2c	8	Cyl.	13 D x 80.6 L	9,900							

(a) Numbers refer to Kaiser Engineers cases, Section 3.4.3.

(b) Numbers refer to Foster Wheeler cases, Section 3.4.2.

(c) D = outside diameter for spheres and cylinders
L = overall length for cylinders

(d) Includes materials, labor, tax, indirect costs, and freight charges for tanks, sodium, and iron.

Note:
I-625 = Inconel 625
AS = *Alloy Steel*
I-800 = Incoloy 800
SS316 = Stainless Steel 316

Table 3.4-6

FACTORY ASSEMBLED STORAGE TANKS SYSTEM COST SUMMARY

Concept:	1	1	1	2	2	2	3	3	4	4
Tank Type	1	2a	2b	3	4a	4b	3,3a	4a,4c	1a	2c
No. of Tanks (Hot)	8	12	12	36	48	48	30	42	6	8
(Cold)	-	-	-	-	-	-	30	42	-	-
Factory Cost[a] ($)	7,170,040	5,077,860	3,656,628	9,838,296	6,308,640	4,190,928	10,290,360	8,469,006	1,792,506	1,128,408
Installation ($)	1,912,000	2,868,000	2,868,000	7,686,000	10,248,000	10,248,000	12.810,000	17,934,000	1,434,000	1,912,000
Iron Balls[b] ($)	12,098,700	18,148,500	18,148,500	-	-	-	-	-	8,065,800	12,098,700
Sodium[c] ($)	326,700	489,720	489,720	4,558,950	6,864,000	6,864,000	4,558,620	6,864,000	217,800	326,480
Total ($)	21,507,440	26,584,080	25,162,848	22,083,246	23,420,640	21,302,928	27,658,980	33,267,006	11,510,106	15,465,588

Field Assembled Case No.:	Vd & Ve	IVd & IVe
Cost of Field Assembled Tanks ($):	5,279,907	5,733,958
Sodium in Field Assembled Tanks ($):	1,519,650	2,228,000
Total ($):	18,309,663	23,487,546

[a]FOB at job site in Southwest.

[b]Low carbon steel at $0.45/lb.

[c]Bulk sodium at $0.33/lb.

3.4.3 FIELD ASSEMBLED VESSEL ANALYSIS

A set of field assembled storage vessels was designed in accordance with Section VIII, Division I (Pressure Vessels) of the ASME Boiler and Pressure Vessel Code.

Results of the design analysis are summarized in Table 3.4-7. A spherical shape was chosen to minimize materials cost for the sodium-only case.

Results of the cost analysis are summarized in Table 3.4-8 through 3.4-13. Detailed cost sheets are contained in Appendix G.

These analyses were originally performed only for Concepts 1 and 2. To obtain cost estimates for Concepts 3 and 4, these figures were scaled as discussed in Appendix G; the scaled results are presented in Tables 3.4-14 through 3.4-17.

Basis and Assumptions

1. The vessels were designed in accordance with Section VIII, Division I (Pressure Vessels) of the ASME Boiler and Pressure Vessel Code; design pressure = 150 psig.

2. All welds will receive 100% radiography; welds are full penetration type; Code formula "E"; joint efficiency = 1.0.

3. No corrosion allowance was made, but wall thickness was rounded off to the next highest one-sixteenth inch.

4. No allowance was made for manufacturer's tolerance in plate thickness.

5. Tank heads are hemispherical.

6. The void space in a static bulk volume of balls is 25 percent.

7. Allowable stress levels per ASME Code Section VIII, Code Case 1325, for Incoloy 800 are 13,000 psi at 1100 F, and 2000 psí at 1300 F; for Incoloy 800 H, 9400 psi at 1100 F and 4600 psi at 1300 F.

8. Density of Incoloy 800 is 0.287 pounds/cubic inch = 495 pounds/cubic foot.

9. Density of liquid sodium at 1100 F is 50 pounds/cubic foot. Density of sodium at 1300 F is 48.92 pounds/cubic foot.

10. Incoloy 625 density is 0.305 pounds/cubic inch = 527.5 pounds/cubic foot, Incoloy 617 is 0.302 pounds/cubic inch = 522.34 pounds/cubic foot; 625 "S" at 1100 F = 19,300 psi and, at 1300 F, "S" = 11,700 psi (ASME Code Case 1409).

Table 3.4-7

SODIUM STORAGE TANKS

Case Number	Unit (ft^3)	Number of Vessels	Shape	Contents	Design (°F)	Material	Wall Thickness (in.)	Vessel Weight (lb)	Diameter (ft)
Ia	79,214	1	Cylinder with Hemisphere Heads	Na-Fe	1300	Incoloy 800	8-15/16 head 18-1/2 cylinder	5.70×10^6	42' 4-1/8"
Ib	79,214	1	Cylinder with Hemisphere Heads	Na-Fe	1300	Incoloy 800H	3-15/16 7-7/8	2.40×10^6	40' 6-7/8"
Ic	39,629	2	Cylinder with Hemisphere Heads	Na-Fe	1300	Incoloy 800	7-1/8 14-11/16	2.86×10^6	33' 7-3/8"
Id	39,629	2	Cylinder with Hemisphere Heads	Na-Fe	1300	800H	3-1/8 6-1/4	1.20×10^6	32'
IIa	119,282	1	Cylinder with Hemisphere Heads	Na-Fe	1100	Incoloy 800	1-9/16 3-3/16	1.2562×10^6	45' 6-3/8" 90'3" Overall Length
IIb	59,392	2	Cylinder with Hemisphere Heads	Na-Fe	1100	Incoloy 800	1-1/4 2-1/2	0.623×10^6	35' 8" 71'6" Overall Length
IIIa	69,074	5	Sphere	Na	1300	800	11-9/16	4.032×10^6	53'
IIIb	69,074	5	Sphere	Na	1300	800H	5	1.707×10^6	52'
IVa	103,626	5	Sphere	Na	1100	800	2-11/16	0.304×10^6	59'
Va	69,074	5	Sphere	Na	1300	Inconel 625	2-11/16	0.240×10^6	51'
VIIa	79,214	1	Cylinder with Hemisphere Heads	Na-Fe	1300	Inconel 625	1-9/16 Head 3-1/16	0.99×10^6	40' 78' Overall Length
VIIb	39,629	2	Cylinder with Hemisphere Heads	Na-Fe	1300	Inconel 625	1-1/4 2-7/8	0.50×10^6	32' 62' Overall Length

Note: L = 2D for cylinder

Table 3.4-8

STORAGE DATA - CASE IIa, CONCEPT 1

Design Pressure	150 psig
Design Temperature	1100 F
Shape	Cylinder with Hemispherical Head
Total Tankage Volume	119,696 ft^3
Number of Vessels	1
Contents	Sodium-Iron
Material	Incoloy 800
Wall Thickness	3-3/16"
Vessel Weight	1.256×10^6 lb
Sodium Weight (lb)	1.48×10^6 lb
Steel Balls (lb)	40.33×10^6 lb
Diameter	45' 6-3/8"
Length	90'3"
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost (including steel balls)	$ 33,495,000
Sodium Cost	$ 489,720
Total Cost	$ 33,984,720

Table 3.4-9

STORAGE DATA - CASE IIb, CONCEPT 1

Design Pressure	150 psig
Design Temperature	1100 F
Shape	Cylinder with Hemispherical Head
Total Tankage Volume	59,392 ft^3
Number of Vessels	2
Contents	Sodium-Iron
Material	Incoloy 800
Wall Thickness	2-1/2"
Vessel Weight	0.623×10^6 lb
Sodium Weight	0.74×10^6 lb
Steel Balls	20.17×10^6 lb
Diameter	35'8"
Length	71'6"
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost (including steel balls)	$ 31,895,700
Sodium Cost	$ 489,720
Total Cost	$ 32,385,420

Table 3.4-10

STORAGE DATA - CASE IVa, CONCEPT 2

Design Pressure	50 psig
Design Temperature	1100 F
Shape	Sphere
Total Tankage Volume	103,626 ft^3
Number of Vessels	5
Contents	Sodium
Material	Incoloy 800
Wall Thickness	2-11/16"
Vessel Weight	0.304×10^6 lb
Sodium Weight	5.200×10^6 lb
Steel Balls	-
Diameter	59'
Length	-
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost	$ 13,769,800
Sodium Cost	$ 6,864,000
Total Cost	$ 20,633,800

Table 3.4-11

STORAGE DATA - CASE Va, CONCEPT 2

Design Pressure	50 psig
Design Temperature	1300 F
Shape	Sphere
Total Tankage Volume	69,074 ft^3
Number of Vessels	5
Contents	Sodium
Material	Inconel 625
Wall Thickness	2"
Vessel Weight	0.240 x 10^6 lb
Sodium Weight	3.454 x 10^6 lb
Steel Balls	-
Diameter	51'
Length	-
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost	$ 14,645,200
Sodium	$ 4,558,950
Total Cost	$ 19,204,150

Table 3.4-12

STORAGE DATA - CASE VIIa, CONCEPT 1

Design Pressure	150 psig
Design Temperature	1300 F
Shape	Cylinder with Hemispherical Head
Total Tankage Volume	79,214 ft^3
Number of Vessels	1
Contents	Sodium-Iron
Material	Inconel 625
Wall Thickness	3-1/16"
Vessel Weight	0.99×10^6 lb
Sodium Weight	0.99×10^6 lb
Steel Balls	26.89×10^6 lb
Diameter	40'
Length	78'
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost (including steel balls)	$ 27,146,100
Sodium Cost	$ 326,700
Total Cost	$ 27,472,800

Table 3.4-13

STORAGE DATA - CASE VIIb, CONCEPT 1

Design Pressure	150 psig
Design Temperature	1300 F
Shape	Cylinder with Hemispherical Head
Total Tankage Volume	39,629 ft^3
Number of Vessels	2
Contents	Sodium-Iron
Material	Inconel 625
Wall Thickness	2-7/8"
Vessel Weight	0.50×10^6 lb
Sodium Weight	0.50×10^6 lb
Steel Balls	13.45×10^6 lb
Diameter	32'
Length	62'
Insulation Type	Kaylo 10
Insulation Thickness	12"
Tank Cost (including steel balls)	$ 28,677,600
Sodium Cost	$ 326,700
Total Cost	$ 29,004,300

Table 3.4-14

STORAGE DATA, CONCEPT 3

	Case IVa	Case IVc*
Design Pressure	50 psi	50 psi
Design Temperature	1100 F	630 F
Shape	Sphere	Sphere
Volume	103,626 ft^3	103,626 ft^3
Number of Vessels	4	4
Contents	Sodium	Sodium
Material	Incoloy 800	2-1/4 Cr - 1 Mo
Diameter	59'	59'
Tank Cost	$ 11,015,840	$ 7,224,124
Sodium Cost	$ 6,864,000	-
Total Cost	$25,103,964	

*Costs scaled from Case IVa.

Table 3.4-15

STORAGE DATA, CONCEPT 3

	Case Va	Case Vc*
Design Pressure	50 psi	50 psi
Design Temperature	1300 F	630 F
Shape	Sphere	Sphere
Volume	69,074 ft^3	$69{,}074^3$
Number of Vessels	4	4
Contents	Sodium	Sodium
Material	Inconel 625	2-1/4 Cr - 1 Mo
Diameter	51'	51'
Tank Cost	$ 11,716,000	$ 5,299,752
Sodium Cost	$ 4,558,950	-
Total Cost	$21,574,702	

*Costs scaled from Case Va

Table 3.4-16

STORAGE DATA, CONCEPT 4

	Case IVd	Case IVe
Design Pressure	50 psi	50 psi
Design Temperature	1100 F	630 F
Shape	Sphere	Sphere
Volume	138,700 ft^3	138,700 ft^3
Number of Vessels	1	1
Contents	Sodium	Sodium
Material	Incoloy 800	2-1/4 Cr - 1 Mo
Diameter	64.6'	64.6'
Tank Cost	$ 3,499,329	$ 2,234,629
Sodium Cost	$ 2,288,000	-
Total Cost	$ 8,021,958	
Cost of Factory Assembled Na + Fe Tanks	$17,722,420	
Total Cost	$25,744,378	

Table 3.4-17

STORAGE DATA, CONCEPT 4

	Case Vd	Case Ve
Design Pressure	50 psi	50 psi
Design Temperature	1300 F	630 F
Shape	Sphere	Sphere
Volume	92,100 ft^3	92,100 ft^3
Number of Vessels	1	1
Contents	Sodium	Sodium
Material	Inconel 625	2-1/4 Cr - 1 Mo
Diameter	56.4'	56.4'
Tank Cost	$ 3,638,039	$ 1,641,368
Sodium Cost	$ 1,519,650	-
Total Cost	$ 6,799,557	
Cost of Factory Assembled Na + Fe Tanks	$15,095,130	
Total Cost	$21,894,687	

3.4.4 PIPING AND VALVE REQUIREMENTS

Piping

Piping sketches for Concepts 1, 2, 3, and 4 are shown in Figures 3.4-17 through 3.4-21. Tables of pipe lengths, fittings, and valves associated with each concept exclusive of the tower were developed. A summary of pipe length, valves, and fittings is shown in Table 3.4-18.

Table 3.4-18

PIPE, VALVE, AND FITTING SUMMARY

Concept	Pipe Length (ft)	Number of Fittings	Valves
1	1195	57	3
2	2175	84	13
3	1305	59	5
4	1755	78	7

The general piping and valve requirements considered in this analysis are specified below.

1. Applicable Codes: ASME Section VIII and ANSI-B31.3 Power Piping Code

2. Valves, piping, and fittings to be trace heated for Na temperature maintenance of >350 F

3. Valves, piping, and fittings to be insulated and covered for weather/ moisture protection to minimize heat loss

4. Design pressure of piping <130 psi in ground loops, 300 psi at base of tower for riser and downcomer piping

5. Flow velocity to be maintained below 25 feet/second

6. Piping/fitting material - 316SS where temperatures exceed 1000 F; 2-1/4 Cr-1 Mo or CS for low temperature regions (<700 F)

Valves

Several large sodium valves are required, the specific quantity depending on the concept selected. Concept 1 requires at least two large 16- or 24-inch sodium valves. Primary regulation of flow would be accomplished by the EM pump. Concept 2 requires the greatest number of valves (at least four in the hot leg and seven in the cold leg). The use of factory fabricated tanks may increase the valve requirements. Grouping tanks together can reduce the use of valves. At least four large sodium valves would be used with Concept 3 (two in the hot leg and two in the cold leg).

Sodium valves require a special seal at the stem to prevent leakage of sodium to the atmosphere. This is commonly done by use of a bellows. In some cases, secondary containment is achieved with a freeze seal. Freeze seal development has been in progress for first line containment. Bellows seals are, however, now recommended because of their widespread use and acceptability.

No 18-inch, 20-inch or 24-inch sodium isolation valves are known to have been built, although check valves of this size are being built by Foster Wheeler for the Clinch River Plant.

The reliability of sodium valves is inherently lower than that of water/steam valves of comparable size. This is particularly true where regulation is required. Sodium valving should therefore be kept to a minimum wherever unless redundancy is feasible. Concepts 1 and 3 have a decided advantage over Concept 2 when considering the number of sodium valves required.

Trace Heating

The liquid metal loop valves and containment vessels require electrical trace heating so that tanks and piping can be preheated prior to filling and maintained at a minimum temperature of 350 F during standby conditions. It is expected that the loops would only be drained during extended downtimes or when the removal of sodium was required for maintenance purposes.

Tubular resistance heaters are commonly applied to piping. In most cases a single heater is considered sufficient. A run of $\sim$17 feet can be accommodated by a single trace heater mounted along or close to the bottom of the pipe. If a double run is necessary, the heaters could be spaced 60° to 90° apart, dividing the vertical centerline of the pipe as shown in Figure 3.4-22. The most effective but most costly method of application uses an annulus around the pipe in which the heater is placed near the bottom and the heat is transferred around the annulus to the entire pipe circumference. The heater also provides for sodium-to-gas leak detection. The more conventional method is to install the tubular heater on the pipe wall and support it along its entire length with a tack-welded sheath as shown in Figure 3.4-22. Tubular heaters of this type are designed for 375 watts/linear foot and are normally operated at $\sim$125 watts/linear foot to assure long life.

Trace heating on tanks can be applied several ways. The spacing between heater strips is dependent on tank volume, wall thickness, and temperature requirements. Tanks with a two-inch wall may have a heater spacing of up to 48 inches. A tank with a one-inch wall may have a spacing of 32 inches. Typical applications are shown in Figure 3.4-23.

Thermocouples can be placed near the heaters to verify heater operation and surface temperature. A spacing of >20 feet is typical in piping.

