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SELECTION AND CONCEPTUAL DESIGN OF AN ADVANCED THERMAL ENERGY STORAGE SUBSYSTEM FOR COMMERCIAL SCALE (100 MWe) SOLAR CENTRAL RECEIVER POWER PLANT

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— Final Report —

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Prepared for

Sandia National Laboratories
Livermore, California

by

BABCOCK & WILCOX
Nuclear Power Group
Nuclear Power Generation Division
P. O. Box 1260
Lynchburg, Virginia 24505

Babcock & Wilcox

Babcock & Wilcox
Nuclear Power Group
Lynchburg, Virginia

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Selection and Conceptual Design of an Advanced Thermal Energy Storage Subsystem for Commercial Scale (100 MWe) Solar Central Receiver Power Plant

Key Words: Thermal Energy Storage, Moving Bed Thermal Energy Storage System, Water/Steam Solar Receivers, Fine Granular Particulate

ABSTRACT

Advanced thermal energy storage concepts were developed and evaluated which are applicable to a 100 MWe solar central receiver plant using water/steam as the working fluid. Operating conditions studied were 510C/10.1 MPa (950F/1465 psia) from the receiver and 299C/2.72 MPa (570F/395 psia) from storage. Three concepts were selected that offered potential for cost and performance improvements over the oil/rock concept presently being installed at the central receiver 10 MWe pilot plant under construction in Barstow, California. From the three concepts selected, the moving bed thermal energy storage system (MBTESS) using a free flowing refractory material as the heat transport and storage media was chosen. A conceptual design was developed, including estimates for cost and performance. Suggestions were made for further development work leading to full scale implementation of the concept.

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1. SUMMARY

Under contract to Sandia National Laboratory Livermore, Babcock & Wilcox performed a six-month concept development study of an advanced solar thermal energy storage system. The study included the following scope of work:

- Select a preferred advanced thermal energy storage concept.
- Prepare a design and budgetary cost estimate for a commercial-scale system.
- Assess the design with respect to potential improvements and limitations.
- Assess the design with respect to higher temperature, higher pressure operation.
- Prepare a plan for construction and testing of a subsystem research experiment in Phase II.
- Outline a development plan and schedule for a commercial-scale subsystem.

The objectives of the project were to prepare a conceptual design that would offer cost/performance advantages over the oil/rock thermocline concept selected for the Barstow Pilot Plant and that would be applicable to solar-repowered electric generating plants.

The following concepts were considered:

- Caloria/granite — thermocline
- Moving sand bed — hot/cold tanks
- Air/rock — thermocline
- Molten salts — hot/cold tanks
- Pressurized hot water — underground hot tanks
- Sodium — hot/cold tanks
- Syltherm 800® /taconite — trickle thermocline

System requirement specifications were provided for design and performance parameters.[1] The receiver working fluid conditions and the storage discharge steam conditions specified were those of a 100-MWe, commercial central receiver solar thermal power system plant[2], which the Barstow 10-MWe Pilot Plant represents. These values were as follows:

Extractable storage capacity, MWht	1710
Charging level, MWt	260
Discharging level, MWt	285
Duration, hours	6
Storage tank operating pressure	Atmospheric
Receiver outlet temperature, C (F)	510 (950)
Receiver outlet pressure, MPa (psia)	10.1 (1465)
Storage outlet temperature, C (F)	299 (570)
Storage outlet pressure, MPa (psia)	2.72 (395)

The concepts were initially screened by analyzing their characteristics, including the following:

- Energy storage density
- Media and containment materials
- Thermal efficiency
- Structural considerations
- Maintenance requirements
- Storage medium deterioration/replacement

Engineering judgment was applied to compensate for a scarcity of conceptual design information for the performance conditions specified. The number of concepts for further evaluation was reduced to four, including

- Air/rock thermocline using air as the heat transport medium and a rock bed as the thermal storage medium.
- Moving sand bed using a fine, free-flowing refractory powder as both heat transport and storage media.
- Molten salt sensible heat using a HITEC® salt (40% $NaNO_2$, 7% $NaNO_3$, and 53% KNO_3 by weight) as the heat transport and storage fluid.
- Oil/rock thermocline using a Caloria oil as the heat transport fluid and granite as the storage medium.

Conceptual designs were prepared for each concept based on the system requirement specifications, including system schematics, subsystem mass balances, temperatures, and heat exchanger and storage media parameters. Costs were compared for selected major system equipment (heat exchangers, storage tanks, pumps/lifts, and thermal storage medium). Selection criteria were developed based on the system requirement specifications and the project objective statements:

- Equipment costs
- Round trip efficiency
- Availability/reliability/maintenance
- Development requirements and risks
- Environmental and safety aspects
- Applicability to higher temperatures

The likelihood of improvements from further optimization was also considered.

The results of the evaluation showed the oil/rock thermocline concept to be the lowest cost option at the Barstow temperature/pressure conditions. However, the 332C (630F) temperature limitations of the Caloria (heat transport fluid) preclude its application to higher temperature operation. The air/rock concept appeared to involve slightly lower capital costs than the moving sand bed or molten salt concepts, but the air/rock round trip efficiency was about 15% lower because of the requirement to drive air circulating fans. As a result, the moving sand bed and molten salt concepts provided lower evaluated costs than the air/rock concept.

The moving sand bed concept was found to have slightly lower capital and operating costs than the molten salt concept. The moving sand bed is capable of operation at storage temperatures above the maximum working temperature and below the freezing temperature of molten salts. It was also concluded that the cost of the moving sand bed concept could be further reduced by optimization of the number of tanks and lengths of the lifts. Therefore, the moving sand bed concept was selected for further conceptual design development.

Optimization of this concept reduced the number of system components and costs of major equipment by $11 million. The number of tanks was reduced from 18 to 12, the number of lifts from 36 to 12. The calculated round trip efficiency was improved from 68.9 to 70.0% due to reduced lift power requirements. As shown in Figure 1-1, the optimized design provides two cold and two hot hopper-type storage bins connected by helical screw lift conveyors, where the outer casing and helical screw rotate as an assembly. The charge and discharge heat exchangers are mounted atop the storage bins for easy access. The particulate material (sand) is also the heat transport medium. Sand flows by gravity over the heat exchanger surface at a low velocity. Special design features, such as inclined tubes, an inclined shell, and the tube arrangement, prevent material stagnation and provide a high sand bed density throughout the heat exchangers for maximum heat transfer. The basic design characteristics of the moving bed thermal energy storage system design are given in Table 1-1.

A budgetary cost estimate was prepared, including all major components and systems. The total system cost was estimated at $26.2 million in June 1980 dollars. A breakdown of energy- and power-related costs is as follows:

MBTESS Energy-, Power-, and Specific-Related Costs (June 1980 $ × 10^3)

Energy related cost, C_s

Item	Direct field cost	Indirect cost	
Excavation	97	7	
Backfill and Compaction	662	125	
Foundations and Footings	92	10	
Storage Structure	3,427	629	
Insulation	362	105	
Medium	1,802	--	
Subtotal	6,442 +	876 =	$7,318
	Contractor's Profit		732
	Engineering		552
		Total	$8,602

Power related cost, C_p

Item	Direct field cost	Indirect cost	
Lifts	7,461	731	
Piping	380	18	
Heat Exchangers	4,978	171	
Aux. Equipment	181	16	
Controls and Inst.	279	--	
Equipment Covers	337	74	
Subtotal	13,616 +	1,010 =	$14,626
	Contractor's Profit		1,463
	Engineering		1,463
		Total	$17,552

Total cost: $26,154

The reference design was assessed with respect to potential limitations and potential improvements. Special equipment design or operating procedures required to control component wear to acceptable levels must be defined by testing. The heat exchanger configuration and performance must also be confirmed by testing. The thermal storage material was assumed to be silica sand (SiO_2) with a 30° angle of repose.

Assessment of safety, environmental, and land use aspects revealed no unique constraints, and none are expected due to the chemically inert medium. Dust control measures will be required, but these methods are considered to be well known.

Assessment of the design at higher storage temperatures revealed substantial improvements in cost and performance. The total system cost was estimated at \$10.5 million in June 1980 dollars compared to \$26 million for the reference design. The design developed is considered applicable to storage temperatures up to 538C (1000F) with only minor modifications. At a 538C (1000F) charge temperature, 62% less storage material and tankage are required due to the increased stored energy density. Savings in power-related equipment also result from reduced material flow rates. These savings are reflected in the breakdown of energy- and power-related costs for 538C (1000F) storage below.

High Temperature (538C Storage) MBTESS Energy-, Power-, and Specific-Related Costs (June 1980 \$ $\times 10^3$)

Energy related cost, C_s

Item	Direct field cost	Indirect cost	
Excavation	38	3	
Backfill and Compaction	258	49	
Foundations and Footings	36	4	
Storage Structures	1,352	245	
Insulation	141	41	
Medium	703	--	
Subtotal	2,528 +	342 =	\$2,870
	Contractor's Profit		287
	Engineering		217
		Total	\$3,374

Power related cost, C_p

Item	Direct field cost	Indirect cost	
Lifts	2,542	249	
Piping	423	10	
Heat Exchangers	2,174	57	
Aux. Equipment	181	16	
Controls and Inst.	115	--	
Equipment Covers	112	25	
Subtotal	5,547 +	357 =	\$5,904
	Contractor's Profit		590
	Engineering		590
		Total	\$7,084

Total cost: \$10,458

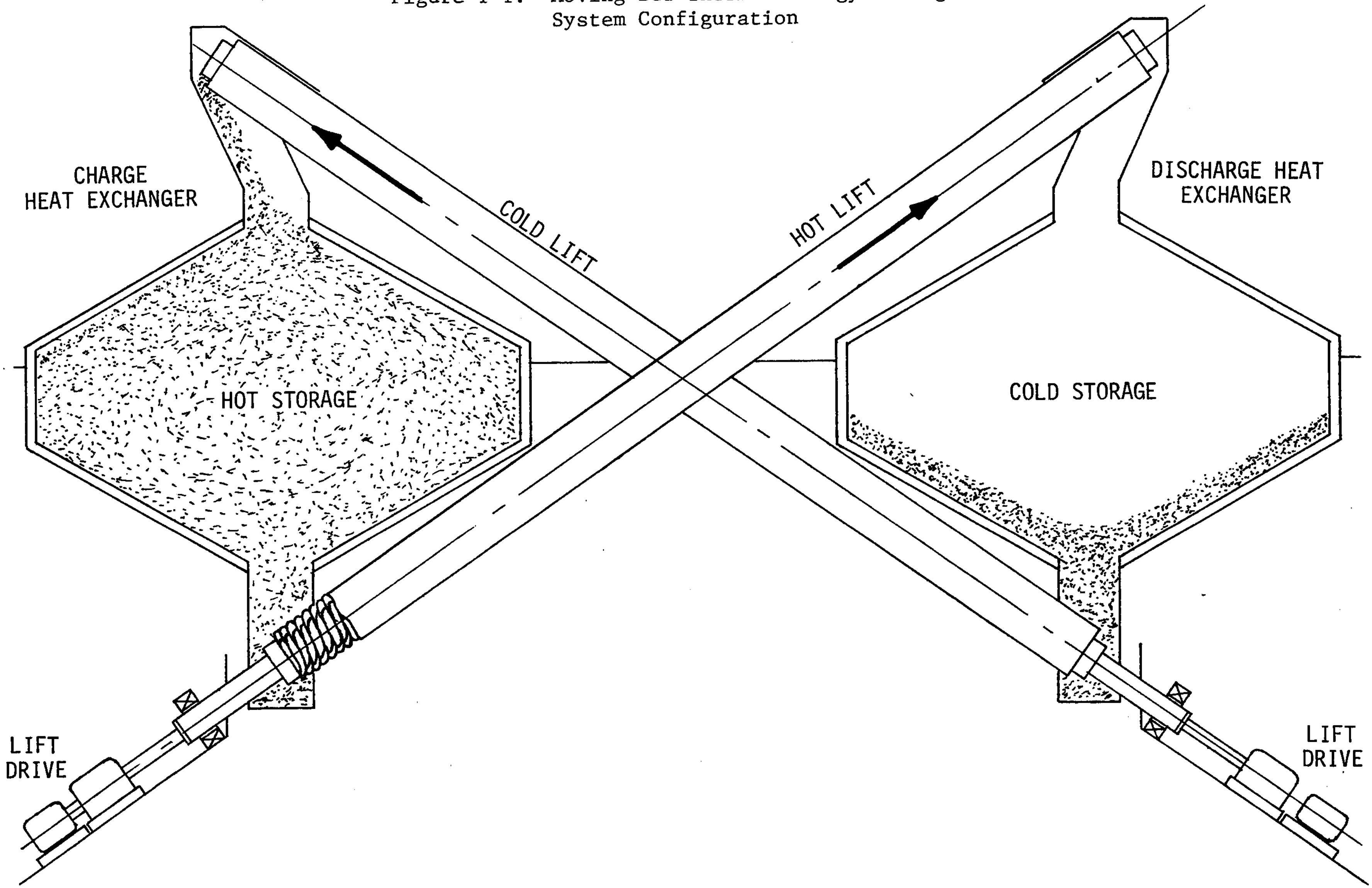
Technical issues in the moving sand bed concept discussed previously require resolution through a program of laboratory subscale experiments to establish bed material behavior in system components, the most important of which is considered to be the heat exchanger. A bed material program was defined to identify candidate materials and characterize their behavior under simulated laboratory conditions. A heat transfer and flow study program was also defined to determine the flow distribution and heat transfer coefficients of the candidate bed materials in the heat exchangers. A development plan was also prepared for a subsystem research experiment leading to a commercial-scale design.

The principal conclusion of this study is that a thermal energy storage concept can be developed which is capable of operation over a wide temperature range and which is compatible with all major receiver working fluids. The system has single-stage storage and the capability to operate at a 538C (1000F) charge temperature and a low discharge storage temperature. This provides a high storage energy density with the absence of considerations associated with media phase change (freezing). The concept offers greatly improved economics for high-temperature applications. At medium temperatures, the system provides a backup to the oil/rock thermocline concept in the event that oil fill/replacement costs should exceed the projected range.

Table 1-1. MBTESS Design Characteristics

Storage Medium		
Material	SiO_2	
Operating range, C (F)	204-332 (400-630)	
Density, kg/m^3 (lb/ft^3)	1522 (95)	
Specific heat, J/kg-°K (Btu/lb-°F)	1030 (0.246)	
Particle size, 10^{-3} mm (10^{-4} in.)	44-74 (17-29)	
Void fraction	0.40	
SiO_2 mass, 10^8 kg (10^8 lb)	0.57 (1.27) working 0.92 (2.02) for costing	
Tanks		
No. of storage tanks	4	
Volume per tank, 10^4 m^3 (10^5 ft^3)	2.51 (8.86)	
Total volume, 10^4 m^3 (10^6 ft^3)	10.1 (3.54)	
Material	ASTM A-53 (carbon steel)	
Design temperature, C (F)	343 (650)	
Insulation material, internal		
Sides and bottom, mm (in.)	SiO_2, 457.2 (18)	
Roof	Thermal Wool II, 114.3 (4.5)	
Heat Exchangers	**Discharge**	**Charge**
Number	6	6
Duty, MWt (10^8 Btu/h)	47.5 (1.62)	43.4 (1.48)
Design flow		
Tube side (W/S), 10^4 kg/h (10^5 lb/h)	6.85 (1.51)	4.05 (8.92)
Shell side (SiO_2), 10^6 kg/h (10^6 lb/h)	1.45 (3.19)	7.94 (1.75)
Surface area, m^2 (ft^2)	819 (8820)	1124 (12,100)
Tube OD, mm (in.)	19 (0.75)	19 (0.75)
Material	Carbon moly	Carbon moly
Lifts		
Number	12	
Length, m (ft)	56.7 (186)	
Outside diameter, m (ft)	1.95 (6.4)	
Material	Carbon steel	
Lift angle	35°	
Capacity/lift, m^3/s (ft^3/s)	0.036 (12.7)	
Horsepower/lift, hp	800	
Lift speed, rpm	29	

Figure 1-1. Moving Bed Thermal Energy Storage System Configuration



2. INTRODUCTION

The development of economical and reliable thermal energy storage is considered a necessity for the successful commercialization of solar thermal power stations. A solar central receiver pilot plant (10 MWe) is being constructed at Barstow, California. The thermal energy storage subsystem (TESS) for this plant is an oil/rock dual-media storage system. As lead laboratory for the thermal energy storage for solar thermal applications (TESSTA) program, Sandia National Laboratories Livermore initiated a development program to produce an advanced TESS (ATESS) offering cost/performance advantages over the oil/rock concept. The development program is divided into three phases.

This report describes the results of the Phase I work performed under Sandia contract No. 20-2990A for identification and conceptual design of an ATESS during the period from September 1980 to February 1981. The objectives of Phase I were to identify a concept that has cost and performance advantages over the oil/rock concept and to develop a commercial-scale (100 MWe) conceptual design supported by budgetary cost estimates.

If authorized, Phases II and III will include the activities necessary for successful operation of a commercial-scale advanced storage subsystem meeting the established cost and performance goals. The activities in Phases II and III must be planned so as to achieve successful operation of the commercial-scale system in the specified five-year time frame from the start of Phase I.

Phase II includes the design, development, and testing needed to resolve technical issues and uncertainties identified in Phase I and to provide the data necessary for detailed design in Phase III. Phase II activities include preliminary design of the storage subsystem and detailed design, analysis, or testing of critical components that may be necessary to support the detailed design of a Subsystem Research Experiment (SRE). Phase II also includes the design, construction, and operation of the SRE. The SRE must be designed to a scale that permits investigation of any critical fabrication techniques and aspects of actual subsystem startup, operation, and shutdown. A concurrent activity in Phase II assesses the effect of the SRE on the design of the commercial-scale system. Phase II also includes cost and schedule estimates for the commercial-scale system in Phase III.

Phase III includes the detailed design, construction, startup, and operation of a commercial-scale storage subsystem. Detailed design activities would include the development of designs specific to the requirements of the site, the receiver subsystem, and the electric power generation subsystem. The design would also reflect the results of the evaluation of SRE data and experience.

2.1. Objectives

The specific objectives for Phase I were as follows:

- Select a preferred ATESS concept.
- Produce a commercial-scale conceptual design and budgetary cost estimate.
- Assess the design with respect to potential improvements and limitations.
- Assess the design with respect to other working fluid conditions (higher temperatures and pressures).
- Develop a plan for construction and testing of an SRE in Phase II.
- Outline a development plan for a commercial-scale system.

2.2. Technical Approach

The following subsections discuss the principles, techniques, and parametric studies employed in accomplishing the major segments of the development activity. The approach selected was chosen to enable early selection of a concept offering cost/performance improvements over previous systems. Time did not permit optimization of all the concepts prior to final selection. Therefore, a proponent of one of the systems not chosen for further study may conclude that important information has been overlooked. This difficulty was recognized at the inception. However, it was B&W's judgment that the study could be of greatest value by providing specific design and cost data on the chosen concept.

2.2.1. System Requirements and Selection Criteria (Task 1)

System requirements defined by Sandia[1] provided the bases for the preliminary concept selection criteria.

2.2.2. Engineering Analysis (Task 2)

The existing literature was reviewed in Task 2 to identify several thermal storage concepts having superior qualities compared to systems already under development. The literature search included the use of the unpublished results of B&W-sponsored research and earlier published surveys.

Several important aspects of thermal energy storage systems were compared to select the systems to be considered for further analysis in Task 3:

- Comparison of required storage capacity based on the thermal energy storage density of the storage media.
- Comparison of storage media and containment materials.
- Comparison of round trip efficiency.
- Comparison of containment vessel structural considerations.
- Comparison of ullage maintenance requirements and storage containment capacity.
- Comparisons of storage fluid deterioration, contamination, purification, replacement requirements, and costs.

A number of storage concepts were considered in the engineering analysis based on the results of prior work. One of the concepts was a molten salt sensible heat design. Both the molten salt sensible heat and the oil/rock concepts have been studied extensively and reported on to provide a comparison base for evaluation of advanced storage systems. A unique storage media that was considered is a moving bed of fine, free-flowing refractory powder, a concept originated by B&W.

2.2.3. Selection of Preferred System (Task 3)

The preferred storage system concept was selected by comparing the systems being considered against selection criteria established by a review of the system requirements provided by Sandia.[1] To ensure that the preferred system was more promising overall and to provide a standard for comparison, the oil/rock system selected for the Barstow plant was also compared to the criteria. This comparison defined a set of criteria against which the alternate systems were judged. The system was chosen that provides the greatest overall promise.

Three levels of screening were used to select the preferred concept. The first was a literature search from which seven candidate ATESS concepts were identified in accordance with the ground rules discussed in section 3.1. In the second screening, this list of seven was further narrowed to three concepts in addition to oil rock. This is reported in section 3.2. These three concepts were analyzed to select a preferred concept as an alternative to oil/rock. The final selection of an ATESS is described in section 4.

2.2.4. Conceptual Design and Cost Estimate (Task 4)

The thermal energy storage concept selected in Task 3 was designed to meet the codes, standards, and operating parameters specified by Sandia.[1]

The subsystem and its component designs were directed toward an optimum combination of low equipment cost, minimum R&D requirements, operating flexibility, and round trip efficiency. The subsystem mass balances, flow balances, and temperatures were calculated to establish the heat exchanger design parameters and the quantity of storage media needed to meet dishcarge heat requirements and heat and pumping losses.

Costs were obtained based on four equipment categories:

- Heat exchangers
- Storage tanks
- Pumps or lifts
- Thermal storage medium

Piping lengths and sizes were estimated in order to calculate pump head requirements. Insulation thicknesses were calculated based on an assumed subsystem target heat loss of 3.6% from the beginning of the charge period to end of the discharge period, assuming a normal operating day.

Storage tank designs considered compatibility of the storage media with insulation, tank support configuration, tank wall thermal gradients, safety requirements, and siting requirements. Heat exchanger designs considered the compatibility of materials with storage media and receiver coolant (water/steam). As the design progressed, the system ullage, leak detection, fire protection, and safety (as applicable) were considered. System performance analysis was carried out for the required modes of operation to ensure the adequacy of the system design. The conceptual design budgetary cost estimate is reported in section 5.

2.2.5. Assessment of Commercial-Scale Subsystem (Task 5)

In Task 5, the conceptual design produced in Task 4 was assessed from the standpoint of potential limitations and future improvements. To maintain objectivity, this assessment was carried out not only by those who produced the design but also by others experienced in the technology and independent of the designers. To accomplish this, technical personnel from B&W's Research and Development and Fossil Power Generation Divisions were utilized through design review. The assessment concentrated on several areas of interest, including improvements in system cost, performance, and fundamental limitations from a technical, environmental, or space standpoint. Where potential improvements were identified, the activities required to realize them were defined. Where potential limitations were identified, the activities required to eliminate or overcome the limitations have been defined. This work is reported in section 6.

2.2.6. Assessment of the Preferred Storage System for Other Receiver Working Fluids (Task 6)

The preferred storage system design was assessed to determine its applicability to receiver designs operating at improved working fluid temperature/pressure conditions. Included in the assessment were the following:

- Establishment of new requirements, if applicable.
- Suitability of the storage medium for operating conditions.
- Design and material selection for heat exchangers for the improved service conditions.
- Adequacy of subsystem component designs.
- Round trip efficiency of the subsystem.

In addition, the design was assessed to determine whether the subsystem could be used as a storage unit delivering steam to the turbine inlet, thereby maintaining high operating efficiency. This work is reported in section 7.

2.2.7. Development Activities for Conceptual Design (Task 7)

The work of the previous tasks was directed toward producing a conceptual design and cost estimate for the preferred concept and assessing the design with respect to potential improvements, potential limitations, and applicability to other working fluid conditions. In Task 7, further activities were defined that are necessary to verify uncertainties in the conceptual design, eliminate or overcome identified limitations, realize potential improvements, and provide the information for detailed design, construction, and operation of a Subsystem Research Experiment (SRE). This is reported in section 8.

2.3. Technical Team

The work was performed by a technical team from the Advanced Energy Systems Engineering Section of B&W's Nuclear Power Generation Division (NPGD). The team was made up of a technical manager and three task leaders supported by additional personnel from NPGD's Engineering Department. Technical support was also provided by B&W's Research and Development and Fossil Power Generation Divisions. Figure 2-1 shows the project organization chart, and Figure 2-2 is the project schedule and task breakdown.

Figure 2-1. Thermal Energy Storage Project Organization

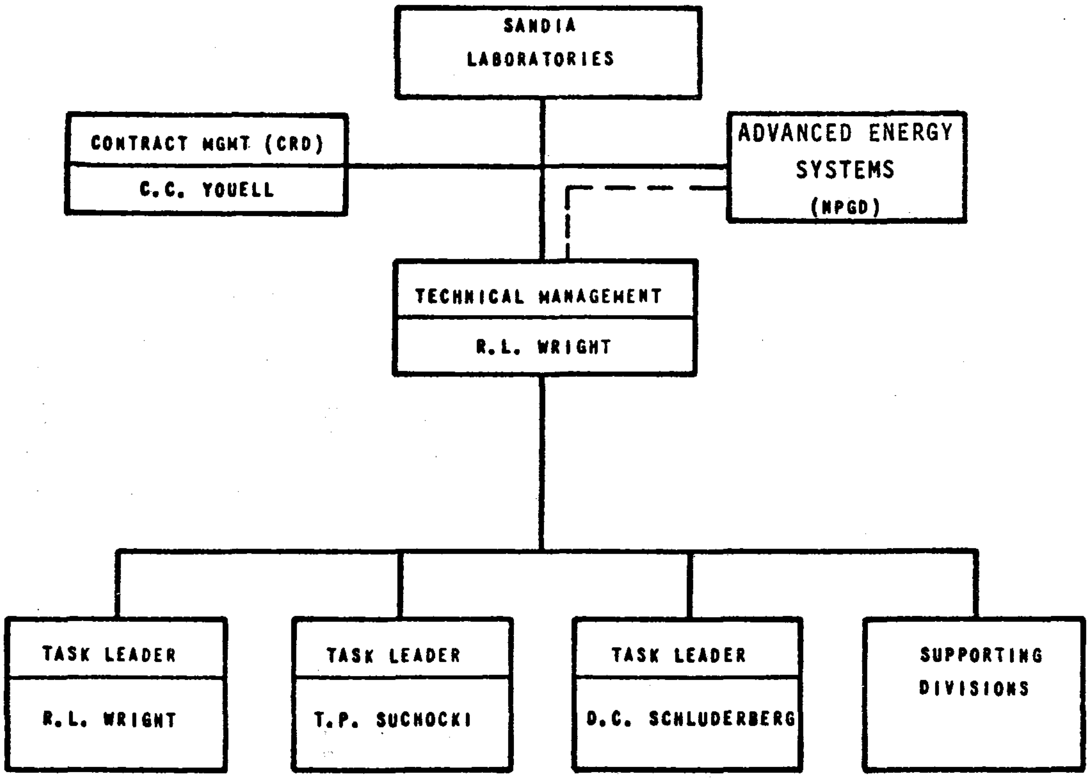
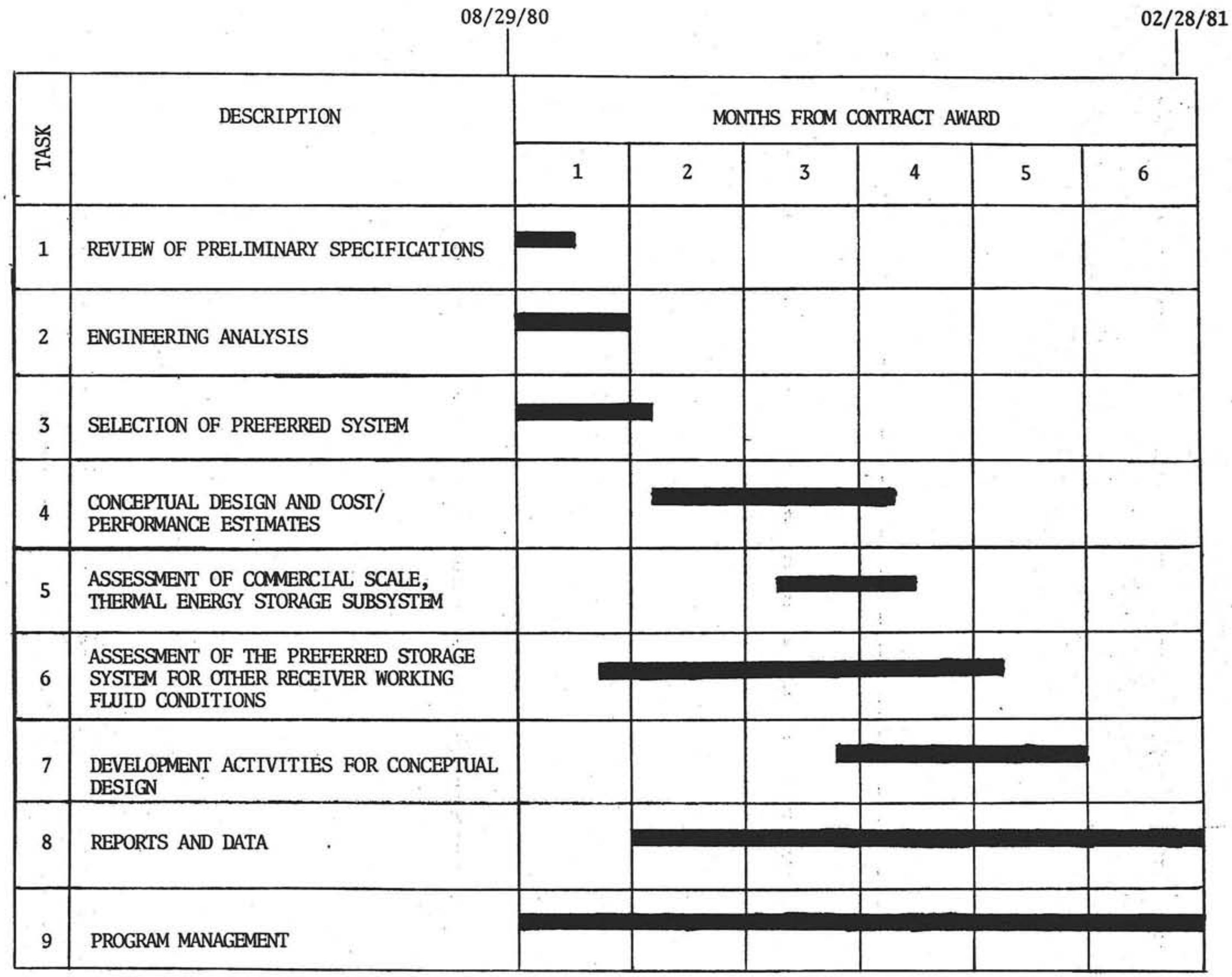


Figure 2-2. Thermal Energy Storage Project — Phase I Schedule



3. INITIAL SELECTION/SCREENING OF CANDIDATE TESS CONCEPTS

3.1. Literature Survey/First Screening

The existing literature on thermal energy storage was reviewed to identify concepts developed prior to this study (see Appendix C). The following ground rules were established to identify candidates from the literature survey:

- Sensible heat storage type only.
- Applicable to Barstow operating conditions.
- Applicable primarily to water/steam receivers.
- Have the potential for upgrading to advanced receiver technology.
- Have simultaneous charge/discharge capability.
- Single-stage storage.