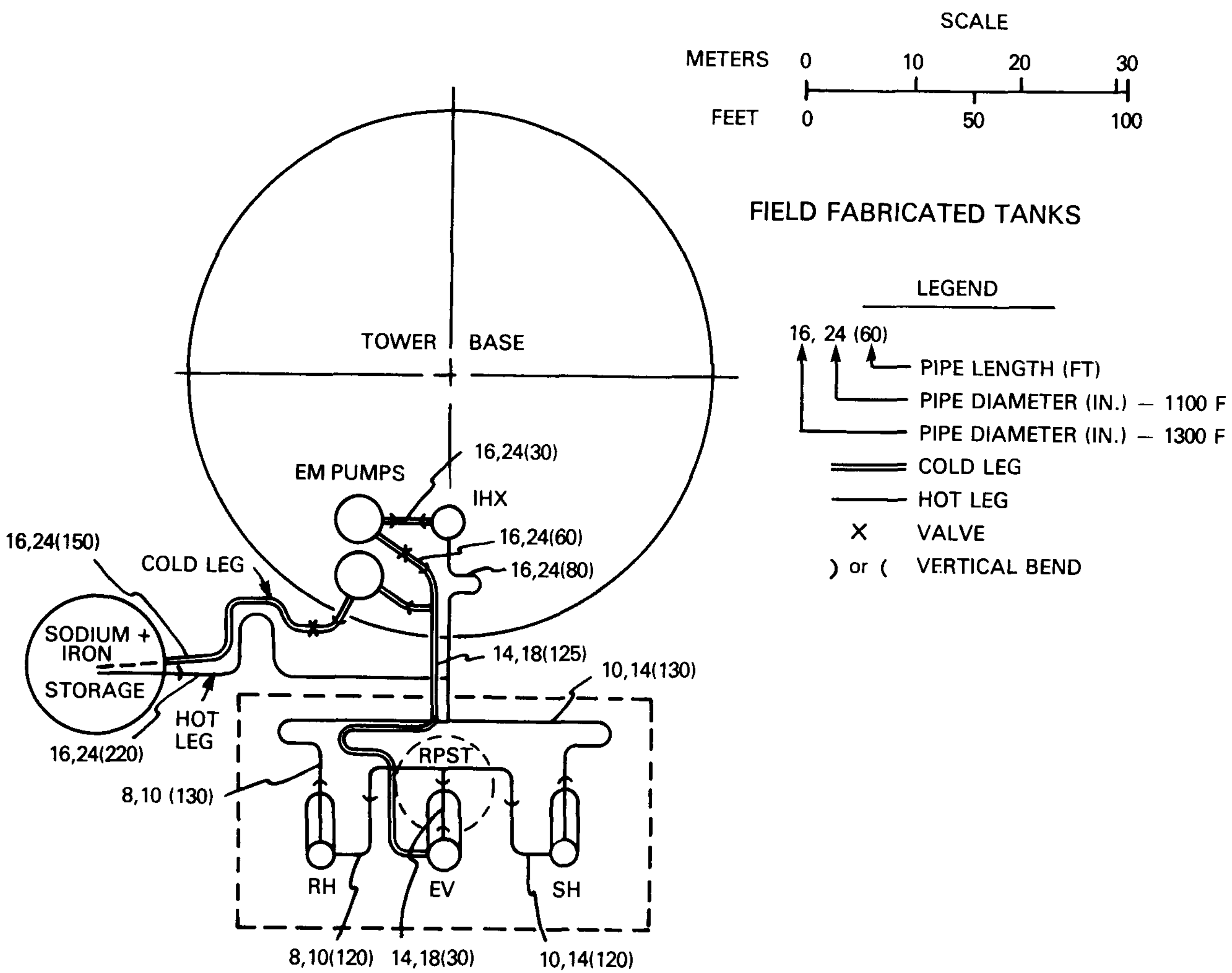


Figure 3.4-17. Piping Layout - Storage Concept 1

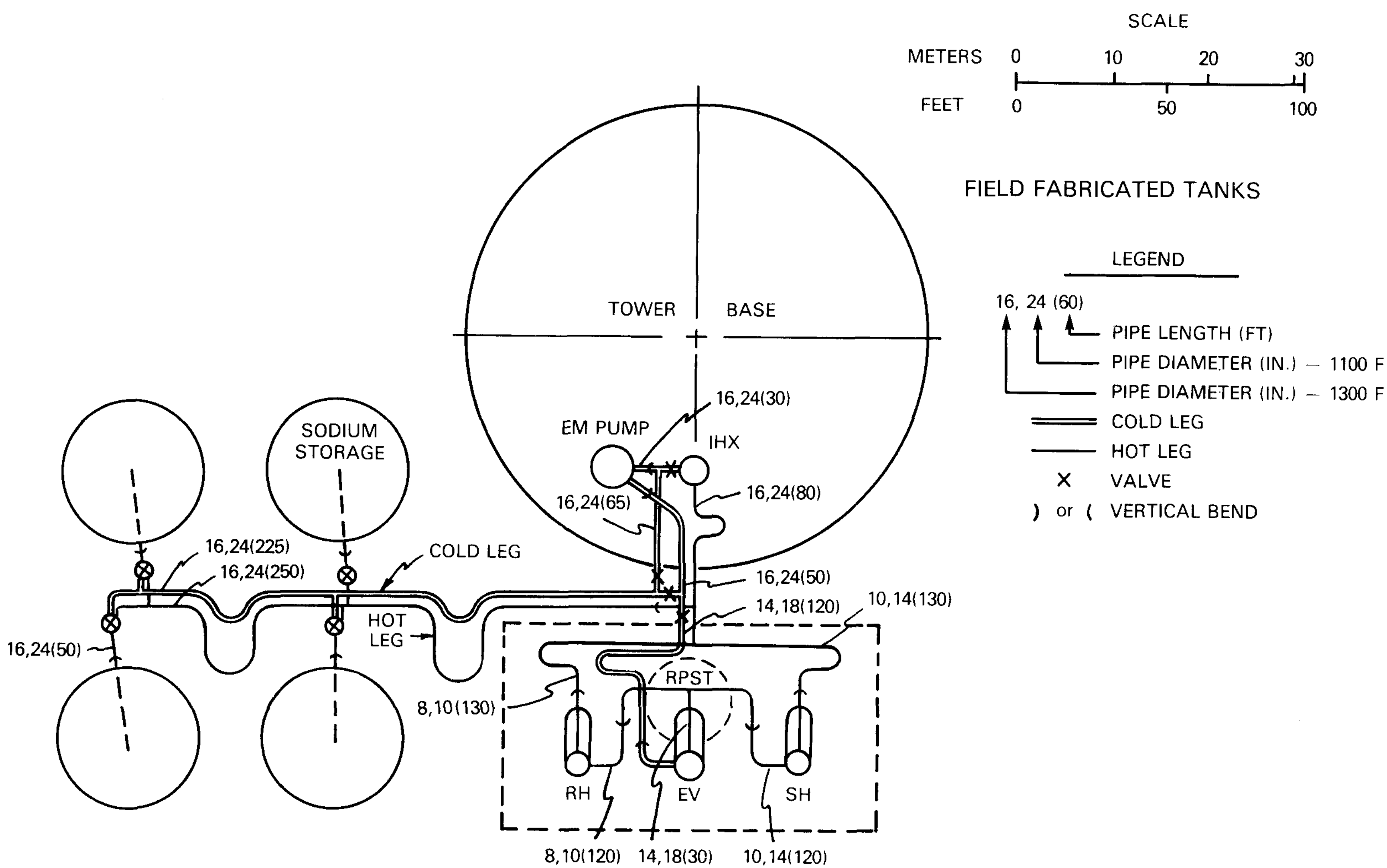


Figure 3.4-18. Piping Layout - Storage Concept 2

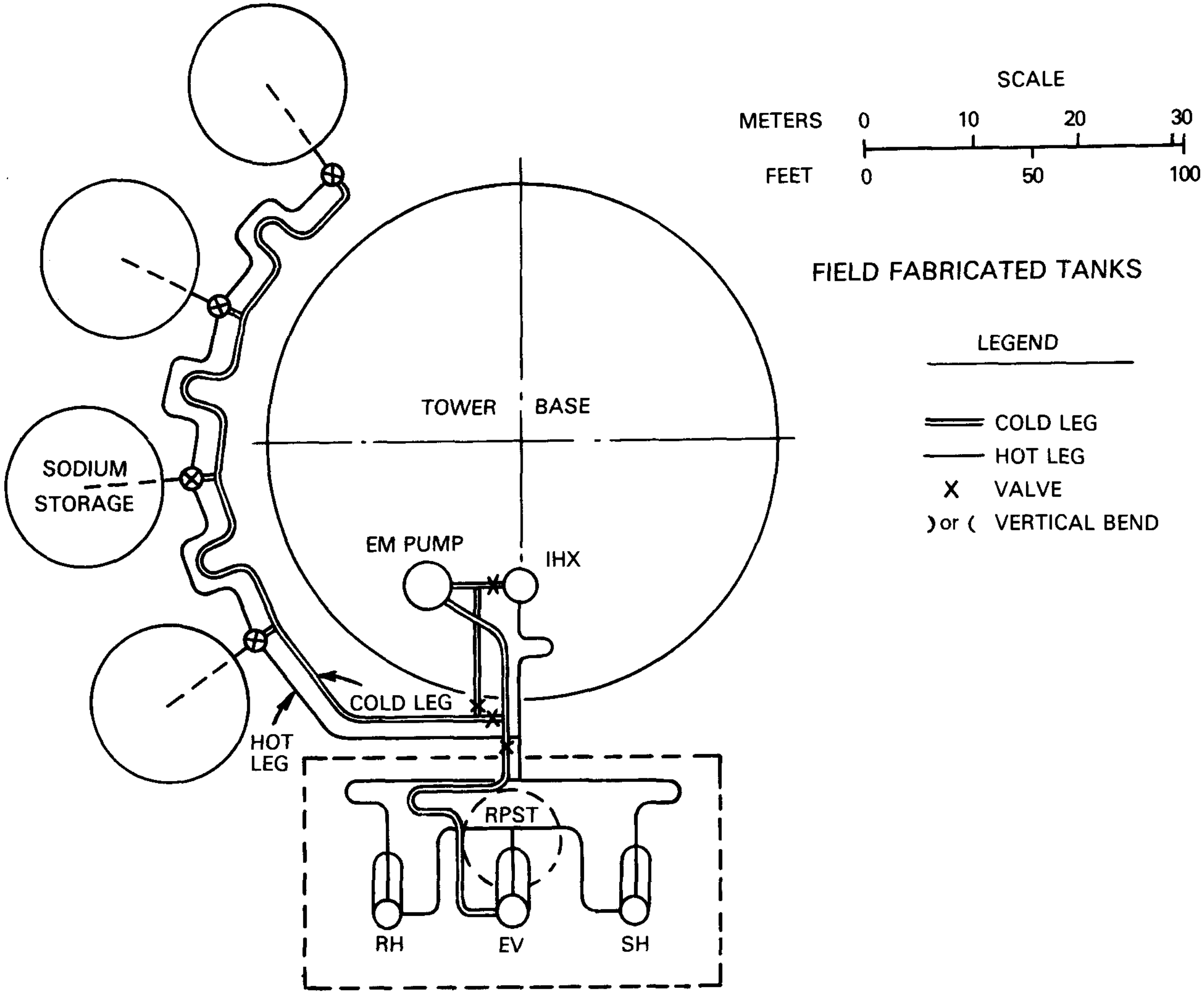


Figure 3.4-19. Piping Layout - Storage Concept 2a

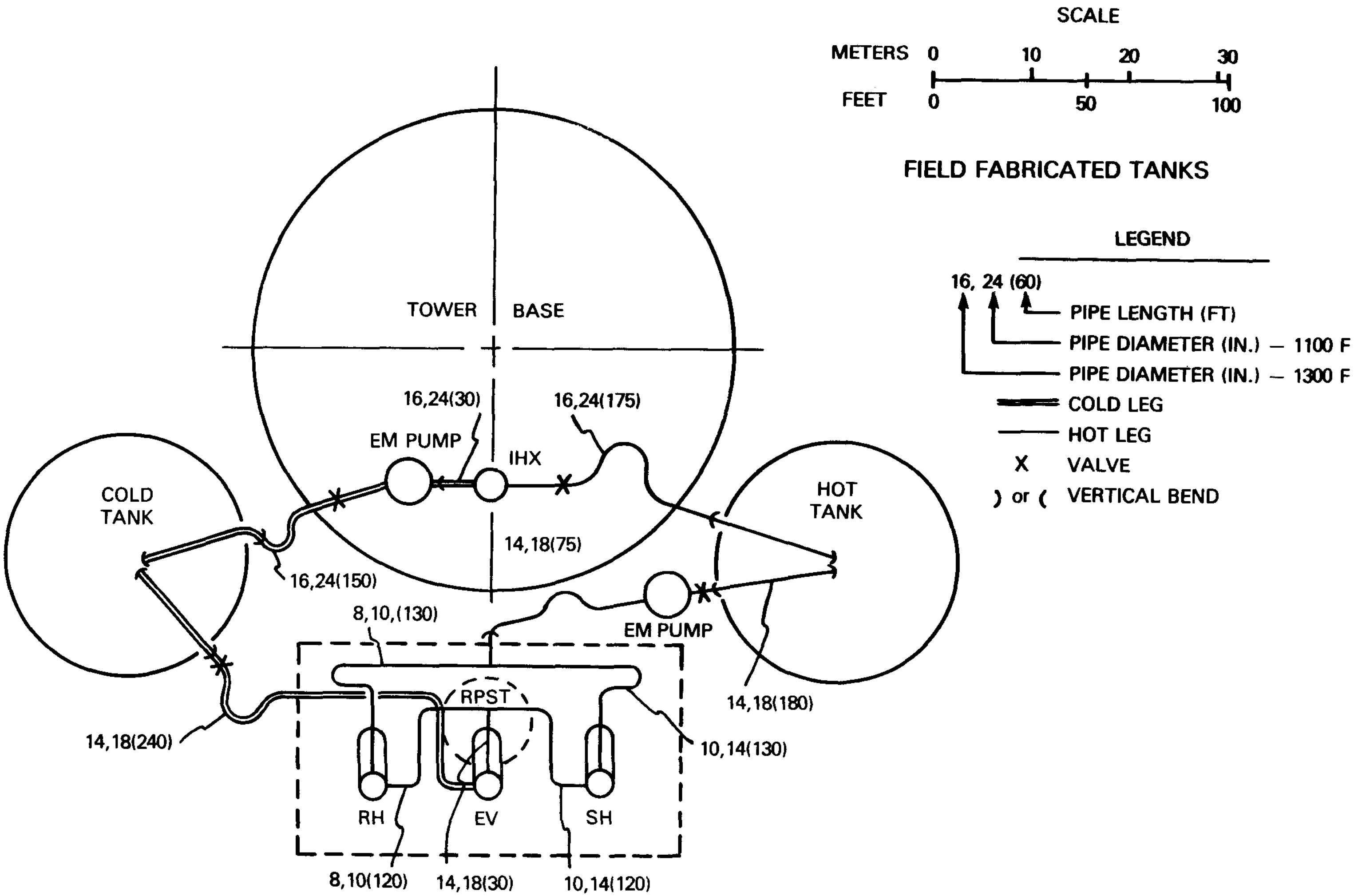


Figure 3.4-20. Piping Layout - Storage Concept 3

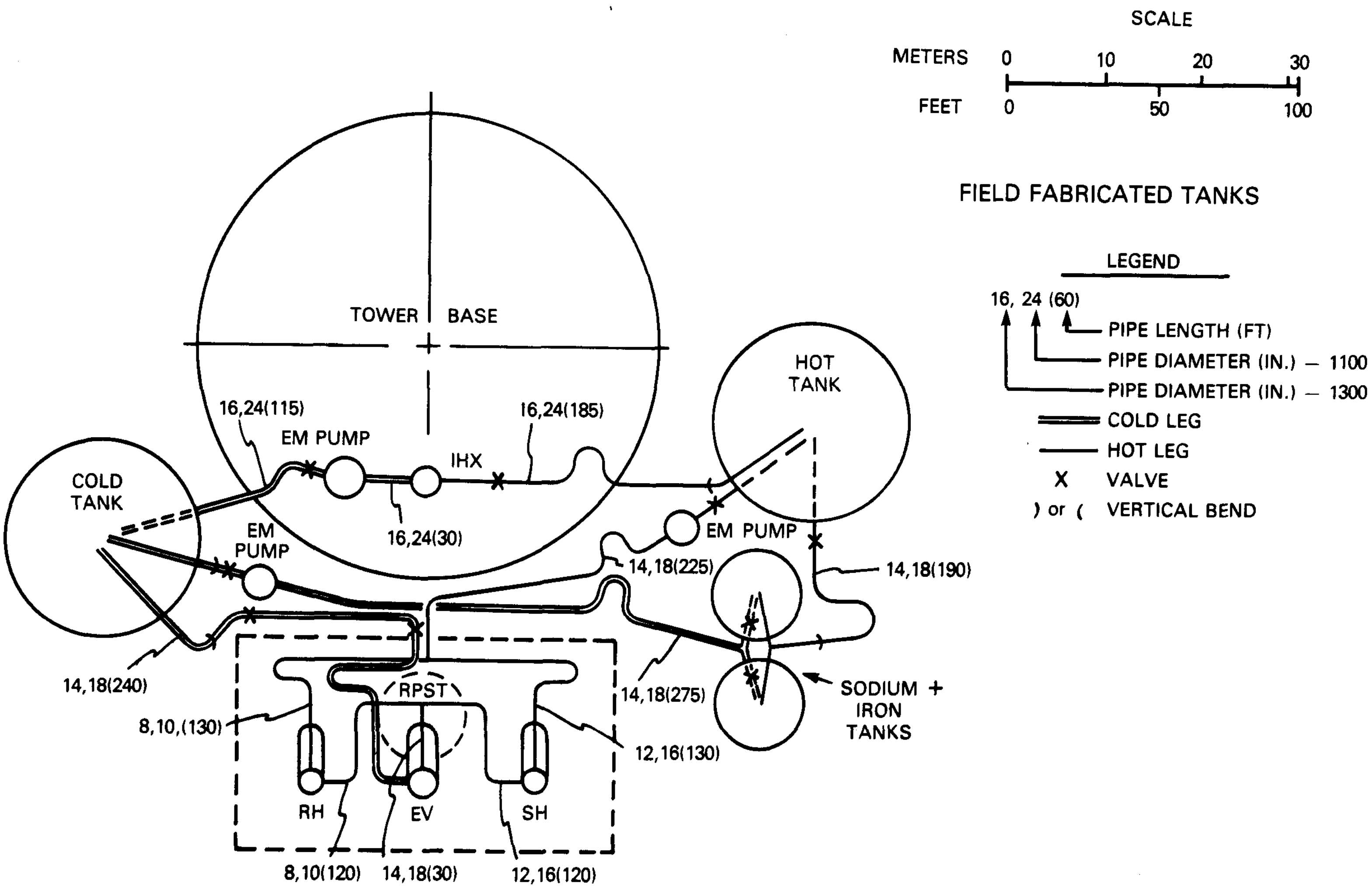


Figure 3.4-21. Piping Layout - Storage Concept 4

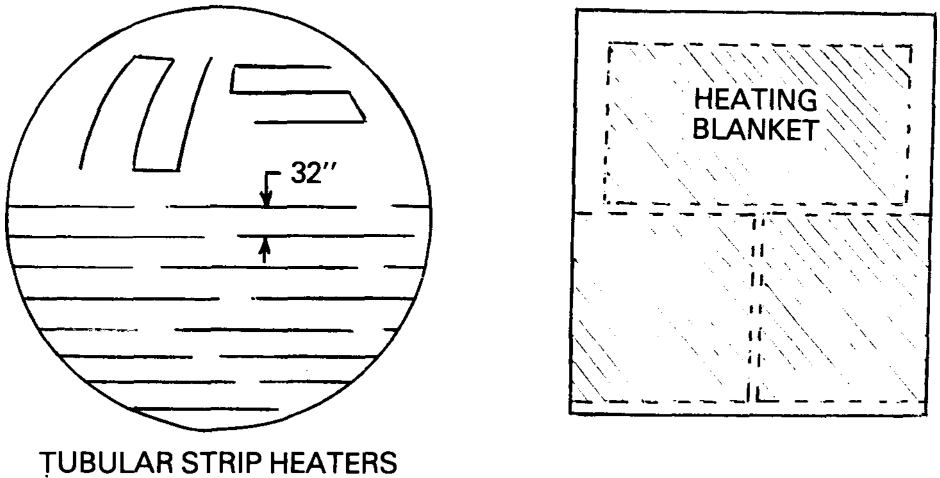


Figure 3.4-22. Trace Heating on Piping

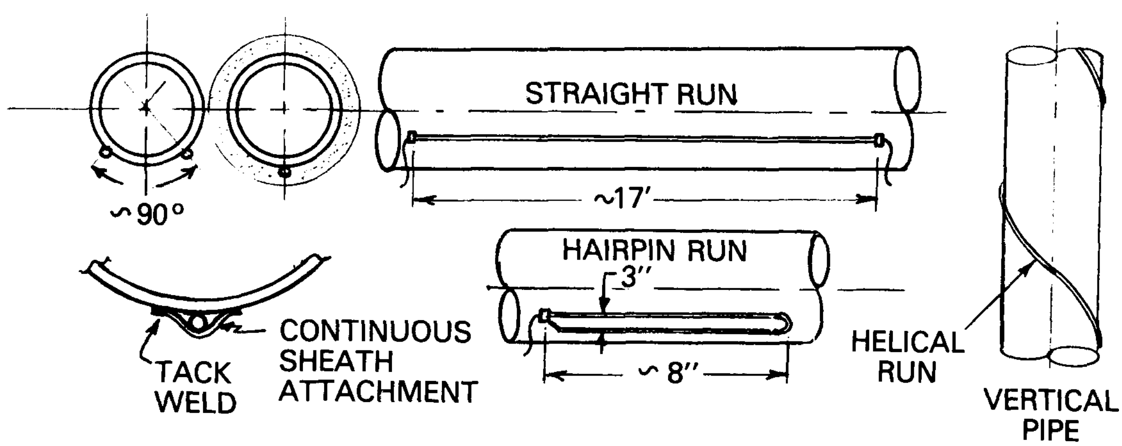


Figure 3.4-23. Trace Heating on Tanks

3.4.5 STORAGE PERFORMANCE ANALYSIS

Insulation Losses From Storage

The system will cycle each day, and, during periods of maintenance or extended cloud cover, downtime may last several days. Nominal storage time is assumed to be 12 hours. For purposes of the parametric study, storage time of 24 hours was used, and an acceptable energy loss of less than one percent was set as an objective; this translates into a loss of less than 6°F for the 1300°F cases.

A series of calculations was performed to determine the energy loss and associated insulation thickness requirements for the various tank configurations. The surface temperature of the insulation was to be maintained at or below the 140°F OSHA standard. The equations used were as follows:

$$\text{temperature of surface } T_S = \frac{T_A\,(h_C + h_R) + \frac{K}{\delta}\,T_{Na}}{(h_C + h_R) + \frac{K}{\delta}} \quad ({}^{\circ}F) \qquad (3.4\text{-}1)$$

$$\text{where } h_c = \frac{0.025\,(V\rho)^{0.58}\,(1 + 0.000576t)}{do^{0.42}} \quad \text{where } t = \frac{T_A + T_S}{2}$$

$$h_R = \sigma\,\varepsilon\,[(T_S + 460)^2 + (T_A + 460)^2]\,[T_S + T_A + 920]$$

$$\text{energy loss } Q + \frac{k}{\delta}\,A_S\,(T_{Na} - T_S) \quad (\text{Btu/hr}) \qquad (3.4\text{-}2)$$

$$\text{temperature loss, } \Delta T = \frac{Q}{MC_p} \quad ({}^{\circ}F/\text{hr}) \qquad (3.4\text{-}3)$$

where
- K = thermal conductivity of insulation (Btu ft/hr^2 °F)
- T_{Na} = temperature of sodium inside tank (°F)
- T_A = ambient air temperature (°F)
- δ = insulation thickness (ft)
- V = air velocity (ft/hr)
- ρ = air density (lb/ft^3)
- do = tank OD
- ε = surface emissivity (0.04 aluminium, 0.90 plaster)
- σ = Stefan Boltzmann constant
- M = mass of tank and contents (lbs)
- C_p = specific heat of tank and contents

Energy loss calculations were made for outside temperatures (T_A) of 0, 32, 65, and 100 F; wind velocities of 0, 8, 40, and 100 mph were used. A summary of the data for the 8 mph and 40 mph, 65 F condition for 1300 F (1260 F sodium temperature) field fabricated tanks is contained in Table 3.4-19. Aluminum and plaster/fiber wrapped insulation were considered. The temperature drop was found to be about 100 percent higher for the plaster/fiber wrap vs. the aluminum. Aluminum or steel also provides a more durable surface for outside application, and, even though initial capital costs are somewhat higher, the aluminum wrapped jacket was determined to be the most cost effective. Results show that the use of 15 or more inches of good quality insulation will keep losses within acceptable limits.

Insulation Requirements and Selection

There are many insulations available for piping and tankage. Requirements established for the Advanced Central Receiver System application are as follows:

- Application temperature: <1100 F, <1300 F
- Thermal conductivity: <1.0 Btu-in./ft^2-hr-^{o}F at mean temperature of 700 F
- Compatible (corrosion, chemical) with 2-1/4 Cr - 1 Mo, 316SS, Incoloy 800, and Inconel 625 containing sodium
- Minimum sag, separation, and settling
- Unaffected by water/moisture
- Suitable for piping and unusual shapes

Five typical insulation materials have been identified:

Kaylo 10 (calcium silicate) <1200 F, Owens Corning

Kaowool (alumina silicate) <2300 F, Babcock & Wilcox

Thermo-12 (calcium silicate) <1500 F, Johns-Manville

Thermal (glass wool) <1000 F, Owens Corning

Holmes Hill Block (mineral fiber) <1900 F, Babcock & Wilcox

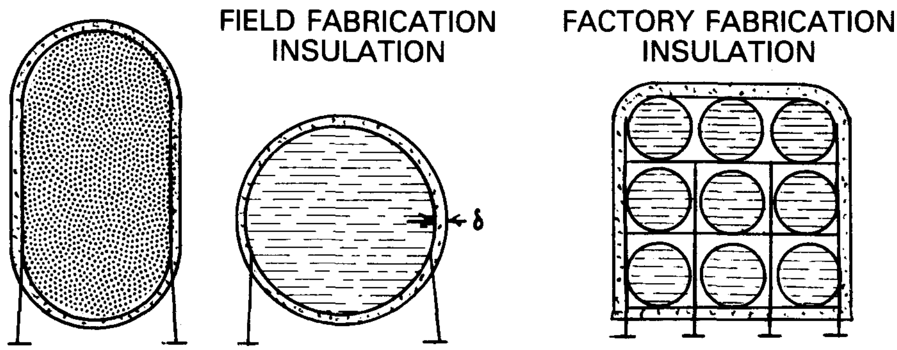
The thermal conductivities of these insulations at mean temperatures of 400, 600, and 800 F are given in Table 3.4-20. Comparative vendor supplied cost data is also provided. Costs have been normalized to account for the differences in thermal conductivity.