The following concepts were identified:

Storage media	Storage arrangement	Tank insulation
Caloria/granite	Thermocline	External
Moving bed	Hot/cold tanks	Internal
Air/rock	Thermocline	Internal
Molten salts	Hot/cold tanks	External
Pressurized hot water	Underground hot tanks	External
Sodium	Hot/cold tanks	External
Syltherm-800*/taconite	Trickle thermocline	External

These combinations of media were considered to be representative of contemporary thinking and best suited for the second screening described in section 3.2.

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*Registered trademark - Dow Corning.

3.2. Second Screening

The concepts identified from the literature as meeting the ground rules defined above and possibly as being competitive with the oil/rock concept were qualitatively assessed. As a result of this assessment, the moving bed, air/rock, and molten salt concepts were selected for a more detailed analysis. These concepts were assessed on the basis of the following considerations:

- High temperature capability (>600F)
- Cost of media, ¢/lb (FOB)
- Safety
- Heat transfer ability
- Operation, maintenance, and availability

The seven concepts were qualitatively assessed on a relative basis of one being poor and six being excellent. The results of this ranking are shown below.

Initial Concept Ranking

Storage media	High temp ability	Est. cost of medium	Safety	Heat trans. ability	Maint and avail	Totals, ranking score
Air/rock	6	6	6	1	6	25
Moving bed	6	5	5	3	5	24
Pressurized hot water	3	6	2	5	3	19
Molten salts	5	4	4	5	2	20
Liquid metal	6	2	3	6	1	18
High temp organic	4	1	2	4	3	14
Oil/rock	1	3	1	2	4	11

An additional factor leading to the exclusion of underground pressurized hot water from the second level of screening was its site-specific nature. None of the other concepts listed above are site-specific.

The air/rock, moving bed, and molten salt concepts were chosen on the basis of this ranking system as the concepts having the greatest possibility of showing cost/performance advantages over the Barstow oil/rock concept. These three concepts were analyzed in greater depth to pick one concept that exhibited the best potential for cost/performance advantages over oil/rock. These analyses are the subject of section 4.

4. SELECTION OF PREFERRED SYSTEM/FINAL SCREENING

Three ATESS concepts (molten salt, air/rock, and moving bed) were chosen as most likely to offer cost/performance advantages over the oil/rock thermocline concept. This section describes the evaluation of these concepts and concludes with the selection of a single concept to be the subject of conceptual design and detailed assessment. The ranking methods and selection criteria were formulated to give a comprehensive, impartial assessment of the three concepts selected. This combination was designed to couple experienced engineering judgment with key economic information developed for each concept.

4.1. Selection Process

The selection of an ATESS was based on two main considerations. The first included those items for which costs could be readily evaluated, e.g., capital costs of major equipment, round trip efficiency, system availability/reliability, and development costs. The other included those items for which costs were difficult to establish, such as evaluation of environmental and safety concerns, applicability to higher receiver working fluid conditions, and the possibility of design improvement. The final selection was based on the judgment of the team, which balanced the definitive and subjective considerations.

4.2. Selection Criteria

The selection criteria used to evaluate the ATESS candidates were:

- Capital costs
- Round trip efficiency
- Availability/reliability, maintenance, inspection, and service life
- Development requirements and risks
- Environmental and safety
- Applicability to higher receiver working fluid conditions
- Possible design improvements.

The first four criteria were combined to give an economic comparison of the alternatives. The last three tempered the final selection of the preferred system. Each of these criteria is discussed below.

4.2.1. Capital Costs

Preliminary air/rock, molten salt, and moving bed conceptual designs were prepared. The thermal energy subsystem for the Barstow commercial scale plant design served as the basis for analysis of the oil/rock concept.[2] Capital costs were estimated for each of the four concepts based on major system components, i.e., storage media, storage tanks, heat exchangers, and pumps/lifts, and are presented in Table 4-1. These estimates were based on information from references 5 and 6 and included transportation and installation costs. The balance of plant (BOP) costs were not included in these estimates. Prices were adjusted to reflect June 1980 costs. These estimates were not made to obtain total capital costs, but rather, incremental costs to establish a basis for selecting a preferred concept. It was considered that the major equipment costs would be representative and sufficient for this purpose[14,29]. The results of the analysis are reported in section 4.4.

4.2.2. Round Trip Efficiency

Round trip efficiency was calculated by the following equation:

$$\eta_{RT} = \frac{E_{out} \cdot \eta_{TESS}}{E_{in} \cdot \eta_{solar}}$$

where

η_{TESS} = 0.27 and η_{solar} = 0.35; since the TESS must supply 285 MWt for 6 hours,

E_{out} = 285 MWT × 6 hours = 1710 MW-ht

E_{in} includes allowances for losses and inefficiencies, which vary between concepts.

$$E_{in} = E_{out} + 0.036E_{out(1)} + E_{PC} + E_{PD} + E_{HT} + E_{L}$$

where

$E_{out(1)}$ = steady-state heat losses from storage,

E_{PC} = parasitic loss to pumps/lifts during charge,

E_{PD} = parasitic loss to pumps/lifts during discharge,

E_{HT} = parasitic loss to heat tracing (molten salt only),

E_{L} = heat loss from lifts (moving bed only).

The results of these calculations are reported in section 4.4, and are tabulated as Table B-2 in Appendix B.

4.2.3. Availability/Reliability, Maintenance, Inspection, and Service Life

Availability/reliability, maintenance, inspection, and service life were grouped together as one criterion to simplify the analysis. The following considerations formed the basis for evaluating the TESS concepts under this criterion:

- Media decomposition/attrition
- Media compatibility with structural materials — corrosion/wear
- Media compatibility with water/steam
- Freezeup/thaw difficulties
- Media spill consequences
- Heat exchangers
- Valves
- Pumps
- Tanks
- Component accessibility (after cooldown)

The results of this analysis are reported in section 4.4.

4.2.4. Development Requirements and Risks

The development requirements and risks of the four major components of each candidate ATESS concept were evaluated. These major components were heat exchangers, valves, pumps, lifts, and storage tanks. A ranking system was used, and based on the outcome of this ranking, the relative costs for this criterion were included in the evaluative cost comparison.

4.2.5. Environmental and Safety

Potential for harm to the environment was evaluated based on four possible external causes:

- Earthquakes
- Missiles (planes)
- Storms
- Sabotage

Safety was evaluated on the basis of the following:

- Storage media flammability
- Storage media reaction with water/steam
- Toxicity of airborne media in particles of vapor
- Media working pressure
- Effectiveness of personnel protective equipment

The results of these analyses are reported in section 4.4.

4.2.6. Applicability to Higher Receiver Working Fluid Conditions

The four candidate ATESS concepts were evaluated for this criterion on the following basis:

- Thermal decomposition
- Compatibility with structural materials
- Insulation
- Ullage
- Safety
- Reliability/availability
- Service life

The inlet steam conditions to the TESS for the commercial-scale Barstow plant are 510C/10.1 MPa (950F/1465 psia). The media were evaluated for their ability to accommodate inlet conditions at 510C/12.5 MPa (950F/1815 psia) and their suitability for second stage storage utilization only. The results of this analysis are reported in section 4.4.

4.3. Advanced Thermal Energy Storage System Candidates

Preliminary system arrangements were developed for the three selected ATESS concepts. Time did not permit optimization of the concepts. These preliminary arrangements in conjunction with the Barstow oil/rock design[2] formed the basis for a comparison to select a preferred concept. The design data generated include the volume of storage medium required, tank sizes required to hold the medium, heat exchanger sizes, and pump requirements. These data are summarized in Table 4-2. In addition, technical concerns relative to each concept were addressed and factored into the selection process.

4.3.1. Oil/Rock

The oil/rock design[2] employs sensible heat storage using dual liquid and solid media for the heat storage in a parallel set of four tanks, each using the thermocline principle to provide high-temperature, extactable energy, as shown in Figure 4-1. Appendix A summarizes the principal characteristic of the subsystem and major components. The oil/rock concept operates as follows: Incoming steam 510C/10.1 MPa (950F/1465 psia) is desuperheated by mixing with water to 360C (680F). This steam is then used to heat oil in the thermal storage heater from 232 to 315.6C (450 to 600F). The oil is stored on the thermocline principle: cool 232C (450F) oil is pumped from the bottom of the storage tank and after being heated is piped to the top of the tank. Upon demand, the hot oil is pumped from the top of the tank to a steam generator where feedwater is heated to produce steam at 299C/2.72 MPa (570F/395 psia). The diagram of temperature versus fraction of heat transferred is shown in Figure 4-2.

The following technical concerns were identified during analysis of the oil/rock concept involving the design of the storage tank to accommodate the high temperature 315.6C (600F) storage fluid:

- Stresses arising from differential thermal expansion of the tank floor.
- Thermal cycling of the tank wall.
- Slumping of the rock media and resultant increases in tank wall stress over the life of the plant (thermal ratcheting).
- Differential thermal expansion between the rock media and the steel tank wall.
- Type of insulation and internal versus external insulation.
- Code to which tank must be designed (API versus ASME).
- Method of support.
- Inspection and maintenance.
- Roof design.

The resolution of these concerns could have an adverse effect on the tank cost. Consequently, these concerns were evaluated (see Appendix D). The subject of tank storage/design was discussed with a major tank engineering and fabrication firm.[11] As a result of this investigation, the following conclusions were reached:

- The tank walls should be strengthened to allow for slumping of the rock media.
- The ASME Code should be used.
- The floor of the tank should be insulated with a sand sandwich-type construction.

Added costs for compliance with the ASME Code and tank floor insulation were included in the tank cost estimates. The cost of strengthening the tank walls to resist bed slumping loads was not included. In addition, no cost was allowed for adding an underground catch basin to combat oil fires.

4.3.2. Moving Bed

A schematic diagram showing the operation of the moving bed concept is shown in Figure 4-3. The bed material (sand) flows over the tubes of the charge heat exchanger. Steam at 510C/10.1 MPa (950F/1465 psia) from the receiver heats the sand from 218 to 332C (425 to 630F). The hot sand is stored in a silo tank and, upon demand, the hot sand flows over the tubes of a discharge heat exchanger to produce steam at 299C/2.72 MPa (570F/395 psia). Figure 4-4 shows the temperature-heat diagram for this operation.

The bed material is transported by an Archimedes-type screw lift. A typical silo-tank, lift, and heat exchanger arrangement is shown in Figure 4-5. Nine such units would be required for the 100-MWe commercial-scale plant, as shown in the plan view of Figure 4-6. To prevent problems associated with differential thermal expansion, internal tank insulation was used as shown in Figure 4-7. A layer of non-flowing sand insulates the tank floor.

Appendix A summarizes the principal characteristics of this subsystem and its major components.

4.3.3. Air/Rock Concept

An air/rock thermal energy storage system was analyzed using the 100-MWe Barstow commercial plant parameters as the basis for the size and relative cost of this concept.

Figure 4-8 is a schematic of the air/rock TESS. Steam from the receiver at 510C/10.1 MPa (950F/1465 psia) passes through a water-air heat exchanger and exits at 249C/9.65 MPa (480F/1400 psia) and is returned to the receiver to be reheated. The air at 1 atm passing through the heat exchanger is heated from 218 to 332C (425 to 630F) and exhausted downward through the rock bed, transferring heat to the bed and thus charging the system. In order to remove energy from the storage system, the air flow is reversed through the rock bed and routed through the discharge heat exchanger. Air enters the discharge heat exchanger at 332C (630F) and exits at 218C (425F), thus transferring its energy to the water entering at 121C/2.76 MPa (250F/400 psia) and exiting as steam at 299C/2.72 MPa (570F/395 psia).

The preliminary air/rock thermal energy storage system consists of nine tank-heat exchanger-fan units as presented in Figure 4-9. A temperature diagram versus the percent of heat transferred for the air rock is shown in Figure 4-10. Appendix A summarizes the principal characteristics of this subsystem and its major components.

The problem of "thermal bed ratcheting" necessitated the tank design to accommodate this phenomenon over the life of the plant. For the purpose of this study a separate segmented, spring-loaded internal shell (girdle) was used to accommodate the differential thermal motions of the rock bed relative to the wall. The evaluation analysis of the tank design is included in Appendix D.

4.3.4. Molten Salt

This molten salt concept utilizes sensible heat for thermal storage. Molten salts have been used for many years in process plants as a heat transfer and heat treatment medium. There are basically two practical salt mixtures available for use — Hi-Tec* and draw salt. The Hi-Tec composition was chosen for this study because it has a lower melting temperature than draw salt, 142C for Hi-Tec versus 220C for draw salt (287 versus 428F). This greatly alleviates concerns of freezeup and reduces the requirements for heat tracing. However, Hi-Tec would be more expensive and more susceptible degradation at high temperatures but at 630F this would be negligible. The Hi-Tec composition is 40% $NaNO_2$, 7% $NaNO_3$, and 53% KNO_3 by weight.

A schematic of the molten salt concept is shown in Figure 4-11. The temperature diagram is Figure 4-12. Appendix A summarizes the principal characteristics of this subsystem and its major components.

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*Registered trademark — Dupont deNemours & Company.

The molten salt charging heat exchanger is a once-through U-tube shell-and-tube heat exchanger. It contains surface areas for desuperheating, condensing, and subcooling. The desuperheating surface is a low-chrome alloy because of the higher temperatures encountered 510C (950F). The remainder is carbon steel. The U-tube shell is used here and in the economizer and boiler of the discharge heat exchanger system to allow for differential thermal expansion.

Separate heat exchangers are used for the economizers boiler, and superheater in the discharging heat exchanger system. A portion of the economizer feedwater flow is recirculated to keep the inlet temperature at 204C (400F). This prevents the salt from freezing at this point. The boiler utilizes the steam drum/recirculation principle to improve operating flexibility and reliability. A relatively small surface area is required in the superheater section (2400 ft^2). In order to maintain a proper ratio of heat exchanger length to shell diameter, the superheater is divided into eight separate shell-and-tube heat exchangers operating in parallel.

Two cylindrical, externally insulated storage tanks, 36.6m (120 ft) is diameter by 18.3m (60 ft) high, were used for this concept (one hot, one cold).

4.4. System Analyses/Results

The four candidate ATESS concepts were analyzed based on the criteria stated in section 4.2. The results of the system analyses are reported below. The development of the rankings, calculation of round trip efficiencies, and determination of cost are shown in Appendix B.

System Analyses

(Major components)	O/R	MB	A/R	MS
Capital costs, 10^6 \$	13.30	25.09	22.64	27.14
η_{RT}, %	73.2	68.9	56.5	71.1
Availability/reliability	28	33	39	21
Development requirements	14	11	14	11
Environmental and safety	16	33	35	18
Applicability — higher temp and pressure	0	28	27	20

The evaluated cost comparison is shown in Table 4-3. This cost evaluation adjusts capital costs of the major ATESS components to take into account the effects of maintainability/availability, round trip efficiency, and developmental costs.

In this table the availability rankings previously determined (and shown on line 1) are normalized to the oil/rock concept. The reciprocal of these numbers (line 3) is then multiplied by 25 (the assumed number of outage days per year for the Barstow plant caused by storage unavailability) to obtain the equivalent outage days above or below the oil/rock concept (line 5). The lost MWe per day, $E_{out(e)}$, (MWhe/day), is then multiplied by the quantities in line 4 to give annual MWhe lost due to storage subsystem outages. If the lost power is worth \$100/MWhe (a typical peak load value), an interest rate of 15% and plant life is 30 years, the equivalent present worth of the gained or lost power over plant life can be computed and is shown on line 7. The major component cost shown on line 8 is thereby adjusted as shown on line 9.

A similar approach is used to express the effect of round trip efficiency, line 10. Values of $E_{in(e)}$ for each of the concepts is compared to the oil/rock concept (630.7 MWhe/day), and the difference is noted in line 12. This value multiplied by the assumed normal number of annual operating days for the Barstow plant (265) combined with the present worth procedure given above yields the values given on line 14. These values are used to make a further adjustment in capital cost as given on line 15. The remainder of the table shows approximate development and FOAK engineering capital cost increments for the major components. These are totaled on line 24 and used to make the final adjustments to the equivalent capital cost as given in line 25.

4.5. Conclusions

The evaluated cost comparison presented in Table 4-4 shows the oil/rock concept to be most cost competitive for the operating parameters of the Barstow plant. This comparison also indicates that the moving bed and molten salt TES concepts have comparable costs and that the air/rock thermocline concept tends to be considerably more expensive.

In comparison with the molten salt concept, the moving bed TESS concept tends to

- Provide an alternate approach to molten salt for TESS operating within a 260 to 621C (500 to 1150F) range.
- Have unique capabilities at storage temperatures above and below the range listed above.
- Benefit materially by a small degree of first-round optimization to simplify components and reduce capital costs.

In addition, in view of environmental and safety considerations and applicability to higher working temperatures the MBTESS is the most viable concept; therefore, it was selected for continued development in the conceptual design and costing tasks.

Table 4-1. Estimated Costs for Major Equipment

	Cost, 10^6 dollars (June 1980 $)			
Component	Oil/rock	Air/rock	Moving bed sand	Molten salt
Tanks	0.76	1.25	0.56	1.23
	4	9	18	2
	3.02	11.25	10.12	2.46
Storage media	4.84	1.063	3.23	22.27
Heat exchangers	5.35	3.77	2.24	2.26
Pump/fan/lift	0.19	6.56	9.50	0.15
Total	13.30	22.64	25.09	27.14

Table 4-2. Concept Comparison Data

	Oil/rock	Air/rock	Moving bed	Molten salt
Tank dimensions (dia × height), ft and number used	105 × 45(4)	105 × 50(9)	53 × 95(18)[a]	120 × 60(2)
Total tank volume, 10^6 ft^3	1.54	3.90	3.38	1.36
Total media volume, ft^3	1.49×10^6[b]	2.48×10^6	1.34×10^6[b]	6.58×10^5
Total heat exchanger surface area, ft^2	2.10×10^5	1.67×10^5[c]	1.12×10^5	7.94×10^4
Total pump horsepower, hp	2.55×10^3	5.00×10^4	1.17×10^4	1.64×10^3
Total system flow rate, lb/h				
Charge	8.74×10^6	1.66×10^7	1.75×10^7	1.15×10^7
Discharge	10.1×10^6	1.82×10^7	1.95×10^7	1.26×10^7

(a)Cone roof.

(b)Working storage media volume.

(c)Area based on ID of tubes.

Table 4-3. Capital-Operating, Maintainability, Availability — Round Trip Cost Evaluation Algorithm (10^6 June 1980 Dollars)

Line	Item	Concept: Air/rock	Moving bed	Molten salt	Oil/rock
1	Availability ranking	39	33	21	28
2	Availbility ranking normalized to Barstow	1.39	1.18	0.75	1.0
3	Reciprocal of line 2	0.72	0.85	1.33	1.00
4	Equiv annual outage days (assume 25 for Barstow for reference)	18	21	33	25
5	Outage days above or below Barstow	-7	-4	+8	0
6	Gained or lost annual MWhe, (5) × E_{out}(e)	-3,232	-1,847	+3,694	0
7	Equiv present worth, $ million[(a)]	-2.121	-1.212	+2.425	0
8	TESS capital cost (key components)	22.64	25.09	27.14	13.30
9	Equiv cost, (7) + (8) [capital + avail.]	20.52	23.88	29.57	13.30
10	Round trip efficiency, η_{rt} %	56.5	68.9	71.1	73.2
11	E_{in}(e) = E_{out}(e)/(10), MWhe/day	817.2	670.1	649.4	630.7
12	(11) - 630.7 MWhe	186.5	39.4	18.7	0
13	Lost annual MWhe, (12) × 265	49,423	10,441	4,956	0
14	Equiv present worth, $ million	32.45	6.85	3.25	0
15	Adjusted capital cost, (9) - (14), $ million	52.97	30.73	32.82	13.30
16	Heat exchanger R&D, $ million[(b)]	0	0.50	0.50	0
17	Heat exchanger FOAK, $ million[(b)]	0	0.25	0.25	0
18	Valve R&D, $ million[(b)]	0	0.20	0.20	0
19	Valve FOAK, $ million[(b)]	0	0.10	0.10	0
20	Pump R&D, $ million[(b)]	0	0.30	0	0
21	Pump FOAK, $ million[(b)]	0.20	0.30	0.30	0
22	Tank R&D, $ million[(b)]	1.0	0	0	1.0
23	Tank FOAK, $ million[(b)]	0.50	0.30	0	0.5
24	$\sum$ (16 through 23)	1.70	1.95	1.35	1.5
25	Total equiv cost, (15) + (24), $ million[(b)]	54.67	32.68	34.17	14.80

[(a)]Present worth based on 30-year plant life, 15% interest rate, and annual payments based on each MWhe being worth $100.

[(b)]Development and FOAK engineering capital cost increment.

Note: Numbers in "Item" column in parentheses are line numbers referring to lines in the "Line" column.

Table 4-4. Evaluation Summary

Criteria	Oil/rock	Air/rock	Moving bed	Molten salt
	$ Millions			
Capital cost estimates[a]	13.30	22.64	25.09	27.14
Round trip efficiency[b]	0	32.45	6.85	3.25
Availability/reliability, maintenance[b], inspection, and service life	0	-2.12	-1.21	2.43
Development requirements and risk	1.5	1.70	1.95	1.35
TOTAL	14.80	54.67	32.68	34.17
	Non-Dollars, Ranking Nos. Only			
Environmental and safety	16	35	33	18
Applicability to higher ranges of working fluid conditions	0	27	28	20

[a] Based on key components.

[b] Normalized to oil/rock.

Figure 4-1. Schematic Flow Diagram of Oil/Rock Thermal Storage Subsystem for 100-MWt Commercial Plant (From Reference 2)

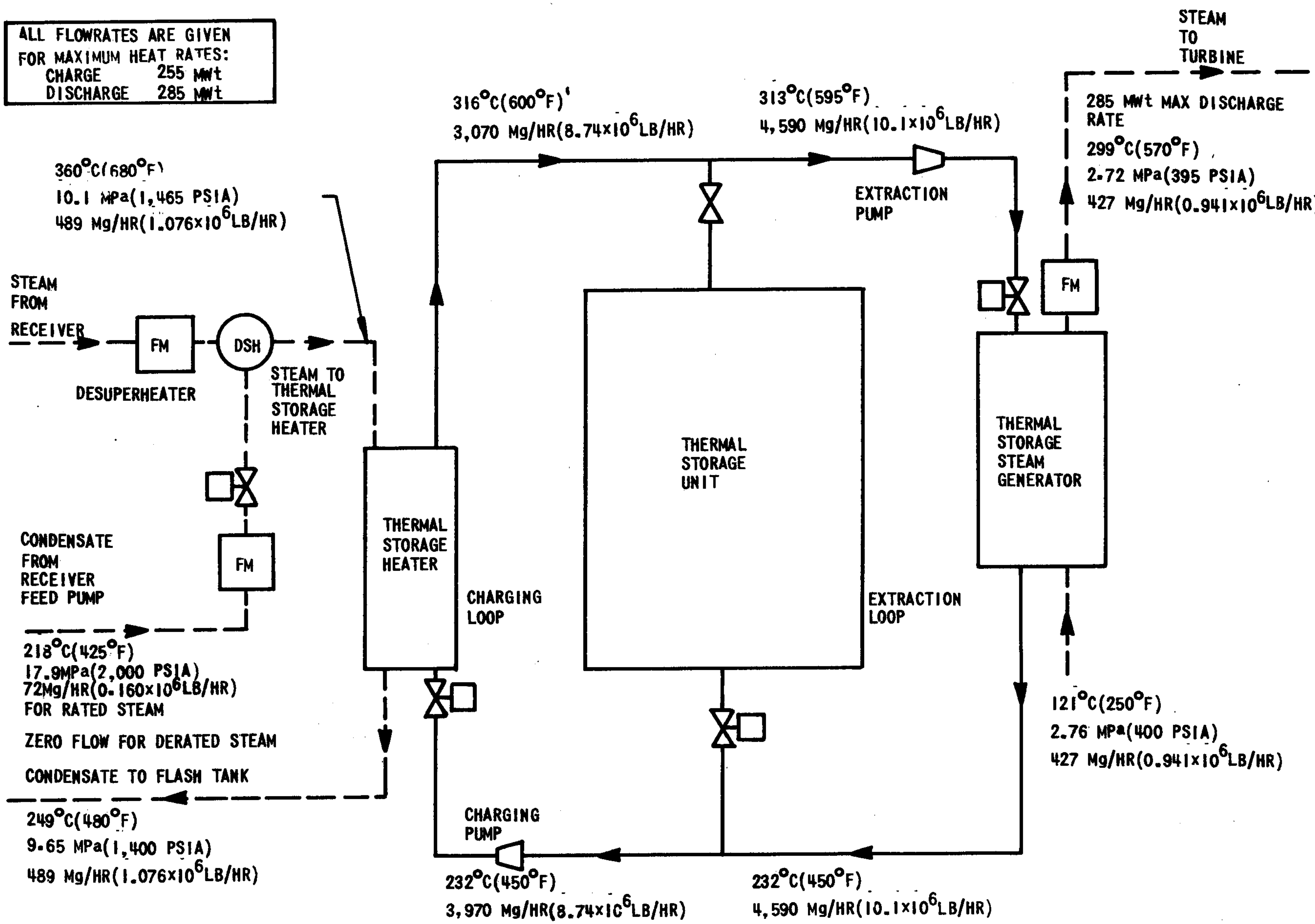


Figure 4-2. Oil/Rock Temperature-Energy Diagram

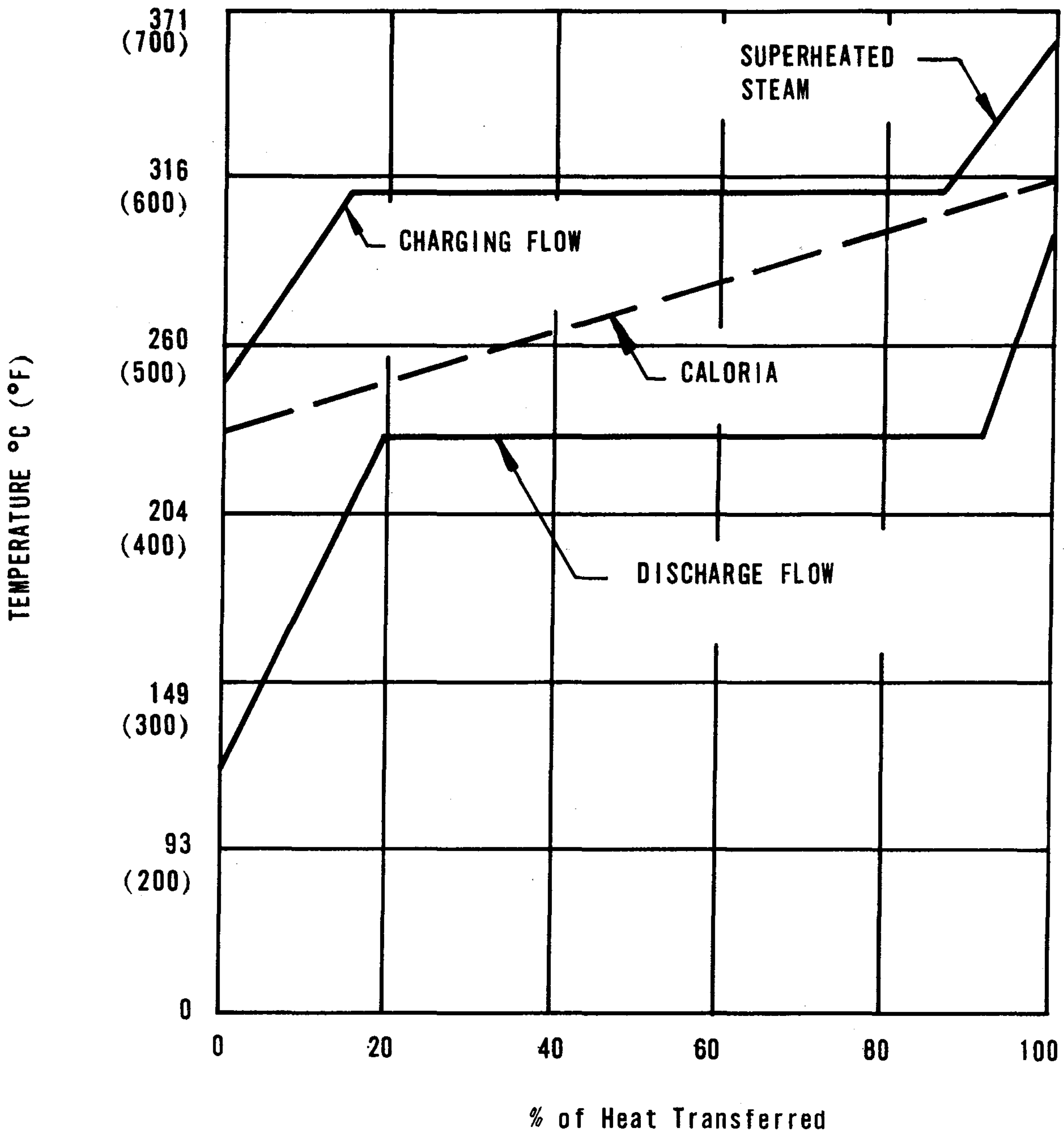


Figure 4-3. Flow Schematic — Moving Bed TESS Concept,
Barstow Commercial Plant

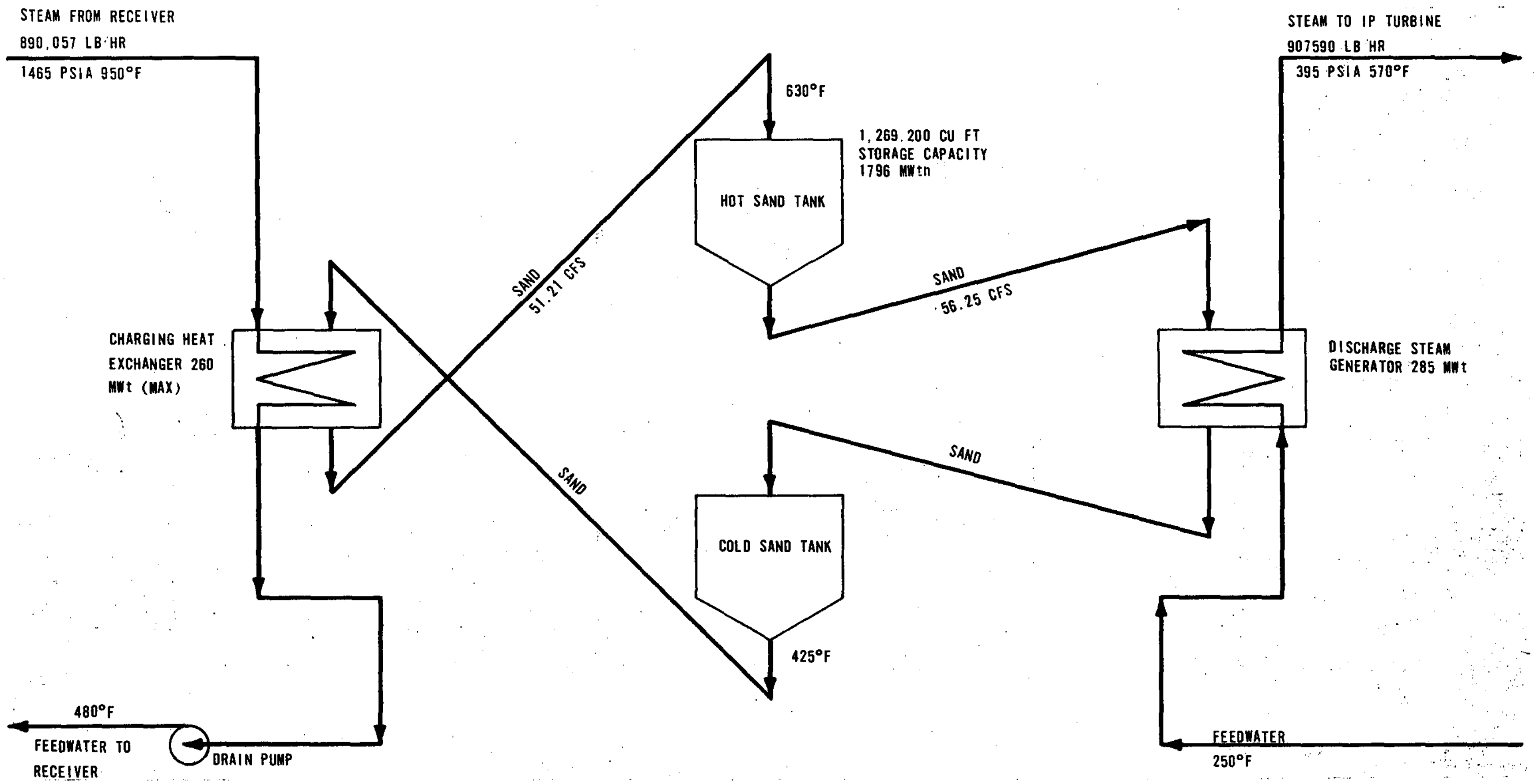


Figure 4-4. Moving Bed Temperature-Energy Diagram

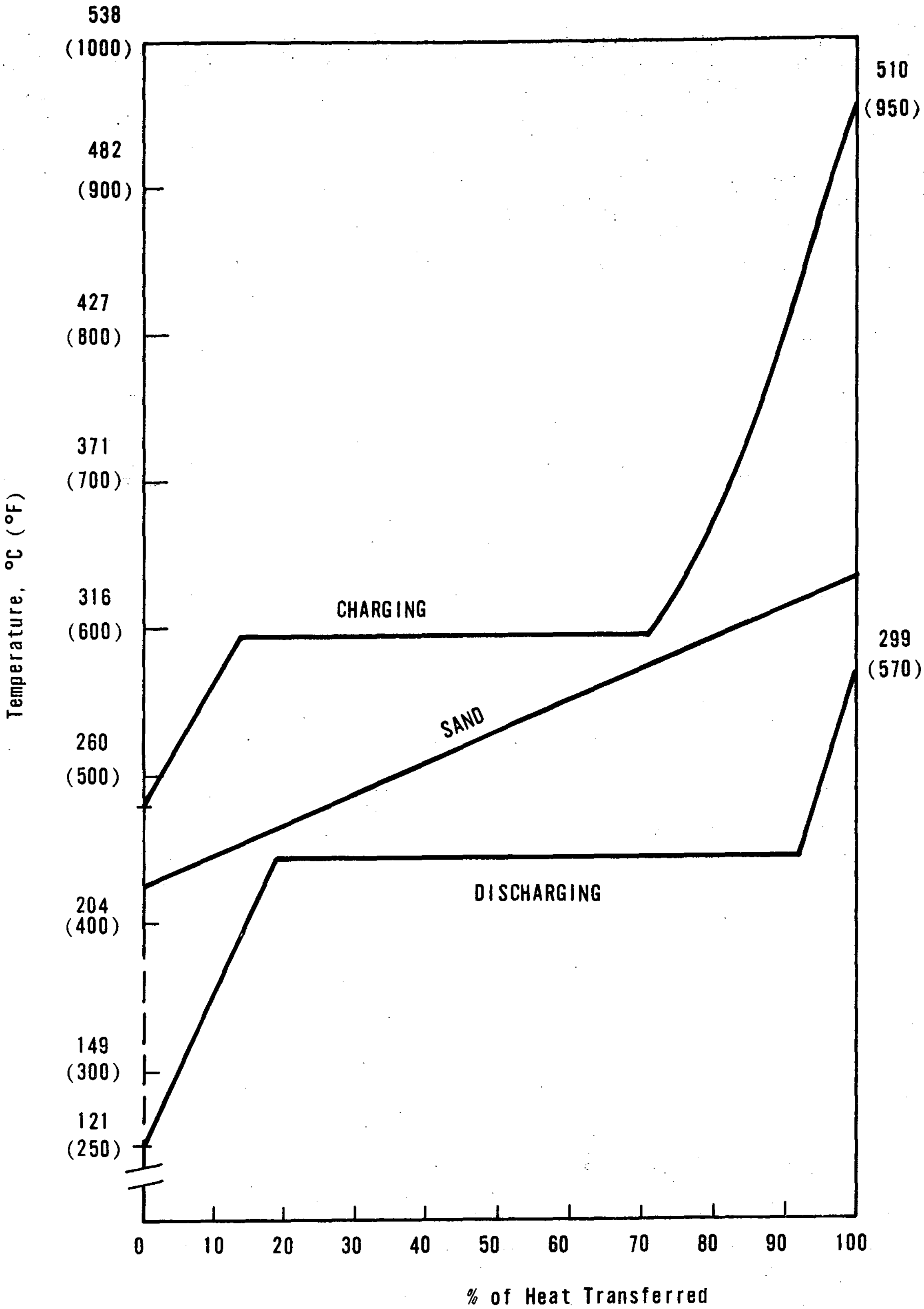


Figure 4-5. Arrangement of Moving Bed TESS

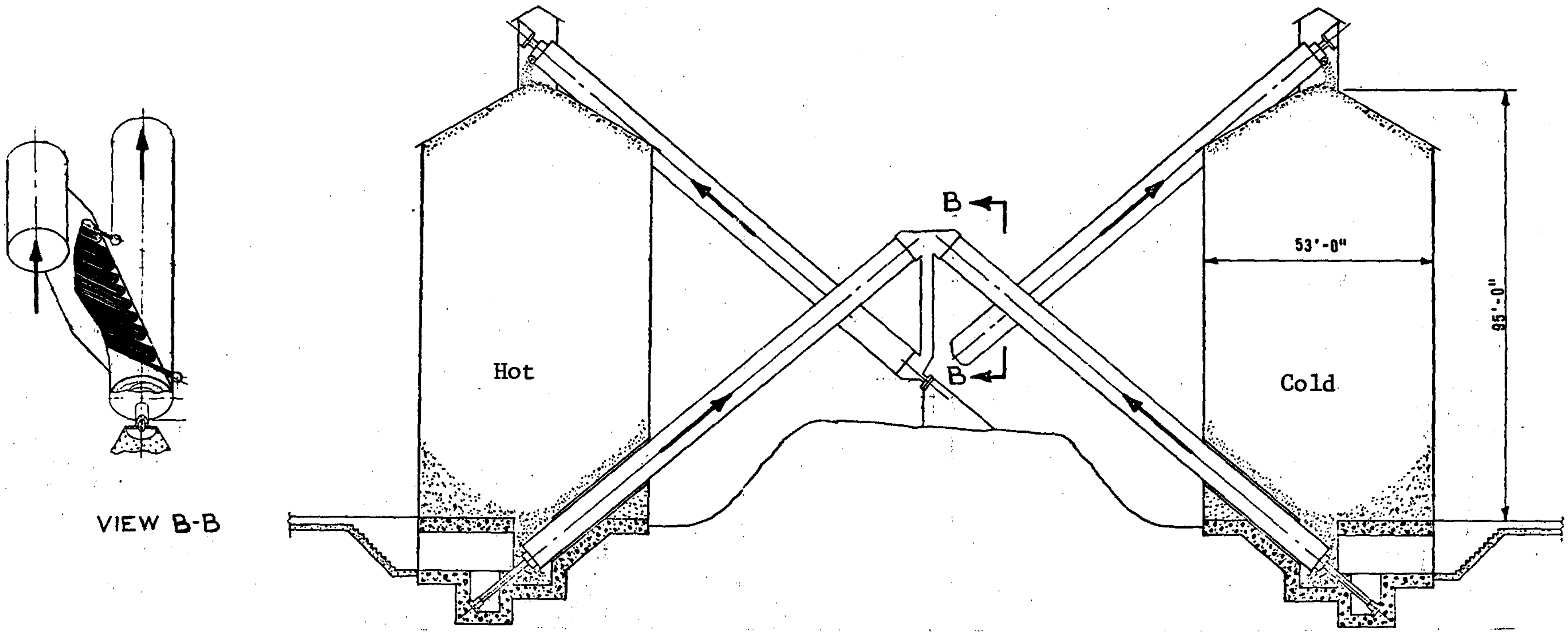


Figure 4-6. Moving Bed Tank Arrangement, Plan View

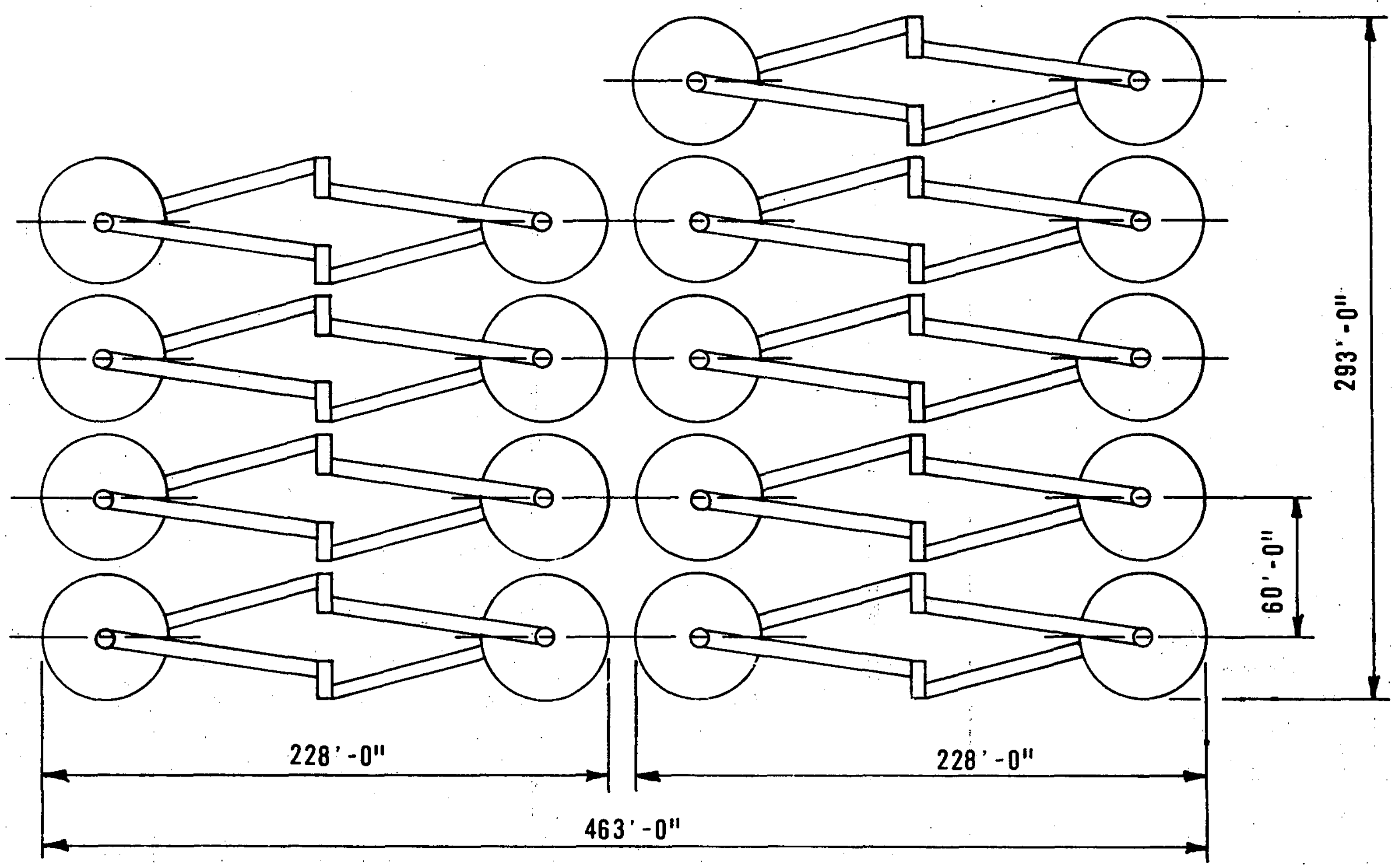
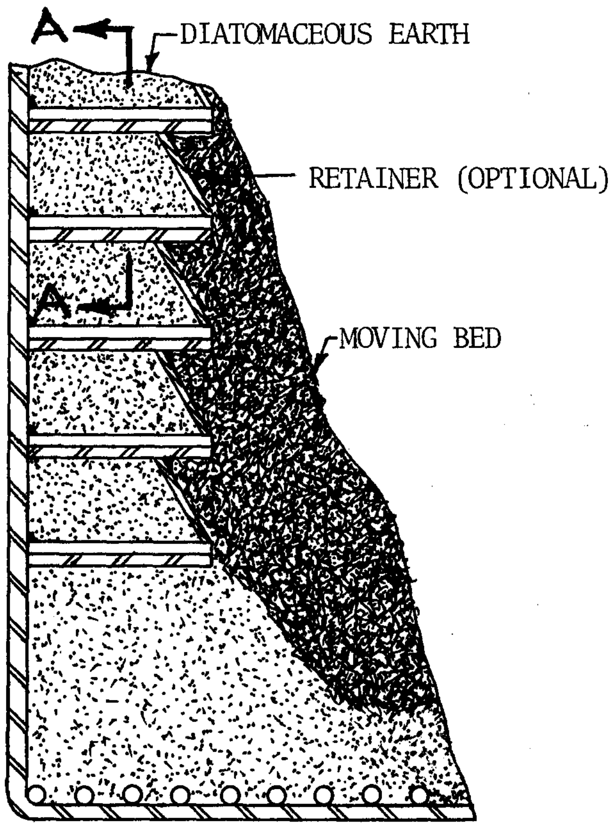
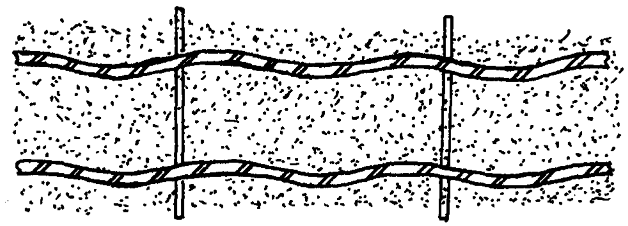


Figure 4-7. Design of Internal Insulation for Moving Bed Storage Tanks





SECTION A-A

Figure 4-8. Air/Rock Flow Schematic, Barstow Commercial Plant

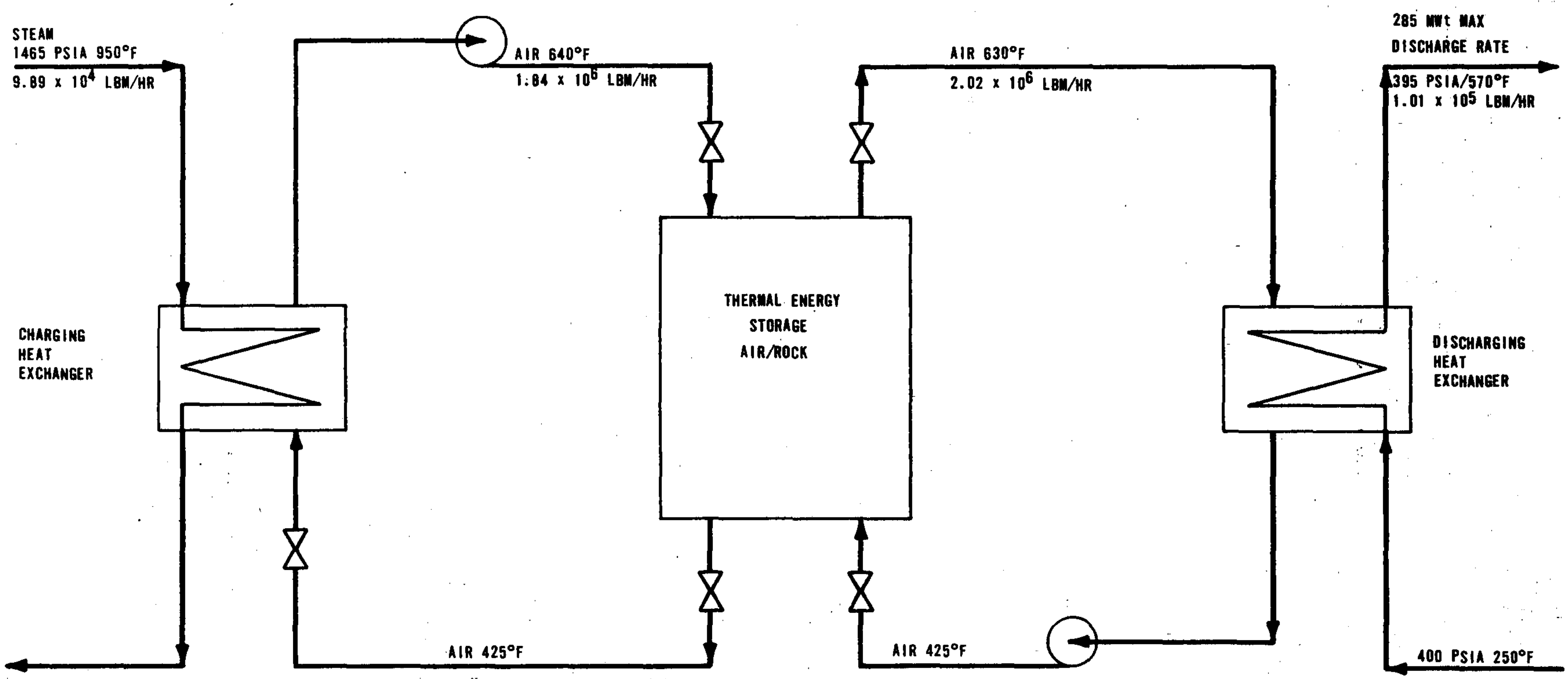


Figure 4-9. Air/Rock Tank-Heat Exchanger-Fan Arrangement — Nine Units Required for 100-MWe Commercial Plant

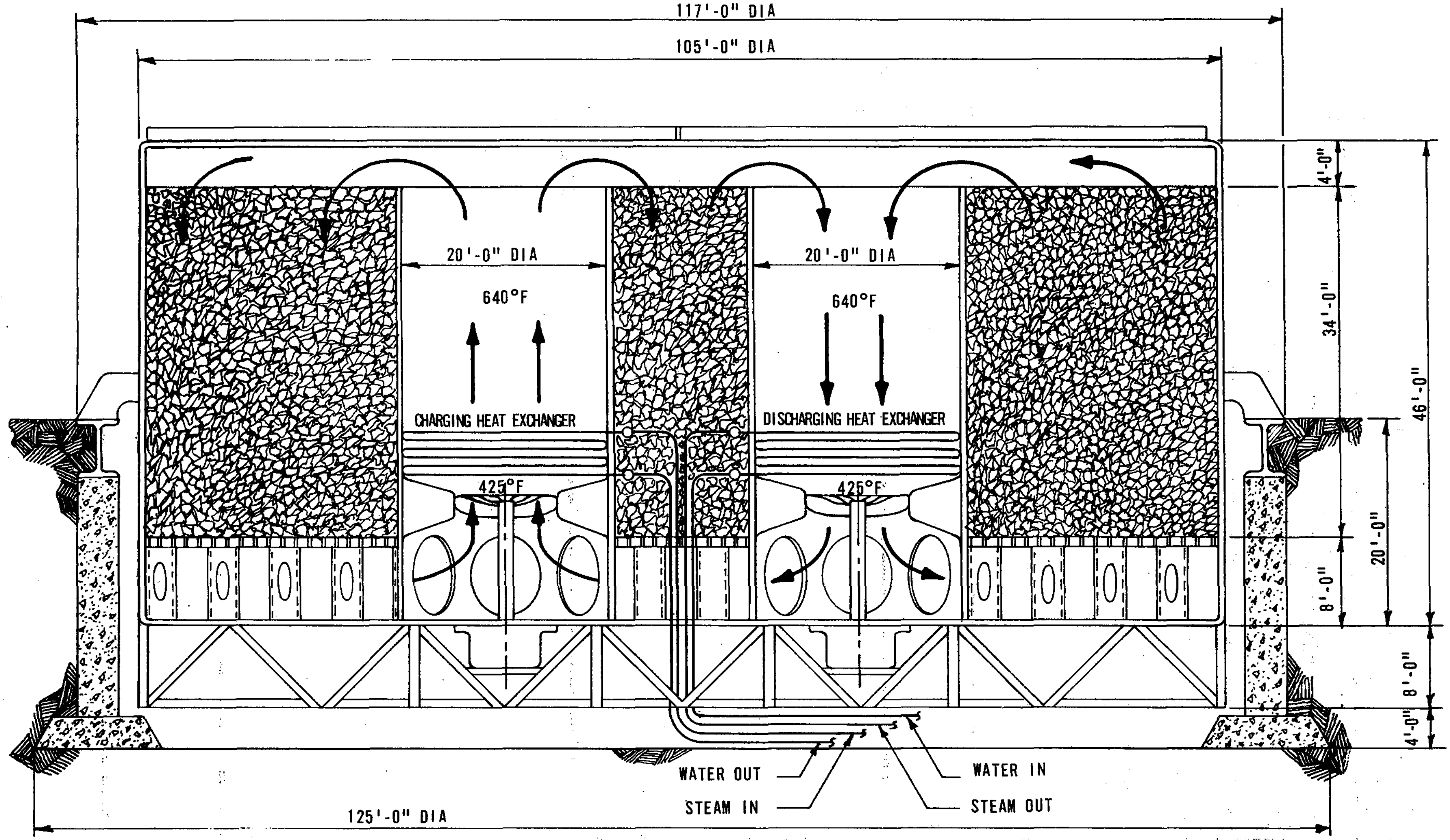


Figure 4-10. Air/Rock Temperature-Energy Diagram

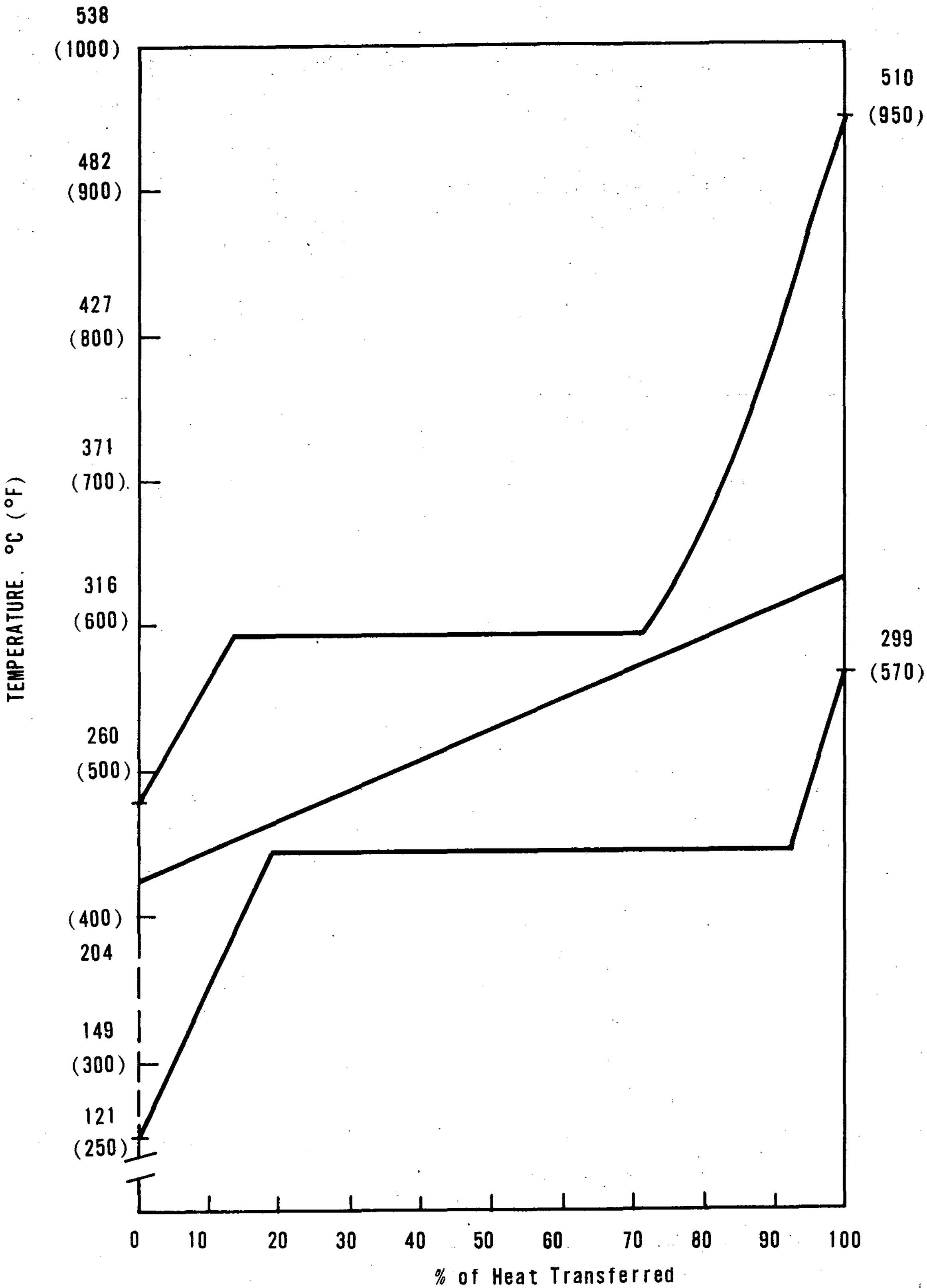


Figure 4-11. Flow Schematic — Molten Salt TESS, Barstow 100-MWe Commercial Plant Parameters