The calcium silicate insulations are candidates and should be used for high temperature applications not exceeding 1200^{o}F (the exception being Thermo-12). Kaylo 10 is a candidate for applications where semi-rigid or rigid insulation can be accepted. Kaylo 10 is not manufactured in blanket form, but it can be purchased in scored block form when large pipes (>36-inch OD) and vessels are to be insulated.

Thermo-12 is also a calcium silicate insulation and is rated at a use temperature of up to 1500^{o}F. According to the technical data, it has a very low thermal conductivity even at the high temperatures. It is now available in scored block and can be purchased with or without an aluminum or stainless steel jacket.

Table 3.4-19

ENERGY LOSS SUMMARY FOR 8 mph and 40 mph, 65 F



		* δ = 15.8 INCHES			* δ = 6 INCHES			
	$h_C + h_R$	Ts' °F	Q'x10⁶ Btu/Hr	ΔT' °F/Hr	Ts'' °F	Q''x10⁶ Btu/Hr	ΔT'' °F/Hr	Wind Vel.
CONCEPT 1 1300°F	A 0.930	137.4	0.938	0.19	250	2.392	0.48	8 MPH
	P 2.000	99.8	0.970	0.20	159	2.607	0.52	
CONCEPT 2	A 0.930	137.4	0.710	0.62	250	1.810	1.59	
	P 2.000	99.8	0.740	0.65	159	1.980	1.74	
CONCEPT 1 1300°F	A 1.303	117.6	0.955	0.19	203	2.503	0.50	40MPH
	P 2.373	94.5	0.974	0.20	145	2.640	0.53	
CONCEPT 2 1300°F	A 1.303	117.6	0.730	0.64	203	1.900	1.67	
	P 2.373	94.5	0.740	0.65	145	2.010	1.76	

* δ = INSULATION THICKNESS
A = ALUMINUM WRAPPED INSULATION
P = PLASTER/FIBER WRAPPED INSULATION
INSULATION — KAOWOOL DENSITY 8#, K = .08333 AT 1260°F
DATA CALCULATED FOR AN INDIVIDUAL TANK A_S = 13930 FT² CONCEPT 1
A_S = 10527 FT² CONCEPT 2

Table 3.4-20

INSULATION COST COMPARABLE DATA

Type		Density	Thermal Conductivity at Mean Temp. Btu/hr sq ft°F/in 400°F	600°F	800°F	Max. Use Temp.	*Cost Data (1978$) $/lin.ft 24" Pipe 2" Layer	** Normalized Costs $/lin. ft.	Ratio
Kaylo-10	Pipe	12.5pcf	.47	.59	~.90	1200°F	= $12.85	$19.40	1.06
	Block	12.5pcf	.47	.59	~.90	1200°F	2.5x7.37 = $18.42	$27.81	1.52
Thermal Wool	Type 1	NA	.76	NA	-	1000°F	0.08x7.37 = $ 0.59	-	-
	Type 2	NA	.48	.78	-	1000°F	0.144x7.37 = $ 1.06	$ 2.12	.12
Thermo-12	Pipe	13pcf	.48	.57	~.72	1500°F	= $13.00	$18.98	1.03
	Block	?				1500°F	1.61x7.37 = $11.87	$17.33	.94
Kaowool	Blanket	4pcf	.35	.50	.72	2300°F	1.31/ft²x7.37= $ 9.65	$12.35	0.67
	Blanket	6pcf	.30	.44	.62	2300°F	1.82/ft²x7.37= $13.41	$15.15	0.83
	Blanket	8pcf	.27	.39	.53	2300°F	2.49/ft²x7.37= $18.35	$18.31	1.0
	Block	14-18pcf	.40	.48	.57	2300°F	8.28/ft²x7.37= $61.02	$75.05	4.09
Holmes Hill Block (Mineral Fiber)		13-17pcf	.50	.60	.74	1900°F	1.40/ft²x7.37= $10.32	$15.89	0.87

* Price based on high volume order

** Normalized $ based on equivalent thermal conductivity at 600°F

Ratio based on Kaowool 8 pcf blanket = 1.0

Note: Aluminum metal jacket cost ~$0.38 per ft^2, ~3.00 per linear ft of 24" pipe.

An alternative high temperature insulation which can be purchased in blanket form and also in molded shapes is an alumina silicate called Kaowool. The advantage of this insulation is that in blanket form it lends itself readily to curved surfaces. A few years ago, it principal disadvantage was its high cost--about 2 to 3 times the cost of the calcium silicates such as Kaylo 10. However, due to supply and demand, the cost of Kaowool blanket has been reduced in the past year; it is now less expensive than Kaylo 10. The maximum blanket thickness available is two inches, and several layers would be required for the Advanced Central Receiver System. The alumina silicates have a very low thermal conductivity; an eight-pound-density blanket provides $\sim$0.5 Btu/in./hr/ft^2-^{o}F at a mean temperature of 700 F (mean temperature is the midpoint between the inside pipe and insulation outer surface, e.g. 1250 F inside, 140 F outside - 700 F mean temperature).

Kaowool can also be purchased in block or molded block form. The molded block form is used for wrapping valves and tees. The maximum available cast/molded thickness is four inches, although if large volumes were foreseen the company would modify its equipment to handle the larger sizes. The company suggests using the ceramic fiber blanket for this application. The blanket could be wrapped around the pipe and unusual shapes and held in place by bands and/or metal jackets. The blanket would have lower cost than a vacuum cast sleeve.

In terms of material cost, Kaowool 4-pcf blanket appears to be quite cost effective. However, it is a very light material compared to the molded block material such as Kaylo 10 and Thermo 12 and therefore may be more susceptible to crushing.

Thermal Wool is listed as a candidate because of its low cost. Thermal Wool Type II blanket would be suitable for the low temperature vessels and in the outer layers of insulation for high temperature applications. Thermal Wool is a fiberglass insulation whose bonding material will maintain bond strength up to 1000 F. Its thermal conductivity increases rapidly with temperature. However, at the lower temperatures associated with the cold leg ($\sim$630 F), the mean temperature would be <400 F and Thermal Wool Type II conductivities would be less than 0.50 Btu/hr-ft^2-^{o}F/in. Its low cost (nearly an order of magnitude lower) and low thermal conductivity at the cold leg temperatures offers considerable economic advantages.

Typical insulation applications are shown in Figure 3.4-24. Maximum batt or roll thickness of Type II Thermal Wool is three inches.

Holmes Hill Block, manufactured by Babcock & Wilcox, was included in the list of candidates to show the cost and applicability of relatively low cost block type insulation. It has application at high temperatures up to 1900 F but due to its outgassing and potential flammability and the irritability of its water repellent additive, its use is not recommended for this application.

Conclusions. The following conclusions were reached regarding the insulation requirements and selection:

1. Block and molded insulation is generally two to three times as costly as blanket material.
2. Blanket material is well suited to pipe and tank insulation; however, multiple layers will require supports/stand-offs.

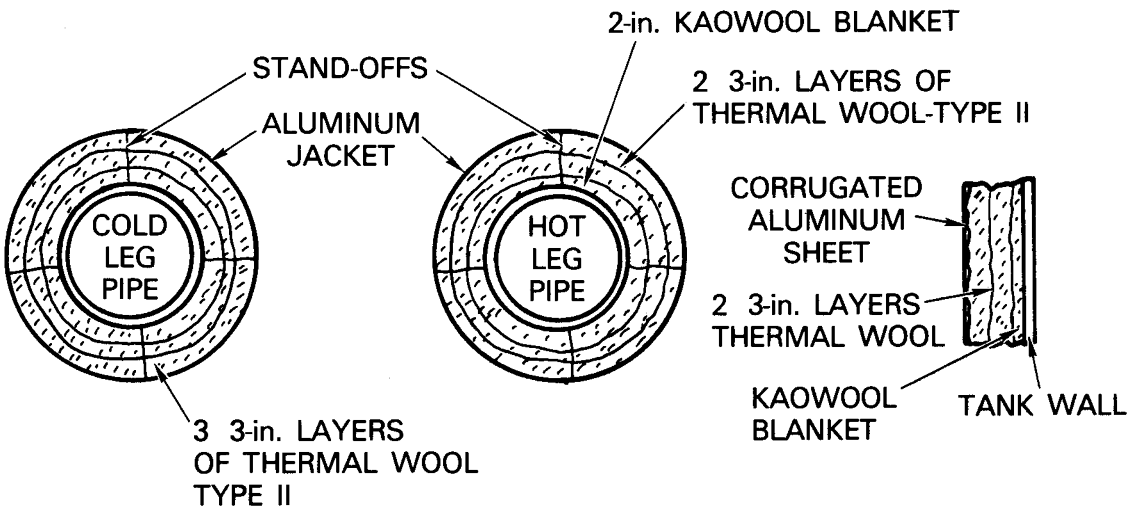


Figure 3.4-24. Insulation Application (Pipes and Vessel Walls)

3. Thermo-12 molded insulation is reasonable in cost; however, where multiple layers are required on the large pipes, scored block material may be necessary.

4. Use of Thermal Wool fiberglass has considerable cost saving potential. An inner layer of high temperature insulation such as Kaowool could be used in conjunction with the Thermal Wool Type II.

5. It is more economical to wrap valves and fittings with a blanket insulation rather than the vacuum molded type.

Recommendation. Wherever possible, use Thermal Wool (up to 1000 F) at a cost of ∿\$0.15/square foot. Where a higher temperature blanket is required, Kaowool can be used at a cost of ∿\$1.30 to \$2.50/square foot. An aluminum jacket would be installed around all insulation at a material cost of ∿\$0.38/square foot.

Thermocline Losses in Sodium-Iron Storage

To estimate thermocline losses of Concepts 1 and 4, the tank outlet temperature was approximated by the curve in Figure 3.4-25. The steam cycle can continue to produce power efficiently until the sodium temperature drops to T_S = 850 F. Thus, the small shaded triangle in Figure 3.4-25 represents the energy in storage which is not recovered due to the thermocline effect. The fraction of energy lost is the ratio of the area of this triangle (a) to the area (A) between $T = T_L$ and the thermocline curve. Using the straight line approximation and triangle similarity arguments, it can be estimated that

$$\frac{a}{A} = \left(\frac{T_S - T_L}{T_H - T_L}\right)^2 \left(\frac{\Delta\tau}{\tau}\right) \qquad (3.4\text{-}4)$$

Based on the detailed analysis of axial conduction and convection effects in thermocline behavior in Appendix F, it was estimated that $(2\Delta\tau/\tau) = 0.174$ for a typical sodium +/iron storage tank. The fraction of energy lost shown in Table 3.4-21 under Conduction and Convection (a/A) is calculated from this estimate. In addition, there may also be local turbulence in the sodium flow which would also tend to spread the thermocline, At this time there is no model for this effect; it has been accounted for here by roughly doubling the convection and conduction loss estimates.

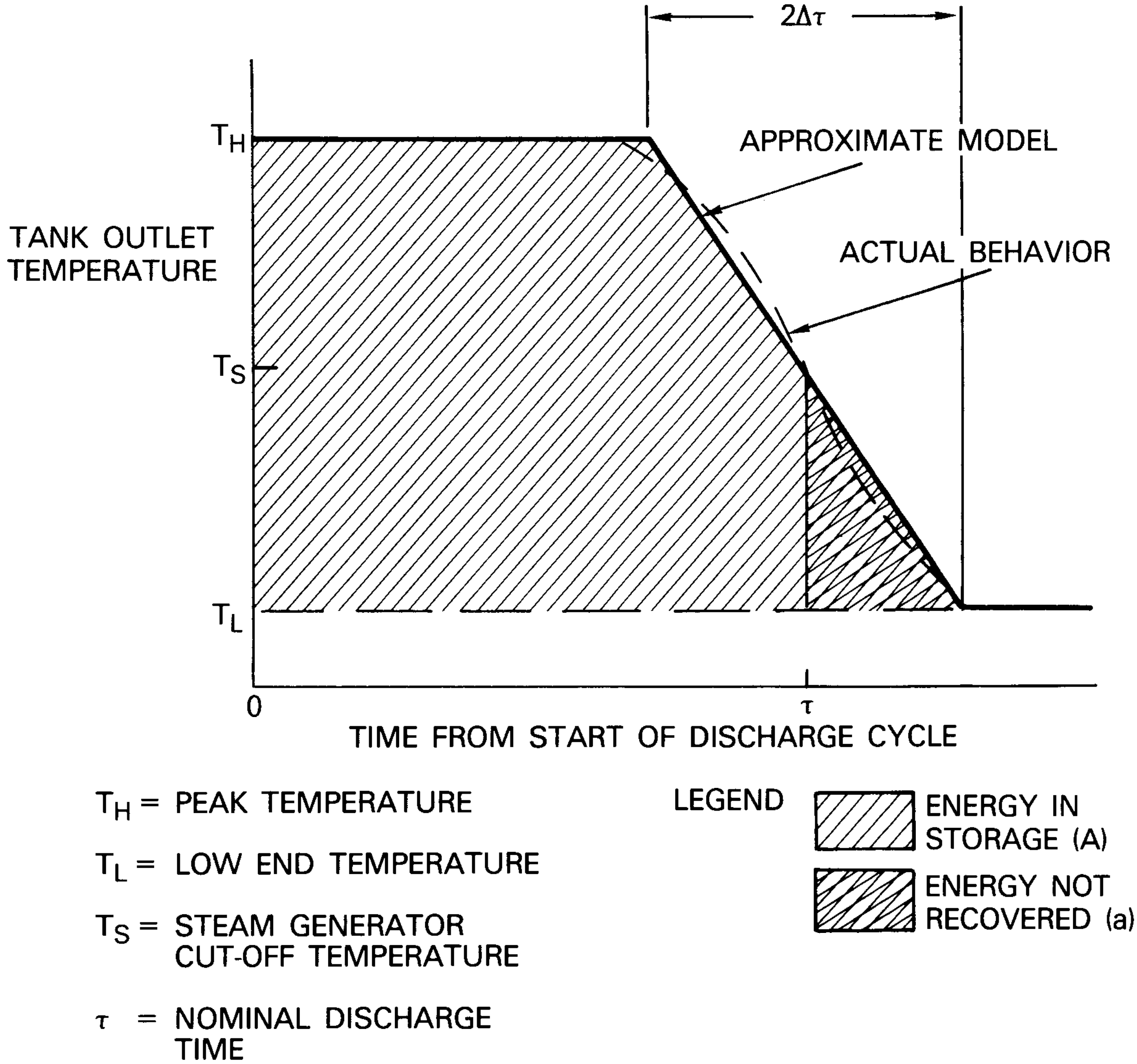


Figure 3.4-25. Thermocline Loss Model

Table 3.4-21

ENERGY LOSSES FROM THERMOCLINE STORAGE

			Fraction of Energy Lost (%)	
T_H (^oF)	T_L (^oF)	T_S (^oF)	Conduction and Convection (a/A)	Total*
1260	630	850	1.1	3
1050	630	850	2.4	5

*Includes effect of flow maldistribution

3.4.6 HEAT EXCHANGER ANALYSIS

Heat Exchanger Equipment

All sodium heat exchanger modules are of the vertically mounted hockey stick type, which uses the hockey stick bend in the pressure vessel to absorb differential expansion between individual tubes and the pressure vessel. Figure 3.4-26 shows an isometric view of the general heat exchanger configuration. Sodium enters at the top inlet nozzle on the side of the vessel, flows up to and through the windows cut in the side of the tube bundle shroud, reverses 180°, and flows down around the tubes in parallel counterflow with the tubes. At the bottom, the sodium exits the tube bundle via a second set of windows in the tube bundle shroud and flows out of the module through an outlet nozzle in the side of the vessel.

The tube-side flow enters through a nozzle in the lower plenum, passes through the tubes, and exits at the top into an outlet plenum and through an outlet nozzle. The tubes are supported in the active flow region with drilled support plates which are attached firmly to the flow shroud. The tubes are arranged in a triangular pitch array with a flow hole drilled between each set of three tubes. In addition to holding the tube support plates in position, the flow shroud directs the flow in the tube region with a minimum of bypass flow. In the hockey stick bend region, slat type tube supports are used which allow vertical movement of the tubes for accommodating thermal expansion, while horizontal movement of the tubes is restrained by the slats.

Special thermal baffles are installed at the tubesheets, these baffles consist of several stacked steel plates. They absorb thermal transients in the sodium and protect the tubesheet from high stresses induced by rapid thermal transients.

Two percent of the flow is bypassed up through the hockey stick region, which is otherwise an inactive heat transfer area, in order to provide a flow sweep of the tubesheet region in case a leak develops. This bypass flow exits through a bleed vent nozzle in the hockey stick head and passes on into a leak detector. Since a leak of high pressure water into sodium produces hydrogen and sodium oxides, two types of small leak detectors are used, a hydrogen diffusion tube and an oxygen meter. The main sodium stream leaving the module is also monitored with an identical small leak detection system. Large leaks (>0.1 pound/second) generate sufficient heat and pressure (hydrogen and steam) that the pressure vessel protection system is activated. This consists of a rupture disc mounted on the module outlet line close to the outlet nozzle. These rupture discs are typically set to rupture at around 350 psi. The relief flow is piped to a reaction products tank where the hydrogen gas is vented through a flare stack and burned. The solid and liquid products fall into the bottom of the tank.

Sodium-water reactions require costly shutdown and repair. Therefore, everything possible is done in the design of steam generators to reduce the probability of a leak. The tube and tubesheet materials are of high quality vacuum arc remelt (VAR) and electro-slag remelt (ESR) manufacture with complete QA procedures at each step of the process. It is believed that the development of tubing material has reached such excellence that through-wall leaks have an extremely low probability.

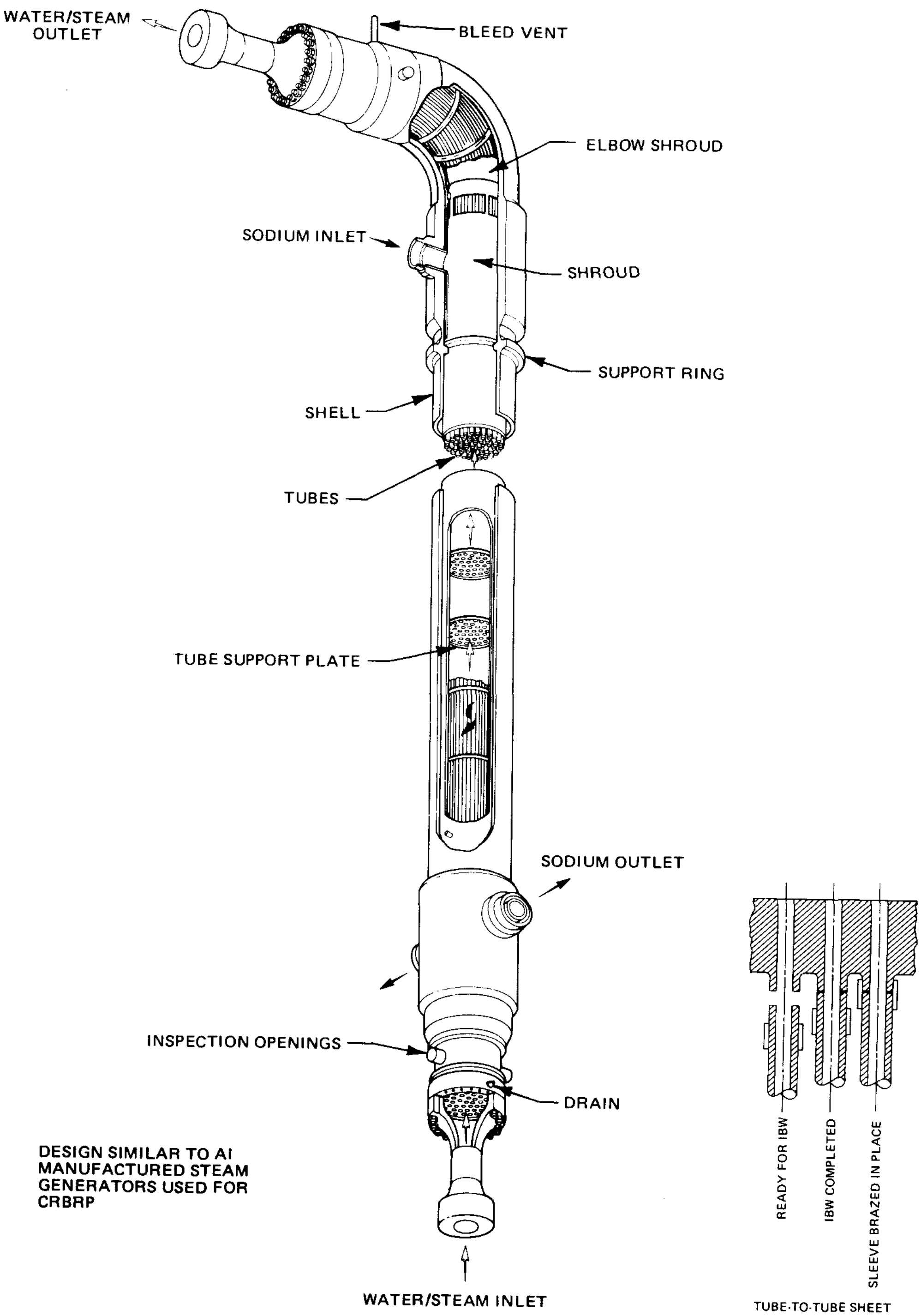


Figure 3.4-26. Steam Generator Module

Producing perfect tube welds is more difficult. The tube weld used is detailed in Figure 3.4-26. A tube boss is machined on the back-face of the tubesheet and the tube is butt welded to the boss using an internal bore weld (IBW) applied with a specially developed, automatically programed welder. When completed, each weld is subjected to the most exacting QC procedures . Of major importance in this inspection program is a special Picker X-ray tube which inserts into the tubesheet hole and takes a 360° image of the weld. The major advantage of the IBW weld is the excellent configuration which it provides for these X-ray inspections.