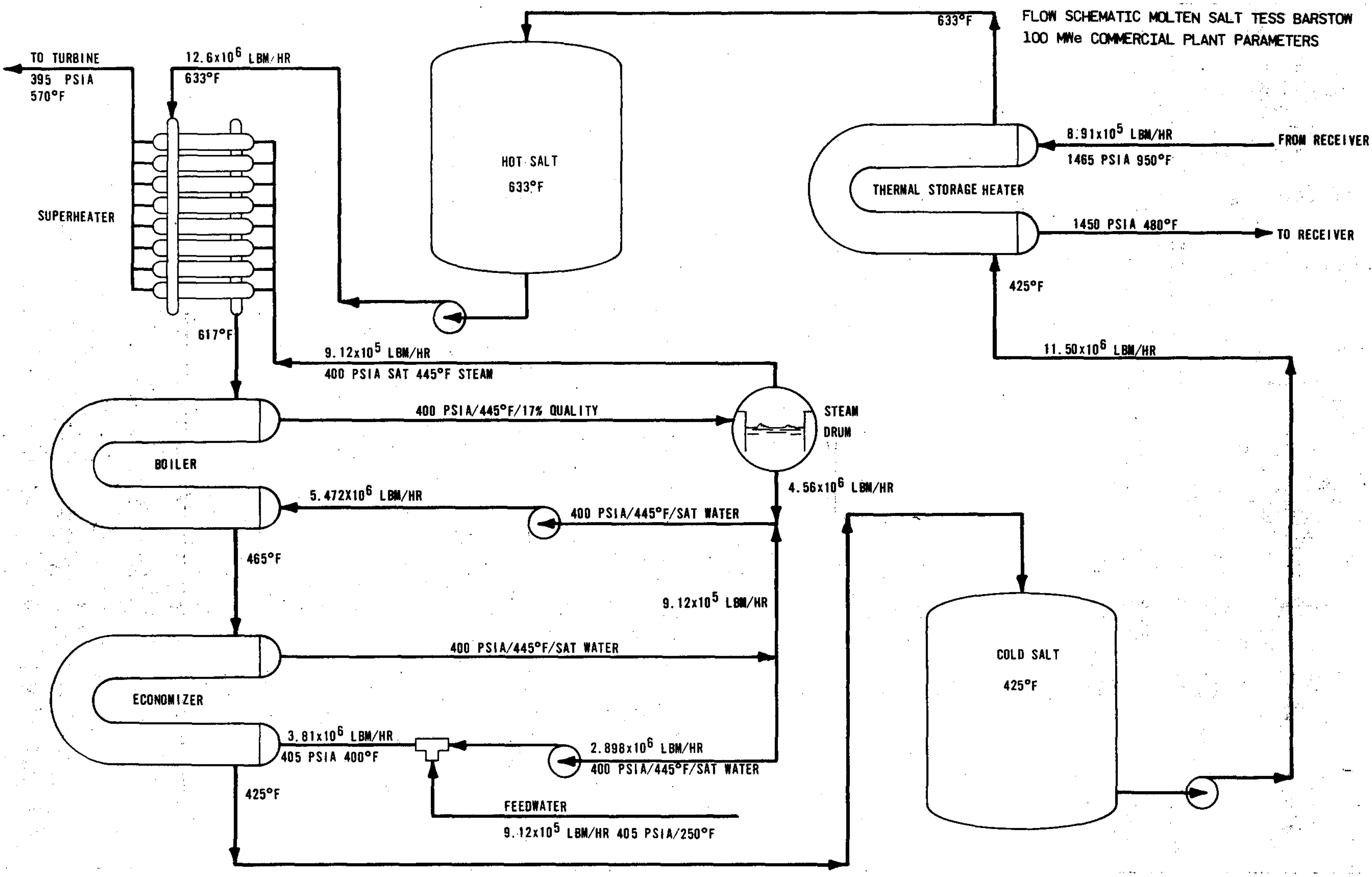
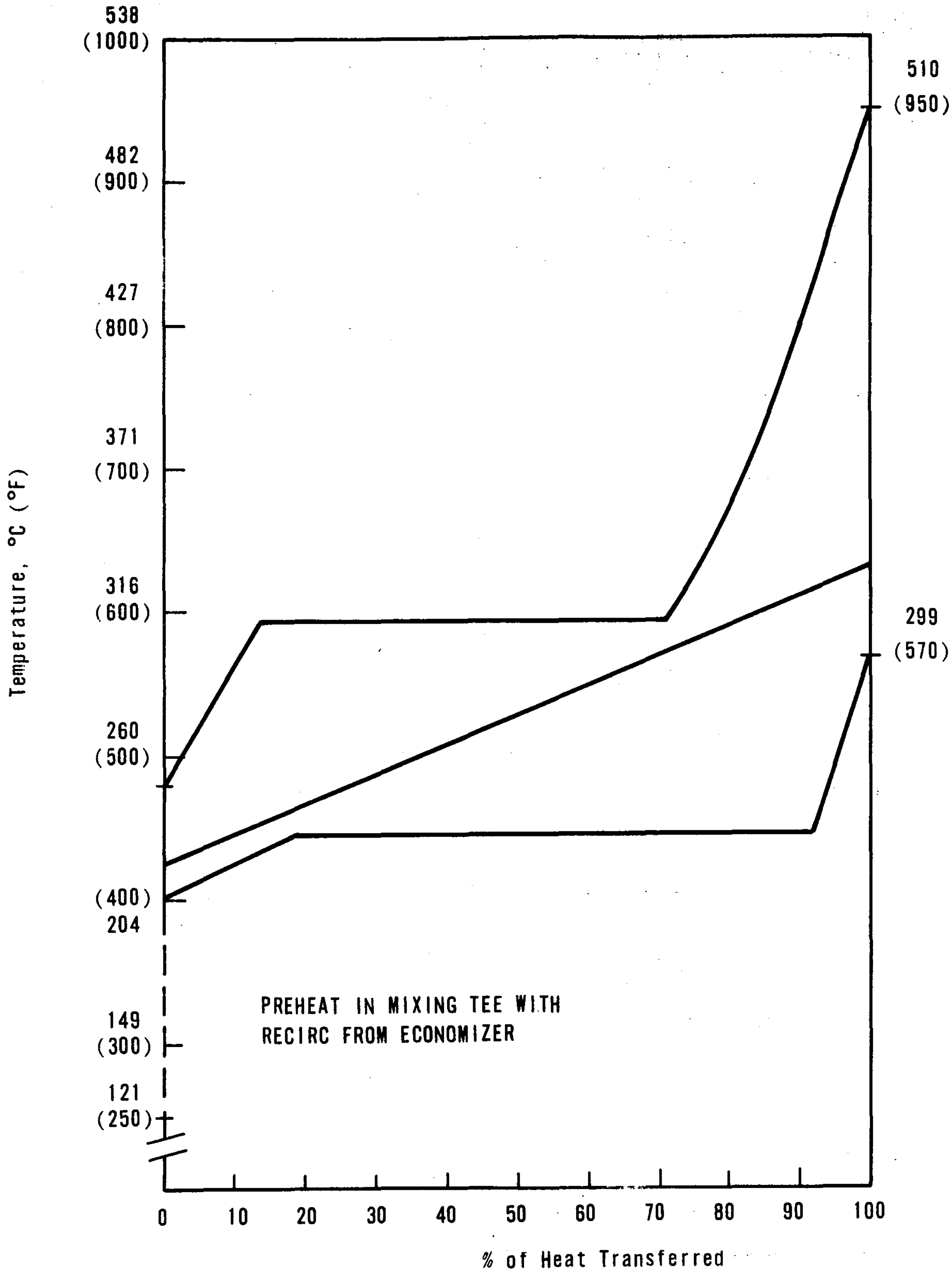


Figure 4-12. Molten Salt Temperature-Energy Diagram



5. CONCEPTUAL DESIGN AND COST/ PERFORMANCE ESTIMATES

This section reports the conceptual design of a moving bed thermal energy storage system (MBTESS), which utilizes a dense bed of particles moving past heating and/or cooling surfaces for heat transfer and heat transport. This concept uses a medium that is relatively low in cost and environmentally acceptable, such as silica sand, olivene, or alumina.

5.1. System Arrangement

A preliminary system arrangement for the MBTESS was developed under Task 2 to verify the feasibility of this concept and to obtain initial cost estimates. That arrangement is shown in Figures 4-5 and 4-6. In Task 4 the development of the commercial-scale conceptual design for the MBTESS was further evaluated with respect to cost and performance. This evaluation produced three configurations, the third of which became the basis for the reference arrangement of the conceptual design and the one considered to be an optimized combination of tanks, lifts, and heat exchangers.

Configuration 1, as shown in Figure 5-1, consists of 10 to 12 silo-type tanks with 9 to 10 of the tanks containing sand and encircled by a sand-water/steam heat exchange complex. Sand carried by Archimedes lifts emerges from the center of this arrangement and is conveyed to the heat exchangers and back to the storage tanks. This configuration was rejected because of operational difficulties associated with the heat exchanger complex, especially during simultaneous charge-discharge operations. In addition, the compactness of components in the central area reduces accessibility and increases maintenance time.

Figure 5-2 shows the second configuration developed; it consisted of cylindrical concrete/steel tanks arranged in a manner similar to the preliminary system arrangement (described above). There are a total of twelve tanks, of which six are empty at any one time. This layout resulted in a reduction in total required lift footage and associated equipment and allowed a total system cost decrement of an estimated $9 million from the cost of the preliminary arrangement.

The third configuration was developed which consisted of four rectangular bin-type steel tanks as shown in Figure 5-3. This configuration was selected to be further developed and analyzed for the commercial-scale conceptual design. From a cost/performance standpoint, it provided the most advantages of all the configurations considered. This arrangement resulted in a reduction of 560 meters (1836 ft) of lift length, thus realizing a major capital cost reduction in the MBTESS. The reduction in lift length resulted in a reduction of parasitic power of the lift drives, improving the round trip efficiency from 68.9 to 72.4%. A decrement of $11 million from the preliminary arrangement cost was considered possible as shown below.

System Arrangement Selection Date

	Estimated major component cost, $ million	Performance
Prelim. system arr. (base)	26	Fair
Configuration No. 1	*	Poor
Configuration No. 2	17	Good
Configuration No. 3	15	Good

*The maintainability, reliability, and inspectability of a system concept are collectively reflected in total plant availability. Because of the complexity of configuration No. 1, it would be expected to have an unacceptably lower availability than the others, and it was thus rejected. A cost calculation was not considered necessary.

This optimization of the system arrangement resulted in the development of the reference design presented in Figure 5-5 and the design specifications shown in Table 5-1.

5.2. System Site Requirements

The MBTESS reference arrangement requires a land area of approximately 4.08×10^4 m^2 (1.34×10^5 ft^2). A total of 108,630 m^3 (1.42×10^5 yd^3) of soil must be excavated; this figure includes access cuts and assumes that the soil is a class 3 material with a maximum angle of repose of 65°. An excavation depth of 6.71 m (22 ft) is required except for equipment cavern locations, where excavation to a depth of 15.85 m (52 ft) is required. After completion of equipment caverns, screw casings, and screw feeder sections, 79,560 m^3 (1.04 $\times 10^5$ yd^3) of excavated soil will be returned and compacted to provide a base for the storage structures themselves. After completion of the structure floors and walls, the remaining excavated soil will be compacted around the walls to make a berm 6.1 m (20 ft) above the original grade.

In general, there are no site restrictions that would be required to protect the general public since the heat storage/transport medium, sand, does not burn, explode, or produce smoke and/or toxic fumes. Spills of the medium would not contaminate the soil or have the potential for contaminating ground water.

5.3. System Design Requirements

The MBTESS reference design described herein will conform to the codes and standards and performance and environmental criteria specified in sections 2, 3, and 4 of reference 1. In addition, the design conforms to the Barstow retrofit requirements.

5.4. System Ullage Requirements

Since the MBTESS does not require a cover gas over the heat transport medium and the system is not pressurized, the ullage requirements tend to be reduced relative to liquid storage systems. The major problem identified concerns the particulate fines normally associated with light-phase particle transport.

These fines, which are produced from particle attrition and wear, must be collected, confined, and properly disposed of. Since this material is thermally stable, compatible with the materials it contacts (non-flammable or explosive), it can be handled easily. The technology used for handling fly ash removal from flue gases can be readily adapted to this system.

5.5. Heat Exchanger Design

The heat exchanger transfers energy and provides the necessary operational flexibility for a responsive system. The system requires two types of heat exchangers — charging and discharging units. The charging heat exchanger transfers energy from the receiver working fluid to the storage medium. The discharging heat exchanger transfers energy from the storage medium to the turbine in the form of steam. The discharge heat exchanger is also required to provide steam for turbine startup and turbine seals during periods of inactivity.

Both types of heat exchangers used for the MBTESS commercial-scale conceptual design are unique in that they use a dense particle bed material, sand, as the heat transport medium. Sand flow is induced by gravity over the heated or cooled heat exchanger surface at a low velocity, approximately 0.15-0.30 m/s (0.5-1.0 fps), to provide good heat transfer without appreciable tube wear or particle attrition.[7]

The basic heat transfer characteristics of dense particle beds flowing by gravity over heated and/or cooled surfaces were investigated.[8] The analyses indicated that adequate heat transfer characteristics could be achieved with crossflow of bed material over tube bundles. However, the tube pattern must be optimized so that little or no bed stagnation or voiding exists around the tube periphery. To ensure that a high bed sand density exists throughout the heat exchangers, the following design features are required:

- A flow restrictor at the bottom of the tube bank to keep the flow passage full of moving bed material.
- A staggered tube pattern designed to reduce flow resistance, minimize stagnant regions at the top of each tube, and reduce the size of the void at the bottom of each tube.
- Inclined tubes at an angle greater than the angle of repose of the bed material. This feature diminishes the probability of flow stagnation at the top of the tube or voiding at the bottom.
- An inclined heat exchanger shell to eliminate stagnation at the tube bundle periphery and to accommodate the longitudinal flow component along the inclined tubes.

The heat transfer correlation used in sizing the heat exchangers herein was obtained from reference 8.

5.5.1. Discharge Heat Exchanger

Consideration was given to operating flexibility, reliability, and manufacturing constraints as they relate to heat exchanger design. This resulted in a minimum of six discharge heat exchangers for the system.

The discharge heat exchanger is a counterflow type having three segments: economizer, boiler, and superheater sections. This heat exchanger transfers heat from the storage medium, sand, which enters at 332C (630F) and exists at

218C (425F), thus heating the water entering at 121C/2.76 MPa (250F/400 psia) and exiting as steam at 299C/2.72 MPa (570F/395 psia).

This heat exchanger utilizes a two-drum "Sterling Boiler" arrangement that is inclined at 40° from the horizontal for the boiler section. Boiling occurs in the inclined tubes, and the steam is collected in a steam drum. The drum level is controlled via a three-element feedwater controller, which maintains proper flow to the steam drum by proportioned level biased by the flow differential of steam to feedwater flow.

The economizer and superheater sections consist of multi-pass tube bundles configured in a serpentine fashion and inclined at an angle of 40° from the horizontal. Figure 5-4 illustrates the discharge heat exchanger, and Table 5-2 describes its specifications. This heat exchanger configuration meets the Barstow inlet and outlet conditions for retrofit capability.

5.5.2. Charge Heat Exchanger

The charge heat exchanger is of the counterflow type having three segments, i.e., a desuperheater, condenser, and subcooler. The water-sand heat exchanger transfers energy from the steam entering at 510C/10.1 MPa (950F/1465 psia) and exiting as water at 249C/9.65 MPa (480F/1400 psia) to the sand which enters at 218C (425F) and exits at 332C (630F).

This heat exchanger design requires a once-through natural circulation condensing section to accommodate the flow direction of the fluids. In the natural circulation condenser condensate flows by gravity down the inclined tubes and then is collected in the lower section of the steam drum. The drum level is controlled with a level sensor, providing input to a flow control valve in the exit of the steam drum. The condensate is then transported to the subcooled section of the heat exchanger by the pressure drop within it.

The desuperheater and subcooler sections consists of multi-pass tube bundles oriented in a serpentine fashion and inclined at an angle at 40° from the horizontal. These sections are similar to those in the discharge heat exchanger. Figure 5-4 illustrates the charge heat exchanger, and Table 5-3 describes its specifications.

5.5.3. Heat Exchanger Technical Concerns

In developing the heat exchangers for the MBTESS reference design, it became evident that the heat transfer coefficient used for SiO_2 is the basic influential parameter of the design. Very little information exists in the literature today on the characteristic of sand flowing over heat exchanger tube banks.

It is necessary to understand the properties of solid particles that directly affect heat transfer from the particle to the surface. In addition, such factors as material shape, temperature, and design considerations must be known. An example is the flow distributer plate/valve configuration and its effect on the flow characteristics of dense bed material.

Consequently, the particle characterization and heat exchange configuration must be evaluated in an experimental program to ensure that the MBTESS is indeed a viable ATESS concept.

5.6. Tank Design

The ATESS tanks act as a storage medium reservoir for thermal energy. Since energy is stored within these tanks, personnel must be protected from potential hazards. Therefore, these tanks are designed in accordance with the codes, standards, and environmental criteria specified in section 2 and 4 of reference 1.

The reference design employs rectangular bin-type steel tanks, each of which contains three separate and independent compartments. The independence of these compartments provides operating flexibility to the system. Four rectangular bin-type tanks are required for the system, thereby providing a total of six hot and six cold compartments for medium storage.

The tank is 28 m wide by 82.3 m long by 21.26 m high (92 ft wide by 270 ft long by 69.75 ft high). In addition, the tanks have a trough-type bottom and a hip roof structure. Figure 5-5 shows the details of the overall tank design and its interface with the system.

The tank floor consists of 6.35 mm (0.25 in.) steel plate covered with a steel honeycomb panel 457.2 mm (18 in.) deep. This structure is not attached to the floor and thus does not develop thermal stresses between the floor and itself. In addition, it inhibits the tendency of the insulating layer of sand to move when the tank (compartment) is being emptied.

The outer vertical tank walls are dual-wall structures containing 457.2 mm (18 in.) of sand for insulation. The walls employ 6.35 mm (0.25 in.) steel sheets attached to support beams on 3.66 m (12 ft) centers attached to a lower concrete footing to accommodate vertical loading and an upper footing to resist an overturning moment.[22]

The tank compartment separation walls consist of interlocking sections of MZ 32 sheet piling, which form an 457.2 mm (18 in.) sandwich structure. The space between the steel structure is filled with sand for insulation.

The hip-roof structure supports itself, but, it does not support the heat exchangers or lifts, which are supported by independent pylons (see Figure 5-5). The roof structure consists of support beams on 3.66 m (12 ft) centers with corrugated sheet steel on each side of the beams with allowance for insulation.

Since the sand side of the MBTESS operates at essentially atmospheric conditions, the tanks were considered as bins rather than pressure vessels and designed accordingly. It is recommended for further analysis (Phase II) that all structural members (beams) of the roof and walls be placed on the outside surface, thereby encountering lower operating temperatures. This design change could result in a cost reduction for the MBTESS reference design.

5.7. Lift Design

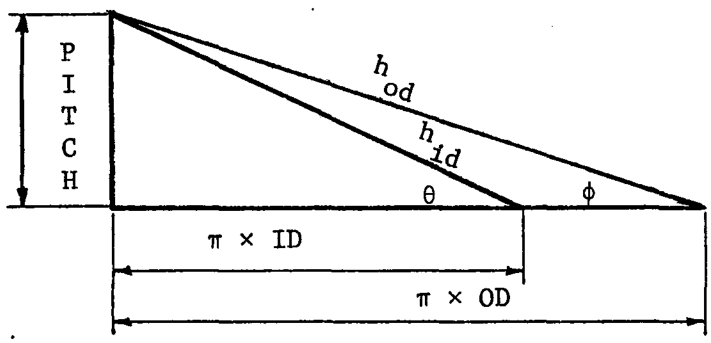
Three basic methods of transporting the bed material were considered:

- Bucket elevators
- Pneumatic fluidizing and conveying
- Archimedes type screw lifts.

Bucket elevators were found to be impractical above 350F due to lubrication problems. The pneumatic conveyer was judged to require too much power and presented problems of heat loss by the transporting air. The Archimedes screw lift as shown in detail "C" of Figure 5-5 was chosen as the best method for moving high-temperature sand. Structurally, it need only be supported at the ends and one or two places along its length depending on the allowable stress and deflection. These points of support are external to and insulated from the high temperatures, thus overcoming the problem of high-temperature lubrication. The following sections present the rationale for the lift used for the reference design with respect to dimensions, structure, drives, and insulation.

5.7.1. Geometry

The relationship between OD, ID, helix angles, and pitch is illustrated below.



$$\tan \phi = \frac{\text{pitch}}{\pi \ \text{OD}}$$

$$\tan \theta = \frac{\text{pitch}}{\pi \ \text{ID}}$$

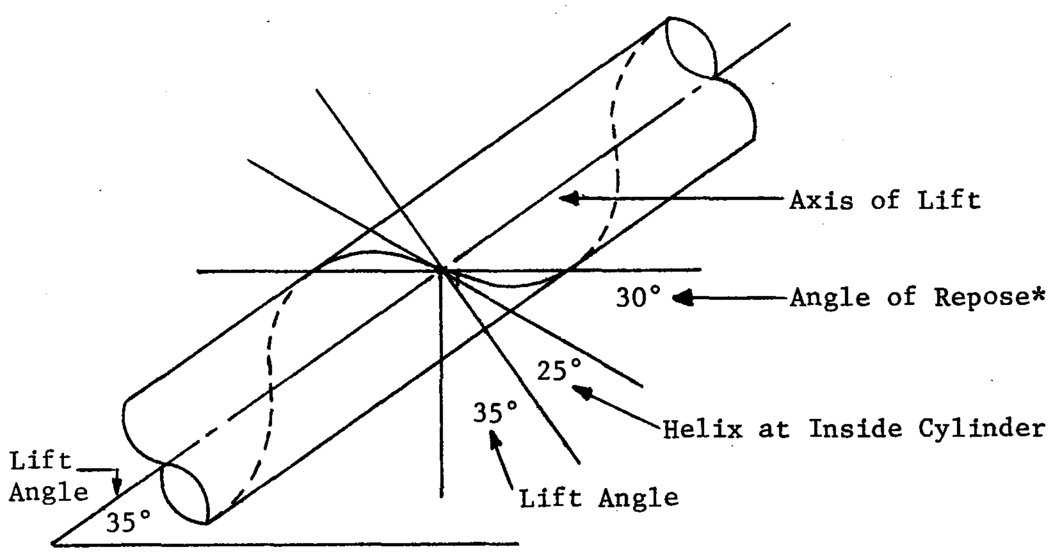
$$\frac{\text{pitch} = \tan \phi \pi \ \text{OD}}{\text{pitch} = \tan \theta \pi \ \text{ID}}$$

$$\tan \phi \ \text{OD} = \tan \theta \ \text{ID}$$

$$\text{ID} = \frac{\tan \phi}{\tan \theta} \ \text{OD}$$

ϕ and θ were chosen as 18 and 25°, respectively; h_{id} and h_{od} represent the helix lengths unraveled.

The helix angles represents the angle a line tangent to the helix makes with a plane perpendicular to the axis of the lift. From previous B&W lift design and modeling activities, the outside helix angle ϕ was determined to be 18°. The helix angle at the ID was picked so that the slope of the helix surface at the inside edge of the surface would be 30° as shown below.



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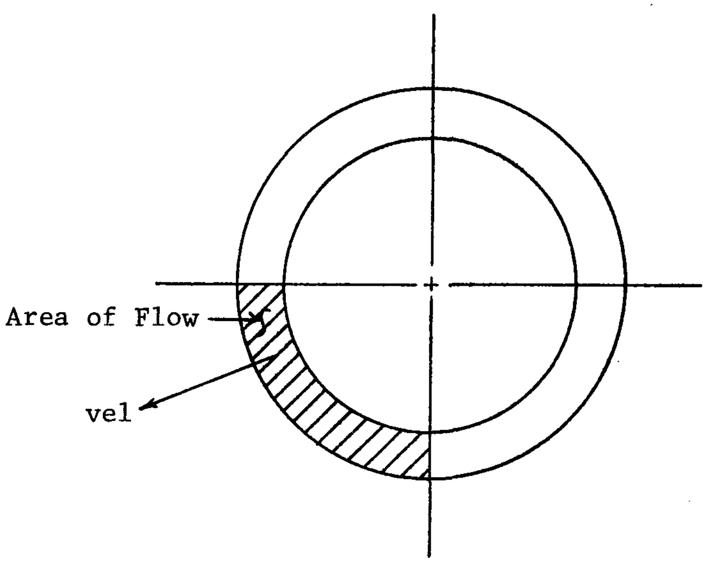
*Assumed to be angle of slide.

The reference angle of repose of the medium was established at 30°. The 35° lift angle was picked based on industrial experience,[30] which shows that the efficiency of the lift drops off quickly above this angle. The lower the lift angle, the longer the lift must be to obtain the required height of lift. Choosing the lift angle as 35° and the angle of repose as 30° constrains the ID helix angle to 25°.

The formula that relates volumetric flow rate to pitch and thus to the ID and OD is derived from that for volumetric flow rate:

$$\dot{vol} = vel \times area$$

where $\dot{vol}$ represents volumetric flow rate, vel is velocity along the axis of the lift, and area is the cross-sectional flow area. This can be visualized in the following sketch



$$\text{Area} = \frac{1}{4}\left[\frac{\pi}{4}\left(OD^2 - ID^2\right)\right]$$

vel represents a vector perpendicular to the plane of the paper.

$$vel = P \times \frac{\omega}{2\pi}$$

where P is the pitch and ω is the rotational velocity in radians/second.

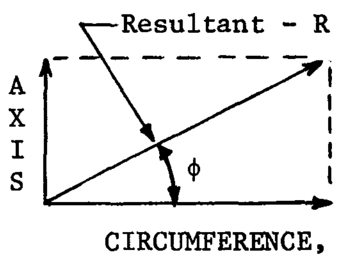
Substituting these values for vel and area and also putting pitch and ID in terms of OD, the volumetric flow rate, lift dimensions, and rotational velocity can be related by

$$\dot{vol} = (\tan \phi \pi\ OD) \times \frac{\omega}{2\pi} \times \frac{1}{4}\left\{\frac{\pi}{4}\left[OD^2 - \left(\frac{\tan \phi}{\tan \theta}\ OD\right)^2\right]\right\}.$$

Simplifying, we have

$$\dot{\text{vol}} = \tan \phi \frac{\pi}{32} \text{OD}^3 \; \omega \left[1 - \left(\frac{\tan \phi}{\tan \theta}\right)^2\right].$$

Because of abrasion, the relative velocity between the sand and the outside wall of the lift must be less than 3.05 m/s (10 fps).[23] The sand has a velocity vector along the circumference and a velocity vector along the axis of the lift. The resultant vector is along the helix. This can be seen in the vector diagram below.



$$\cos \phi = \frac{C}{R}$$

(Note: This ϕ is the same as OD helix angle.)

The circumferential velocity is $\omega \times \text{OD}/2$. The resultant, which must be less than 10 fps, is

$$R = \frac{\omega(\text{OD}/2)}{\cos\phi} .$$

We have then that

$$10 \text{ fps} = \frac{\omega \times \text{OD}}{\cos \phi \times 2};$$

solving for ω and substituting into the expression for $\dot{\text{vol}}$

$$\omega = \frac{2 \times \cos \phi}{\text{OD}} \; 10 \text{ fps},$$

$$\dot{\text{vol}} = \frac{\sin \phi}{\cos \phi} \frac{\pi}{32} \text{OD}^3 \frac{2 \times \cos \phi}{\text{OD}} \; 10 \text{ fps} \left[1 - \left(\frac{\tan \phi}{\tan \theta}\right)^2\right].$$

The volumetric flow rate required is 0.036 m^3/s (12.7 ft^3/s) per lift. Therefore solving for the OD:

$$OD = \left\{ \frac{16\ \dot{vol}}{\sin \phi\pi\ 10\ fps}\ \frac{1}{\left[1 - \left(\frac{\tan \phi}{\tan \theta}\right)^2\right]} \right\}^{\frac{1}{2}}$$

$$= \left\{ \frac{(16)\ (12.7\ ft^3/s)}{\sin 18°\ \pi\ (10\ fps)}\ \frac{1}{\left[1 - \left(\frac{\tan 18}{\tan 26}\right)^2\right]} \right\}^{\frac{1}{2}}$$

$$OD = 1.95\ m\ (6.4\ ft)$$

$$ID = 1.34\ m\ (4.4\ ft)$$

$$pitch = 1.98\ m\ (6.5\ ft)$$

The length of each lift was determined to be 56.7 m (186 ft), as seen in Figure 5-5. The screw has three starts (or flights — the helix surfaces are termed flights by the screw industry). The number of flights was established after consultation with a screw lift manufacturer.[12] It appears that trouble in feeding the lift arises if more than three flights are used; as one start fills, it partially blocks the trailing one. This is a dynamics problem that will best be resolved through further development (see section 8).

5.7.2. Structural Supports and Analysis

Carbon steel plate, 6.35 mm (0.25 in.) thick, was chosen as the construction material for both the inner and outer cylinders of the lift and the flights for the screw or helix. The lift would be supported by a combination thrust and radial bearing at the bottom end and cradle-type supported at the middle and the top end (Figure 5-5). The bearing supports are designed to limit the temperature of the bearings to below 148.9C (300F) and to allow for thermal growth of the lift.

A simple analysis of the lift shows that when nominally loaded with sand, it will deflect about 20.3 mm (0.8 in.) at the center of the 28.4 m (93 ft) span between supports. The maximum stress at the outer fibers of the lift will not exceed the yield stress of the material with a 25% safety factor, which is considered adequate for this study. The maximum cyclic stress also occurs at the midpoint of the 28.4 m (93 ft) span as the lift rotates. It was calculated by cyclic stress analysis method as outlined in reference 31 that the lift will not experience fatigue failure during the life of the plant (30 years). The natural frequency of the lift was investigated and found not to be a problem.

The lift resonant frequency in torsion was calculated using a simplified model in which the polar moment of inertia was lumped at the end of a wire assumed to represent the pure torsional spring constant. The natural frequency was found to correspond to 538 rpm, which is much faster than the maximum operational speed of the lift (29 rpm). Also, the calculation did not take into account the damping effect of the sand. Thus, no problems should arise from operation at resonant frequencies.

5.7.3. Drive Selection and Power Requirements

In order to have the capability to vary the flow of sand through the lifts in the reference design, various types of drive equipment were investigated. Information was obtained through consultations with various motor and drive manufacturers.[23-28]

Various a-c equipment was considered and eliminated for the following reasons:

- Constant-speed a-c motor with gearbox. The main disadvantage of single-speed drive is the time lag between input and output response, which is undesirable in the charging mode.

- a-c motor with inverter drives. This system was found to be very expensive.

- a-c motors with hydroviscous drives. This arrangment was considered the simplest but the least reliable mechanically.

- a-c motors with eddy-current drives. At very low rpm these drives are very inefficient.

A d-c motor with variable speed control was investigated. Although comparable in cost to the a-c motor inverter drive, the d-c drive would be simpler to operate and maintain, and would probably be more reliable. The d-c motor drive arrangement is shown in Figure 5-5.

The d-c motor drive was chosen for the reference design. However, in further design phases of the program, a possible strategy for procuring the drives would be to define the operating parameters and request proposals showing life cycle costs substantiated by operating data.

The lift power requirements consisted of two factors — friction horsepower and water horsepower.

The sliding of sand within the lifts is the major contributor to friction horsepower. Other friction losses, such as those incurred within the bearings, are negligible whan compared to the rubbing of sand and were not considered. Water horsepower is the power required to transport the sand to the desired elevation, i.e., the potential energy change.

The frictional horsepower, HP_f was calculated using the following:

$$HP_f = \frac{W \times f \times V_c}{550}$$

where W = weight of sand (52,000 lb_f),
f = coefficient of friction (0.5),
V_c = relative velocity between sand and steel (10 fps).

The water horsepower, HP_w was calculated as

$$HP_w = \frac{\dot{W} \times \Delta Z}{550}$$

where $\dot{W}$ = weight flow rate of sand (890 lb_f/s),
ΔZ = change in elevation through the lift (107 ft).

A drive efficiency factor of 80% was assumed to apply to the total lift horsepower required. The power required for one discharge lift was calculated to be 800 hp. Note that the friction horsepower calculation is conservative; it is assumed that the sand moves as a rigid body when, in fact, there will be some internal circulation. This should be quantified in future developmental programs.

5.7.4. Insulation

The entire lift would be enclosed in a carbon steel tunnel structure, which provides backfill support to the underground segment and weather protection above ground. A 6-inch gap is formed between the lift OD and the ID of the tunnel. The interior of the tunnel structure would be lined with 1.5 inches of calcium silicate insulation. This would limit heat losses from the lift to about 0.2%.

5.8. Control Philosophy

Control systems must provide for stable operation and acceptable transient response characteristics. In contrast, a control philosophy encompasses a much larger scope. In developing this philosophy for controlling a system, one must consider the interplay between such factors as safety, reliability, economics, and the man/machine interface. Considered collectively, these factors can be blended together through suitable tradeoffs, yielding an overall control philosophy for the system.

5.8.1. Interface Requirements

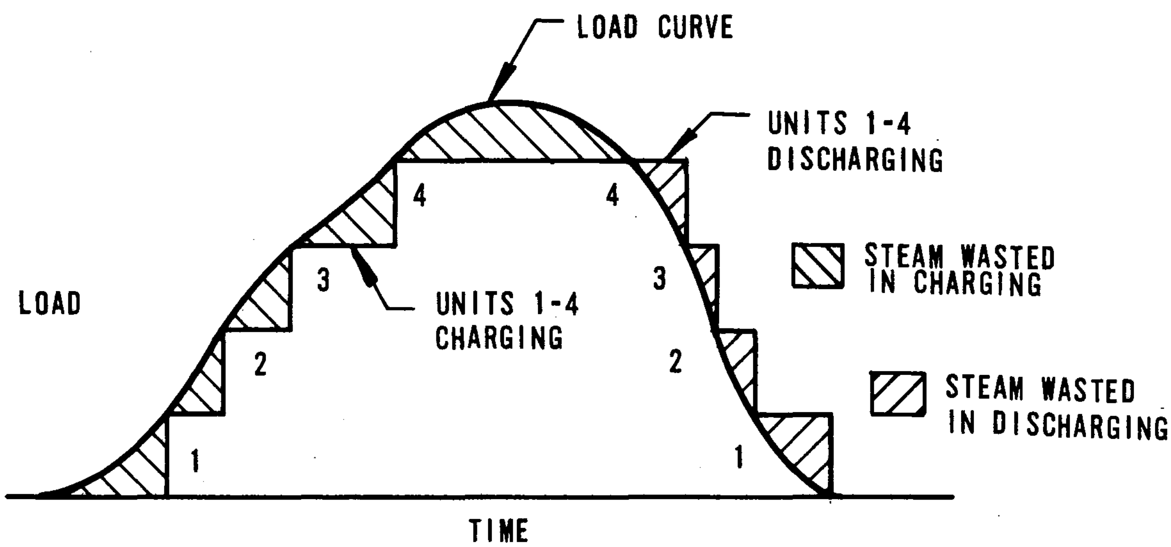
A thermal energy storage device must provide the buffer between the energy input from the receiver and the electrical load. A control system for energy storage cannot be designed until the fluctuations of the interfacing plant are known. For the conceptual design of the MBTESS, it is assumed that there are energy fluctuations on both sides of the storage subsystem. These fluctuations can be slow-acting (i.e., load-follow) or rapid (i.e., intermittent cloud cover or turbine trip) in nature. The control philosophy should be developed to accommodate both of these transient conditions in addition to startup, shutdown, and steady-state operation.

5.8.2. Discrete Vs Continuous Load Follow

As stated above, there are energy fluctuations on both sides of the storage subsystem that the control system must accommodate. These fluctuations, being both slow- and fast-acting, demand that the control system be versatile and accurate. A compromise then arises between cost and the degree of accuracy and versatility desired for the MBTESS. The choice between discrete and continuous load-follow represents such a compromise. The continuous load-following concept has the desired degree of accuracy and verstility, along with higher cost when compared to the discrete load-following concept. This estimated cost differential is overshadowed when considering the differences in versatility between the two load-following concepts.

The MBTESS is designed to accept a maximum charging rate of 260 MWt and a maximum discharging rate of 285 MWt. Since the MBTESS has six sets of hot and cold tanks, these rates correspond to 43.3 MWt/tank charging rate and 47.5 MWt/tank discharging rate. A control philosophy in which discrete charging and discharging rates are used to load-follow has many economic advantages over a system that continuously load-follows. In the discrete mode, MBTESS storage units are either on or off. Cost savings are realized since control systems are simpler and require less instrumentation, single-speed a-c drive motors can be used on the lifts instead of variable-speed d-c motors, and accurate sand valving and flow measurement problems can be avoided.

However, three disadvantages exist for the discrete mode of operation. First, this system has no way of compensating for changes in steam inputs from the receiver or any other perturbation that might originate in the storage subsystem itself or the other subsystems with which it interfaces. In the discharging mode, these perturbations could create undesirable steam conditions for the turbine. Second, a discrete system will have a poor load factor and round trip efficiency. A plot of a typical demand curve, along with the corresponding load-follow curve of the storage system operating in the discrete mode, is shown below.

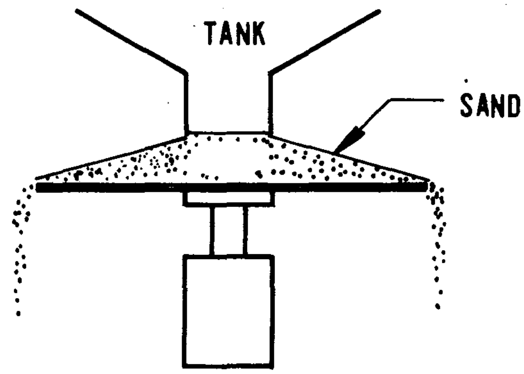


In the charging mode, a storage unit can only be put on line when an adequate steam supply exists. Excess steam from the receiver, meant for storage, will have to be blown down or wasted between the discrete switching points. When discharging, a storage unit can only drop off the line if the load has reduced below the discrete level of the next storage unit. Again, steam is blown down between the discrete levels of the storage units. The idle units may also have a longer and more complex heatup procedure since they have suffered heat losses for at least one day. The first two advantages of the discrete load follow storage concept cited above lead to poor flexibility and make this concept unacceptable. Therefore, the continuous load-follow concept was chosen to form the basis for the control philosophy of the MBTESS.

5.8.3. Instrumentation

Instrumentation will provide the necessary information for the plant operators and the data acquisition and control systems. Temperatures can be measured with thermocouples in the storage tanks and the charging and discharging heat exchangers. Since the MBTESS is not pressurized, no measurements of pressure on the sand side are necessary. The level of sand in the storage tanks and the heat exchangers must be known; a photocell arrangement or mechanical oscillation probe can be used. Since sand does not assume a flat surface, level indications will have to be corrected assuming a shape for the sand pile. This shape will vary as a function of height, the angle of repose, and whether charging or discharging is occurring. The energy stored (MWht) in any tank can be computed from the inputs of the tank thermocouples and level indicators. The levels in the heat exchangers must be controlled to maintain a dense phase of sand around the heat exchanger tubes to maximize heat transfer.

The sand flow rate can be measured in either of two ways. One is to place a deflection plate in the path of the sand; by measuring the force exerted by the sand on the plate, the flow rate can be determined by change of momentum considerations. Another approach would be to run pre-startup tests to determine the lift capacity versus speed and do without any on-going flow rate measurement during operation. The sand may be valved by gate or butterfly valves or by one that makes use of the angle of repose. Gate valves are inherently hard to control and would also tend to bind due to the abrasive sand. Butterfly valves would be simpler but would tend to leak. A concept for such a valve that makes use of the angle of repose is sketched below. Whenever the plate is rasied above the position shown, the flow stops; as the plate is lowered, flow increases. This type of valve is favored over the others because it can be controlled and would have acceptable leakage.



5.8.4. Controls

Figure 5-6 is a schematic of the controls and instrumentation for the charging subsystem. The flow of steam from the receiver is controlled by a control valve (CVSTC), so that the steam flow can be balanced appropriately between the operating units. The flow meter (FSTC), thermocouple (TSTC), and pressure transducer (PSTC) provide the necessary inputs to the charging controller, from which it can compute the incoming available energy. The charging controller then computes the required sand mass flow rate, sending a speed command to the lift drive motor (LTCH) and a position command to the angle of repose valve (CVCH). The deflection flow meter (FCH), positioned at the exit of the lift, provides feedback in determining the sand flow rate. The difference between the thermocouple (TCH) temperature and the reference outlet temperature produces an error signal, which is integrated to provide trimming to the flow rate determination. The level sensor (LCH) will provide feedback to the control valve (CVCH) to maintain proper heat exchanger level, ensuring dense-phase heat transfer.

The controls and instrumentation for the discharging subsystem are also shown in Figure 5-6. From a demand signal, steam flow and sand flow rates are computed by the discharging controller. Condensate is pumped (PDS) and the flow rate is controlled by the control valve (CVDS). Flow rate is monitored by flowmeter (FSTD) and signals from thermocouples (TSTD) and pressure transducers (PSTD) are compared to their setpoints 299C/2.72 MPa (570F/395 psia) to create the necessary feedback. The discharge controller sends a speed control signal to the lift drive motor (LTDS) and a position command to the angle of repose valve (CVDS). The deflection flowmeter (FDS), positioned at the exit of the lift, provides feedback in determining the sand flow rate. The level sensor (LDS), will provide feedback to the control valve (CVDS) to maintain proper heat exchanger level, ensuring dense-phase heat transfer.

5.9. Cost Estimates

The budgetary cost estimate prepared for the MBTESS reference design included all major components and systems. The estimate can be divided into the following areas:

- Site preparation
- Sand storage and equipment structures
- Sand moving equipment and support structures
- Heat exchangers and piping
- Auxiliary equipment and instrumentation
- Job-indirect costs
- Medium costs

Site preparation costs comprised excavation costs and backfill and compaction costs; labor, equipment, and equipment moving charges were included. Clearing and grubbing was not included because the TESS site will overlap the power plant site. The soil description given in reference 2 was used, and all rock strata were assumed to lie below the 15.85 m (52 ft) excavation depth. Reference 9 was used as a source for the cost estimates.

Cost estimates for storage and equipment structures were based on structures described in section 5.6. Steel plate and structural members were used for the sand storage tanks and steel reinforced concrete and steel reinforced concrete block in the equipment caverns. Sand and Thermal Wool II insulation were included. Cost estimates include material costs and shipment, fabrication costs, erection costs, and equipment and consumables costs. References 6, 9, 10, 11, and 15 were the source of the cost data.

Sand moving equipment and support structures priced were sand lift screws, bearings, drive motors, reduction gears, eddy-current drives, screw casings, bearing casings, screw shaft extensions, support columns, and pads. References 9, 10, 11, 12, 16, and 17 were used to estimate material and shipping charges, assembly, and erection costs.

Heat exchangers and piping were estimated with information from reference 9, 13, 15, 18, and 19. Material, shipping, assembly, erection, and insulation costs were included. Valves, fittings, and pipe were priced on the assumption that steam and feedwater lines ran within 30.5 m (100 ft) of the TESS.

Auxiliary equipment and instrumentation costs were estimated for obvious components, and an overall estimating factor was used for the remaining items. References 9, 13, and 20 were used for these estimates.

Job-indirect costs were based on total contracted manhours expended in 12 different job classifications. These indirect costs, calculated by reference 9, include the following:

Jobsite Overhead

1. Construction Equipment, Owned or Rented, With a Value Over $500.00

Trucks	Generators	Hoists
Autos	Compressors*	Scaffolding
Cranes		

2. Job Organization

Project manager	Safety man	Secretary
Superintendent	Master mechanic	Material checkers
Asst. Superintendent	Timekeepers	Tool shed keeper
Engineer		

3. Temporary Facilities

Contractor's office	Furniture and fixtures	Storage shed
Architect-owner's office	Janitor service	Carpenter shop
Electric service	Signs	Saw shed
Water service	Temporary toilets	Field plan tables
Heating	Tool shed	Temporary stairs
Cooling		

4. Supplies

Photographs	Drinking water	Cups
Stationery	Ice	First aid equipment
Postage	Dispensers	

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*Except earthwork equipment used by the general contractor.

5. Temporary Protection and OSHA Requirements

Dust control — Protect sills, corners, stairs
Noise control — Rails at openings, slab edges
— Rain protection
— Protect existing property/trees

6. Telephone — Communications

Telephone-telegraph	Loudspeaker

7. Insurance — Bonds — Sales Tax

Truck and auto	Special risk
Public liability	
Builder's risk	

8. Insurance and Taxes on Labor Paid by Contractor

Workmen's compensation	Bodily injury	Social security

9. Progress Reports and Scheduling

Progress reports	Certified payrolls

10. Expendables — Which is Any Tool or Consumable Costing $500.00 or less

Hammers	Slings	Clamps
Blades	Bars	Fuels
Bits	Cutters	Lubricants
Shovels		

Home Office Overhead

1. Salaries

Officers or owners	Engineers	Clerical
Estimators		

2. Office Rent or Mortgage

3. Furniture, Fixtures, Machines

4. Stationery — Supplies — Postage

5. Utilities

6. Telephone

7. Insurance

8. Business Taxes — Licenses

9. Legal and Consulting Fees

10. Sales Promotion — Education

Entertainment	Conventions	Manuals
Associations	Study courses	Travel
Seminars	Textbooks	

11. Equipment Purchases or Mortgages

Vehicles	Cranes	Loaders, etc.

12. Yard Expense

13. Loss of Interest on Retainages

14. Interest Expense

Because of the uncertainty surrounding media requirements a range of cost was developed. At the low range ($1.8 million), construction sand is obtained and processed at the plant site. The high estimate ($4.7 million) is based on purchase and shipment or preprocessed silica flour. References 6, 9, 20, and 21 were used to estimate this cost.

In all estimates, minimum wages payable on federal and federally assisted construction projects in the Los Angeles area during summer of 1980 were used. These wages included health and welfare insurance and pension and vacation funds but did not reflect rates for apprentices or premium rates for overtime.

Table 5-4 is a summary of the estimated costs broken down into the areas discussed above. Contractor's profit, engineering cost, and media cost are included. A second breakdown, shown in Table 5-5, is based on materials. These figures also show the construction cost variations between the Houston and Barstow areas. Table 5-6 is a more detailed version of Table 5-4, giving costs for major components and systems. Table 5-7 provides a breakdown of the energy related, power related, and specific total costs of the MBTESS.

All of these estimates are based on a moving sand bed system that would meet Barstow requirements. The estimated cost of the commercial-scale conceptual design is $26 million.

Table 5-1. MBTESS Reference Design Specifications

Design Characteristics	
Properties of storage medium	
Material	SiO_2
Operating range	204/332C (400/630F)
Density	1522 kg/m^3 (95 lb/ft^2)
Specific heat	1030 J/kg-°K (0.246 Btu/lb-°F)
Particle size	44-74 × 10^{-3} mm (17-29 × 10^{-4} in.)
Void fraction	0.40
SiO_2 storage medium mass - working	0.57 × 10^8 kg (1.27 × 10^8 lb)
- for costing	0.92 × 10^8 kg (2.02 × 10^8 lb)
Tank characteristics	
No. of storage tanks	4
Tank geometry	Figure 5-5
Tank volume: per tank	2.51 × 10^4 m^3 (8.86 × 10^5 ft^3)
Total	10.1 × 10^4 m^3 (3.54 × 10^6 ft^3)
Tank material	ASTM A-53
Design temperature	343C (650F)
Tank mass: per tank	5.17 × 10^5 kg (1.14 × 10^6 lb)
Total	2.07 × 10^6 kg (4.56 × 10^6 lb)
Tank surface area	
Top/tank	2.78 × 10^3 m^2 (2.88 × 10^4 ft^2)
Top, total	1.07 × 10^4 m^2 (1.15 × 10^5 ft^2)
Side/tank	1.12 × 10^3 m^2 (1.21 × 10^4 ft^2)
Side, total	4.50 × 10^3 m^2 (4.84 × 10^4 ft^2)
Bottom/tank	2.78 × 10^3 m^2 (2.88 × 10^4 ft^2)
Bottom, total	1.07 × 10^4 m^2 (1.15 × 10^5 ft^2)
Tank insulation material (internal)	
Sides and bottom	SiO_2
Roof	Thermal Wool II
Insulation thickness	
Sides and bottom	457.2 mm (18 in.)
Roof	114.3 mm (4.5 in.)
Operating Characteristics	
Extractable capacity	1710 MWht (5.84 × 10^9 Btu)
Charging	
Maximum	260 MWt (887 × 10^6 Btu/h)
Design	260 MWt (887 × 10^6 Btu/h)
Minimum	TBD[(a)]

Table 5-1. (Cont'd)

Discharging	
Maximum	285 MWt (973×10^6 Btu/h)
Design	285 MWt (973×10^6 Btu/h)
Minimum	TBD
Duration	
@ max discharge rate	6 hours
@ design discharge rate	6 hours
@ min discharge rate	TBD
Storage medium operating temperatures	
Hot	332C (630F)
Cold	218C (425F)
Storage tank operating pressure	Atmospheric
Storage ramp rate, % of max discharge power/minute	TBD

(a) TBD: to be determined in future development efforts.

Table 5-2. MBTESS Commercial-Scale Conceptual Design: Discharge Heat Exchanger Specifications

Number of heat exchangers required[a]	6
Duty	47.5 MWt (1.62 × 10^8 Btu/h)
Tube side conditions (steam/water)	
Inlet	121C/2.76 MPa (250F/400 psia)
Outlet	299C/2.72 MPa (570F/395 psia)
Minimum flow	TBD[b]
Maximum flow	6.85 × 10^4 kg/h (1.51 × 10^5 lb/h)
Fouling conductance	0.057 MWt/m^2 °C (10^4 Btu/h-ft^2-F)
Tube material: superheater	Carbon Moly. SA-209 Grade T1A
boiler	Carbon Moly. SA-209 Grade T1A
economizer	Carbon Moly. SA-178 Grade T1C
Tube OD	19 mm (0.75 in.)
Tube ID	16 mm (0.62 in.)
Shell side conditions (S_iO_2) @ atmospheric pressure	
Inlet	332C (630F)
Outlet	218C (425F)
Minimum flow	TBD
Maximum flow	1.45 × 10^6 kg/h (3.19 × 10^6 lb/h)
Surface area	819 m^2 (8820 ft^2)
Duty cycle	Daily
Service life	30 years

[a]The data in this table are for one of the six parallel heat exchangers.
[b]TBD: to be determined in future development efforts.

Table 5-3. MBTESS Commercial-Scale Conceptual Design: Charge Heat Exchanger Specifications

Number of heat exchangers required[a]	6
Duty	43.4 MWt (1.48×10^8 Btu/h)
Tube side conditions (steam/water)	
Inlet	510C/10.1 MPa (950F/1465 psia)
Outlet	249C/9.65 MPa (480F/1400 psia)
Minimum flow	TBD
Maximum flow	4.05×10^5 kg/h (8.92×10^5 lb/h)
Fouling conductance	0.57 MWt/m^2-°C (10^4 Btu/h-ft^2-F)
Tube material	
Desuperheater	Croloy 1 SA-213 Grade T12
Condenser	Carbon Moly. SA-209 Grade T1A
Subcooler	Carbon Moly. SA-209 Grade T1A
Tube OD	19 mm (0.75 in.)
Tube ID	15 mm (0.59 in.)
Shell side conditions (S_iO_2) @ atmospheric pressure	
Inlet	218C (425F)
Outlet	332C (630F)
Minimum flow	TBD
Maximum flow	7.94×10^6 kg/h (1.75×10^7 lb/h)
Surface area	1124 m^2 (12,100 ft^2)
Duty cycle	Daily
Service life	30 years

[a]The data in this table are for one of the six parallel heat exchangers.
[b]TBD: to be determined in future development efforts.

Table 5-4. Cost Estimate Summary — MBTESS

CLIENT Barstow, Ca. DESCRIPTION MSB System CONT. NO:
LOCATION Moving Sand Bed TESS Table 5.9-1 MADE BY:
PROJECT APPROVED:

Account: 5700 Energy Storage System

	ITEM & DESCRIPTION	MANHOURS	ESTIMATED COST (June 1980 $) LABOR	SUBCONTRACTS	MATERIALS	TOTALS
	Excavation & Backfill	22,212	491,000	269,000		760,000
	Concrete	11,896	179,000	124,000	151,000	454,000
	Structural Steel	94,964	1,635,000	688,000	1,690,000	4,013,000
	Machinery & Equipment	43,486	741,000	664,000	5,318,000	6,723,000
	Insulation	9,469	132,000	389,000	199,000	720,000
	Piping & Heat Exchangers	14,657	263,000	228,000	4,817,000	5,308,000
	Instrumentation			279,000		279,000
	Media & Processing Costs	4,480	67,000	134,000	1,601,000	1,802,000
(1)	Direct Field Costs	201,000	3,508,000	2,775,000	13,776,000	20,059,000
(2)	Indirect Costs					1,886,000
(3)	Contractors Profit					
	(10% of (1+2))					2,195,000
(4)	Engineering					
	(10% of (1+2 -$1,802,000)					2,014,000
	TOTAL					26,154,000

Table 5-5. Cost Summary — Material Breakdown

CLIENT SANDIA DESCRIPTION Summary CONT. NO:

LOCATION Barstow and Houston Areas MADE BY:

PROJECT Moving Sand Bed TESS Table 5.9-2 APPROVED:

Account: 5700 Energy Storage System

ITEM & DESCRIPTION	MANHOURS	ESTIMATED COST (June 1980 $) LABOR	SUBCONTRACTS	MATERIALS	TOTALS
Field Cost Summary Barstow Area					
Excavation & Backfill	22,212	491,000	269,000		760,000
Concrete	11,896	179,000	124,000	151,000	454,000
Structural Steel	94,964	1,635,000	688,000	1,690,000	4,013,000
Machinery & Equipment	43,486	741,000	664,000	5,318,000	6,723,000
Insulation	9,469	132,000	389,000	199,000	720,000
Piping & Heat Exchangers	14,657	263,000	228,000	4,817,000	5,308,000
Instrumentation			279,000		279,000
Media & Processing Costs	4,480	67,000	134,000	1,601,000	1,802,000
TOTAL	201,164	3,508,000	2,775,000	13,776,000	20,059,000
Field Cost Summary -					
Houston Area					
Excavation & Backfill	22,212	353,000	269,000		622,000
Concrete	11,896	160,000	124,000	151,000	435,000
Structural Steel	94,964	1,389,000	688,000	1,690,000	3,767,000
Machinery & Equipment	43,486	645,000	664,000	5,318,000	6,627,000
Insulation	9,469	95,000	389,000	199,000	683,000
Piping & Heat Exchangers	14,657	221,000	228,000	4,817,000	5,266,000
Instrumentation			279,000		279,000
Media & Processing Costs	4,480	49,000	134,000	1,601,000	1,784,000
TOTAL	201,164	2,912,000	2,775,000	13,776,000	19,463,000

Table 5-6. Estimate Summary — Detail Breakdown

ESTIMATE SUMMARY

PROJECT: Moving Sand Bed TESS

ESTIMATE NO.

SHEET NO. 1 of 4

PREPARED BY: DATE:

CHECKED BY: DATE:

ACCOUNT: 5700 Energy Storage System

MAIN ACCOUNT	DESCRIPTION	QUANTITY	MATERIALS UNIT PRICE	MATERIALS AMOUNT	LABOR UNIT M.H.	LABOR TOTAL M.H.	LABOR RATE	LABOR AMOUNT	SUB-CONTRACT UNIT PRICE	SUB-CONTRACT AMOUNT	TOTAL
							MH				
2-18	Excavation	142,000CY			.0081/CY	1,152	27.56	$ 31,800	$.450/CY	$ 64,000	
	Equipment Moving									1,600	
	TOTAL										$ 97,000
2-21	Backfill & Compaction	142,000CY			.148/CY	21,060	21.82	$ 459,000	$139/CY	$198,000	
	Equipment Moving									5,200	
	TOTAL										$ 662,000
2-43	Aggregate Base Under Floor	2,840 T	6.00/T	$17,000	2.71/CY	84	22.92	$ 1,925	1.84/CY	1,308	$ 20,200
	Equipment Caverns										
3-5,3-6,3-20	Concrete	236 CY	$45/CY	$10,600	1.19MH/CY	282	14.17	$ 3,996			$ 14,600
3-5,3-20	Screeds & Forms		Various	$ 1,500	Various	271	16.60	$ 4,500			6,000
3-5,4-1	Reinforcing Steel								3.8/100lb	98,300	98,300
3-5,7-0	Moisture Barrier	75,500 ft^2	.16/ft^2	11,800	.019/ft^2	1,416	13.98	$ 19,800		1,600	33,200
4-1	Concrete Block & Morter	111,000 Blk	0.74/Blk	81,800	0.077/Blk	8,560	15.00	$ 128,500			210,300
5-7	Steel Deck & Beams	Various		9900	Various	103	17.46	$ 1,800		$ 12,300	24,000
	TOTAL										$ 386,400
	Foundations & Footings										
3-1,3-3	Concrete	516 CY	45/CY	$23,260	1.16/CY	597	14.12	$ 8,410			$ 31,700
3-1,3-3	Reinforcing Steel									22,200	22,200
3-1,3-3	Screeds & Forms			$ 4,900		657	16.58	$ 11,200			$ 16,100
	TOTAL										$ 70,000
	Structures										
8-31	Floor & Wall Plates	2,084,000lbs	25¢/lb	521,000	.015/lb	35,864	17.46	$ 626,200			$1,147,200
8-31	Structural Shapes	1,868,800 "	25.1¢/lb	469,100	.025/lb	32,108	17.46	$ 560,000			$1,029,700
8-31	Corrugated Roof Sheets	587,600 "				2,026			70.7¢/lb	415,400	$ 415,400
8-31	16 Gauge Floor Baffles	107,000				369			70.0¢/lb	74,970	$ 74,970
5	MZ 32 Sheet Piling	856,640 "		299,800		8,489	17.46	$ 171,300			$ 471,100
5	Center Columns	24WF36x230	$.251/lb	131,600		4,505	17.46	$ 78,700		78,600	$ 288,900
	TOTAL										$3,427,000

Table 5-6. (Cont'd)

ESTIMATE SUMMARY

PROJECT: Moving Sand Bed TESS

ESTIMATE NO.