But in spite of the high quality possible with the IBW welds and QC procedures, it is this region of the heat exchanger where most leaks are expected to occur. Therefore General Electric has proposed a backup device for the weld. After satisfactorily passing inspection, the IBW weld is dressed down at the tube OD to permit the sliding of a short sleeve up over the weld. The sleeve is then brazed in place to provide a second leak barrier. This procedure is illustrated in Figure 3.4-26. It is expected that such a steam generator design will be virtually 100 percent leakproof over the life of the plant. This is of great importance since a faulty steam generator design can easily prevent a plant from providing economical power due to unavailability associated with shutdown, leak location, leak repair, damage assessment, and requalification of the steam generator module involved.

Steam Generator System Configuration

The steam generator system consists of an evaporator module operating in the feedwater recirculation mode with a steam drum, a superheater module which superheats the saturated steam from the drum, and a reheater module. Consideration was given to the idea of using a simple, less expensive once-through steam generator in place of the separate evaporator and superheater modules. However, further study of the plant duty cycle indicated that the once-through concept would be very difficult to shut down and restart, whereas the recirculating steam drum can be held in a hot standby mode overnight and is ready to be used the next morning. Thermal storage in the steam drum also helps in riding through turbine transients. Additional steam is always immediately available through flashing off the drum, whereas the once-through system must activate a control system and takes time to generate the steam.

The selected steam generator system configuration is illustrated in Figures 3.4-27 and 3.4-28. Note that ten percent of the flow is removed from the steam drum and blown down through regenerative heaters to the condenser where it is mixed with condensate and returned. Full flow demineralizers process all water leaving the condenser plus the ten percent blow-down flow and return it to the steam generator at the highest water quality possible to avoid any water chemistry problems in the steam generators which might lead to tube leaks.

The steam generators are supplied with hot sodium from storage (Figures 3.4-27 and 3.4-28). After extracting the heat from the hot sodium, the sodium is returned to cold storage. A large electromagnetic (EM) pump is used to circulate the sodium in the steam generator subsystem.

A constant speed recirculation pump is required to maintain flow from the steam drum to the higher pressure evaporator inlet. The ratio between the

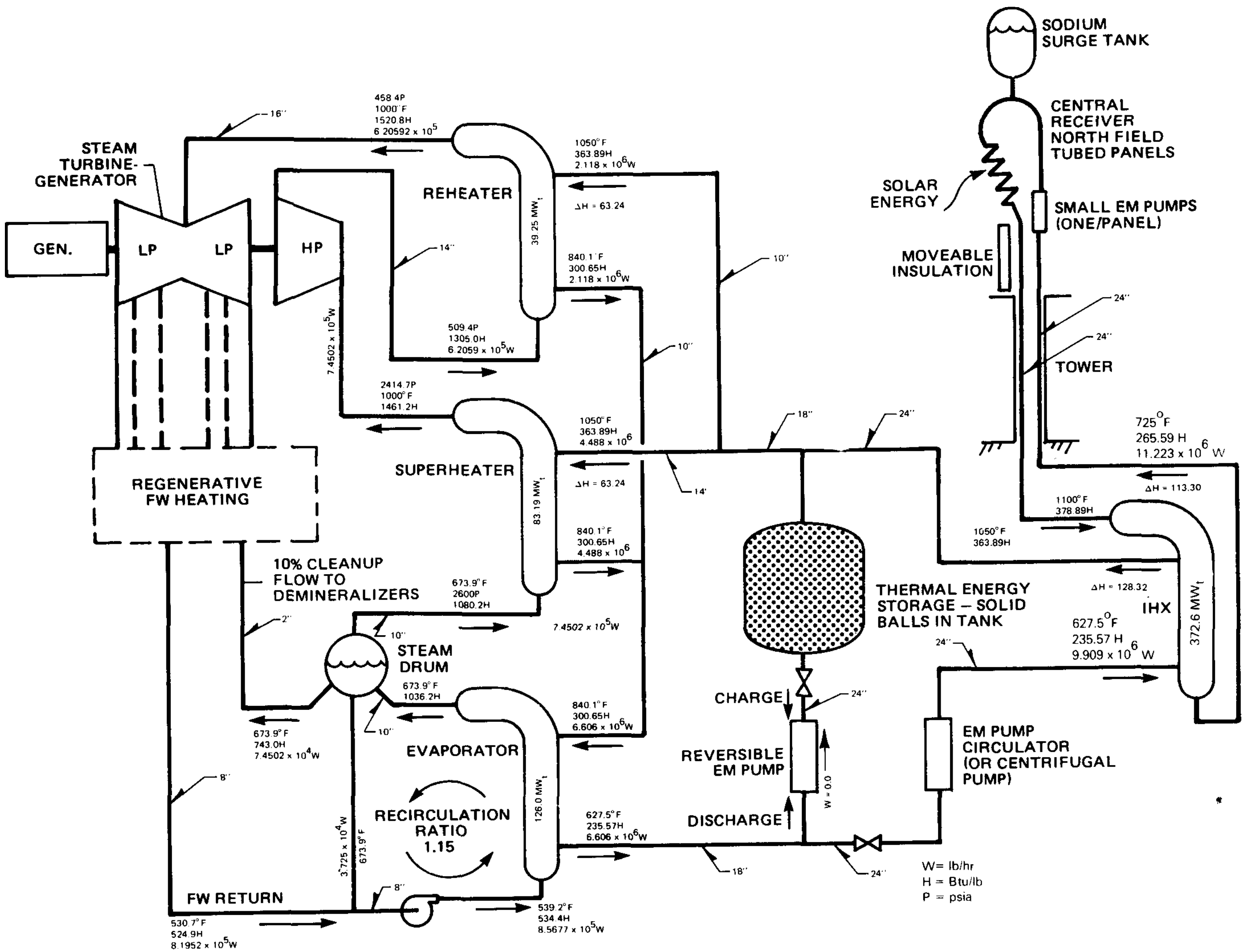


Figure 3.4-27. Reference Advanced Central Receiver Power System-1100 F System

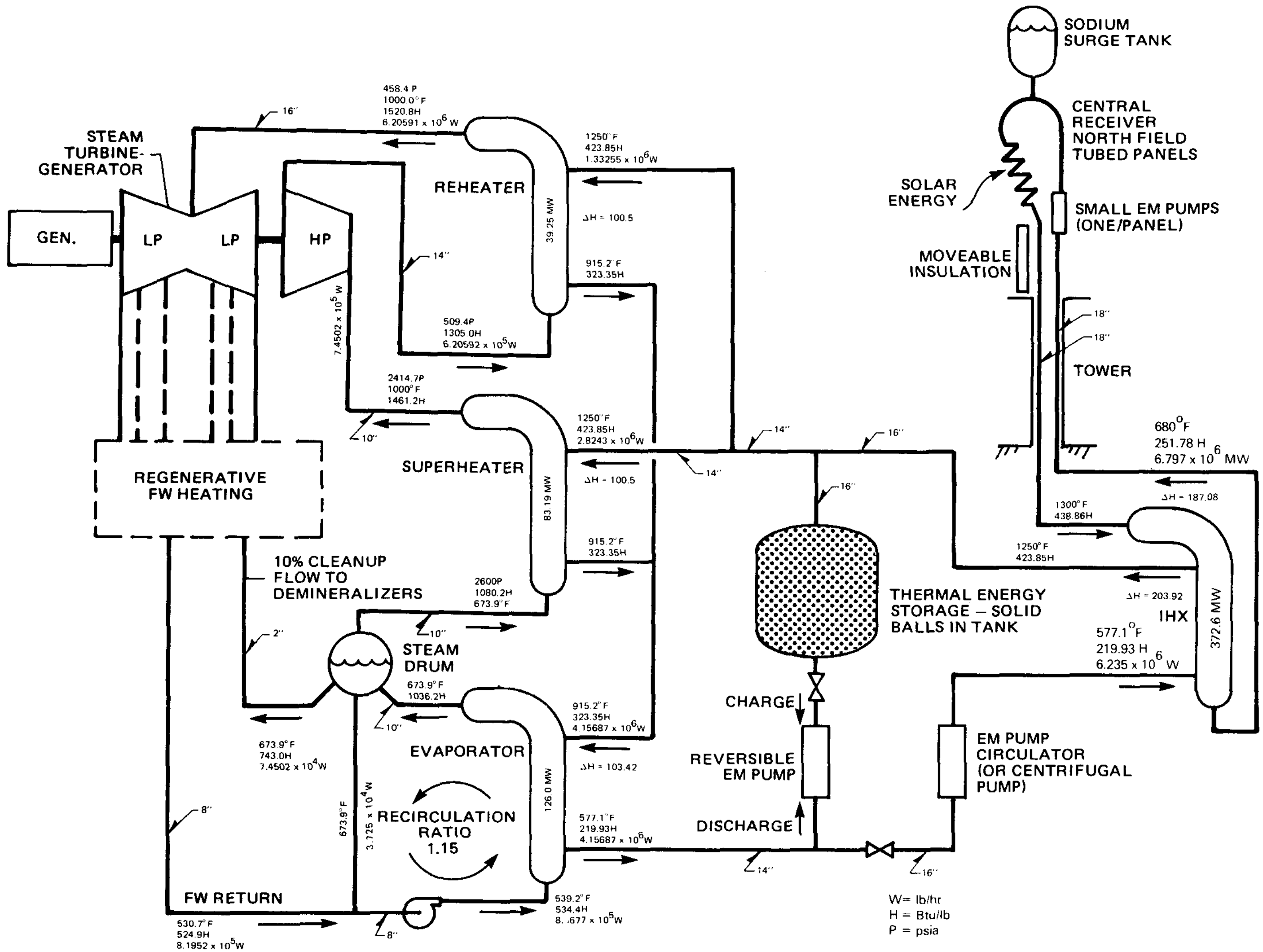


Figure 3.4-28. Reference Advanced Central Receiver Power System - 1300 F System

evaporator flow and steam flow from the drum is called the recirculation ratio. In this design, a recirculation ratio of 1.15 was used. This means that the concentrated impurities in the 15 percent water exiting the evaporator are very effectively removed by the 10 percent blow-down flow from the steam drum. In a once-through steam generator, this water would be fully evaporated and the concentrated impurities resulting from the boil off process would be deposited on the tube walls.

Intermediate Heat Exchanger (IHX)

The IHX module is identical in all respects to the steam generator modules except that the leak detection and large leak protection devices are unnecessary since the fluids are both sodium. Leaks in the IHX are of no consequence unless the maintenance of the desired sodium inventory becomes a problem in the tower loop. Because the operating pressures on both sides of the IHX tubes are low, the tube walls can be made thin, resulting in efficient heat transfer since the tube wall resistance controls the heat transfer process.

The IHX isolates the storage subsystem and steam generator subsystems from the sodium pressure at the tower base. The tube side of the IHX carries the tower flow and pressure. Since the tower sodium flows in a closed loop, the pump used to circulate the flow must only overcome the loop friction loss and the riser/downcomer differential densities. This pump can be placed anywhere in the loop. The reference configuration shown in Figures 3.4-27 and 3.4-28 uses the small EM pumps on the absorber panels as the loop circulation pumps. A small sodium surge tank on the top of the tower maintains the desired pressure levels.

System Heat Balance

A heat balance was performed on the steam generators to establish the flow rate and low end temperature, and the solar multiple was applied to these results to arrive at the flow rates in the IHX. A heat balance on the IHX then yielded the tower flow rates and temperatures. The results of these calculations are given in Figures 3.4-27 and 3.4-28; the details of the calculation are described in Appendix H.

Sizing the Steam Generators

The flow rates and temperatures computed in Appendix H along with the material selections from Section 3.3.1 provide all data required to size the heat exchanger modules. There were eight modules (two each of evaporator, superheater, reheater, and IHX) to design for the parametric analysis. All eight designs were studied parametrically to determine where the design limits occur. To do this, three tests were applied:

- Minimum cost design
- Overall length limit that can be shipped
- Reasonable or optimum pressure loss (in some instances, system dictated pressure loss limits)

The design tool used was a General Electric computer code (STMGEN) which determines active tube length once the number of tubes, tubing diameter, tube wall thickness, and tube pitch are specified, along with the thermodynamic properties of the shell side fluid and tube side fluid as determined by the heat balance.

The capital costs of each heat exchanger were obtained by breaking down known costs for the steam generator into separate categories covering material costs and fabrication/inspection costs. Each separate cost item was given a scaling factor to account for sizes larger or smaller than the base case, plus adders to allow for the use of thin wall tubing, more expensive material, more difficult metal working, and different design pressures on tube and shell side. The tooling and material handling charges were made proportional to the sum of the material costs and the fabrication costs.

Each heat exchanger was analyzed for cost and where possible, the design with minimum cost was chosen. In the case of the evaporator, the minimum-cost design was too long to be shipped; therefore, the design shown is the maximum size that can be shipped. The superheater was constrained to give a tube side pressure drop consistent with the system heat balance* at the same time that the tubing was constrained at 0.625 inch OD with 0.109 inch wall. The same size tubing was also used on the evaporator. This is considered the minimum practical tubing diameter that can be internal bore welded. Time and budget did not permit a study of larger diameter tubing or alternate design concepts for the superheater and evaporator in an attempt to determine if the costs could be reduced. Therefore, neither the evaporator nor superheater is necessarily fully optimized for cost. The reheater and IHX required larger diameter thin wall tubing. Therefore, an extensive parametric study was made for these designs with the result that the reheaters and IHX for both the 1100 F and 1300 F systems are close to minimum cost designs. In addition, the tube side pressure loss in the reheater was constrained to that dictated by the turbine. The line losses between the turbine and reheater were assumed to be 20 psi, leaving 31 psi across the reheater. The tube side pressure loss in the IHX was arbitrarily set at 10 psi to minimize the tower pumping cost. These assumptions on pressure loss could be subjected to further optimization.

The design parameters for the IHX, evaporator, superheater, and reheater are given in Tables 3.4-22 through 3.4-25 for the 1100 F and 1300 F systems. Table 3.4-26 shows the final costs for all eight designs. Two conclusions become evident. First, the differential cost between the 1100 F and 1300 F systems is small and relatively independent of whether an IHX is used in the design or not. The reason for this is the high cost penalty for Inconel 625† in the 1300 F system and the poor heat transfer in the 1300 F evaporator. Secondly, the $15 x 10^6 for the IHX is so high that it is necessary to reconsider eliminating the IHX by using a throttle on the downcomer to remove the tower pressure from the ground equipment.

The cost of the IHX might be reduced by increasing the tube side pressure loss or by increasing the tower flow rate which has the effect of increasing

*A 50 psi line loss between the superheater and the turbine was assumed giving a 135 psi loss across the superheater.

†Inconel 625 is six times more costly per pound than 2-1/4 Cr - 1 Mo.

the LMTD and heat transfer coefficients. Since these two options increase the tower pumping costs, it is unlikely that the overall costs would drop by more than \$1 to 2 x 10^6.

Table 3.4-22

EVAPORATOR DESIGN PARAMETERS

	1100°F SYSTEM	1300°F SYSTEM
PERFORMANCE DATA		
Power Rating, MWt	126.0	126.0
Active Heat Transfer Area Ft^2	9412	14320
Heat Transfer Uncertainty Factor, %	13.7	13.7
HEAT TRANSFER COEFFICIENTS ($BTU/HR\text{-}FT^2\text{-}°F$)**		
Tube I.D. ($x10^{-3}$)	2.51/22.6/2.49*	1.76/23.4/1.25*
Tube O.D. ($x10^{-3}$)	7.03/6.90/6.77	5.25/5.08/4.92
Tube Wall ($x10^{-3}$)	1.81/1.80/1.77	1.82/1.79/1.76
Fouling ($x10^{-3}$)	3.72/3.72/3.72	3.72/3.72/3.72
Overall ($x10^{-3}$)	0.74/0.98/0.72	0.63/0.94/0.54
SHELL SIDE PARAMETERS		
Sodium Inlet Temp., °F	840.1	915.2
Sodium Outlet Temp., °F	627.5	577.1
Sodium Flow Rate, Lbs/Hr	$6.606x10^6$	$4.157x10^6$
Vessel O.D./Length, Ft	3.98/97	4.90/97
Tube Pitch/Diam. Ratio	1.95	1.95
Shell Side Pressure Loss, psi***	12.74	6.39
TUBE SIDE PARAMETERS		
Tube O.D./Thickness, Inches	0.625/0.109	0.625/0.109
Number of Tubes	719	1094
Tube Active Length, Ft	80	80
Water Inlet Temp., °F	539.2	539.2
Steam Outlet Quality	87%	87%
Steam Flow Rate, Lbs/Hr	$8.578x10^5$	$8.578x10^5$
Tube Side Pressure Loss, psi	86	46
MATERIAL OF CONSTRUCTION	2-1/4 CR-1 Mo	2-1/4 CR-1Mo

*Preheat/Nucleate/Film (Listed at Mid-zone)

**Based on Tube O.D.

***Friction Loss - No Static Head Included

Table 3.4-23

SUPERHEATER DESIGN PARAMETERS

	1100°F SYSTEM	1300°F SYSTEM
PERFORMANCE DATA		
Power Rating, MWt	83.19	83.19
Active Heat Transfer Area, Ft^2 **	4732	1967
Heat Transfer Uncertainty Factor, %	6.5	6.5
HEAT TRANSFER COEFFICIENTS ($BTU/HR-FT^2-°F$)**		
Tube I.D. $(x10^{-3})$	2.59/1.01*	2.91/1.22*
Tube O.D. $(x10^{-3})$	6.21/5.89	5.57/5.12
Tube Wall $(x10^{-3})$	0.91/1.02	0.92/1.07
Overall $(x10^{-3})$	0.605/0.468	0.648/.514
SHELL SIDE PARAMETERS		
Sodium Inlet Temp., °F	1050	1250
Sodium Outlet Temp., °F	840.1	915.2
Sodium Flow Rate, Lbs/Hr	$4.488x10^6$	$2.824x10^6$
Vessel O.D./Length, Ft	3.60/66	3.25/43
Tube Pitch/Diam. Ratio	1.95	1.95
Shell Side Pressure Loss, psi***	9.6	5.3
TUBE SIDE PARAMETERS		
Tube O.D./Thickness, Inches	0.625/0.109	0.625/0.109
Number of Tubes	600	479
Tube Active Length, Ft	48.2	25.1
Steam Inlet Temp., °F	673.9	673.9
Steam Outlet Temp., °F	1000	1000
Steam Flow Rate, Lbs/Hr	$7.450x10^5$	$7.450x10^5$
Tube Side Pressure Loss, psi	135	135
MATERIAL OF CONSTRUCTION	Incolloy 800	Inconel 625

*Inlet Value/Outlet Value

**Based on Tube O.D.

***Friction Loss - No Static Head Included

Table 3.4-24

REHEATER DESIGN PARAMETERS

	1100°F SYSTEM	1300°F SYSTEM
PERFORMANCE DATA		
Power Rating, MWt	39.25	39.25
Active Heat Transfer Area, Ft^2	4006	1589
Heat Transfer Uncertainty Factor, %	6.5	6.5
HEAT TRANSFER COEFFICIENTS ($BTU/HR\text{-}FT^2\text{-}°F$)**		
Tube I.D.	334.9/360.4*	405.0/444.5*
Tube O.D.	2946/2777	2784/2526
Tube Wall	2422/2693	2478/2863
Overall	267.5/285.2	309.4/333.9
SHELL SIDE PARAMETERS		
Sodium Inlet Temp., °F	1050	1250
Sodium Outlet Temp., °F	840.1	915.2
Sodium Flow Rate, Lbs/Hr	2.118×10^6	1.333×10^6
Vessel O.D./Length, Ft	4.27/69.4	3.87/40.4
Tube Pitch/Diam. Ratio	1.95	1.95
Shell Side Pressure Loss, psi ***	3.95	2.71
TUBE SIDE PARAMETERS		
Tube O.D./Thickness, Inches	1.05/0.05	1.00/0.05
Number of Tubes	303	271
Tube Active Length, Ft	48.1	22.4
Steam Inlet Temp., °F	468.9	468.9
Steam Ohtlet Temp., °F	1000	1000
Steam Flow Rate, Lbs/Hr	6.206×10^5	6.206×10^5
Tube Side Pressure Loss, psi	31	31
MATERIAL OF CONSTRUCTION	Incolloy 800	Inconel 625

*Inlet Value/Outlet Value

**Based on Tube O.D.

***Friction Loss - No Static Head Included

Table 3.4-25

IHX DESIGN PARAMETERS

	1100°F SYSTEM	1300°F SYSTEM
PERFORMANCE DATA		
Power Rating, MWt	372.6	372.6
Active Heat Transfer Area, Ft^2 *	14,525	14,891
Heat Transfer Uncertainty Factor, %	10	10
HEAT TRANSFER COEFFICIENTS (BTU/HR-FT^2-°F)*		
Tube I.D.	6779	6982
Tube O.D.	4097	4360
Tube Wall	2785	2472
Overall	1333	1287
SHELL SIDE PARAMETERS		
Sodium Inlet Temp., °F	627.5	557
Sodium Outlet Temp., °F	1050	1250
Sodium Flow Rate, Lbs/Hr	9.909×10^6	6.235×10^6
Vessel O.D./Length, Ft	7.45/61	6.5/65
Tube Pitch/Diam. Ratio	1.75	1.75
Shell Side Pressure Loss, psi**	7.201	7.794
TUBE SIDE PARAMETERS		
Tube O.D./Thickness, Inches	0.970/0.045	0.790/0.045
Number of Tubes	1300	1500
Tube Active Length, Ft	44	48
Sodium Inlet Temp., °F	1100	1300
Sodium Outlet Temp., °F	725	680
Sodium Flow Rate, Lbs/Hr	11.223×10^6	6.797×10^6
Tube Side Pressure Loss, psi	10	10
MATERIAL OF CONSTRUCTION	316 SS	Inconel 625

*Based on Tube O.D.
**Friction Loss - No Static Head Included

Table 3.4-26

PARAMETRIC ANALYSIS
HEAT EXCHANGER COST SUMMARY - (M$)

	1100 F			1300 F		
	Labor	Material	Total	Labor	Material	Total
Evaporator	2.114	1.463	3.577	2.900	2.379	5.279
Superheater	1.906	2.473	4.379	1.440	1.644	3.084
Reheater	2.070	3.037	5.107	1.673	2.032	3.705
Total	6.090	6.973	13.063	6.013	6.055	12.068
IHX	5.467	9.760	15.227	4.745	10.280	15.025
Total	11.557	16.733	28.290	10.758	16.335	27.093

3.4.7 STORAGE COST ESTIMATES

Storage subsystem cost estimates, using 1978 dollars, were originally developed for the parametric study for two system temperatures (1300 and 1100 F) and the four arrangement concepts listed below. Both factory fabricated and field fabricated storage tanks were considered.

Concept 1 - Sodium Iron-Storage

Concept 2 - Sodium Storage (Five tanks, One Empty)

Concept 3 - Sodium Storage (Hot and Cold Tanks)

Concept 4 - Hybrid of Concepts 1 and 3

These concepts are shown schematically in Section 3.4.1.

A molten salt storage system for 1100 F maximum temperature with a Concept 3 storage arrangement was also evaluated by scaling the comparable sodium system costs. The scaling relationships are described in Appendix J; they were derived on the basis of a detailed comparison of the thermal and hydraulic behavior of salt with respect to sodium. All material selections were the same as those of the sodium system except that the cold leg tanks and piping were carbon steel.

In addition to the nine cases above which all include an Intermediate Heat Exchanger (IHX), eleven other cases were developed by scaling to evaluate the possible benefits of eliminating the IHX.

Tables 3.4-27 and 3.4-28 give the cost estimates for the systems with IHX, including the molten salt case. Table 3.4-29 describes the method for computing the incremental costs for Cases S10 through S22, using Case S3 as the base for 1100 F systems and Case S8 as the base for 1300 F systems.

The sources of these cost estimates are discussed below.

Thermal Energy Storage Tanks

Storage tank costs were developed from cost estimates for factory fabricated tanks as discussed in Section 3.4.2 and for field fabricated tanks as described in Section 3.4.3. The cost comparison shown in Table 3.4-5 of Section 3.4.2 indicates that factory fabricated cylindrical tanks are most economical for Concept 1, field fabricated spherical tanks are most economical for Concepts 2 and 3, and a combination of factory fabricated and field fabricated tanks is most economical for Concept 4. These are the combinations quoted in Tables 3.4-27,3.4-28, and 3.4-29. Costs for the small high pressure tanks in the two-pressure storage cases were estimated by scaling the field fabrication costs for larger tanks.

Pipe Fittings and Valves

Piping costs include pipe, direct labor, trace heating, and insulation. Pipe material costs were developed from vendor supplied prices. Valve costs were developed from recent quotes adjusted for differences in material and specified standards.

Pumps

The large pump costs were developed from recent manufacturing experience with large EM pumps. Comparison with previous cost data indicates that costs for centrifugal pumps, including the motor drive system, should be equivalent to EM pumps with the same flow, head, and temperature requirements. Power required for these pumps is listed for all twenty cases in Table 3.4-30 (see also Table 3.3-13 in Section 3.3.2).

IHX and Steam Generators

Estimated capital costs for the IHX and steam generator units are presented in Section 3.4-6. These costs were developed from previous estimates of steam generator costs by breaking down the cost items into separate categories covering material, fabrication, and inspection. Each cost item was given in a scaling factor to account for size change, material differences, different design pressures, and increased cost due to use of thin wall tubing. Tooling and material handling costs were made proportional to the sum of the material and fabrication costs.

Dump Tank and Surge Tank

These tank costs were based on recent tank procurements for ASME Boiler Code Section VIII tank designs of similar size.

Sodium and Molten Salt

Sodium costs were estimated at $0.33/pound for large quantity commercial grade sodium. Reactor grade sodium is generally more expensive than this because much more stringent impurity control is required to prevent the development of radioactive species. This quality of sodium refining is not required for solar applications.

Molten salt was costed on the basis of a verbal quote from Park Chemical Corporation of $0.20/pound.

GENERAL ELECTRIC

Table 3.4-27

STORAGE SUBSYSTEM COST SUMMARY - 1100 F SYSTEM

Item		S1	S2	S3	S4	S5
	Case:	S1	S2	S3	S4	S5
	Concept:	1	2	3	4	Molten Salt
	Tank Type Factory:	2a	--	--	2c	
	Tank Type Field:	--	IVa	IVa & IVc	IVd & IVe	
Item		Cost $\$ \times 10^6$	Cost $\$ \times 10^6$	Cost $\$ \times 10^6$	Cost $\$ \times 10^6$	Cost $\$ \times 10^6$
1. Thermal Energy Storage Tanks[a]		7.946	13.770	18.240	8.774	5.833
2. Pipe, Fittings, and Valves[b]		0.881	2.829	1.155	1.115	0.693
3. Pumps		10.400	10.250	8.750	11.750	6.153
4. IHX		15.227	15.227	15.227	15.227	43.839
5. Evaporator		3.577	3.577	3.577	3.577	5.845
6. Superheater		4.379	4.379	4.379	4.379	6.713
7. Reheater		5.107	5.107	5.107	5.107	7.901
8. Dump Tank		0.400	--	--	--	--
9. Surge Tank		0.100	--	--	--	--
10. Sodium/Molten Salt[c]		0.490	6.864	6.864	2.614	3.382
11. Iron Balls[c]		18.148	--	--	12.099	--
Total (M$)		66.655	62.003	63.299	64.642	80.362

[a] Includes material, assembly, and installation costs for tank and iron balls.

[b] Pipe costs shown do not include contractor indirect costs.

[c] Material cost only, without tax.

Table 3.4-28

STORAGE SYSTEM COST SUMMARY - 1300 F SYSTEM

Item		S6	S7	S8	S9
	Case:	S6	S7	S8	S9
	Concept:	1	2	3	4
	Tank Type – Factory:	1	--	--	1a
	Tank Type – Field:	--	Va	Va and Vc	Vd and Ve
		Cost $\$x10^6$	Cost $\$x10^6$	Cost $\$x10^6$	Cost $\$x10^6$
1.	Thermal Energy Storage Tanks	9.082	14.645	17.016	8.507
2.	Pipe Fittings and Valves*	0.55	1.27	0.67	0.86
3.	Pumps	7.20	8.75	5.75	9.50
4.	IHX	15.03	15.03	15.03	15.03
5.	Evaporator	5.28	5.28	5.28	5.28
6.	Superheater	3.08	3.08	3.08	3.08
7.	Reheater	3.70	3.70	3.70	3.70
8.	Dump Tank	0.35	--	--	--
9.	Surge Tank	0.10	--	--	--
10.	Sodium	0.327	4.559	4.559	1.738
11.	Iron Balls	12.099	--	--	8.066
	Total	56.798	56.314	55.085	55.761

Table 3.4-29

SUMMARY OF INCREMENTAL COSTS FOR STORAGE SUBSYSTEM PARAMETRIC CASES S10 THROUGH S20 (NO IHX)

	1100 F Full Flow Throttle Valve			Jet Pump		Two-Pressure Storage					
	S10	S11	S12	S13	S14	S15	S16	S17	S18	S19	S20
Cost Reductions with Respect to Cases S3 (1100 F) and S8 (1300 F)	Small Pumps	Large and Small Pumps	Large Pump and Valves	Large and Small Pumps	Large Pump and Valves	Pump Cold/Hot 1100 F	1300 F	Pump Hot Only 1100 F	1300 F	Cold/Hot Smaller Pumps 1100 F	1300 F
IHX	15.32	15.32	15.32	15.32	15.32	15.32	15.03	15.32	15.03	15.32	15.03
Large EM Pump	5.00	5.00	5.00	5.00	5.00	8.75	5.75	8.75	5.75	8.75	5.75
Primary Dump and Surge	0.26	0.26	0.26	0.26	0.26	0.26	0.19	0.26	0.19	0.26	0.19
Piping	0.20	0.20	0.20	0.20	0.20	--	--	--	--	--	--
Small EM Pumps (Low ΔP)	10.90	10.90	10.90	10.90	10.90	--	--	--	--	--	--
24-in. Riser	0.30	--	--	--	--	--	--	--	--	--	--
Smaller Storage	3.70	3.70	3.70	3.70	3.70	3.70	1.90	3.70	1.90	3.70	1.90
Smaller Tower Pumps	--	--	--	--	--	1.30	0.70	1.30	0.70	1.30	0.70
Δ Storage in High Pressure Tanks	--	--	--	--	--	1.41	1.28	1.11	1.08	1.41	1.28
Cold Leg Storage Tanks (Low Pressure)	--	--	--	--	--	--	--	3.56	2.40	--	--
Total	35.68	35.38	35.38	35.38	35.38	30.74	24.85	34.00	27.05	30.74	24.85
Cost Additions											
17 8-in. Risers	1.40	--	--	--	--	--	--	--	--	--	--
Small EM Pumps (High ΔP)	17.40	4.30	--	4.30	--	--	--	--	--	--	--
Large High ΔP Pump	--	8.75	9.63	5.75	6.63	13.43	11.06	5.85	4.89	9.97	8.50
Small EM Trim Pumps	--	--	--	--	--	--	--	--	--	--	--
Trim Control Valves	--	--	0.72	--	0.72	--	--	--	--	--	--
Small High Pressure Tanks	--	--	--	--	--	4.26	3.78	14.68	11.02	4.26	3.78
Large Low ΔP Pump	--	--	--	--	--	3.60	2.75	3.60	2.75	3.60	2.75
Control Valves	--	--	--	--	--	0.43	0.25	.13	0.08	1.33	0.76
Extra Pressure on Steam Generators	--	--	--	1.31	1.31	1.31	1.21	1.31	1.21	1.31	1.21
Total	18.80	13.05	10.35	11.36	8.66	23.03	19.05	25.57	19.95	20.47	17.00
Net Savings	16.88	22.33	25.03	24.02	26.72	7.71	5.80	8.43	7.10	10.27	7.85
Base Cost ($M)	63.30	63.30	63.30	63.30	63.30	63.30	55.09	63.30	55.09	63.30	55.09
Total Cost ($M)	46.42	40.97	38.27	39.28	36.58	55.59	49.29	54.87	47.99	53.03	47.24

Table 3.4-30

STORAGE LOOP PUMPING POWER AT DESIGN POINT (MW_e)

Case	Peak Pump Power	IHX Pump	Steam Generator Pump	Storage Pump	Additional* Tower Pumping	A** Pump	B** Pump
S1	3.00	--	1.50	1.50			
S2	3.16	2.11	1.05	--			
S3	2.05	1.00	1.05	--			
S4	3.05	1.00	1.05	1.00			
S5	1.17	0.57	0.60	--			
S6	1.88	--	0.94	0.94			
S7	2.26	1.57	0.69	--			
S8	1.26	0.63	0.63	--			
S9	1.89	0.63	0.63	0.63			
S10	6.21	--	1.05	--	5.16		
S11	6.21	--	1.05	--	5.16		
S12	7.19	--	1.05	--	6.14		
S13	2.03	--	1.05	--	0.98		
S14	3.02	--	1.05	--	1.97		
S15	4.10, 1.62	--	1.05	--	--	1.13, 0.21	1.92, 0.36
S16	3.01, 1.07	--	0.63	--	--	0.95, 0.18	1.43, 0.26
S17	2.18, 1.26	--	1.05	--	--	1.13, 0.21	--
S18	1.58, 0.81	--	0.63	--	--	0.95, 0.18	--
S19	2.97, 1.62	--	1.05	--	--	--	1.92, 0.57
S20	2.06, 1.07	--	0.63	--	--	--	1.43, 0.44

*Accounts for added pumping required to drive flow through throttle valve or jet pumps.

**These pumps handle only 18.5% of the sodium flow on the average, but they must be designed for peak flow conditions. Power shown is for peak and average respectively.

3.5 ELECTRIC POWER GENERATION SUBSYSTEM

3.5.1 STEAM CYCLE HEAT RATES AND COSTS

Six different steam cycle configurations were evaluated in the parametric analysis; these configurations are described in Table 3.5-1. The first three cycles represent typical configurations found in fossil fired steam plants of the 100 MW_e size range. They have regenerative feedwater heating involving five heaters fed by extractions from the reheat and low pressure turbines.

Case EP4 has two feedwater heaters added to the high temperature end of the heater train which are fed by extraction points on the high pressure turbine (heaters above the reheat point - HARP). The low pressure turbine has also been fitted with larger last-stage blades to increase the flow annulus and decrease kinetic energy losses on the turbine exhaust into the condenser. These additions improve the steam cycle efficiency, but are typically too costly for fossil applications.

Cases EP5 and EP6 are two further attempts to improve steam cycle efficiency by going to higher temperatures, and in EP6, to a second reheat. Case EP5 has the HARP feature and enlarged exhaust annulus area; EP6 does not have HARP because the cold reheat temperature is too high for effective feedwater heating. As can be seen in Table 3.5-1, these design variations generally result in higher efficiencies at higher cost. EP6 has a poorer heat rate than EP5 because the addition of the second reheat reduced the high pressure turbine efficiency, increased steam seal losses, and eliminated the HARP. The question is: how much additional cost is justified by the collector, receiver, and storage savings that result from these efficiency improvements? This is not a simple question because the sodium flow rates and storage requirements are tied not only to the thermal power required at the steam generators, but also to the sodium low end temperature as it exits from the steam generators. Thus, it is possible to reduce the thermal power to steam and actually increase the storage volume required. The reasons for this paradox are explained in the analysis below.

3.5.2 STEAM GENERATOR HEAT BALANCES

A version of the steam generator analysis described in Section 3.4.6 was used to evaluate the sodium flow rates and low end temperatures for Cases EP1 through EP6.

Figure 3.5-1 describes a simplified heat transfer model for these heat exchangers. Sodium enters the superheater and reheater at $\sim$1100 F, the flow split between these two exchangers is adjusted so the outlet sodium temperature (T^*) is the same for both. These flows are mixed and put into the evaporator/economizer. The sodium flow rate and low end temperature are constrained by the amount of heat which must be delivered and the maintenance of a fixed temperature pinch at the onset of nucleate boiling. If the evaporation temperatures is reduced by going to a lower pressure steam cycle, then it is generally possible to achieve a lower sodium low end temperature. This increases the operating temperature difference for the sodium and tends to decrease the sodium flow rate and storage volume. However, low steam pressure gives poorer efficiency, requiring larger heat input which tends to increase

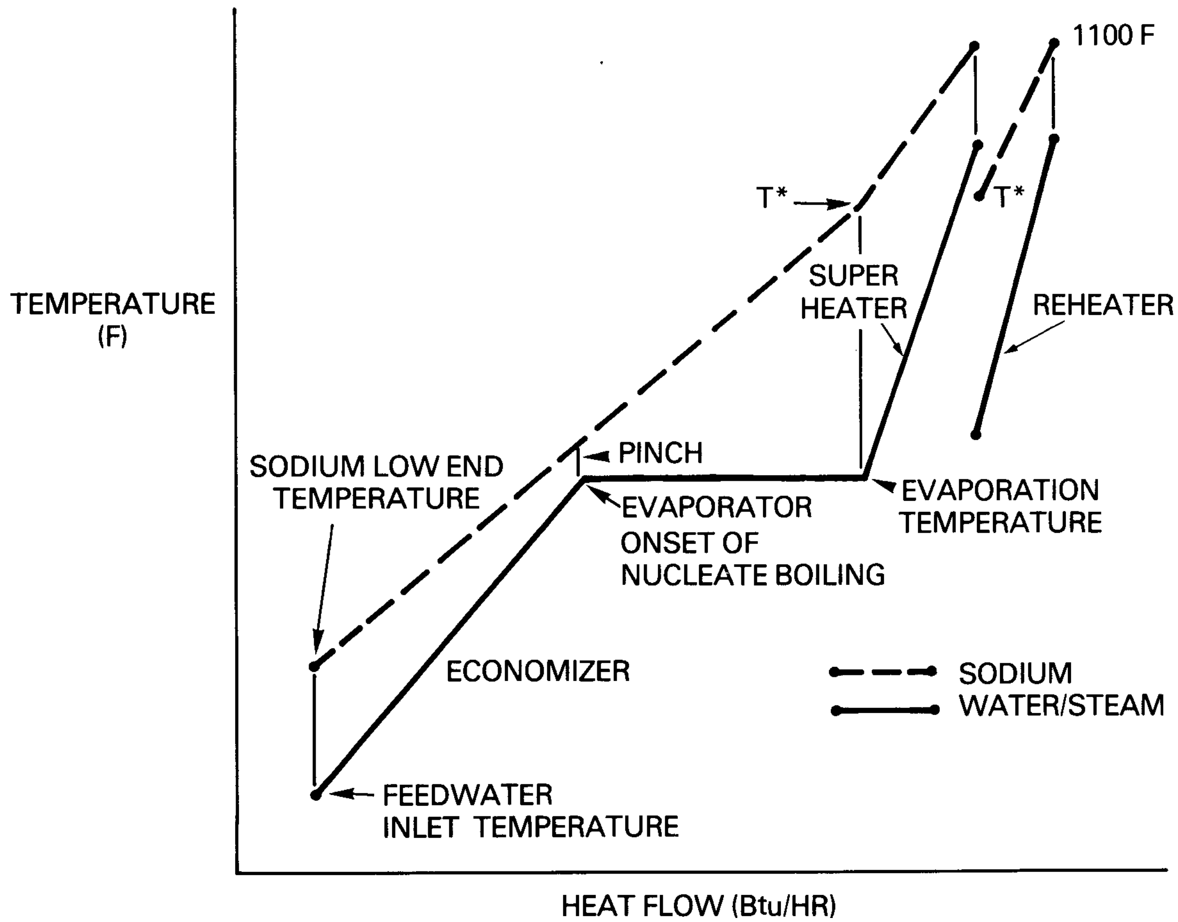


Figure 3.5-1. Steam Generator Heat Balance

flow rate and storage capacity. Which of these effects dominates depends on the details of the heat exchange process.

The equations which model this process are given in Appendix K, and the calculations of Cases EP1 through EP6 at 1300 F and 1100 F are described there. The results of this analysis are summarized in Table 3.5-2 where Case EP2 has been taken to be the reference case at both 1100 F and 1300 F. Note that although the heat to steam decreases in moving from EP1 to EP5 the sodium flow rate increases and decreases erratically due to the interaction described in Figure 3.5-1.

3.5.3 EFFECT OF STEAM CYCLE ON POWER PLANT COST AND PERFORMANCE

The steam cycle selection affects the performance and cost of the entire power plant. Since costs for the whole plant were not generated as part of this parametric analysis, it was necessary to use data from another study to assess the benefits of Cases EP1 through EP6. The data in Table 3.5-3 were taken from the recently published work on sodium-cooled receivers by Atomics International, McDonnell Douglas, and the University of Houston. The costs were divided into those which scale as the heat to steam, those which scale as the sodium flow rate, and those which are constant. The EPGS was broken out as a separate item, and the incremental costs in Table 3.5-1 were applied to this item. The heat exchanger costs do not, in fact, scale as the sodium flow because changing the steam system pressure changes the relative duty in the evaporator, superheater, and reheater. However, a detailed calculation of these effects indicates that sodium flow scaling is a relatively good approximation. Distributables were proportionally divided among the four categories to yield the overall plant cost of 211.26 M$.

In the Atomics International study, a 2000 psi steam cycle was employed which had a heat rate comparable to Case EP2 at 1800 psi in the present study. The plant costs were therefore scaled with EP2 as the base case.

Plant output from the Atomics International study is summarized in Table 3.5-4, and the scaling relationships for the auxiliaries are identified.

Applying the relative sodium flows and heat rates in Table 3.5-1 to these cost and performance scaling laws gives the result plotted in Figure 3.5-2.

Based on this calculation case, EP5 appears to be the most cost-effective choice. This selection would not be appropriate for a fossil fired plant because the gain in efficiency and fuel savings could not offset the increased capital investment. However, the high cost of the solar heat supply justifies this choice for the advance central receiver power plant.

The 1100 F sodium cases are less sensitive to the cycle selection than the 1300 F cases because at 1100 F the increases in sodium flow rate tend to offset the decreases in heat rate.