SHEET NO. 2 of 4

PREPARED BY: DATE:

CHECKED BY: DATE:

ACCOUNT: 5700 Energy Storage System

MAIN ACCOUNT	DESCRIPTION	QUANTITY	MATERIALS UNIT PRICE	MATERIALS AMOUNT	LABOR UNIT M.H.	LABOR TOTAL M.H.	LABOR RATE	LABOR AMOUNT	SUB-CONTRACT UNIT PRICE	SUB-CONTRACT AMOUNT	TOTAL
	Structural Insulation										
2-18, 2-21	18" Layer of Sand	11,620tons	7.70/T	89,500	.04 MH/T	465	13.64	6360	.252/T	2920	$ 98,800
15-82	Roof Thermal Wool Type II	124,000	.88/ft²	109,500		9004	13.98	125,900			$235,400
	Column insulation (Column Silicate)	2500							11.2/ft²	28,000	$ 28,000
	TOTAL										$362,000
	Sand Screws										
100	Screw	48-465'Sect	67,200	3,225,600		31,271	17.46	546,000		546,000	4,317,600
5	Screw Casing	5.537x10⁵lbs	.37$/lb	226,500		5,470	17.46	95,500		95,500	417,500
15	Screw Casing Insulation								6.80/ft²	358,000	358,000
5	Supports		Various	32,100		6,030	16.75	101,000		11,350	144,000
100	Bearings			420,600		1,132	17.46	22,900		117,900	561,000
100	Drives			1,512,000		9,834	15.36	151,000			1,663,000
											7,461,000
3	Center Pillar Foundations			561		29	14.12	406		576	$ 1,500
	Piping										
15-43	Pipe 16" sch 120	840'	$90/ft	75,600		336	21.31	7,200			$ 82,800
15-43	Pipe 14" std.	840'	54.57/ft	45,800		244	21.31	5,200			51,000
15-43	Pipe 12" sch 120	740'	63.34/ft	46,800		260	21.31	5,600			52,400
15-43	Pipe 10" std.	740'	40.48/ft	30,000		174	21.31	3,800			33,800
15.43	Tecs	24	various	7,600		503	21.31	10,700			18,300
15.43	Gate Valves	24	various	89,500		108	21.31	2,300			91,800
	TOTAL										$ 330,100
	Heat Exchangers										
100	Charging	6	$479,000	2,874,000		6,516	17.48	114,000		114,000	3,102,000
100	Discharging	6	275,000	1,648,000		6,516	17.48	114,000		114,000	1,876,000
	TOTAL										$4,978,000

Table 5-6. (Cont'd)

ESTIMATE SUMMARY

PROJECT: Moving Sand Bed TESS

ESTIMATE NO.:
SHEET NO. 3 of 4
PREPARED BY: DATE:
CHECKED BY: DATE:

MAIN ACCOUNT	DESCRIPTION	QUANTITY	MATERIALS UNIT PRICE	MATERIALS AMOUNT	LABOR UNIT M.H.	LABOR TOTAL M.H.	LABOR RATE	LABOR AMOUNT	SUB-CONTRACT UNIT PRICE	SUB-CONTRACT AMOUNT	TOTAL
	Auxiliary Equipment										
100	5 Ton Hoist	12	5,590	67,000		989	16.95	16,770			84,000
100	Ullage Screens	2		50,000		160	16.95	2,700			52,700
100	Ullage Troughes & Conveyors	1		43,000		100	16.95	1,700			44,700
											181,000
	Control & Instrumentation										
100	Transducers & Detectors									138,000	
100	Controllers									24,000	
100	Recorders & Panel									17,000	
100	Microprocessors									100,000	
											279,000
101	Media & Processing Equip.										
	500 HP Reversible Impactors	4		166,000		200	16.95	3,400			169,400
	500 HP Motors	4		80,000		56	16.95	900			80,900
	Trough Conveyors	4								134,100	134,100
	Sand	101,000T	13.55/T	1,355,000							1,355,000
	Processing					4,224	14.87	62,800			62,800
											1,802,000
	Job Indirect Costs										
	Account 2					22,761	5.90	134,000			
	Account 3					3,252	7.58	25,000			
	Account 4					8,560	6.71	57,000			
	Account 5					24,597	15.48	381,000			
	Account 8					70,367	6.08	428,000			
	Account 15					10,629	11.35	121,000			
	Account 100					56,518	13.09	740,000			
	Account 101					4,480	0	0			
											1,886,000
	TOTAL										21,944,000

Table 5-6. (Cont'd)

CLIENT SANDIA DESCRIPTION Estimate Summary CONT. NO:
LOCATION Totals MADE BY:
PROJECT Moving Sand Bed TESS Table 5.9-3 APPROVED:

Account: 5700 Energy Storage System

	ITEM & DESCRIPTION	MANHOURS	ESTIMATED COST (June 1980 $) LABOR	SUBCONTRACTS	MATERIALS	TOTALS
	Site Preparation	22,212	491,000	269,000		760,000
	Sand Storage & Equipment Structures	104,829	1,749,000	736,000	1,782,000	4,267,000
	Sand Moving Equipment & Supports	53,737	916,000	1,129,000	5,417,000	7,462,000
	Heat Exchangers & Piping	14,657	263,000	228,000	4,817,000	5,308,000
	Aux. Equipment & Instrumentation	1,249	21,000	279,000	160,000	460,000
	Media & Processing Costs	4,480	67,000	134,000	1,601,000	1,802,000
(1)	Direct Field Costs	201,164	3,507,000	2,775,000	13,777,000	20,059,000
(2)	Indirect Costs					1,886,000
(3)	Contractor's Profit (10% of 1 and 2)					2,195,000
(4)	Engineering (10% of (1+2 - $1,802,000))					2,014,000
	TOTAL					26,154,000

Babcock & Wilcox

Table 5-7. MBTESS Energy-, Power-, and Specific-Related Costs (June 1980 $ × 10³)

Energy related cost, C_s

Item	Direct field cost	Indirect cost	
Excavation	97	7	
Backfill and Compaction	662	125	
Foundations and Footings	92	10	
Storage Structure	3,427	629	
Insulation	362	105	
Medium	1,802	--	
Subtotal	6,442 +	876 =	$7,318
	Contractor's Profit		732
	Engineering		552
		Total	$8,602

Power related cost, C_p

Item	Direct field cost	Indirect cost	
Lifts	7,461	731	
Piping	380	18	
Heat Exchangers	4,978	171	
Aux. Equipment	181	16	
Controls and Inst.	279	--	
Equipment Covers	337	74	
Subtotal	13,616 +	1,010 =	$14,626
	Contractor's Profit		1,463
	Engineering		1,463
		Total	$17,552

C_{s_e} = \$8,602/(70,000 KWe × 6 h) = \$20.48/KWhe

C_{s_t} = \$8,602/(285,000 KWt × 6 h) = \$5.03/KWht

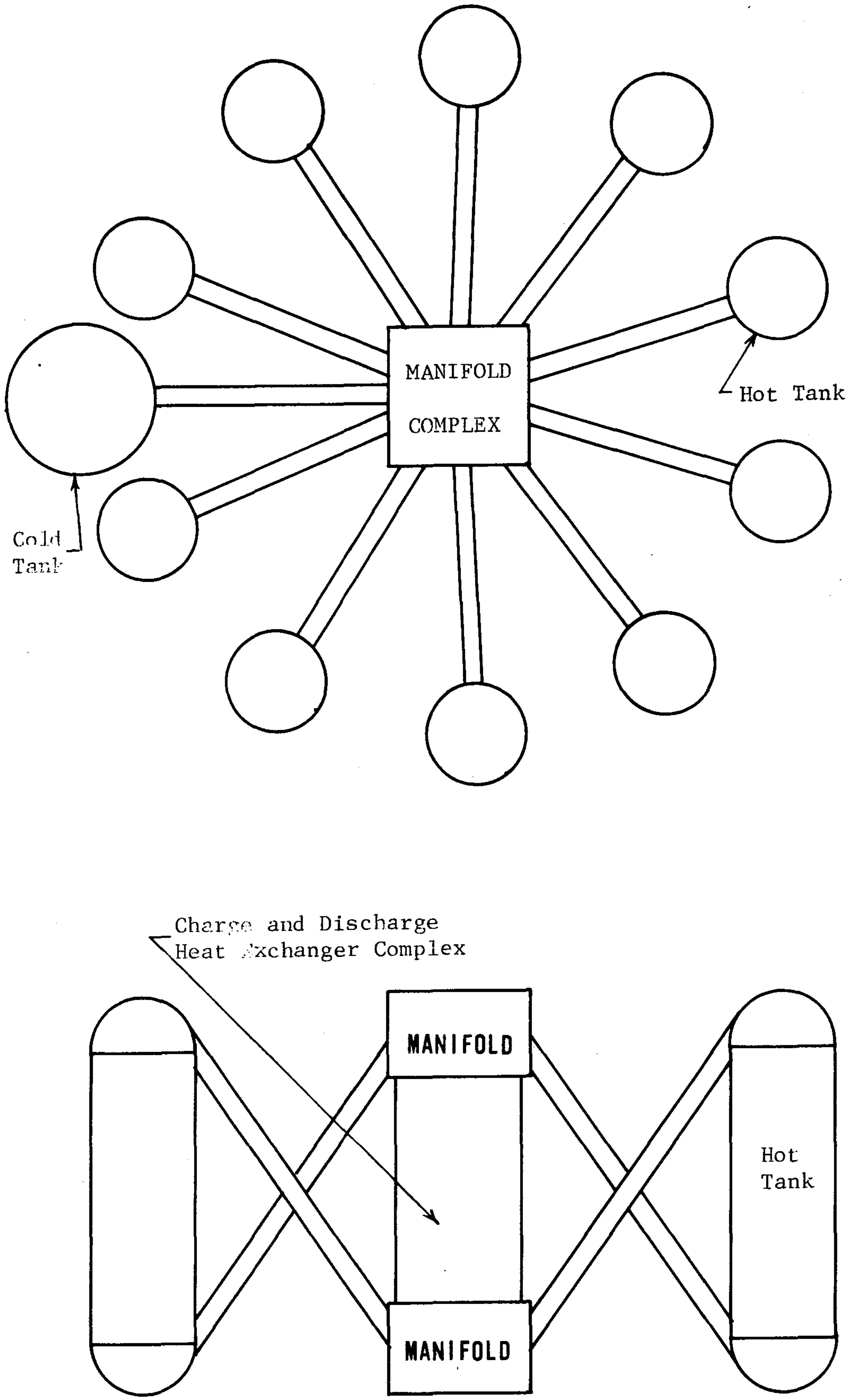
C_{p_e} = \$17,552/70,000 KWe = \$250.74/KWe

C_{p_t} = \$17,552/285,000 KWt = \$61.59/KWt

$C_T = (C_p + (C_s \cdot h))/h$ = \$62.27 KWhe

= \$15.30 KWht

Figure 5-1. Configuration 1



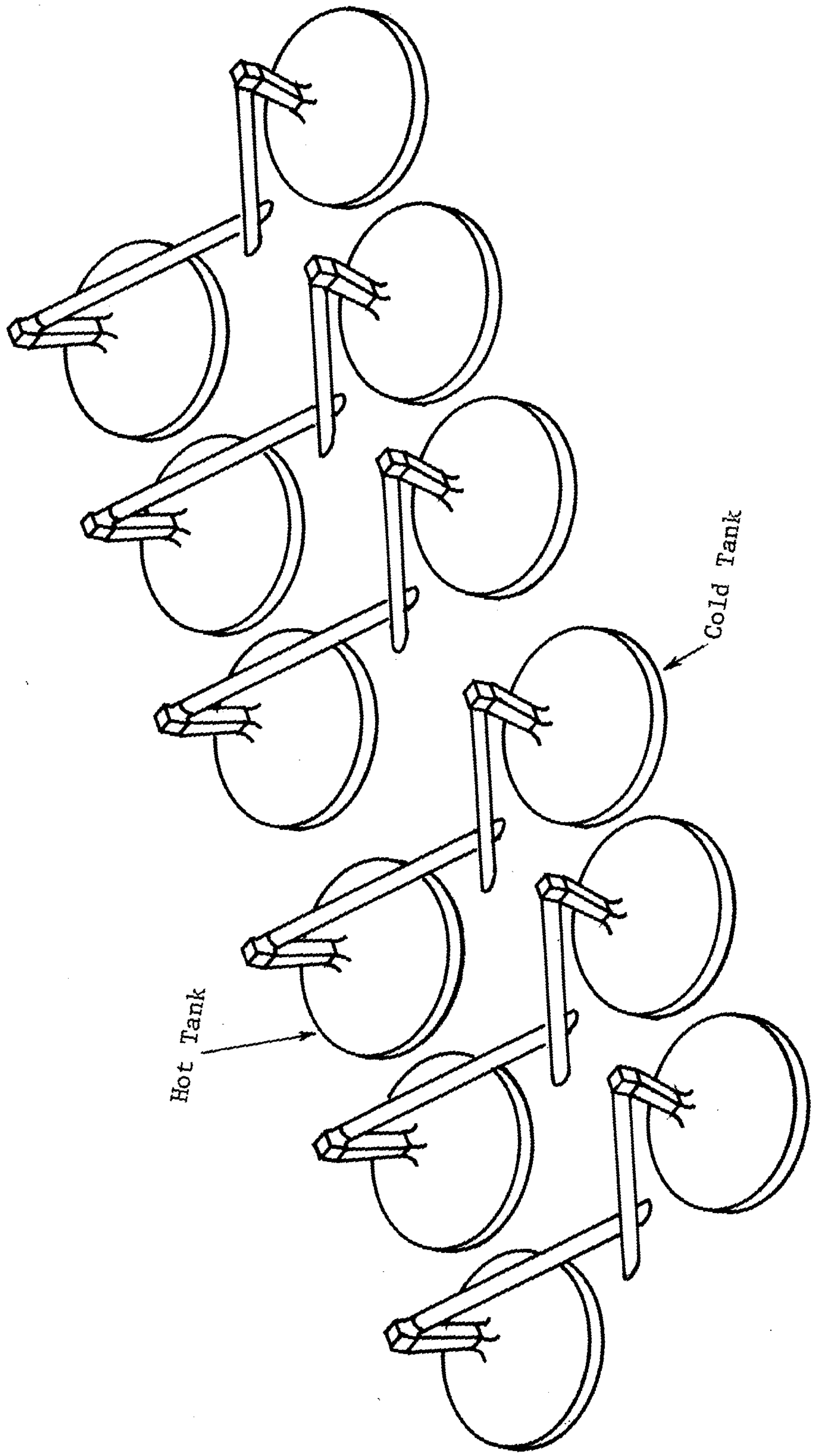


Figure 5-2. Configuration 2

Figure 5-3. Configuration 3

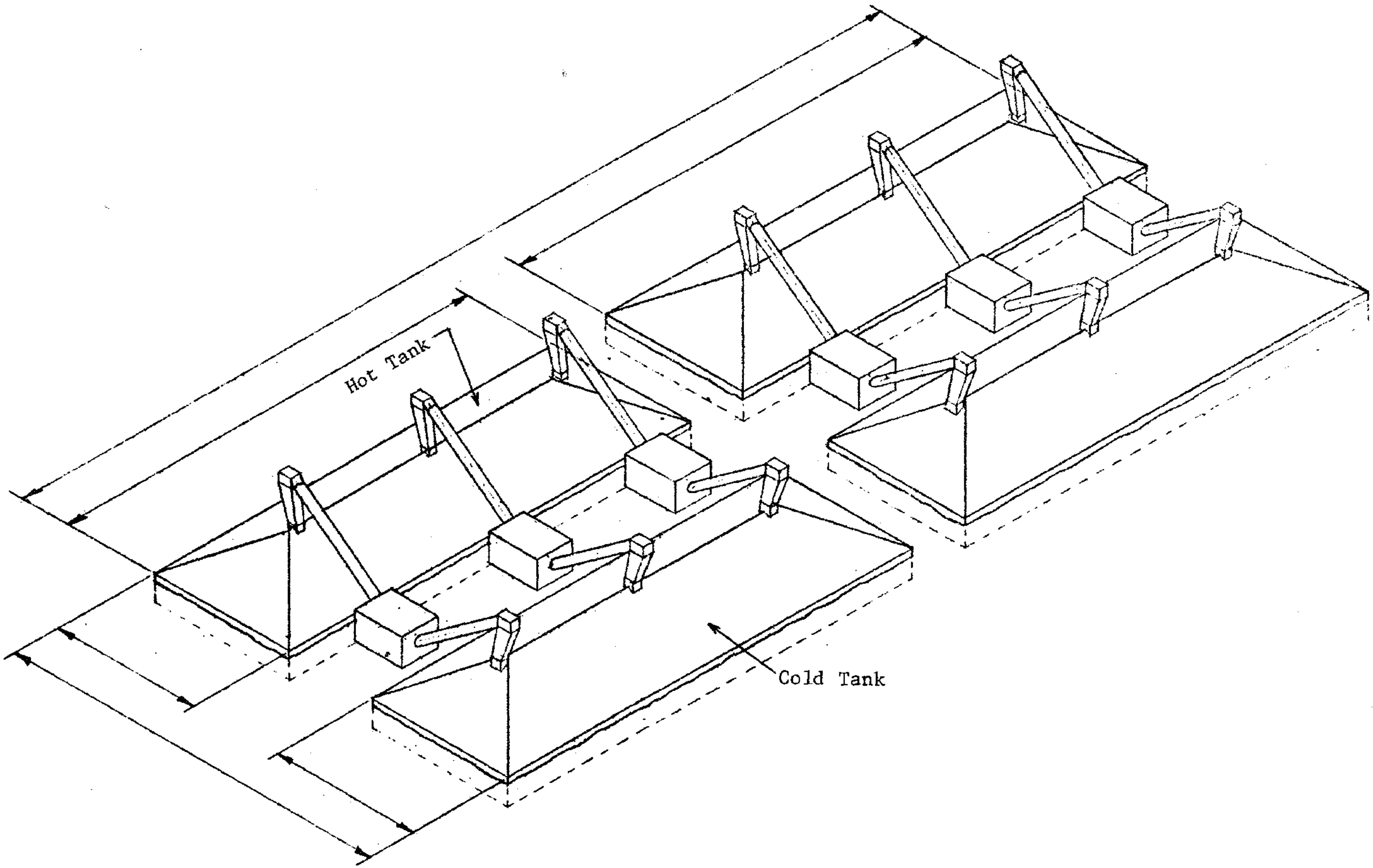
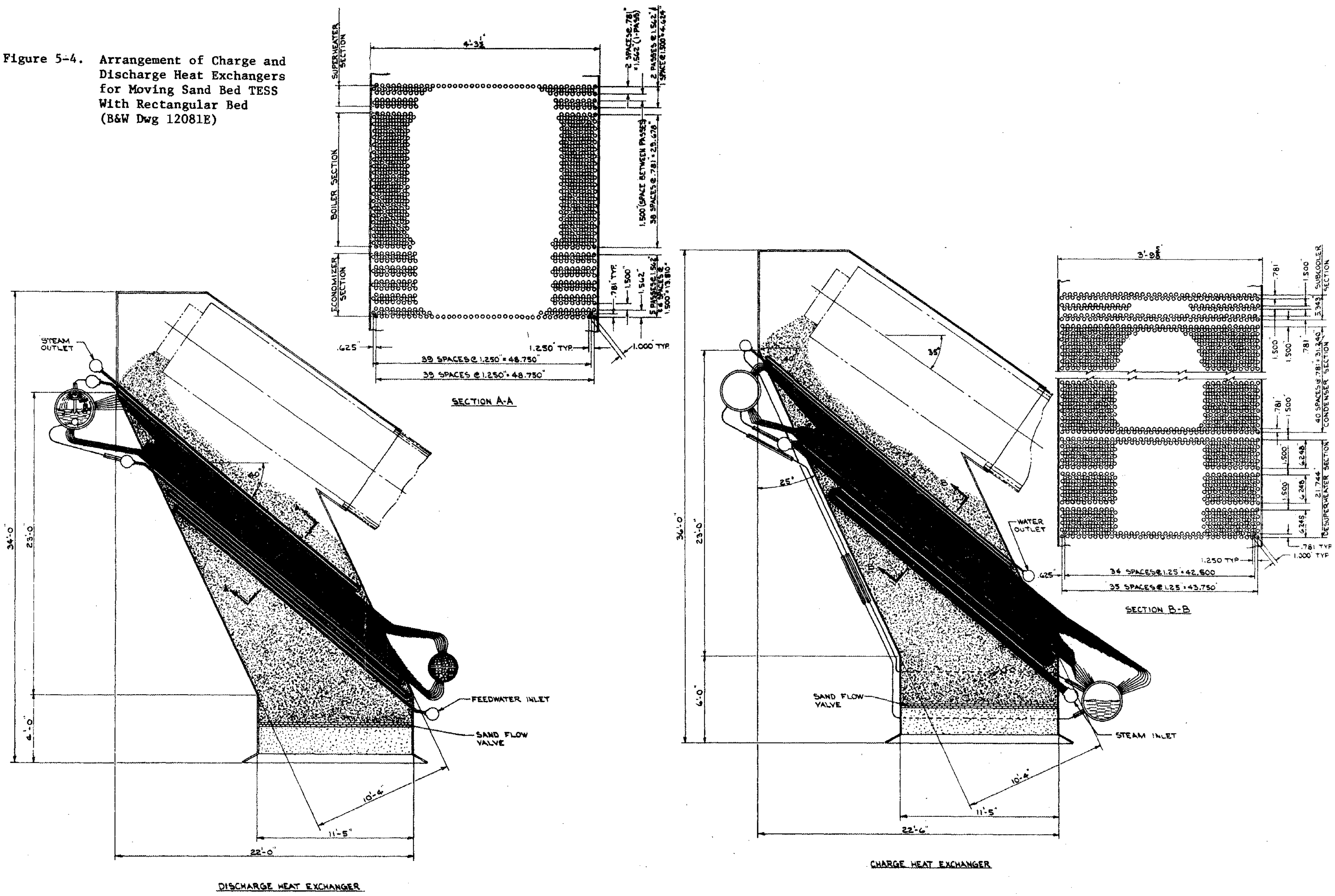


Figure 5-4. Arrangement of Charge and Discharge Heat Exchangers for Moving Sand Bed TESS With Rectangular Bed (B&W Dwg 12081E)



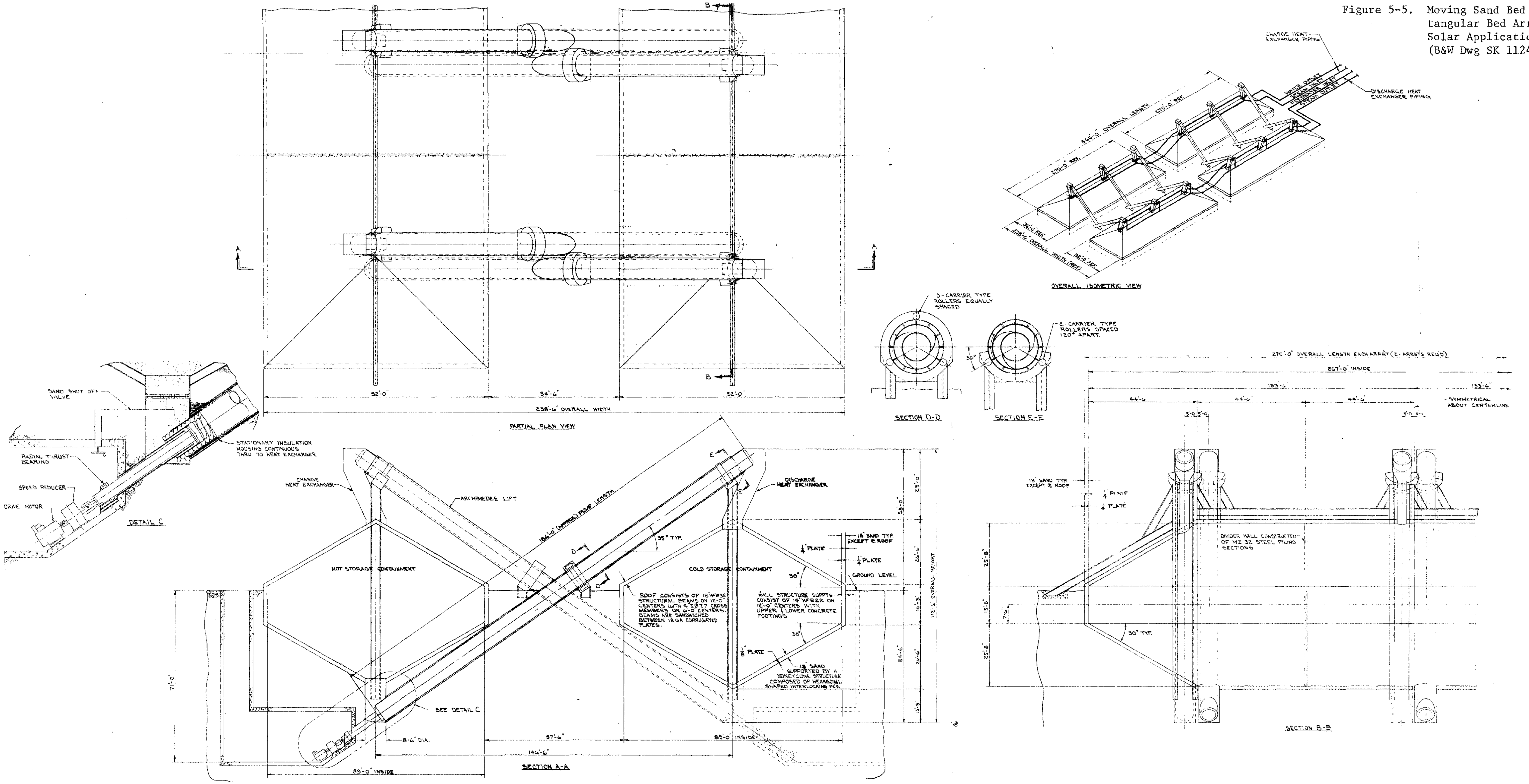
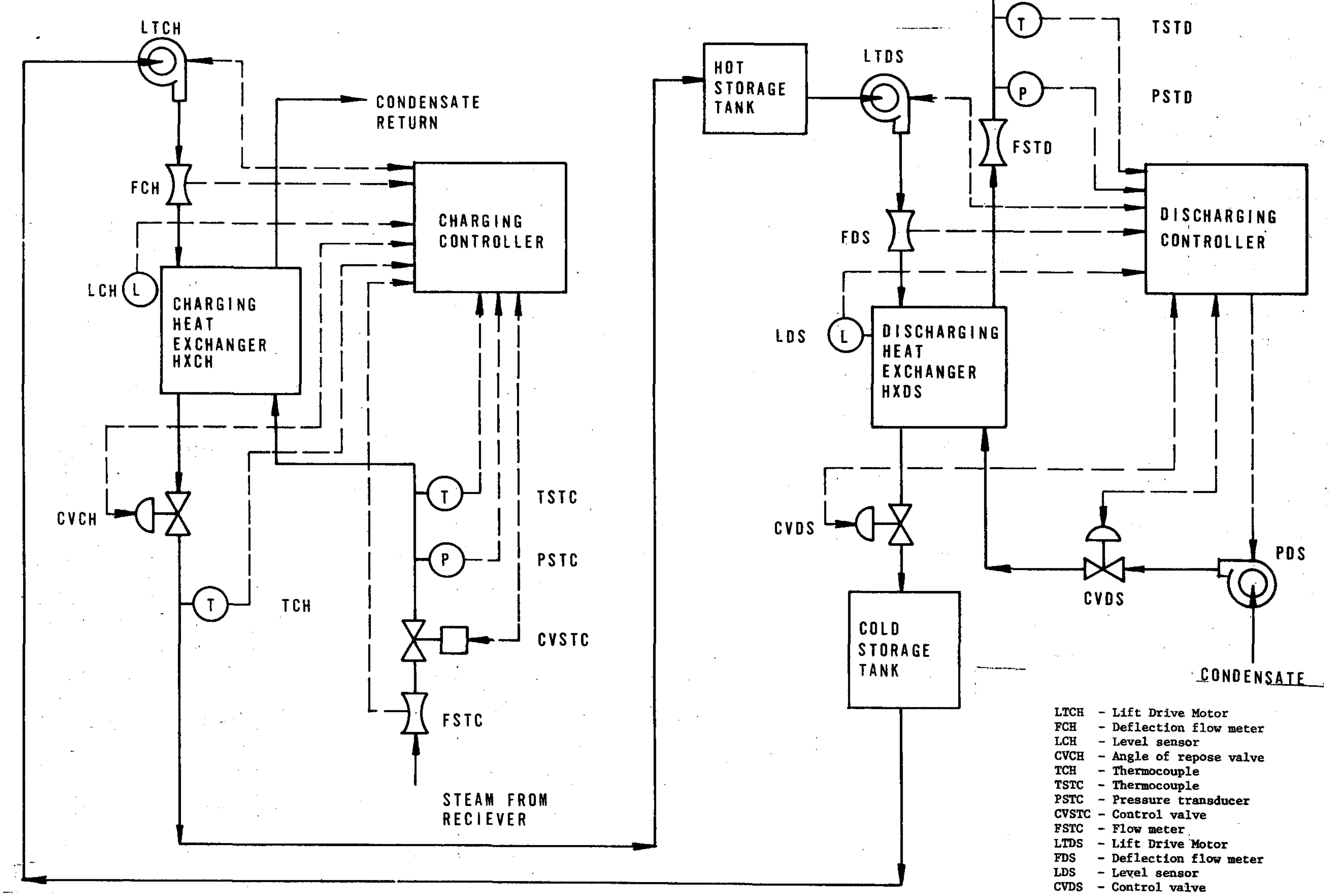


Figure 5-5. Moving Sand Bed TESS With Rectangular Bed Arrangement for Solar Application (B&W Dwg SK 112480F-3)

Figure 5-6. Controls Schematic



LTCH - Lift Drive Motor
FCH - Deflection flow meter
LCH - Level sensor
CVCH - Angle of repose valve
TCH - Thermocouple
TSTC - Thermocouple
PSTC - Pressure transducer
CVSTC - Control valve
FSTC - Flow meter
LTDS - Lift Drive Motor
FDS - Deflection flow meter
LDS - Level sensor
CVDS - Control valve
FSTD - Flow meter
TSTD - Thermocouple
PSTD - Pressure transducer
CVDS - Angle of repose valve
PDS - Condensate pump

6. ASSESSMENT OF COMMERCIAL-SCALE TESS

The MBTESS conceptual design was assessed for future improvements that could potentially reduce the cost of the system. In addition, potential limitations were defined.

6.1. Potential Improvements

6.1.1. Performance

System performance was evaluated in terms of round trip efficiency, η_{RT}, as defined in section 4.2.2, where

$$\eta_{RT} = \frac{E_{out} \cdot \eta_{TESS}}{E_{in} \cdot \eta_{solar}}$$

E_{out} is the required thermal energy output. E_{in} includes E_{out} plus the system heat losses and parastic power required for the lifts. The quantity, η_{TESS}, is the thermal to electric conversion efficiency when utilizing steam from storage. The quantity, η_{solar}, is the thermal to electric conversion efficiency when using steam solely from the receiver. Therefore, η_{RT} can only be increased by reducing E_{in} or increasing η_{TESS}. Since E_{out} is normally fixed, reducing η_{solar} is counterproductive in terms of overall plant efficiency. Reducing the heat losses from the tanks and lifts and reducing parasitic power loss to the lifts will reduce E_{in}.

Further analysis is required to determine whether the cost of additional insulation to reduce thermal losses would be justified. Improvements in the efficiency of the method of transporting sand between storage and the heat exchangers would also be beneficial. The use of steam turbine-driven lifts or bucket elevators, provided the technology is available for these elevators to operate above 177C (350F), should be investigated.

6.1.2. Cost Reduction

Refinements to the conceptual design would probably reduce the life cycle cost of the thermal energy storage subsystem. This suggests the need for a design-costing algorithm in which the plant design could be improved to arrive at the lowest life cycle cost. The development of such an algorithm is presented in section 8 of this report.

Raising the pressure of the receiver working fluid, a steeper temperature gradient is allowed. The larger temperature gradient, in conjunction with the same amount of thermal energy stored, would reduce the required volume of the tanks and the lift capacity. Therefore, a reduction in system costs should be realized and the turnaround efficiency improved. This would need to be balanced against the increased cost of the receiver and associated piping due to higher-pressure operation.

6.1.3. Economics of Scale

The reference storage system design described in section 5 is for a nominal 100-MWe plant. Moving bed storage subsystems for other power levels and capacities were not designed. Therefore, effects of scale size on subsystem costs cannot be addressed at this time.

6.2. Potential Limitations

6.2.1. Status of Material and Heat Exchanger Technology

The heat storage medium (sand) in a circulating system at temperature may require special equipment and/or operating parameters to control system wear and particle attrition. Particle size has been held to below 100 microns, and bed velocities are 0.15-0.30 m/s (0.5-1.0 fps) over the heat exchanger tube surface and 3.0 m/s (10 fps) or less in the Archimedes lifts (relative sliding velocity)[7]. These constraints have been placed on the system based on experience with fluidized bed heat transfer and light-phase transport of fluid catalyst particles, in order to minimize system wear and particle attrition.

The technical concerns stated in section 5.5.3 apply. Heat transfer characteristics, particle characterization and heat exchanger configuration must be evaluated in an experimental program to ensure that the MBTESS is a viable ATESSS concept. This program must be accomplished early in the development phase; it is described in detail in section 8.

6.2.2. Commercial Availability of Storage Material, Equipment, and Instrumentation

The TES material used in the reference design is SiO_2 (sand) that has a nominal 30° angle of repose. This material is readily available and relatively inexpensive; however, purity requirements may have to be established.

The major components, i.e., tanks, lifts, and heat exchangers, are not off-the-shelf items, but no special technology should be required for their construction.

Instrumentation to measure sand level in the storage tanks and heat exchangers and sand flow instrumentation will have to be developed. Other instrumentation to measure temperatue and pressure and associated controllers are readily available.

6.2.3. Safety and Environmental Constraints

The MBTESS is expected to present no unique safety and/or environmental constraints.

In general, the system does not require site restrictions to protect the public since the heat storage/transport medium, sand is benign. Burn hazards and dust control are of concern; however, methods of control have been established and are well known.

6.2.4. Land Use Constraints

The MBTESS reference design was designed with the storage tanks partially buried. A location with a high water table would affect this design. If it is necessary to have the total system above ground, then re-evaluation of the plant cost would be required.

7. APPLICABILITY TO OTHER RECEIVER OPERATING CONDITIONS

The MBTESS reference design was assessed for its applicability for higher-temperature applications, such as process heat or power generation applications of solar energy. This assessment addressed the following conditions:

- Increase storage operating temperature from 332 to 538C (630 to 1000F). This condition is considered independently of the receiver working fluid, it is assumed that the receiver working fluid provides energy for 538C (1000F) storage. (A potential good match would be to use a sodium receiver to charge storage.)
- Increase water-cooled receiver conditions from 510C/10.1 MPa (950F/1465 psia) to 510C/12.5 MPa (950F/1815 psia) and 510C (950F) reheat.

7.1. MBTESS Utilizing 538C (1000F) Peak Medium Storage Temperature

Application of the MBTESS with a peak storage medium temperature of 538C (1000F) will provide improvements in performance and cost.

The increase in storage temperature ΔT ($T_{hot} - T_{cold}$) to 278C (500F) or greater will materially reduce the size and cost of the tanks, storage medium, lift pumping power, and heat loss. This system provides storage ΔT values (in a single stage) that are typical of current two-stage system designs. Avoiding two-stage storage will allow use of an improved discharge power cycle yielding improved round trip efficiencies and reduced size and cost of storage equipment and media.

Temperature diagrams for candidate power cycles are shown in Figures 7-1 and 7-2. Figure 7-1 illustrates a relatively simple 510C/10.1 MPa (950F/1465 psia) cycle without reheat operating in both the charge and discharge modes. Figure 7-2 shows a more advanced steam cycle with 510C/16.55 MPa (950F/2400 psia) and 510C (950F) reheat. Round trip efficiency for either cycle is estimated to be better than 90%.

The increase in storage operating temperature to 538C (1000F) will reduce the storage requirements approximately 61% for storage of 1796 MWh (6.13×10^9 Btu) as required for the MBTESS reference design. This reduces the number of lifts and their associated equipment. In addition, the turnaround efficiency will improve due to reduced pumping power and heat losses. The MBTESS reference design is adaptable to 538C (1000F) storage operation with minor modifications e.g., the following:

1. Storage tanks — The internal structure of the tank remains essentially unchanged; however, the main roof beams should be placed on the top side of the roof, thereby reducing the temperature gradient seen by the beams, and additional support will be required for the inner skin of the roof.

   The main roof beams should be run to a separate footing beyond the side walls of the tank structure, thus eliminating the lateral load footing on the side wall.

   Provide insulation cooling for the heat exchanger support column, or support the heat exchanger with external trusses on top of the roof.

   Provide active cooling for the storage tank bottom and its foundations.

2. Heat Exchangers — A reduction in the number of heat exchangers will be realized since the increase in storage ΔT will reduce the required storage volume for the same amount of energy by the reference design. Additionally, the heat exchanger will have to be constructed of type 304 stainless steel instead of carbon steel as used in the reference design.

3. Lifts — Operation at 538C (1000F) will require the lifts to be constructed of type 304 stainless steel. Operation at this temperature will reduce the number of lifts and their associated equipment by approximately 66%; however, the individual lift capacity will have to be increased by approximately 15%.

Increasing the medium storage temperature to 538C (1000F) has resulted in a cost decrement of $12 million in the reference design cost. Table 7-1 lists the estimated costs for such a system. Table 7-2 provides a breakdown of energy related, power related, and specific related costs for the high temperature MBTESS.

Storage ΔT can be increased with various combinations of discharge cycle conditions. If all the advantages are taken in order to reduce storage size and cost when considering a discharge cycle of 482C/3.22 MPa (900F/468 psia), then it is possible to attain a single-stage MBTESS with a storage ΔT of as large 347C (625F) as shown in Figure 7-3.

7.2. MBTESS Using 510C/12.5 MPa (950F/1815 psia) Receiver Conditions With 510C (950F) Reheat

Application of the MBTESS with improved receiver operating conditions has the potential for improving energy storage economics. The improved receiver operating conditions would allow the MBTESS to operate over a greater temperature range, i.e., 191 to 399C (375 to 750F). This temperature increase will reduce the storage requirements by 54% for storage of 1796 MWh (6.13×10^9 Btu) as required for the MBTESS reference design. A reduction in the number of lifts would be realized as well as a reduction in system heat losses.

This design utilizes a temperature versus heat transferred diagram presented in Figure 7-4. A schematic for the turbine arrangement and heat exchangers is shown in Figures 7-5, 7-6a, and 7-6b, respectively.

This design would require replacement of the reheater surface in the solar receiver by additional superheater surface. High-pressure steam [1.32×10^5 kg/h (290,640 lb/h)] circulated through this additional surface augments the 3.07×10^5 kg/h (675,970 lb/h) to push the charge pinchpoint to the left (Figure 7-4), thus allowing a greater storage ΔT. This extra steam (cooled to saturation) is circulated back to the superheater inlet by a steam circulator (Figure 7-6b). In addition, the 510C/3.06 MPa (950F/445 psia) reheat is replaced by 266C/1.03 MPa (510F/150 psia) reheat. This is accomplished by condensing saturated steam at 12.5 MPa (1815 psia) during charge. During the discharge cycle, reheat takes place in the moving bed heat exchanger complex as shown in Figure 7-6a.

On the basis of 3.1×10^5 Kg/h (685,800 lb/h) steam to the low-pressure (LP) turbine during discharge, the LP turbine inlet flow will be 4.36×10^5 Kg/h (960,000 lb/h) or about 40% higher. This extra flow could be diverted to a separate LP turbine to avoid the control problems associated with dual admission to both 1.03 MPa (150 psia) and 5.15 MPa (750 psia). However, if the dual admission control problems can be solved, a single LP turbine may be feasible since the amount of LP steam flow modulation above or below the average flow of 3.73×10^5 Kg/h (822,000 lb/h) (17%) is similar to that anticipated for peak load cycling power plants operating with or without large amounts of extracted steam to achieve the desired power modulation.

The high-pressure steam condensing reheater used during the charging mode of operation is similar in function to reheaters used in nuclear power plant steam cycles. Current problems associated with some designs of this equipment appear to be avoidable with proper charger modifications.

The steam compressor could be turbine- or motor-driven. In the latter case, such a pump would have a vertical-shaft, wet-rotor, water-cooled, two-pole induction motor with two water-lubricated bearings driving a multi-stage steam compressor having water-cooled outboard bearings. This unit would have no shaft seals. Special units of this type are reported to have been built and operated.[32]

The gross thermal efficiencies during charge and discharge were calculated to be 41.2% and 38.4% respectively. Therefore, improved performance and reduced cost can be realized with this option.

Table 7-1. Cost Estimate for 538C (1000F) Storage System

CLIENT SANDIA DESCRIPTION Estimated Cost of CONT. NO:
LOCATION MSB System if Media Temperature MADE BY:
PROJECT Moving Sand Bed TESS Cycles between 500°F & 1000°F APPROVED:

Account: 5700 Energy Storage System

	ITEM & DESCRIPTION	MANHOURS	ESTIMATED COST (June 1980 $) LABOR	SUBCONTRACTS	MATERIALS	TOTALS
	Site Preparation	8,663	191,000	105,000		269,000
	Sand Storage & Equipment Structures	40,281	672,000	281,000	688,000	1,641,000
	Sand Moving Equipment & Supports	18,271	316,000	384,000	1,842,000	2,542,000
	Heat Exchangers & Piping	4,886	88,000	76,000	2,433,000	2,597,000
	Aux. Equipment & Instrumentation	590	10,000	171,000	115,000	296,000
	Media and Media Processing	1,748	26,000	52,000	625,000	703,000
1	Direct Fiels Costs		1,303,000	1,069,000	5,703,000	8,075,000
2	Indirect Costs					699,000
3	Contractor's Profit (10% of 1+2)					877,000
4	Engineering (10% of (1+2 - 703,000)					807,000
	TOTAL					10,458,000

Table 7-2. High Temperature (538C Storage) MBTESS Energy-, Power-, and Specific-Related Costs (June 1980 $ × 10^3)

Energy related cost, C_s

Item	Direct field cost	Indirect cost	
Excavation	38	3	
Backfill and Compaction	258	49	
Foundations and Footings	36	4	
Storage Structures	1,352	245	
Insulation	141	41	
Medium	703	--	
Subtotal	2,528 +	342 =	$2,870
	Contractor's Profit		287
	Engineering		217
		Total	$3,374

Power related cost, C_p

Item	Direct field cost	Indirect cost	
Lifts	2,542	249	
Piping	423	10	
Heat Exchangers	2,174	57	
Aux. Equipment	181	16	
Controls and Inst.	115	--	
Equipment Covers	112	25	
Subtotal	5,547 +	357 =	$5,904
	Contractor's Profit		590
	Engineering		590
		Total	$7,084

$C_{s_e} = \$3,374/(70,000 \text{ KWe} \times 6) = \$8.03/\text{KWhe}$

$C_{s_t} = \$3,374/(285,000 \text{ KWe} \times 6) = \$1.97/\text{KWht}$

$C_{p_e} = \$7,084/70,000 \text{ KWe} = \$101.00/\text{KWe}$

$C_{p_t} = \$7,084/285,000 \text{ KWt} = \$24.86/\text{KWt}$

$$C_T = (C_p + (C_s \cdot h))/h = \$24.86/\text{KWhe}$$
$$= \$6.11/\text{KWht}$$

Figure 7-1. Temperature Diagram for High-Temperature, Single-Stage Storage Using 1450 psi, 950F Steam Cycle in Charge and Discharge Modes

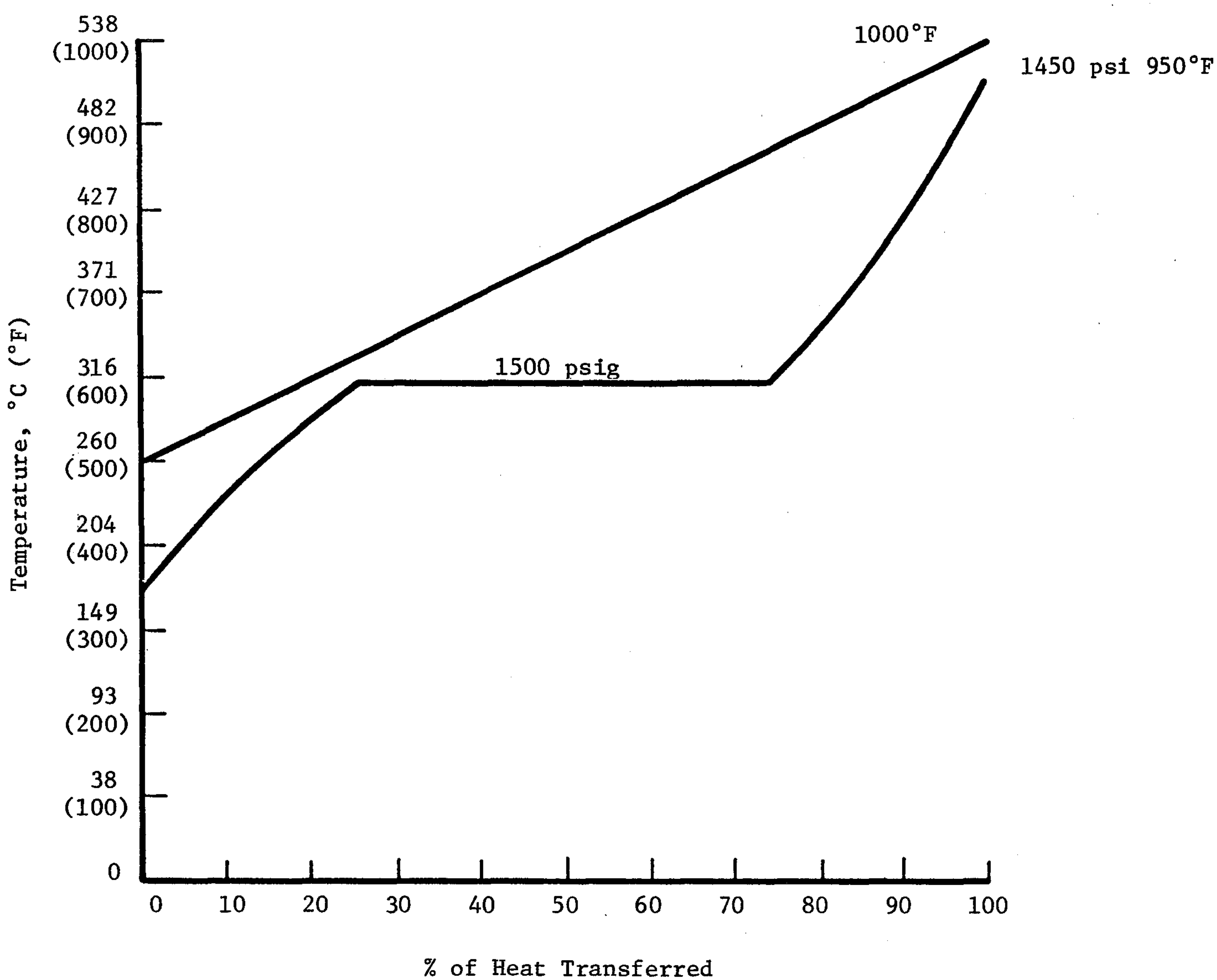


Figure 7-2. Special Temperature Diagram for Solar Plant Using High-Temperature Thermal Storage and Reheat Steam Cycle With Reduced Final Feed Temperature and Two-Stage Reheat in Charge and Discharge Modes

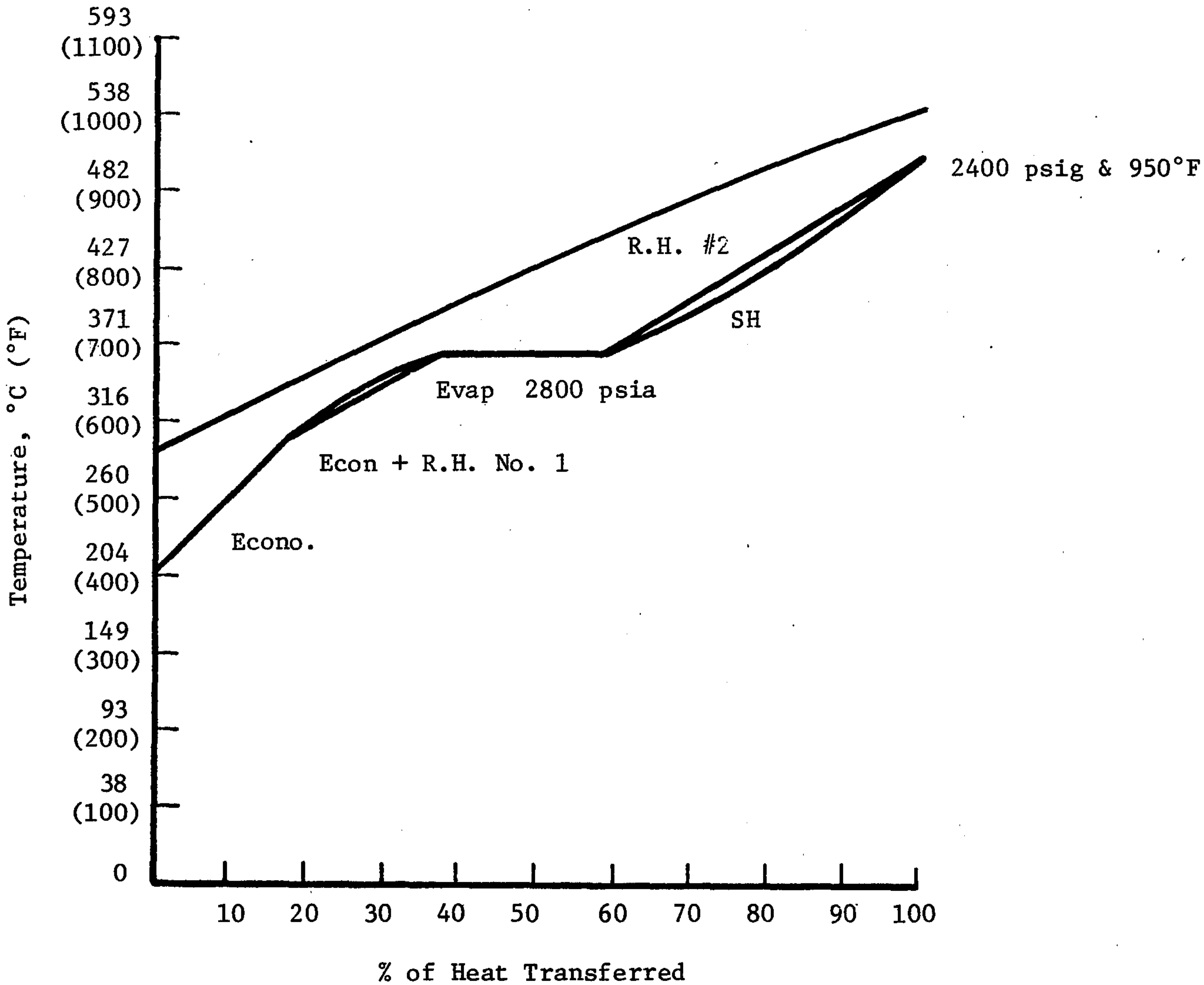


Figure 7-3. Temperature Diagram for Single-Stage Moving Bed Thermal Energy Storage, 468 psia, 900F Steam Cycle