Table 3.5-1

STEAM CYCLES HEAT RATES AND COSTS

Case	Steam Conditions*	Heat Rate† (Btu/kWh)	Incremental†† Cost (M$)	Gross Output (MW_e)
EP1	1450P/1000F/1000F	8152	-0.19	113
EP2	1800P/1000F/1000F	8016	0 (Base)	113
EP3	2400P/1000F/1000F	7856	+0.40	113
EP4	2400P/1000F/1000F HARP**	7662	+1.76	113
EP5	2400P/1050F/1050F HARP**	7543	+2.29	113
EP6	2400P/1050F/1050F/1050F	7548	+4.44	113

*Throttle Pressure (psig)/Throttle Temperature (F)/Reheat Temperature (F)/Second

**HARP - feedwater heater above reheat point, also the leaving losses have been reduced by increasing the last stage annulus size

†Heat to steam ÷ gross electrical output

††Includes costs for turbine/generator and heat rejection system (1978 dollars)

Table 3.5-2

STEAM CYCLES SODIUM FLOW RATES AND HEAT INPUT TO STEAM

Case	Steam Conditions	Relative Sodium Flow rate	Relative Heat to Steam
	1100 F Sodium		
EP1	1450P/1000F/1000F	0.981	1.017
EP2 (ref.)	1800P/1000F/1000F	1.000	1.000
EP3	2400P/1000F/1000F	1.024	0.980
EP4	2400P/1000F/1000F HARP	1.001	0.956
EP5	2400P/1050F/1050F HARP	1.011	0.941
EP6	2400P/1050F/1050F/1050F	1.060	0.942
	1300 F Sodium		
EP1	1450P/1000F/1000F	1.000	1.017
EP2 (ref.)	1800P/1000F/1000	1.000	1.000
EP3	2400P/1000F/1000F	0.996	0.980
EP4	2400P/1000F/1000F HARP	1.037	0.956
EP5	2400P/1050F/1050F HARP	0.976	0.941
EP6	2400P/1050F/1050F/1050F	1.024	0.942

Table 3.5-3

SODIUM COOLED CENTRAL RECEIVER POWER PLANT COSTS

Costs Scale As:

	Heat Rate (M$)	Sodium Flow (M$)	Constant (M$)	ElectGen System (M$)
Land	1.45			
Buildings			4.81	
Collector	94.58			
Receiver		11.50		
Heat Exchanger		6.63		
Pipe		6.11		
Tower	8.84			
Storage		14.99		
Media		11.05		
Turbine-Generator				17.48
Elect Plant			3.27	
Controls			1.78	
Totals	104.87	50.28	9.86	17.48
Distrubutables	16.53	7.93	1.55	2.76
Sum	121.40		11.41	20.24
Overall	211.26 Base Case, 1800 psig/1000 F/1000 F			

Reference: Liquid Metal Cooled Solar Central Receiver Feasibility Study and Heliostat Field Analysis, Final Report, Part 1, University of Houston, Houston, Texas, Report ORO 5178-78-1- UC 62, October 1977.

Table 3.5-4

SODIUM-COOLED RECEIVER POWER PLANT OUTPUT

Auxiliaries	Gross 113 MW_e
Field[(1)]	1.78
Storage Pump + Tower Pump[(2)]	5.89
BOP	5.33
	13.00
Net Output	100 MW_e

(1) Scales as Heat Flow Required at Steam Generators

(2) Scales as Sodium Mass Flow

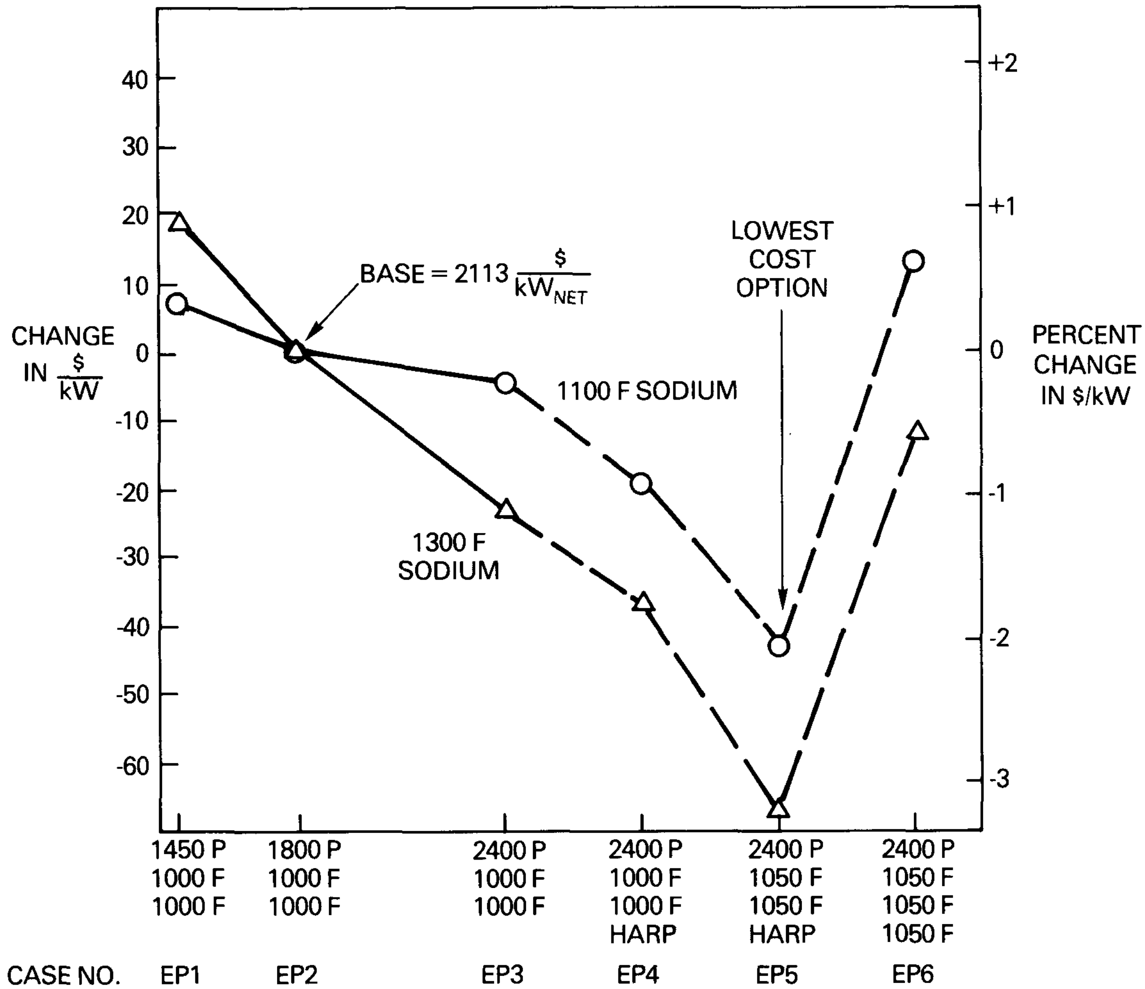


Figure 3.5-2. Variation of Plant Capital Cost with Steam Cycle Selection

3.6 MASTER CONTROL SUBSYSTEM

The controls effort for the parametric analysis involved an assessment of the plant subsystem control strategies with the intent of characterizing the differences among the various parametric concepts. The plant master control, an essential element in the characteristics of plant operation, was thus not fully developed during the parametric analysis. However, a general master control configuration has been assumed in order to complete the control structure and permit the necessary assessment

The operation of the plant as part of an electric utility network is of particular importance in the evaluation and selection of plant components and configuration. The evaluation of control characteristics has been conducted with emphasis on the design of a plant with performance characteristics which will maximize its acceptability and attractiveness to a utility as an element in its operations. Accordingly, a brief discussion of the environment in which a power plant must operate precedes the discussion of specific proposals for subsystem and plant configurations.

3.6.1 POWER PLANT OPERATION ON A UTILITY NETWORK

Operating Environment

The power system environment is characterized by:

- Fluctuating and changing levels of connected load, aggravated by unusual loads or weather.
- Changing power plant status, both planned and unplanned, resulting in varying levels of available generation.
- Network disturbances due to unplanned loss of generation or change in network configuration due to line switching.
- Necessity to over- or under-generate to correct system time or adjust for inadvertent energy exchange on the lines.

Load changes which are random, i.e., small in magnitude and occurring over relatively short time intervals, cause small deviations in generation load balance and result in small frequency deviations(from + 0.02 Hz to -0.04 Hz in the United States). Hence one requirement for plant response is the necessity of frequency regulation.

Over longer time spans (i.e., days, weeks, years), variation in load occurs within larger tolerance; this variation is predictable as a function of time of day, day of week, and time of year. The necessity to maintain a match of generation with load over these larger variations requires that at least some plants have load-following capability.

The variability in connected load, together with the economics of plant and system operation, creates the need to take units fully out of service, either daily or for more extended periods. Maintenance of the plant also creates a need to shut down and start up plants on a planned basis. Hence plant characteristics in startup and shutdown are an important operating consideration.

A mild emergency in a power system results from the unexpected loss of generation within a control area. The sudden in-flow of power over tie-lines from adjacent control areas may exceed the thermal capacity of transmission line conductors. A small drop in frequency on the entire interconnection also results. To restore the tie-line loadings to normal and eliminate the frequency deviation, the remaining generation in the area must be increased by the amount lost. Industry standard practices specify that this shall be accomplished within ten minutes, and, in most utilities, this additional power must be provided by units already synchronized to the system. This requirement is referred to as tie-line backup.

Major system emergencies may result in a loss of all ties to the remainder of the power system, leading to the isolation of a part of the system, i.e., the formation of an island. The isolated segment may be generation-rich (with a rise in frequency and a need to reduce generation) or generation-deficient (with a need for an increase in generation, load shedding, or a combination to restore frequency and permit resynchronization with the interconnection).

A special case of islanding is the sudden disconnection of a unit or plant from the rest of the system, leaving the unit with no connected load. Referred to as load rejection, the unit is usually shut down by its overspeed protection. However, it is desirable (where possible) to cut back on unit power output to the level of the plant auxiliaries' load. It is usually desirable to resynchronize and reload the unit as quickly as possible if the load rejection has been caused by a malfunction external to the plant.

Desirable Plant Response Characteristics

Operation of an interconnected power system places certain control and maneuverability requirements on the aggregate generation in the interconnection; these requirements must be imposed ultimately on the individual units. Since individual units vary in their relative ability to maneuver and play different roles in the overall economics of day-to-day operation, it is not possible to unequivocally define absolute response requirements. However, there are some general guidelines which can be used in the design of power plants for use on electric utility networks.

General guidelines which are desirable objectives in the design of a new plant concept are:

1. Each generating unit and its controls should be inherently stable under all combinations of possible manual and automatic control while connected to the system. That is, under no circumstances should the stable operation of any unit depend on the characteristics of other units.
2. It is highly desirable that each unit, if called upon, be able to assume its proportionate share of load regulating and/or frequency regulating duty.
3. Generating unit controls, in responding to external stimuli (such as frequency deviation or automatic generation control signals), should not impose on the unit an excursion which would cause the unit to lose

control or to trip off the line. That is, control action should be limited to the amount of control to which the unit can respond without exceeding limits on process variables (such as water level, pressures, or temperatures).

A unit should have the capability for startup from hot conditions during a system emergency (e.g., the Consolidated Edison blackout) and to operate on-line as spinning reserve under unusual circumstances.

Based on these guidelines, it is possible to specify quantitative goals for the control of individual generating units from an analysis of the aggregate system needs. The aggregate system needs are quantified below and the resulting requirements for response of individual units are stated (Ref. 3.6-1).

Frequency Regulation. The requirements for frequency regulation are essentially those for speed governing of the prime mover. They are defined in industry standards (Refs. 3.6-2 a,b,c) and may be summarized as follows:

1. A prompt stable response in change of power output of +1.3 percent or -0.7 percent of MW rating, with at least 30 percent of total change within the first two seconds.
2. A maximum deadband of 0.06 percent frequency (0.036 Hz on a 60 Hz system).
3. A steady-state regulation of 5 percent (i.e., 20 percent change in power output for each 1 percent decrease in frequency).

These specifications apply only to the speed control and assume that the energy supply is capable of meeting the demands made upon it as defined above. In plants where the energy supply is complex, the overall plant control will respond to frequency deviation and will exercise a coordinated control over both prime mover and energy supply to meet the speed/load demand.

Load Following. For those generating units called upon to adjust output to follow long term load variations, a typical expectation is the ability to go from 100 percent power to 50 percent power at rates of 1 percent to 2 percent per minute over much of this range, and to make the total excursion over a 2 hour period and return in the same elapsed time. Peaking units (normally combustion turbines) are expected to load and unload over a range of 70 percent of rating in periods of 10 to 20 minutes.

Tie-Line Backup. Increase in generation for tie-line backup is generally provided for spinning reserve (units already synchronized to the system). The one to two percent per minute response rate cited for load-following duty is generally adequate for tie-line backup.

Startup - Shutdown of Plant. Just as different requirements for load following exist for different types of units, there is a distinction made for startup and shutdown rates. Peaking units, most likely combustion turbines, are often used for non-spinning reserve to meet unexpected sudden load increases; as such, these units should be capable of start-to-full load in 30 minutes or less. For intermediate range steam units, start-to-full load in one to two hours is desirable. Base load units could take from two to four hours for a start following a brief shutdown and six to ten hours following a more extended shutdown. Shutdown rates comparable to startup rates would be permissible.

A plot of response rate in percent MW/minute vs. the number of minutes at which this rate can be sustained, plotted on log-log coordinates (Ref. 3.6-3), helps to present the data above. Figure 3.6-1 illustrates the transition from excursion-limited response to rate limits over the range of normal operation conditions. Typical system emergency requirements are also shown. Load-rejection capabilities are omitted from this figure since they do not represent the same kind of requirement in specification and design of unit response.

Operation of the Advanced Central Receiver Power Plant

The economics of operation of the advanced central receiver power plant, together with the characteristics of the solar energy supply indicate that it will probably be operated at maximum output as a base load plant while it is on-line and sunlight is available. Its daily startup/shutdown cycle will make it more like a peaking unit, considering its energy contribution to the system. Perhaps it would be more accurate to view it as a peak load shaver rather than a more conventional peaking plant.

From this perspective, its desired operating characteristics will be a combination of those described earlier for base load and peaking units. It should be capable of contribution to frequency regulation, but will not ordinarily be called upon for load following beyond its daily on-off cycle. It is also apparent that it will more closely resemble an intermediate range steam unit in its startup and shutdown capability.

3.6.2 GENERAL CONFIGURATION FOR ADVANCED CENTRAL RECEIVER PLANT MASTER CONTROL

Although a formal design for the master control subsystem was not developed during the parameter analysis, assumptions have been made as to its form, functions, and interactions with the plant subsystems and the utility system control operations. The plant master control is assumed to function primarily in a supervisory role with respect to the subsystem controls. Information from the utility dispatch center will be input to the master control, which will be used to effect coordinated control of plant and the subsystems. By monitoring the subsystems, plant condition, and status, appropriate information will be communicated by master control to the utility dispatch control center.

A further assumption is that each subsystem control has a stand-alone capability and can operate automatically, independent of the master control from operator manual inputs. Thus, the plant will have three levels of control: fully automatic, semi-automatic with the master control out of service, and fully manual in a conventional manner. In the latter two modes, coordination of control is, of necessity, provided by the plant operator.

This configuration is illustrated in Figure 3.6-2. It is the basis for the evaluation of the proposed subsystem controls in the following section.

3.6.3 ADVANCED CENTRAL RECEIVER PLANT SUBSYSTEMS CONTROL EVALUATIONS

Collector Subsystem Control

Whether the heliostats are glass or enclosed, and whether the field is north or circular, the control of the individual heliostats and the field in total will be the same. Since the control is essentially self-contained and

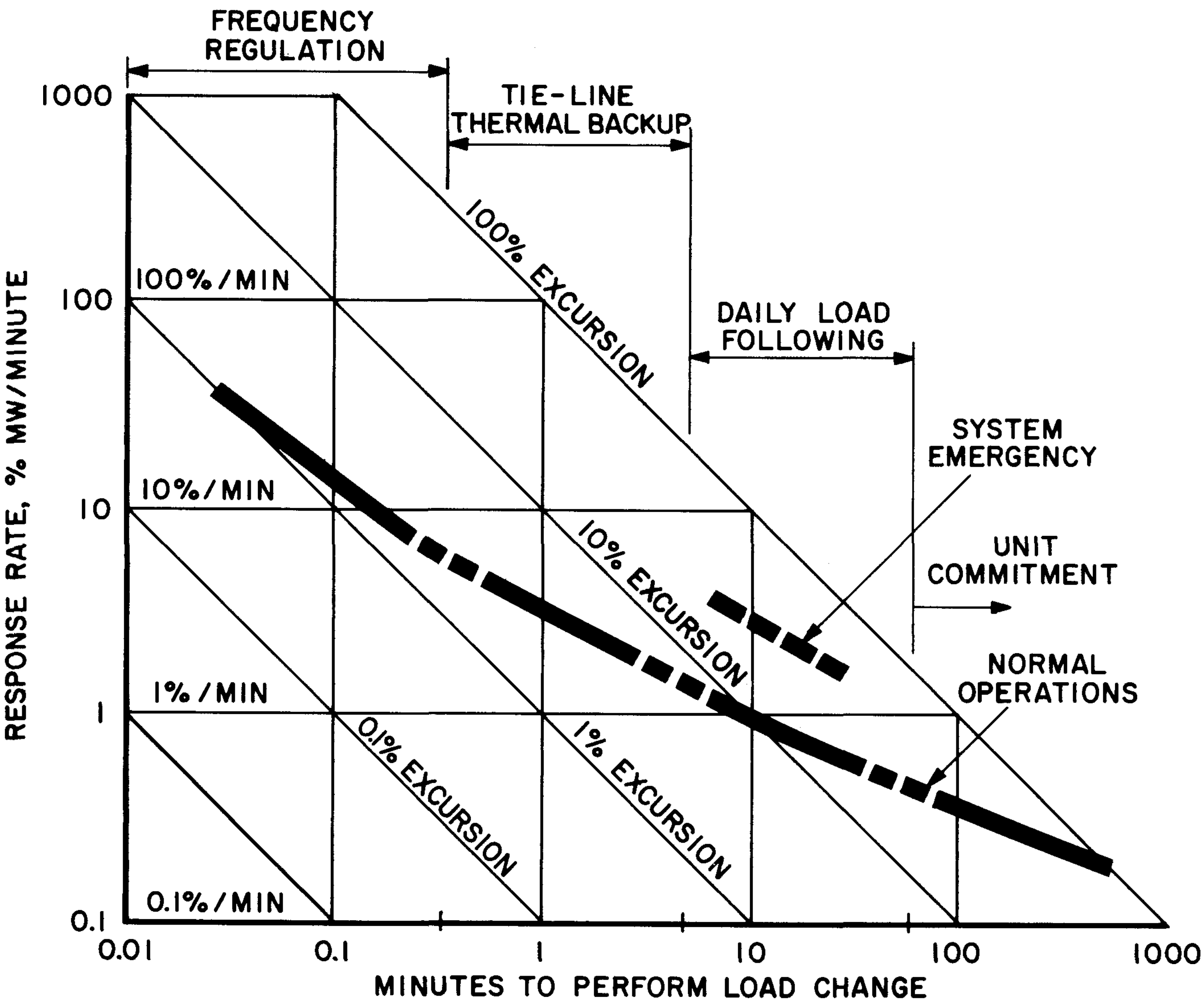


Figure 3.6-1. Maneuvering Requirements of Generating Units for Utility System Operation

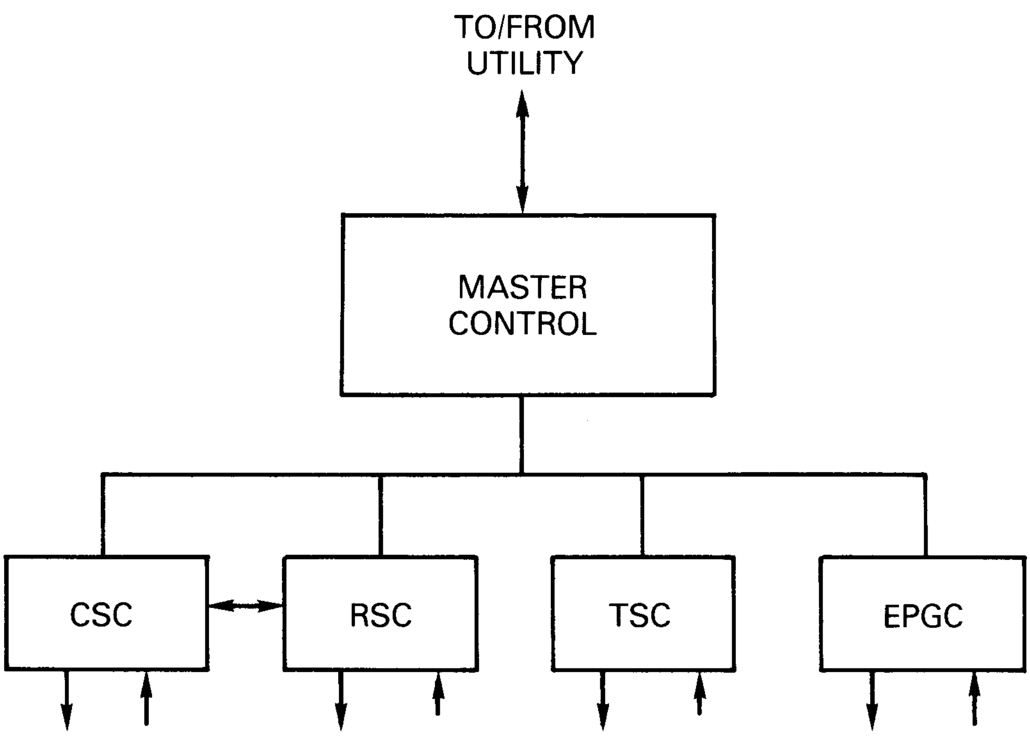


Figure 3.6-2. Control System Configuration

must by itself meet the requirements for aiming and tracking, the master control exercises only a supervisory role in turning the field on or off, according to the demands for plant operation. This includes a decision-making function to maintain collector operation or to shutdown during cloud passage with reduced total insolation. Another master control action will be necessary when the thermal storage is fully charged and the total available solar power exceeds the scheduled or rated plant output.

In one instance, it seems desirable for a direct communication between subsystem controls. If an overheated receiver panel is sensed and cannot be corrected by the receiver controls, that information would best be communicated directly to the collector subsystem control. This ensures prompt action by the latter, and provides for continued plant operation should the master control be inoperable.

Information available on the collector subsystem control strategy and implementation indicates that there should be no problems from the interface with the master control and no adverse effects on the operating characteristics of the plant.

Receiver Subsystem Control

The only supervisory control function that the master control subsystem exerts on the receiver control occurs during startup or shutdown and is associated with decisions based on environmental and utility demand inputs.

Thermal Energy Storage Subsystem

Four basic storage concepts, shown in Figure 3.6-3 were proposed:

1. Sodium - Iron Storage. The storage of thermal energy in a combination of hot liquid sodium surrounding iron or steel balls has serious drawbacks on the operation of the plant. To minimize the thermocline spread, it must be charged and discharged at a single flowrate. Moreover, it is not possible to partially charge or discharge the stored energy without appreciable loss of temperature due to conduction. Thus these requirements and limitations cannot be matched to the changes in solar availability and anticipated utility demand without degradation of plant performance. It was not recommended for this reason.

2. Empty Tank Storage. The empty tank storage concept offers great flexibility in terms of charging and discharging stored energy. However, it does require a greater number of control points due to the multiplicity of valves. This, in turn, will require a dispatch logic in the energy storage control subsystem and the capability for rapid switching of flows to and from one tank to another. Because of the complexities of control, this is not a recommended selection.

3. Hot and Cold Tank Storage. Extreme flexibility of operation is attainable from this concept. With full capacity in both hot and cold tanks, the storage system acts as an isolating medium permitting fully

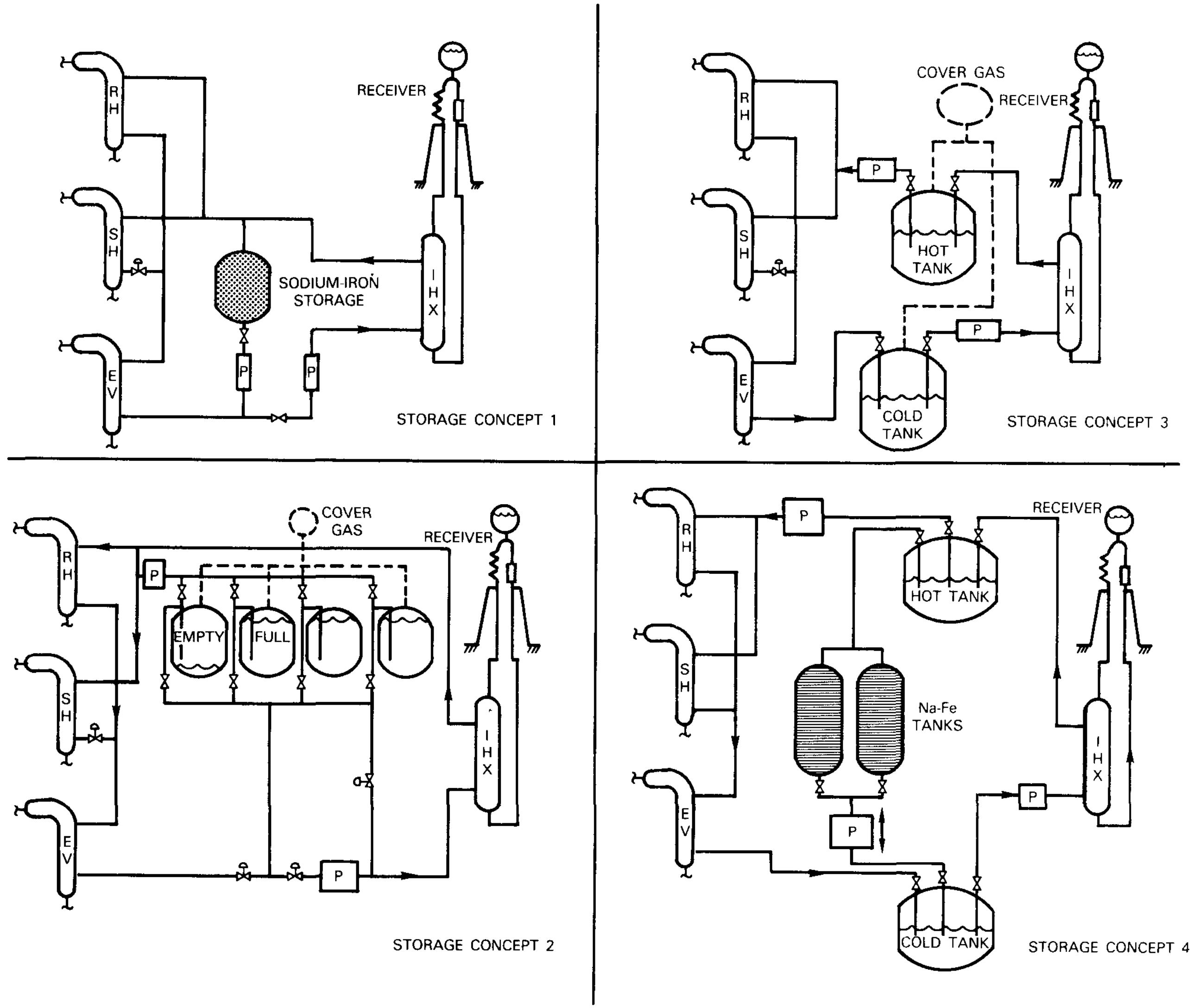


Figure 3.6-3. Proposed Storage Concepts

independent operation of the collector/receiver subsystems and the electric power generation subsystem. There is no requirement for a single rate of energy charging or discharging, nor the necessity for full charge or discharge. Solar energy input can be directed solely to storage (up to the capacity of the storage system) or readily divided between charging storage and generating power in any proportion. Energy in storage can be partially withdrawn and the remainder held without additional losses until needed again, or it can be recharged to any level up to full capacity. The hot tank - cold tank configuration acts as an energy surge tank between collector/receiver and power generation.

From the standpoint of design of either subsystem or master control strategies, this concept seems to be the most desirable; it affords the most favorable characteristics for desired plant performance.

4. Hybrid Sodium-Iron/Hot Tank - Cold Tank System. This concept possesses some of the favorable and unfavorable characteristics previously enumerated for Concepts 1 and 3. As a consequence of combination, the control strategy for the energy storage subsystem would be more complex. The sodium-iron tanks would have to be charged at the single-optimum rate, requiring a predicted assurance of the availability of the full volume of hot sodium during the charging interval. Conversely, the discharge at the optimum rate must be scheduled to meet power output demands, within the capacity of the hot tank.

   Given the enumerated additional requirements on subsystem control, we forsee no other potential impact on the master control or plant performance. However, unless there are cost benefits available from this configuration, there are no reasons to select this concept in preference to the simple hot tank - cold tank system.

The elimination of the IHX does not appear to alter the observations and conclusions respecting the control requirements for each of the concepts. The major characteristic of the hot tank - cold tank concept, the energy surge tank effect, is retained, so this configuration is the preferred choice for best plant operation and control.

REFERENCES FOR SECTION 3.6

3.6-1. D.N. Ewart, M.H. Dawes, R.P. Schulz, A.S. Brower, "Power Response Requirements of Electric Utility Generating Units," American Power Conference, April 24, 1978, Chicago, Illinois.

3.6-2a. Joint ASME-AIEE Committee, "Recommended Specification for Speed-Governing of Steam Turbines Intended to Drive Electric Generators Rated 500 KW and Larger," IEEE Standard 122 (AIEE No. 600), December 1959.

b. Joint IEEE-ASME Subcommittee, "Proposed IEEE Recommended Specification for Speed Governing and Temperature Protection of Gas Turbines Intended to Drive Electric Generators," IEEE No. 282, August 1968.

c. Joint AIEE-ASME Committee, "Recommended Specification for Speed-Governing of Hydraulic Turbines Intended to Drive Electric Generators," IEEE No. 125 (AIEE No. 605), September 1950.

3.6-3. IEEE Working Group, "MW Response of Fossil Fueled Steam Units," IEEE Transactions, Vol. PAS-92, No. 2, March/April 1973, pp. 455-463.

Section 4

SELECTION OF PREFERRED PLANT CONCEPT

The advanced central receiver plant concept selected as a result of the parametric analysis is shown in Figure 4-1. It consists of a 360° field of enclosed plastic heliostats focusing onto a cylindrical receiver operating at a peak sodium temperature of 1100 F. The heliostats are focused to slant range to achieve higher fluxes and lower receiver losses. Pressure reduction at the base of the downcomer is accomplished using a throttle valve complex.

Hot and cold sodium is stored in separate arrays of field assembled tanks, and sodium is supplied to the steam generators by a pumping loop which is isolated from the absorber loop by the tanks.

Separate steam generator modules provide the evaporation, superheat, and reheat functions required to deliver steam to a high efficiency steam cycle (2400 psi/1000 F/1000 F with heaters above the reheat point).

4.1 SELECTION REVIEW PROCESS

The process by which the commercial plant concept was selected consisted of a review of the parametric study results and conclusions at three different levels.

First the GE technical team met to present and integrate results from the parametric analysis and to rank the different subsystem and component options in accordance with a predetermined set of selection criteria to be discussed below. Of paramount concern were the issues of reducing plant cost and minimizing development requirements.

The overall system configuration shown in Figure 4-1 was selected at this meeting; however, a peak temperature of 1300 F was chosen to achieve cost reductions that had been noted for this temperature in the parametric cases. The material selected for this application (Inconel 625) was recognized to have undesirable sodium corrosion characteristics and a tendency to embrittlement during long term high temperature operation. However, another material (Inconel 617) seemed to offer hope for a solution to these problems, so it was decided to pursue 1300 F in Phase 1 and Phase 2 subsystem research experiments, retaining 1100 F as a backup option based upon Incoloy 800 and 316 stainless steel.

This plant concept and the selection logic were then presented to a technical review panel consisting of representatives from a utility and six organizations involved in utility equipment manufacture and product development (Table 4-1 identifies the GE Advanced Central Receiver Technical Review Panel).

The panel generally agreed with the conclusions of the GE technical team, but commented on three specific areas. First, the panel felt that an extended study using a range of heliostat costs corresponding to the high and low limits of cost would be required to make a conclusive decision about the most cost-effective heliostat design (for further discussion of this issue, see Section 4.3). Second, the utility representative noted that the design approach was based on

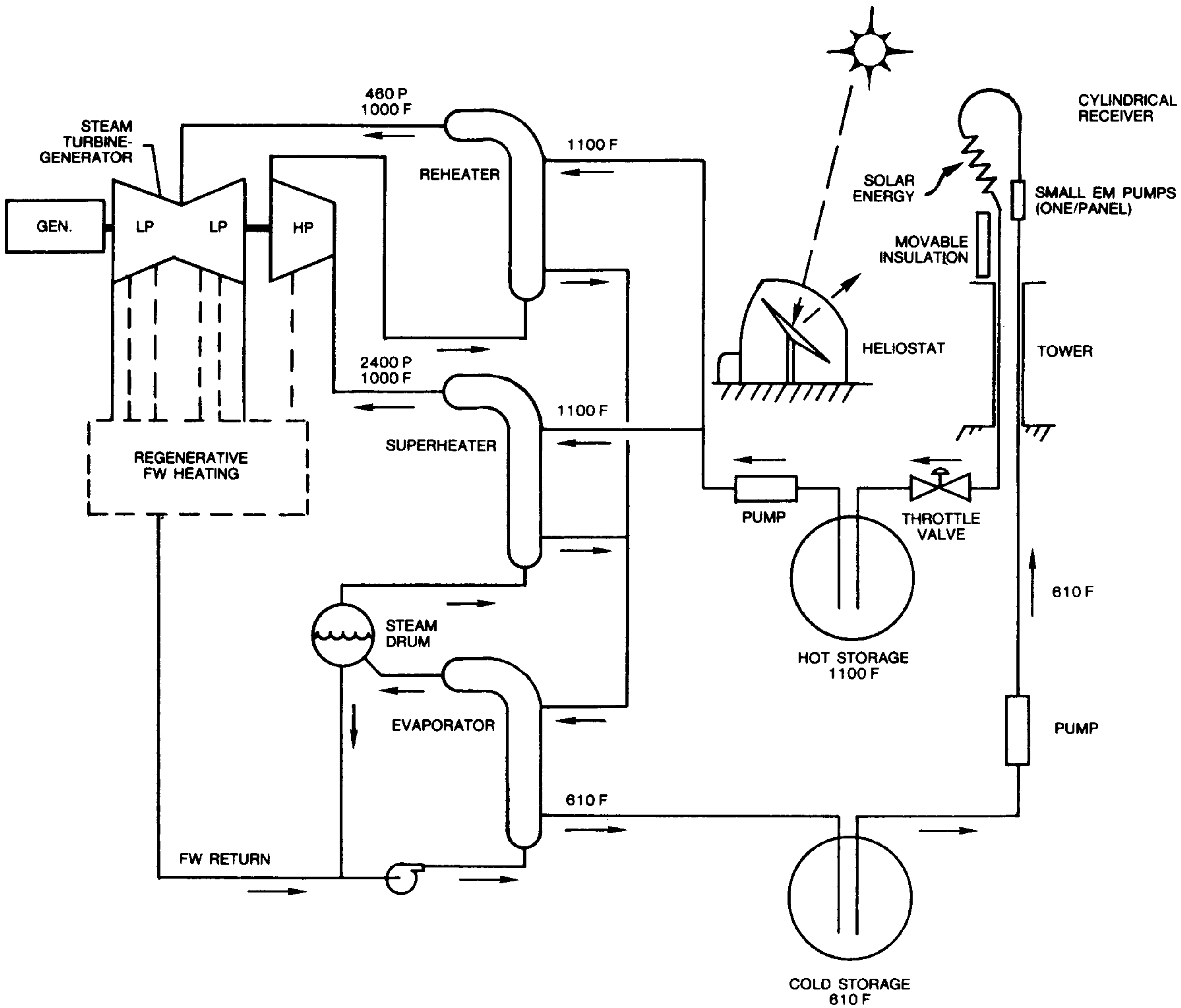


Figure 4-1. Commercial Plant Configuration

Table 4-1

STRUCTURE OF GENERAL ELECTRIC ADVANCED
CENTRAL RECEIVER TECHNICAL REVIEW PANEL

Chairman:	General Manager Electric Utility Systems Engineering Department General Electric Company
Members:	Chairman of the Board Foster Wheeler Development Corporation
	General Manager Center for Energy Systems General Electric Company
	Manager Development Engineering Advanced Reactor Systems Department General Electric Company
	General Manager Energy Systems Programs Department General Electric Company
	Director Research and Development Southern California Edison Company
	Manager Advanced Energy Programs Space Division General Electric Company

steady, full output turbine operation during the day, without regard for the possible use of the turbine as a load-following device to offset gas turbine peaking requirements. He was concerned that not addressing load following directly might lead to inappropriate choices in the Task 4 conceptual design, although he could find no limitations on load-following capability in the proposed concept. Finally, the panel expressed concern about the selection of the 1300 F temperature level, pointing out that the cost advantages noted might be illusory because of uncertainty about manufacturability, creep fatigue, and corrosion.

The concept was next evaluated by the management of General Electric's Advanced Reactor Systems Department and Corporate Research and Development. Of particular concern in this review was the temperature level and its impact on plant cost and development risk. Although 1300 F did seem to offer a cost improvement, it was concluded that Inconel 617 was a high-risk material because of its potentially poor resistance to sodium corrosion. The testing required to evaluate the sodium corrosion was judged to be inconsistent with DoE near-term goals for the Advanced Central Receiver program. As a consequence, 1300 F was dropped and 1100 F was selected.

4.2 SELECTION CRITERIA

Both the initial concept selection and the technical review panel evaluation were performed by comparing the parametric options with respect to a list of selection criteria described in this section. This list was derived in part from a methodology developed by General Electric to assist the Electric Power Research Institute in its evaluation of advanced fossil and nuclear power generation concepts.* The list also includes all of the criteria noted in the Advanced Central Receiver Request for Proposal.

Criteria from both sources were combined and screened to retain only those which could assist in discerning differences among the parametric options. For instance "Plant Construction Time" appeared in the EPRI-derived list, and this item is important in comparing a gas turbine combined cycle (three years construction) with a nuclear plant (seven to nine years construction). However, all solar options considered in this study have about the same construction time, so this criterion was dropped from the list. Similarly, "Water Requirements" was dropped from the list, not because it is unimportant, but because it did not assist in separating the parametric options in this study.

The selection criteria which remained after this screening are listed in Table 4-2. As noted above, the intent of these criteria is to assist in selecting a plant concept from among the parametric options, not to judge this plant concept against an absolute standard as will be done in the assessment of the commercial plant concept in Task 5. To help focus on this more narrow objective, definitions were developed for each criterion (Table 4-3). Each definition describes the utility objective with respect to the power plant attribute addressed by that criterion. The definitions also describe the action required to measure compliance with the objectives.

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*Comparative Study and Evaluation of Advanced Cycle Systems, EPRI Research Project 235-1, Final Report EPRI AF-664, General Electric Company, February 1978.

Table 4-2

ADVANCED CENTRAL RECEIVER
SELECTION CRITERIA

A. Capital Cost

B. O&M Requirements

C. Control Characteristics

D. Forced Outage Rate

E. Startup Power Requirements

F. Potential for Improvements in Cost and Performance

G. Environmental Intrusion

H. Land Requirements

I. Hardware Materials Availability

J. R&D Required

K. Industrial Capability of Manufacture

L. Plant Safety

Table 4-3

SELECTION CRITERIA DEFINITIONS

A. Capital Cost

Objective: Minimize Capital Contribution to Cost of Electricity

Action: Compute Figure of Merit, FOM

$$FOM = \frac{\text{Cost (M\$)}}{\text{Net Energy (MWh/Year)}}$$

Subsystem	Cost Includes	Net Energy Measurement Location
Receiver/Collector (RC Cases)	Heliostats Land, Wire, Absorber, Tower	At Tower Base (MW_th/Year
Storage (S Cases)	Field, Receiver, Pipes, Storage, Heat Exchangers	At Steam Generators (MW_th/Year)
EPGS (EP Cases)	Field, Receiver Storage EPGS	At Transmission Line (MW_eh/Year), but Capacity Factor Assumed the Same for All Options; Actual Selection Is Based on Net Full Load Power (MW_e)

B. O&M Requirements

Objective: Minimize Operating and Maintenance Contributions to Cost of Electricity

Action: Identify Significant O&M Operations Required

Table 4-3 (Cont'd)

C. Control Characteristics

Objective: Design Plant to Provide Rapid, Stable Response to External Events

Action: Assign Ratings (Good, Fair, Poor) to Concepts Based on Ability to Respond to the Following Events:

Event	Modes of Operation
Sunrise	Cold Start, Hot Start
Sunset	Shutdown, Hold
Cloud Passing	Rapid Storage Response
Sun Following	Slow Storage Response
Utility Demand	Rapid Storage Response Hot Start Without Sun
Load Rejection	Rapid Storage Response

D. Forced Outage Rate

Objective: Maximize Plant Availability

Action: Identify Unpredictable Failure Modes in Each Concept

E. Startup Power Requirements

Objective: Minimize Electrical Demand During Startup

Action: Estimate Auxilliary Equipment and Trace Heating Loads for Cold Start

F. Potential for Improvements in Cost and Performance

Objective: Evaluate Concepts on Basis of Optimized Form

Action: Comment if Case Which Represents Concept Is Suboptimal

G. Environmental Intrusion

Objective: Minimize Adverse Effects on Environment

Action: Identify Environmental Effects Due to Normal Construction, and Operating and Maintenance Operations (Excluding Accidents)

Table 4-3 (Cont'd)

H. Land Requirements

Objective: Minimize Land Use to Improve Siting Flexibility

Action: Estimate Land Area and Shape Required

I. Hardware Materials Availability

Objective: Minimize Usage of Materials Which are Scarce Within the United States

Action: Estimate the Weight of Scarce Materials Used in Each Concept

	U.S. Imports* % of 1972 Consumption	Alloy Steel (%)	316 SS (%)	Incoloy 800 (%)	Inconel 625 (%)
Chromium	100	2.25	18	21.0	21.5
Molybdenum	50	1.00	-	-	9.0
Nickel	92	-	8	32.5	61.0

J. R&D Required

Objective: Achieve Early Commercialization of Concept and Minimize Development Cost

Action: Identify Barrier Problems and Critical Experiments

K. Industrial Capability of Manufacture

Objective: Minimize Construction of New Manufacturing Facilities

Action: Identify Those Manufacturing Operations Required Which are not State-of-the-Art

L. Plant Safety

Objective: Minimize Hazards to Plant Personnel and Neighborhood Due to Accidents

Action: Identify Major Accidents Possible

*"Energy Conversion Alternatives Study, "General Electric Phase II Final Report, Vol. II, Part 4, NASA-CR 134949, General Electric Company December 1976

Another important concern is the relative weighting applied to these criteria. Although no weighting was proposed to the various selection groups, certain patterns of emphasis did evolve during the course of the evaluation. Capital Cost (A), Control Characteristics (C), Forced Outage Rate (D) and R&D Required (J) received greater emphasis than the other criteria, while Startup Power Requirements (E) and Hardware Materials Availability (I) were considered less critical. It should be noted that materials availability refers to the scarcity of certain alloying elements and not to the availability of suitably strong or corrosion resistant alloys. This latter concern was addressed under Forced Outage Rate (I) and Industrial Capability of Manufacture (K).

4.3 COMPARISON OF PARAMETRIC OPTIONS

As discussed in Section 3.1, the parametric options were analyzed and evaluated in three separate groups. As shown in Table 4-4, five receiver/collector cases were considered to select the receiver geometry and heliostat type. Table 4-5 identifies the twenty different storage/sodium loop options which were used to select the peak temperature level, storage concept, tank fabrication technique, sodium loop configuration, and receiver flow control method. Table 4-6 lists the EPGS cases analyzed to select the optimum steam cycle. The following sections describe the application of the selection criteria to these thirty cases.

4.3.1 Evaluation of Receiver/Collector Cases

An external cylindrical absorber with enclosed plastic heliostats in a surrounding field arrangement was selected. Data and comments with respect to this selection are listed below.

A. Capital Cost. Table 4-7 lists capital cost, annual energy, and figure of merit for the five cases evaluated. Comparing cases RC1 and RC2 shows that the cavity is not cost effective, even though it improves receiver efficiency by as much as two percentage points. This is principally because the cost for flow control pumps increases as absorber panels are added to make the cavity. Also, as the cavity increases in depth, the convective losses increase and these losses eventually negate the decrease in radiative losses.

Comparing RC1 vs. RC3 and RC4 vs. RC5 seems to indicate that cylindrical receivers with surrounding fields are less expensive than flat receivers with north fields. Finally, comparing RC1/RC3 vs. RC4/RC5 indicates a possible saving available by using enclosed heliostats.

The few cases studied indicated a trend toward the enclosed heliostat design being of lower cost than the glass design. More cases would be required to investigate the sensitivity of this conclusion to uncertainties in the heliostat cost estimates. These additional calculations were not performed because it is not the intent of the present study to perform a complete evaluation of low cost heliostat designs. Rather Cases RC1 through RC5 are intended to determine whether there are differences between the glass and enclosed heliostats which would make one suitable for use with the sodium-cooled receiver while the other is not. For instance, it was discovered that the enclosed, nonfocusing heliostat does not produce suitably high fluxes on a cylindrical receiver, so a focused version has been used in this study. Beyond this, there is little functional difference between the two heliostats.

Table 4-4

PARAMETRIC CASES - RECEIVER/COLLECTOR

For Selection of: Heliostat
Receiver Geometry

Case No.	Heliostat	Receiver Geometry	Temperature (°F)
RC1	Glass	Flat	1100
RC2	Glass	Cavity	1100
RC3	Glass	Cylinder	1100
RC4	Enclosed	Flat	1100
RC5	Enclosed	Cylinder	1100

Table 4-5

PARAMETRIC CASES - SECONDARY LOOP

For selection of: Temperature Level
Storage Concept
Tank Fabrication Technique
Sodium Loop Configuration
Receiver Flow Control

Case No.	Temp. (°F)	Storage Concept	Pressure Reducer	Tank Assembly	Tower Flow Control
S1	1100	1	IHX	Factory	Small EM Pumps on Absorber Panels
S2	1100	2	IHX	Field	Small EM Pumps on Absorber Panels
S3	1100	3	IHX	Field	Small EM Pumps on Absorber Panels
S4	1100	4	IHX	Factory Field	Small EM Pumps on Absorber Panels
S5	1100	3/Salt	IHX	Field	Small EM Pumps on Absorber Panels
S6	1300	1	IHX	Factory	Small EM Pumps on Absorber Panels
S7	1300	2	IHX	Field	Small EM Pumps on Absorber Panels
S8	1300	3	IHX	Field	Small EM Pumps on Absorber Panels
S9	1300	4	IHX	Factory Field	Small EM Pumps on Absorber Panels
S10	1100	3	Valve	Field	Small EM Pumps at Tower Base, Multiple Risers
S11	1100	3	Valve	Field	Large Pump at Tower Base - Small Panel Pumps
S12	1100	3	Valve	Field	Large Pump at Tower Base - Panel Valves
S13	1100	3	Jet Pump	Field	Same as S11
S14	1100	3	Jet Pump	Field	Same as S12
S15	1100	Two Pressure Storage		Factory Field	Small EM Pumps on Absorber Panels
S16	1300	High/Low Hot and Cold Tanks		Factory Field	Small EM Pumps on Absorber Panels
S17	1100	Two Pressure Storage		Factory Field	Small EM Pumps on Absorber Panels
S18	1300	High/Low Hot and Cold Tanks		Factory Field	Small EM Pumps on Absorber Panels
S19	1100	Same as S15/S16		Factory Field	Small EM Pumps on Absorber Panels
S20	1300	Added Valves to Eliminate One Pump		Factory Field	Small EM Pumps on Absorber Panels

Table 4-6

PARAMETRIC CASES - EPGS

For Selection of: Steam Conditions

Case No.	Steam Conditions
EP1	1450 P/1000 F/1000 F
EP2	1800 P/1000 F/1000 F
EP3	2400 P/1000 F/1000 F
EP4	2400 P/1000 F/1000 F HARP
EP5	2400 P/1050F/1050 F HARP
EP6	2400 P/1050 F/1050 F/1050 F

Table 4-7

COLLECTOR/RECEIVER CAPITAL COSTS

	RC1	RC2[a]					RC3	RC4	RC5
Receiver Type	External	Cavity	Cavity	Cavity	Cavity	Cavity	External	External	External
Receiver Shape	Flat	-	-	-	-	-	Cylinder	Flat	Cylinder
Heliostat Type	Glass	Glass	Glass	Glass	Glass	Glass	Glass	Enclosed	Enclosed
Field Type	North	North	North	North	North	North	360°	North	360°
Cost[b] (M$)	70.4	70.4	75.3	80.0	84.7	90.6	73.0	68.5	64.1
Net Energy[c] (TWh/yr)	0.846	0.846	0.864	0.865	0.863	0.857	0.886	0.835	0.863
Figure of Merit ($/MWh)	83	83	87	92	98	106	82	82	74
Land (km^2)	2.9	2.9	2.9	2.9	2.9	2.9	2.8	6.8	5.7
Depth of Cavity (m)	-	0	4.9	9.8	14.6	20.8	-	-	-
Receiver Efficiency(%)	-	91.4	93.3	93.5	93.2	92.6	-	-	-

[a] Scaled from RC1, Appendix A

[b] Heliostats, land, wire, tower, receiver, riser/downcomer, pumps and site preparation

[c] Delivered at the base of the tower, net, after thermal equivalent of field and receiver auxiliaries have been deducted

B. O&M Requirements. The glass heliostat requires washing; the enclosed heliostat is not washed but requires replacement of the panels in the enclosure once every 15 years and cleaning of the blower filter once a month. No differences were noted between O&M for cylindrical vs. flat receivers.

C. Control Characteristics. No differences were noted among heliostat and receiver alternatives.

D. Forced Outage Rate. The glass heliostat has poorer aiming accuracy in high winds, thus reducing plant output. The flat receiver has higher flux levels, aggravating the low cycle fatigue problem and possibly leading to poorer reliability.

E. Startup Power Requirements. No differences were noted.

F. Potential for Improvements in Cost and Performance. Nonfocusing enclosed heliostats did not provide high enough flux levels to properly utilize the sodium-cooled cylindrical receiver. Cases RC4 and RC5 were reevaluated using focused heliostats. The heliostat costs used in this analysis correspond to the mature technology at high production levels; little improvement is expected.

Selective coatings would greatly improve both receivers.

G. Environmental Intrusion. Enclosed heliostats require the clearing of more land, while glass heliostat washing may contaminate the environment. No differences were noted with respect to the receiver concepts.

H. Land Requirements. Table 4-7 lists the land area required for each case. The enclosed heliostat uses significantly more land than the glass concept because of the larger number of heliostats and the additional spacing required to reduce shading and blocking by the enclosure.

I. Hardware Materials Availability. No differences were noted for the heliostats. Receiver differences are summarized below:

	Weight (thousands of pounds)	
Material	Flat Receiver	Cylinder Receiver
Inconel 625	233	421
2-1/4 Cr - 1 Mo	150	120
316SS	330	516

J. R&D Required. The enclosed heliostat requires testing of plastics for enclosures and reflectors to determine life. The nonenclosed heliostat needs a large supply of low-iron float glass to meet reflectivity goals. No differences were noted for the receiver concepts.

K. Industrial Capability of Manufacture. Glass heliostats require an on-site manufacturing facility; enclosed heliostats are shipped ready for field installation. No differences were noted for the receivers.

L. Plant Safety. No differences were noted for heliostats. The flat receiver is more likely to produce damaging reflection because of the high incident flux.

4.3.2 Evaluation of Storage Subsystem Cases

Case S11 was selected; it has the following characteristics:

Peak temperature:	1100 F
Storage concept:	sodium in separate hot and cold tanks
Sodium loop configuration:	no IHX; pressure reducer is a throttle valve
Receiver flow control:	small EM trim pumps on absorber panels

The data and logic used in this selection are described below according to selection criteria.

A. Capital Cost. Table 4-8 summarizes the cost and annual energy balance data used to compute the figure of merit. Collector costs, C_C, were scaled from Case RC3 by the equation:

$$C_C = C_{CB} \left(\eta_{RB}/\eta_R\right) \qquad (4\text{-}1)$$

where: C_{CB} = Base collector cost = 47.5 M$

η_{RB} = Base receiver efficiency = 90.6%

η_R = Receiver efficiency

Receiver costs were obtained by scaling the results of Section 3.3.4 for a tower height of 170 meters and a receiver size of 12.5 meters (diameter) by 12.5 meters (height). These costs are somewhat different from Case RC3 because RC3 has no IHX and higher pumping costs. Incremental pump costs for the no IHX cases in Table 4-7 are included in the storage loop cost (taken from Section 3.4.7).

Net annual energy, E_N, was computed by the following equation:

$$E_N = Q\,[x(e-1) + 1] - P_S \times 8009.2 \qquad (4\text{-}2)$$

where: Q = net annual energy at base of tower (includes deduction for tower pumps and field auxiliaries) = 0.386 TW_th (Case RC3)

x = fraction of energy passed through storage = 18.5%

e = storage turnaround efficiency (insulation, turbine start, thermocline)

P_S = power for storage loop pumps plus incremental pumping due to throttle valve

Equation (4-2) assumes that the storage pumps run on a 40 percent capacity factor.

The figure of merit, FOM, given by

$$FOM = \frac{C_T \times 10^6}{E_N} \qquad (4\text{-}3)$$

Table 4-8
STORAGE SUBSYSTEM CAPITAL COSTS

Storage Subsystem Cases:			S1	S2	S3	S4	S5	S6	S7	S8	S9	S10	S11	S12	S13	S14	S15	S16	S17	S18	S19	S20
	Temperature	(°F)	1100	1100	1100	1100	1100	1300	1300	1300	1300	1100F	1100F	1100F	1100F	1100F	1100F	1300	1100	1300	1100	1300
	Storage Concept		1	2	3	4	3(Salt)	1	2	3	4	3	3	3	3	3	3	3	3	3	3	3
	Tank Type		Factory	Field	Field	Fact./Field	Field	Factory	Field	Field	Fact./Field	Field	Field	Field	Field	Field	Field	Field	Field	Field	Field	Field
	Pressure Reducer		IHX	IHX	IHX	IHX	IHX	IHX	IHX	IHX	IHX	Valve	Valve	Valve	Jet Pump	Jet Pump	Two Pressure Storage			Two Pressure Storage		
	Receiver Efficiency	(%)	90.6	90.6	90.6	90.6	90.6	89.5	89.5	89.5	89.5	90.6	90.6	90.6	90.6	90.6	90.6	89.5	90.6	89.5	90.6	89.5
Cost	Collector[a]	(M$)	47.5	47.5	47.5	47.5	47.5	48.1	48.1	48.1	48.1	47.5	47.5	47.5	47.5	47.5	47.5	48.1	47.5	48.1	47.5	48.1
	Receiver[b]	(M$)	17.4	17.4	17.4	17.4	17.4	16.8	16.8	16.8	16.8	17.4	17.4	17.4	17.4	17.4	17.4	16.8	17.4	16.8	17.4	16.8
	Storage Loop	(M$)	66.6	62.0	63.3	64.6	80.4	56.8	56.3	55.1	55.8	46.4	41.0	38.3	39.3	36.6	55.6	49.3	54.9	48.0	53.0	47.2
	Total	(M$)	131.5	126.9	128.2	129.5	145.3	121.7	121.2	120.0	120.7	111.3	105.9	103.2	104.2	101.5	120.5	114.2	119.8	112.9	117.9	112.1
Losses	Storage Loop Pumps	(MWe)	3.0	3.2	2.1	3.1	1.2	1.9	2.3	1.3	1.9	6.2	6.2	7.2	2.0	3.0	1.6	1.1	1.3	0.8	1.6	1.1
	Storage Efficiency	(%)	93	98	98	93	98	95	98	98	95	98	98	98	98	98	98	98	98	98	98	98
Net Annual Energy to Steam[c]		(TW_th)	0.850	0.857	0.866	0.850	0.873	0.863	0.864	0.872	0.872	0.833	0.833	0.825	0.867	0.859	0.870	0.874	0.872	0.876	0.870	0.874
Figure of Merit		($/$MW_th$)	155	148	148	152	166	141	140	138	138	134	127	125	120	118	139	131	137	129	136	128

[a]Assumes 360° field, 1100 F/1300 F, 170 m tower (Case RC3). Base values are C_{CB} = 47.50 M$, η_{RB} = 90.6%, Q = 0.886 TW_th

[c]Assumes that solar multiple of 1.5 implies that 18.5% of annual energy passes through storage; also assumes 40% capacity factor on pumps.

[b]Based on costs in Section 3.3.4.

where C_T = Collector, receiver, and storage loop total cost total cost

Comparing the two groups--S1 through S4 and S6 through S9--shows a clear trend toward lower cost for the 1300 F systems. Within these groups there is not much variation in figure of merit, so none of the storage concepts appears to have a clear cost advantage over the others.

Molten salt increases the cost of heat exchanges in the secondary loop in Case S5, making this concept noncompetitive.

The throttle valve cases, S10 through S12, show clearly that lower system costs are achieved when the IHX is eliminated.

The jet pump cases appear to be the least expensive systems. This may be illusory, however, since these systems may require expensive backup pumps for times when jet pump output does not meet the system flow demand.

Two-pressure storage (Cases S15 through S20) seems to offer a cost advantage with respect to the IHX cases at both 1100 F and 1300 F; however, this concept is not as effective in reducing costs as the throttle valve concepts (Cases S10 through S12).

B. O&M Requirements. The higher temperature concepts (1300 F) are likely to require more maintenance to monitor sodium corrosion effects (see Appendix L) and to maintain calibration of control instrumentation.

Storage Concepts 1, 2, and 4 all involve thermal cycling of some storage elements; this will increase maintenance of the insulation and piping over Concept 3. Concept 2 has a large number of valves which will increase maintenance over the other concepts. Case S12 also involves a large number of valves on the control panels. Jet pumps are a low maintenance item.

C. Control Characteristics. Concept 1 is rated poor because of the requirement that the tanks be charged and discharged at only one rate which corresponds to minimum thermocline dispersion. Concept 2 is rated fair because it requires a dispatch logic to find the appropriate tank for charge and discharge and there are many valves to control. Concept 3 is rated good because it permits completely independent operation of the receiver and steam generator loops.

Concept 4 is an improvement over Concept 1, but like Concept 2 it requires a dispatch logic and forecasting of input and output to decide whether to convert energy from liquid storage to solid storage or vice versa (rated fair).

Eliminating the IHX and using a throttle valve for pressure reduction will complicate the tower flow control . Thus, from a controls point of view, Cases S10 through S12 are not as desirable as S3 and S8, and are rated fair.

The jet pumps may not be able to match system demands at all operating points and are rated poor (Cases S13 and S14).

Two-pressure storage has the desirable tower flow control and storage flexibility of the IHX cases, S3 and S8, so Cases S15 through S20 are rated good.

D. Forced Outage Rate. The primary concerns here relate to temperature level, control valves, and storage thermal cycling.

At 1100 F the selected materials are 316 stainless steel and Incoloy 800. As discussed in Appendix L, there is enough sodium experience with these materials to have good assurance of a reliable design. This is especially important in the absorber panels where large thermal stresses are found with frequent cycling and long hold times.

The material selected for 1300 F service, Inconel 625, has not been tested in 1300 F sodium. However, tests with similar materials (see Appendix L) indicate the possibility of grain boundary attack. This material also ages at temperature in steam and loses ductility. An alternate material, Inconel 617, has been identified which does not appear to age and is expected to display better compatability with sodium due to its different composition, but these claims have not been demonstrated. Stainless steel (316) has been tested at 1300 F in sodium and its creep/fatigue characteristics are known at that temperature. However, 316 SS would be too expensive to use for storage tanks because it has low strength at 1300 F and cannot be used in the superheater due to stress corrosion on the steam side.

Control valves have been a reliability problem in sodium systems because of galling and seizing of the moveable parts. This makes them undesireable for applications where frequent inspection and maintenance are not possible, such as the valves required on the absorber panels in Cases S12 and S14. The throttle valve required at the downcomer base for Cases S10 through S12 can be made reliable by redundant design and frequent inspection.

Storage concepts 1, 2, and 4 all require that storage vessels cycle between 1100 F (or 1300 F) and the low end temperature of about 630 F. This can cause thermal stress damage and unpredictable failure. In Concept 3 the tanks do not change temperature; this concept is preferred.

E. Startup Power Requirements. This plant will probably experience only one or two truly "cold starts" in its lifetime. The sodium in storage can be circulated around the system to keep components hot with minimal trace heating. This could continue for several months before the sodium temperature would fall to 350 F where increased electric heating would be needed to prevent freezing. Thus only the initial plant startup and a startup after a six-to eight month outage would require the massive amounts of energy involved in bringing the storage media from ambient temperature up to 350 F. Therefore, power required for a cold start is not an important consideration in selecting the storage concept.

F. Potential for Improvements in Cost and Performance. The cases considered here demonstrate the cost advantages of two temperature levels (1100 F and 1300 F) which correspond to two completely different classes of materials (high alloy steels and nickel-base alloys respectively). These cases have not been fine-tuned to select the optimum temperature for each class of materials. For instance there is a gain in the strength of Inconel 625 in dropping from 1300 F to 1250 F which might offset the increase in storage volume and achieve lower cost.

Concept 4 may have an optimum split between liquid and solid storage capacity; this has not been investigated.

Pipe flow speed has been maintained at about 20 feet/second. There may be a more economical choice since the flow speed represents a trade-off between piping and pumping costs.

G. Environmental Intrusion. All systems studied have the same effects on the environment.

H. Land Requirements. All systems can be located within the tower exclusion area.

I. Hardware Materials Availability. Three cases were taken as being typical; these cases are summarized below:

Material	Weight (millions of pounds) Case S3	Case S8	Case S6
Inconel 625	—	2.88	1.38
Incoloy 800	3.66	—	—
2-1/4 Cr - 1 Mo	3.66	2.88	—

Cases S3 and S8 show the change in materials when going from 1100 F to 1300 F with Concept 3. Case S6 shows that sodium + iron storage, Concept 1, saves the high alloy material.

J. R&D Required. Concept 1 (sodium + iron) requires testing to demonstrate thermocline stability and to investigate the effects of repeated thermal expansion cycles.

The throttle valve required by the no-IHX cases requires testing to establish reliability.

Inconel 625 and the alternate Inconel 617 are unproven materials for sodium service and require extensive testing to prove their suitability for this application and establish code case data.

The development required by sodium + iron storage and the 1300 F concepts is probably inconsistent with the objective of installing a pilot plant in the early 1980s. Throttle valve development could be performed in this time span.

K. Industrial Capability of Manufacture. Piping elbows, tees, and valves are not currently available in Inconel 625 and this could limit the commercialization of the 1300 F systems.

The longitudinal joints between adjacent tubes in the absorber panels must be welded or brazed. Welding such thin wall tubing (0.035 inch) can be done with radio frequency techniques but remains to be demonstrated in this geometry. Brazing a structure this size would require a large furnace with precise atmosphere and temperature control. Furnaces like this have been built but are not common.

The additional brazed sleeves protecting the steam generator tube/tubesheet welds require effort to perfect the installation process.

L. Plant Safety. Large quantities of sodium could be released if a tank ruptures; this would be less of a problem with Concept 1 which has the minimum sodium inventory.

4.3.3 Evaluation of EPGS Cases

Section 3.5 describes the effect of steam cycle selection on plant capital cost. The lowest cost option is Case EP5, which has throttle steam at 2400 psig and 1050 F, with reheat to 1050 F.

Cost is the dominant consideration in this selection, since all turbines considered had approximately the same reliability and load-following capability. Therefore, the only criterion operative in selecting the EPGS was "Capital Cost."

Case EP5 was an extension of the parametric analysis to a new set of steam conditions and was completed after EP4 had been selected as the conceptual design basis. These throttle steam conditions are 2400 psig and 1000 F with reheat to 1000 F. Two feedwater heaters above the reheat point are used.

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