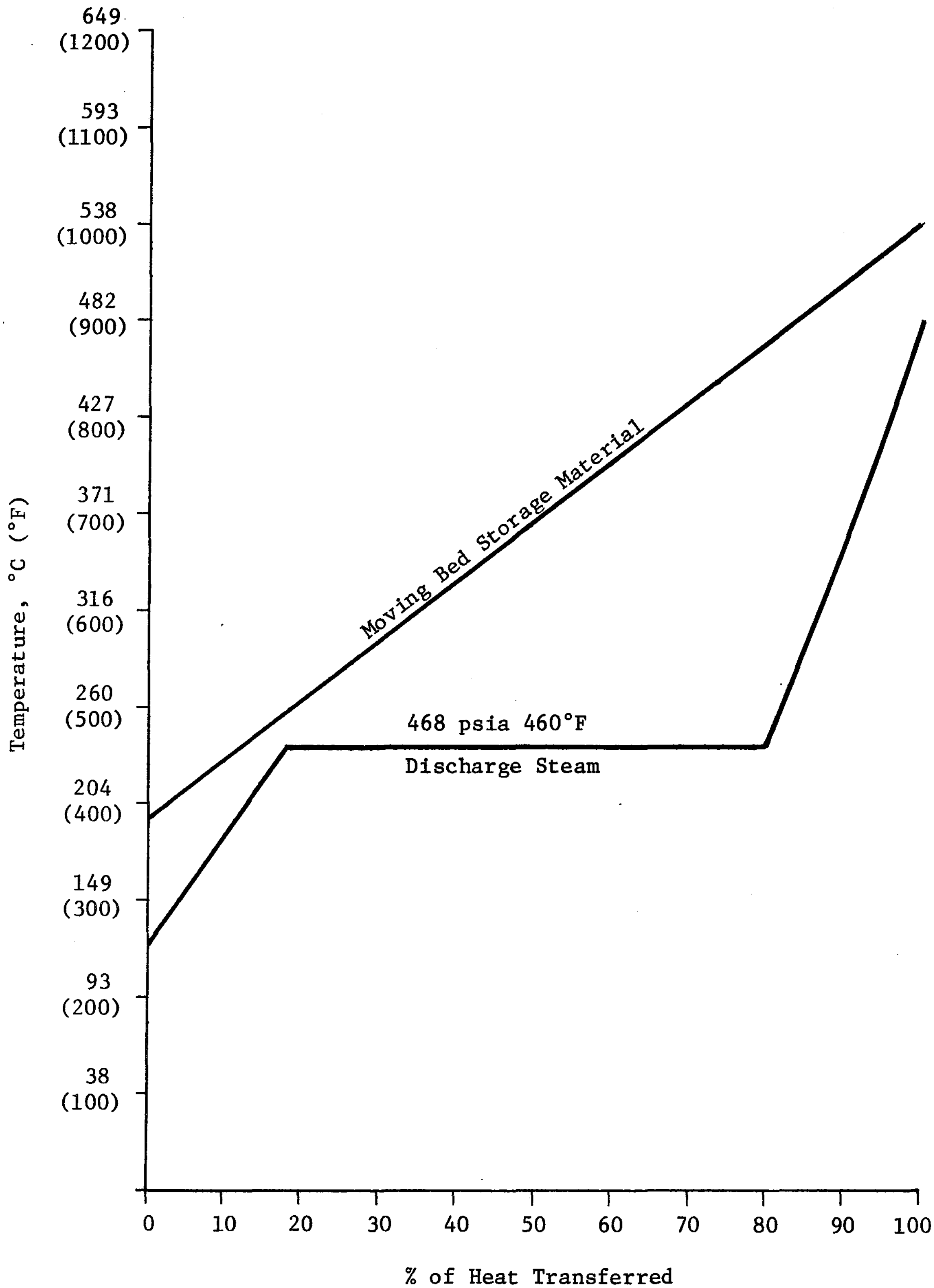


Figure 7-4. Charge/Discharge Temperature Diagram

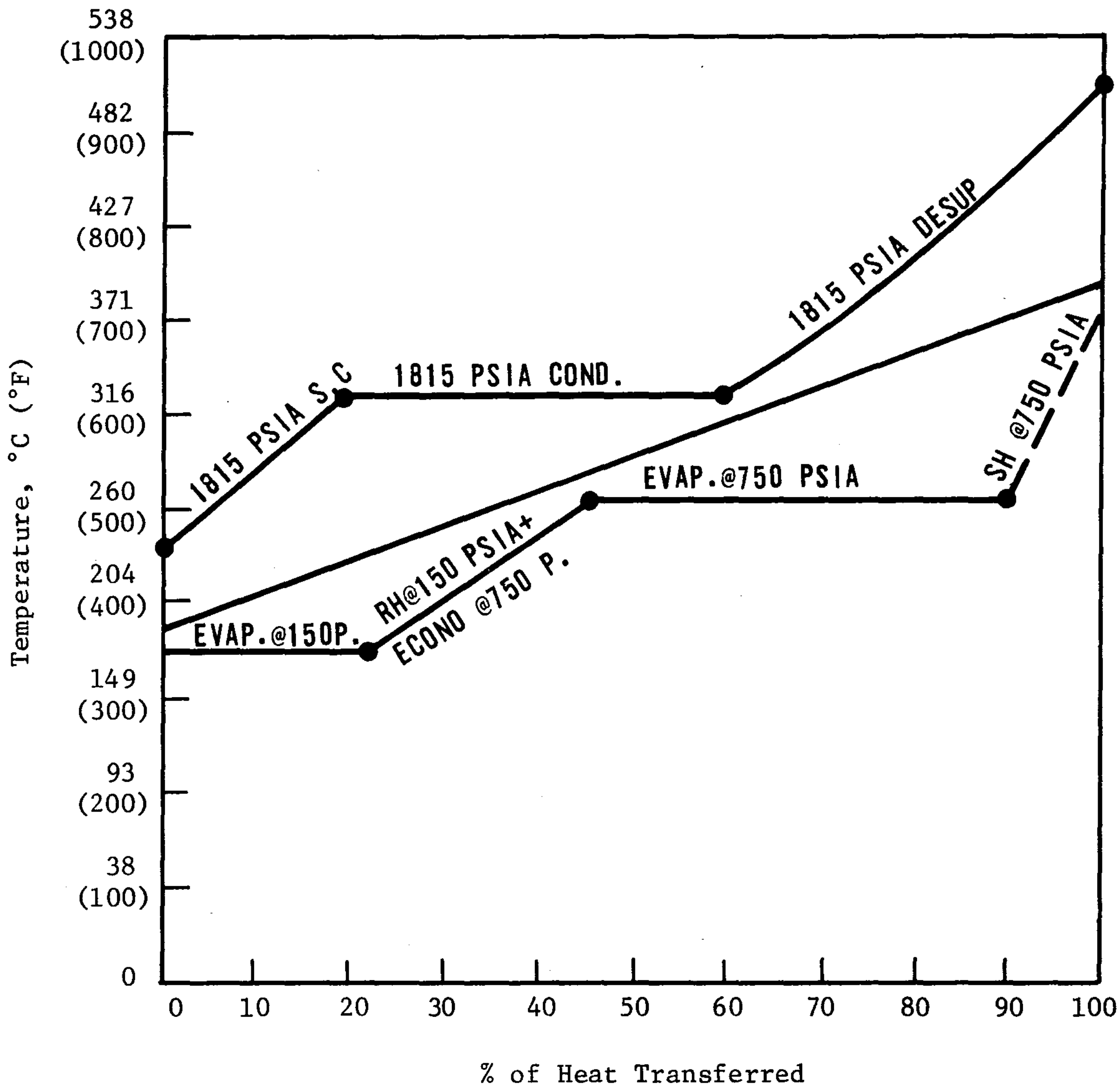


Figure 7-5. Turbine Flow Diagram

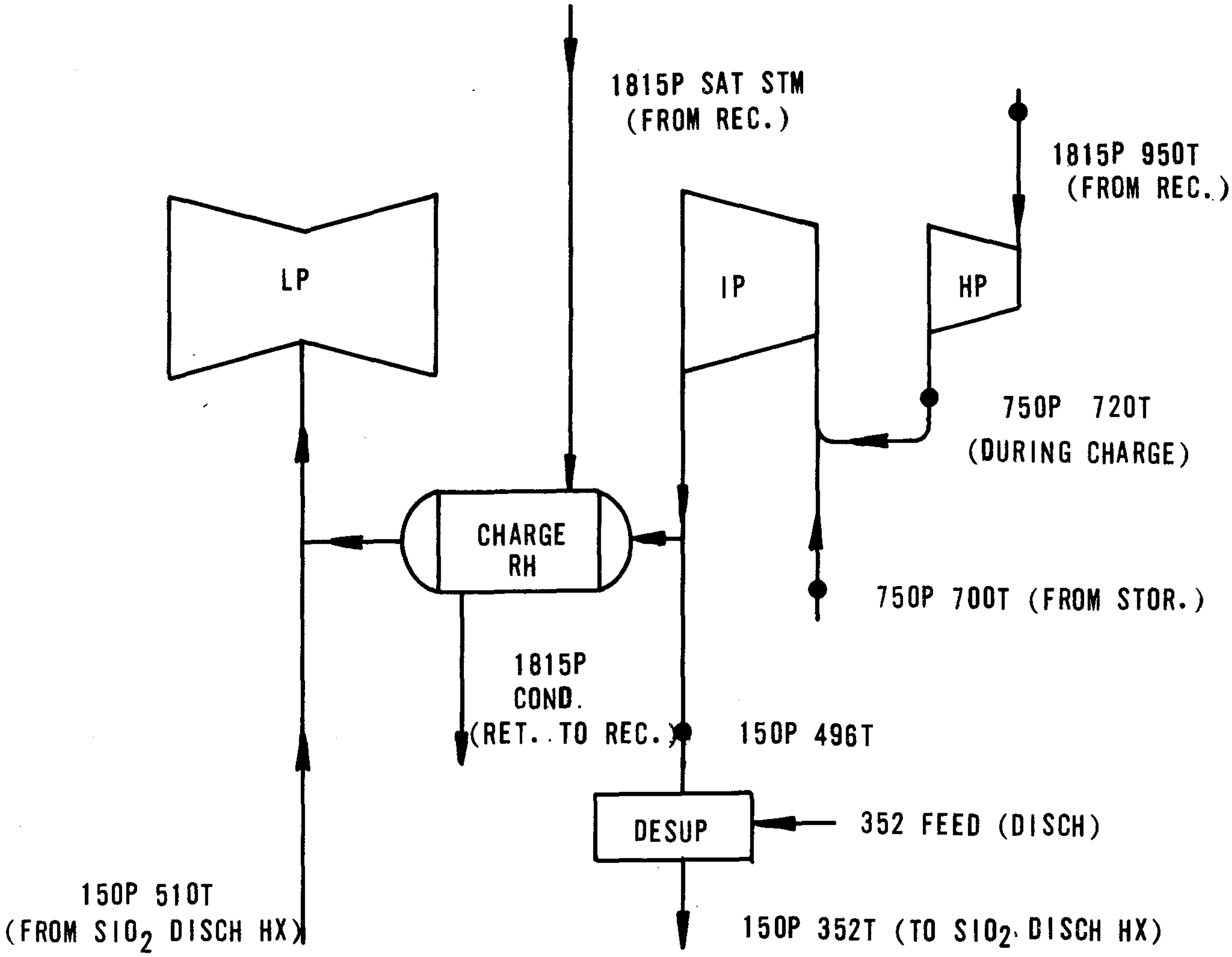


Figure 7-6a. Discharge Heat Exchanger Diagram

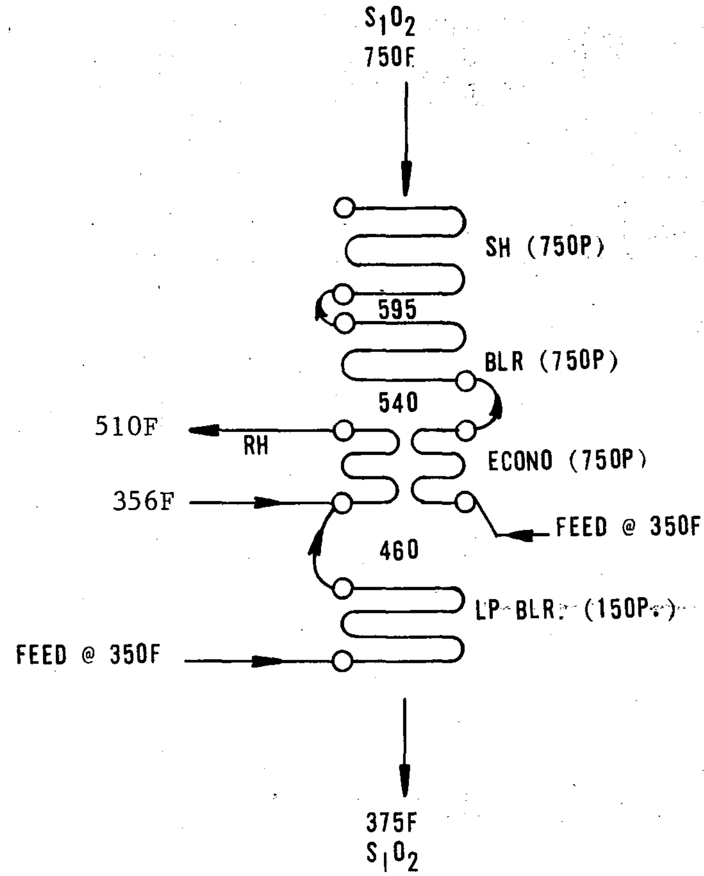
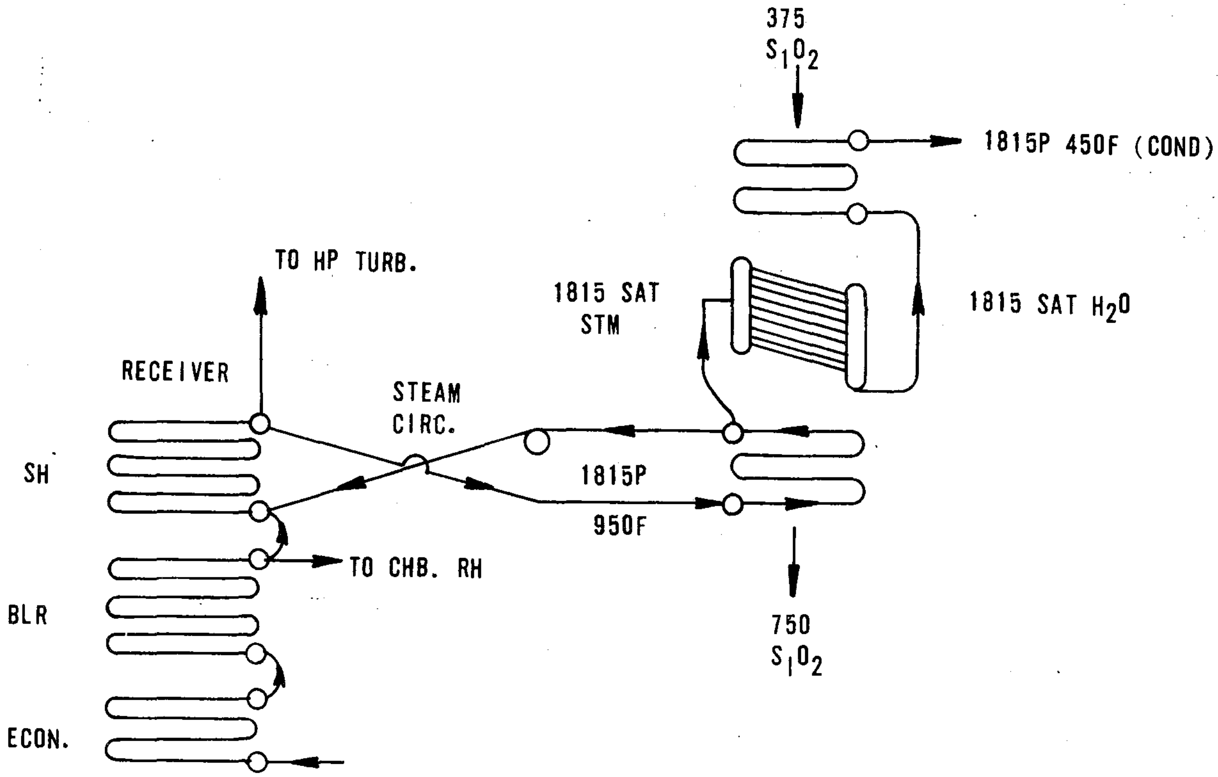


Figure 7-6b. Charge Heat Exchanger Diagram



8. DEVELOPMENT ACTIVITIES FOR CONCEPTUAL DESIGN

The objective of this task is to identify additional analyses and experiments required to bring the conceptual design of Task 4 to a state of readiness for final design and construction. These developmental programs must be designed to resolve any performance uncertainties that have surfaced during the Phase I design effort of the MBTESS. Uncertainties have been identified in the following areas:

- Bed material characterization
- Heat transfer and flow studies
- Lift development
- Other system components

8.1. Bed Material Characterization

Bed material selection, characterization, and property control are necessary to achieve useful results in all other phases of developmental activities. Work efforts in Phase I have identified three areas of development:

- A combination of bed material properties and equipment design must be developed to provide adequate flow of the dense bed material in the tank, transfer equipment, and heat exchanger to provide acceptable heat transfer efficiency.
- The flow characteristics and attrition behavior of candidate moving bed materials must be established and optimized.
- The thermal, mechanical, and chemical stability of naturally occuring and processed candidate materials must be determined under realistic conditions representative of this process.

An R&D program has been defined which would identify candidate materials for this application and characterize their behavior under simulated laboratory conditions. Activities would lead to the development of material recommendations for a heat transfer study and for first-generation design of the total moving bed system. A study would be conducted to (1) identify and select potential candidate materials along with cost estimates for bulk supply (2) characterize their flow properties using standardized tests and simulated operating conditions, (3) characterize thermal cycling effects up to 1100F on material flow properties and attrition, (4) determine the performance of selected materials using special tests for a measure of flowability, attrition, and wear, and (5) recommend materials for the heat transfer study. This program would provide information for the selection of optimum materials to be used in a moving bed TESS and would have an additional benefit of developing information that could contribute to the system design and operating criteria.

The following section outlines a five-task program to identify and ultimately specify a material as the heat transfer medium in a moving bed thermal energy storage system. Emphasis would be placed on identifying and selecting materials as soon as possible to support the heat transfer and other design activities. Existing or available procedures and equipment would be sought to help reduce the time and costs to do this work.

8.1.1. Task 1 — Identification and Test Procedures of Candidate Materials

Candidate materials would be identified and selection criteria developed through literature searches and contacts with material vendors and consultants. To provide a means of screening candidate materials, procedures will be identified to evaluate the following characteristics at temperatures up to 593.3C (1100F):

- Thermal properties
- Flowability and angle of repose
- Medium attrition
- Interaction of medium with structural components/wear
- Other physical properties affecting system performance

The last item would include such parameters as flowing bed density, tap density, particle size and morphology, mineralogy, chemistry, and physical stability after themal cycling bewteen ambient temperature and 593.3C (1100F).

Possible candidates would include silica sand, alumina, zircon sand, mine tailings, olivine, calcined clay, and glass beads. Other materials would be identified and a summary of available physical, chemical, and mechanical property data — as well as cost and availability — will be compiled. This should result in a number of candidate materials to be evaluated in Task 3.

8.1.2. Task 2 — Design and Construction of Test Equipment

Based on procedures identified in Task 1, test equipment would be designed and constructed to permit medium property evaluation at elevated temperatures and under variable conditions of humidity. Specialized tests would be needed to characterize flowability, medium attrition, wear of structural components, and medium density. If needed, equipment would be designed to simulate the conditions that are expected to exist during operation of the MBTESS.

8.1.3. Task 3 — Characterization of Materials

The materials selected would be characterized according to the procedures identified in Task 1 to permit the recommendation of preferred media candidates for the MBTESS. Tests would be conducted in a manner designed to minimize the overall effort and to provide timely information for concurrent heat transfer studies, i.e., some candidates may be eliminated based on thermal stability, flowability, and short-term attrition and wear tests.

Some characteristics of the media can be expected to change substantially with extended attrition and wear. For example, chemistry, particle size, particle morphology, and flow characteristics may be dependent on the extent of attrition of the media and the degree of contamination by wear of metallic structural components. Further, it may become apparent during these tests that some candidates are unsatisfactory in their commercially available form, but after benefication or other processing, they could represent promising materials. Should this situation become apparent, limited efforts would be necessary to evaluate laboratory processed materials.

The effective density (ρ), thermal conductivity (k), and specific heat (c) are key thermal properties of the bed materials. The volumetric heat storage (ρc) is a common criterion for judging sensible heat storage materials, while the thermal diffusivity ($k/\rho c$) is a measure of the dynamic energy transport potential. These properties are a function of the bed porosity and solid and gas constituents. In particular, the effective thermal conductivity (kc) includes the combined effects of radiation transport and conduction through the solid particles and interstitial gas. These transport mechanisms can be predicted by a fundamental heat transfer model of insulations. This type of model predicts the heat flow by conduction and radiation in solid, porous, multi-constituent media. The model could be a finite-difference formulation of one-dimensional, steady-state heat transfer, with separate models for conduction and radiation. For conduction, heat flow through the medium can be visualized through the electric analogy in which the conducted heat flow is a sum of components through the gas alone, through series gas-solid phase interactions and through the solid alone. Radiation heat transfer might be modeled with the basic radiative transport equation. This equation would account for the simultaneous multiple scattering, absorption, and remission of energy across porous solid interfaces. The following parameters would be key program inputs:

- Thermal conductivity of solid particles
- Particle size distribution and volume fraction of each constituent (for mixed beds)
- Bed porosity and nominal pore diameter
- Index of refraction of the solid constituent

Such a model could be used to study the effects of key parameters on the thermal properties of a particle bed.

Initially, in concert with the bed material characterization program described earlier, 10 candidate bed materials would be screened to find two for detailed heat transfer and flow testing. The initial screening will be done by comparing the thermal properties mentioned above. These might be obtained by one or a combination of the following methods:

- Literature search
- Direct measurement via the hot wire technique
- Predictions via the heat transfer model

Some property data should be available in the literature; however, some would have to be obtained by predictions or measured using the hot wire technique described in Appendix E.

Selected bed materials would be measured for thermal properties via the hot wire technique. All bed parameters (particle size, porosity, etc.) would be identical to those selected for detailed heat transfer and flow testing.

The transient hot wire method (Appendix E) could be used for measuring the thermal conductivity and diffisivity of the particle bed. The principle advantages of this technique are as follows:

- Both thermal conductivity and thermal diffusivity can be obtained from the same data. Other techniques require separate heat transfer experiments to obtain these thermal properties. The errors are also reduced with this simple technique. Specific heat is simply obtained from the measured diffusivity and conductivity with an independently measured density.

- This is a macroscopic technique for determining total effective thermal properties. The transient hot wire method will measure the average thermal conductivity and diffusivity of a relatively large bed specimen.

- Provides essentially isothermal thermal property measurements. The wire temperature only increases approximately 16C (30F) above the average bed temperature.

- The technique is conceptually simple and requires standard laboratory equipment for setup.

This technique will be used to test specimens at mean temperatures that cover the bed operating temperature range. This information would be used to develop material and operating specifications, provide a ranking of candidate materials, and identify materials for heat transfer studies.

8.1.4. Task 4 — Support for Heat Transfer Study

Two different materials would be recommended for the heat transfer studies. This would involve technical, economic and commercial considerations. Support would be provided for the heat transfer study as needed on a continuing basis. Depending on the results of this study, additional testing may be identified for Task 3.

8.1.5. Task 5 — Recommendations for Further Work

The tests performed in Task 3 would use specially designed laboratory equipment and therefore may not necessarily simulate the conditions of the actual full-scale system. Thus, it might be desirable to define further tests that take into account the effects of larger volumes of material and the resulting high pressures. The data generated in Task 3 would be assessed and a recommendation will be made regarding the need for further work.

8.2. Heat Transfer and Flow Studies

In an MBTESS, heat is stored in a bed of particulate material. Heat is transferred to or from the material by allowing it to flow by gravity over a heated or cooled bank of tubes. The proposed heat exchanger configuration consists of an inclined bank of tubes arranged in a staggered pitch. The tubes will be inclined at an angle equal to or greater than the angle of repose of the bed material in order to prevent a stationary layer of particles from developing on top of the tubes.

The bed material will be selected through a separate program outlined in the previous section. This program should narrow the number of candidate bed materials to two. These materials would then be used in the flow and heat transfer tests. Before the moving bed thermal storage concept can be confirmed, the following information must be obtained:

- Gravity flow of particulate matter over tube banks must be better understood to facilitate design of the flow distribution and flow control devices for the moving bed.

- No heat transfer data exist for flows of moving beds over banks of tubes. Therefore, reliable data for the particulate side heat transfer coefficient is needed before the heat exchanger design can be finalized.

The recommended test program is described in the following subsections.

8.2.1. Task 1 — Design and Construction of Test Equipment

In this task the test apparatus for the flow and heat transfer studies will be designed and constructed. More detailed descriptions of the test apparatus and arrangement are given in the sections that follow.

8.2.2. Task 2 — Flow Studies

The flow tests would be completed before any heat transfer tests are performed. The objectives of the flow tests are as follows:

- To determine the effect of the tube bank geometry (tube pitch, spacing, etc.) on particulate flow rate and flow distribution.

- To determine the feasibility of using flow distribution and control devices at the inlet and/or outlet of the tube bank to maintain even particle distribution and a given particle flow rate.

Test Apparatus and Parameters

A schematic of the flow test apparatus is shown in Figure 8-1. The apparatus could be constructed of a clear plastic material, allowing observation of the flow patterns through the tube bank. A particle hopper with dampers will be positioned above the test section in order to maintain a steady flow of particles over the tube bank. A segmented box placed at the outlet of the test section could determine the flow distribution at the exit of the tube bank. To determine the particulate flow rate, the weight of the bed material falling into the test section per unit time could be measured. From the literature, it has been shown that gravity flow of particulate matter through a restriction is a function of the geometry of the restriction and the particulate properties (bulk density, angle of repose, etc.).

Referring to Figure 8-1, two inlet configurations will be tested. For Configuration A, the particle flow will be distributed across the complete inlet of the test apparatus. For Configuration B, the particle flow will be introduced into one side of the test apparatus. The latter configurations would be tested to determine the effects of nonuniform inlet flow on the flow rate and flow distribution through the tube bank.

The ranges of test parameters for the flow studies are given below.

Test Parameters for Flow Studies

Parameter	Range
Bed material*	Two candidates
Material size, distribution*	Two per material (start and end of life)
Bed material temperature	Ambient
Inclination of tube bank	Three angles: angle of repose, +5°, +10°
Tube pitch/diameter ratio	1.24, 1.5, 2.0
Tube diameter (OD), in.	0.75
Number of tube rows	5, 10, 20
Range of bed material flow velocity, fps	0.25 to 2.0

*Based on selection and results of the bed material characterization study.

8.2.3. Task 3 — Heat Transfer Studies

The optimum tube configurations (inclination and tube pitch/diameter ratio) that result in the most uniform particle flow distribution would be chosen from the flow tests and used in the heat transfer tests. The objective of the heat transfer tests is to determine the heat transfer coefficient for two candidate bed materials flowing over a fixed tube geometry. Because of the large number of test parameters, it is assumed that the optimum tube configuration would result from the flow studies presented in the previous section.

Test Apparatus and Parameters

This test would be run in the steady-state, once-through mode. Figure 8-2 is a schematic of the heat transfer test rig. A measured flow of particulate material enters the top of the rig from a particle hopper. The inlet particle temperature is measured by a grid of thermocouples at the inlet. The particles then flow through an electrically heated tube bank. The temperature of the particles is measured again at the tube bank exit by another grid of thermocouples. The heated particles are then discharged into a storage container near the exit of the tube bank.

The power input to each tube row will be independently controlled. Thermocouples mounted around the circumference of one or two tubes in each tube row would determine an average tube wall temperature for that row. These temperature measurements could be used in calculations to determine a local heat transfer coefficient for each tube row as well as an average coefficient for the entire heat exchanger.

The laminar analogy for single-component fluid flow in the literature suggests that the heat transfer coefficient would be higher for the first tube row and that it would reach a lower asympotic value about 10 tube rows down through the heat exchanger.

The ranges of test parameters for the heat transfer studies are given on the following page.

Test Parameters for Heat Transfer Studies

Parameter	Range
Bed material*	Two candidates
Material size, distribution*	Two per material (start and end of life)
Bed material temperature at inlet	Ambient, +37.7C (+100F)
Inclination of tube bank Tube pitch/diameter ratio	Best configuration based on results of flow studies
Tube diameter, in.	0.75
Number of tube rows	12
Range of bed material fluid velocity, fps	0.25 to 2.0

*Based on the selection and results of the bed material characterization study.

8.3. Lift Development

Three areas of development have been identified from the Phase I conceptual lift design effort:

- Optimization of the design configuration
- Wear
- Loads

A program to answer these questions would follow the material characterization and heat transfer and flow studies mentioned previously, making use of the data derived from these programs.

8.3.1. Optimization of Design Configuration

Additional design work is required to optimize the design configuration of the lift. An in-depth analysis of geometric parameters would provide for a design optimization that would yield the highest capacity lift possible, one of the major requirements for the MBTESS. Typical geometric parameters to be considered are as follows:

- Number of flights
- Pitch to diameter ratio, P/OD

- Inside to outside diameter ratio, ID/OD
- Helix angles

This optimization could also reduce the starting torque, operational torque, and horsepower requirements of the lift, thus improving the turnaround efficiency and providing lower cost at the same time.

8.3.2. Wear

The wear data from the material characterization test would be factored into the design of the lift. These data would set limits or guidelines in the following areas:

- Lift speed
- Liners to retard wear
- Sacrificial liners
- Maintenance schedules

Since the lift is the most active of the MBTESS components, wear should be a major consideration in its final design.

8.3.3. Loads

Additional design work is required to calculate the deadweight, thermal, seismic, normal, and off-normal operational loads for the lift. The manner in which these loads are transferred to the bearings must also be determined. A finite-element model for determining the various loads on the lift should be developed. This model should consider the supports, bearings, and lift drives, as well as the completes internal structure of the lift itself.

8.4. Other System Components

Additional design work is required for various system components, such as instrumentation, lift drives, valves, lift bearings and seals, heat exchangers, etc. To facilitate this detailed design effort, the development of a dynamic simulation model of the MBTESS is desirable. This model will allow for parameterization studies, component sizing, further development of a control philosophy, and dynamic response studies to be completed on the MBTESS. The model will be developed in a modular fashion, so that individual component models can be easily updated when new data are available from the laboratory tests (material characterization and heat transfer and flow studies). This simulation tool will be valuable in designing a subsystem research experiment (SRE) facility whose development would follow the major programs outlined thus far. A dynamic simulation would provide pretest predictions and would allow for identification of key focus areas in the design of such a facility.

8.5. Conclusion

A diagram of the development programs required to bring the conceptual design of Phase I to a state of readiness for design and construction is shown in Figure 8-3. The programs outlined in this test have described the developmental activities required for the time span covering the near term to the design of the SRE facility. Considerable attention should be given to the development and design of the SRE facility to make the transition from SRE to pilot plant a logical extension.

Figure 8-1. Flow Test Apparatus, Conceptual Arrangement

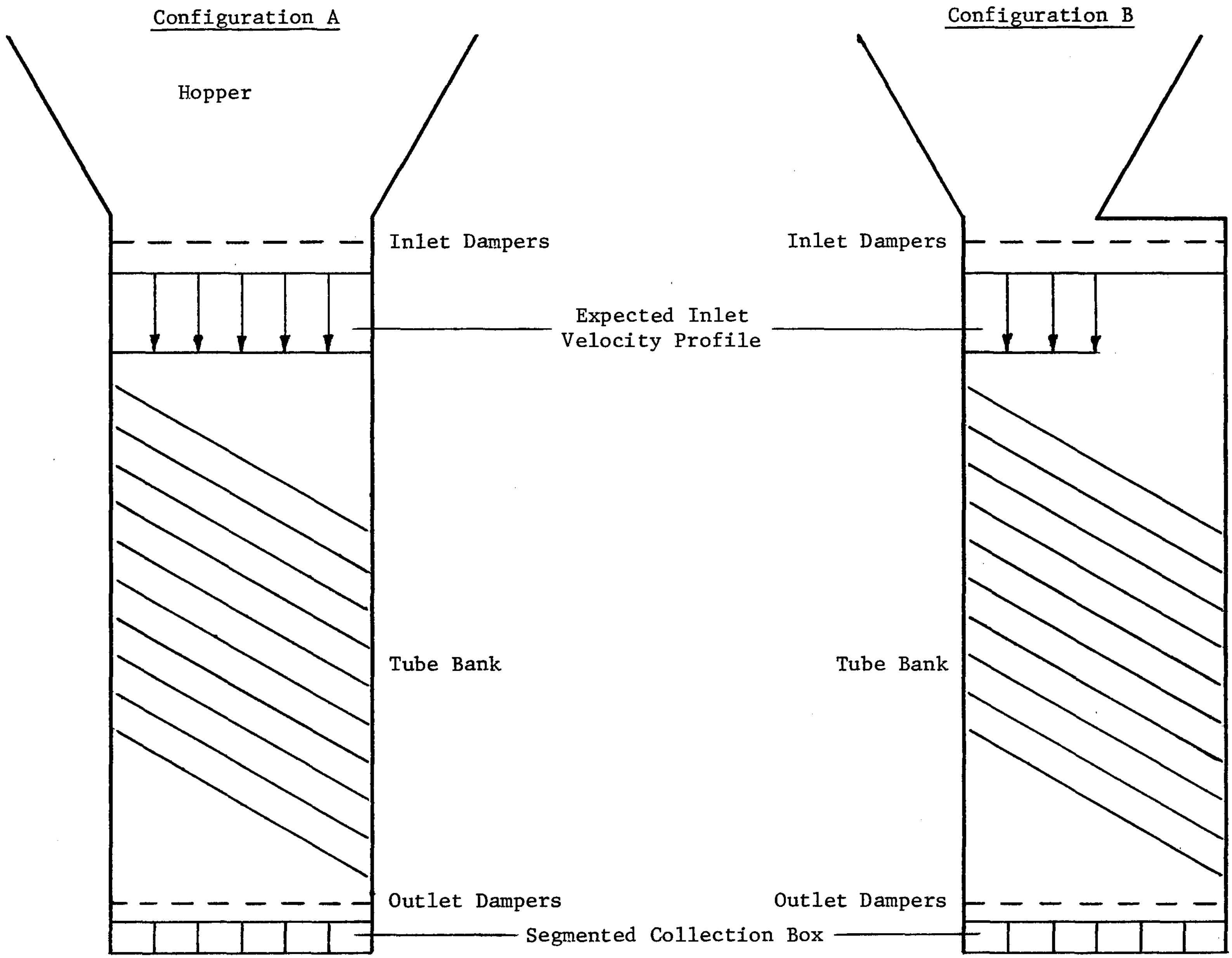


Figure 8-2. Heat Transfer Test Apparatus, Conceptual Arrangement

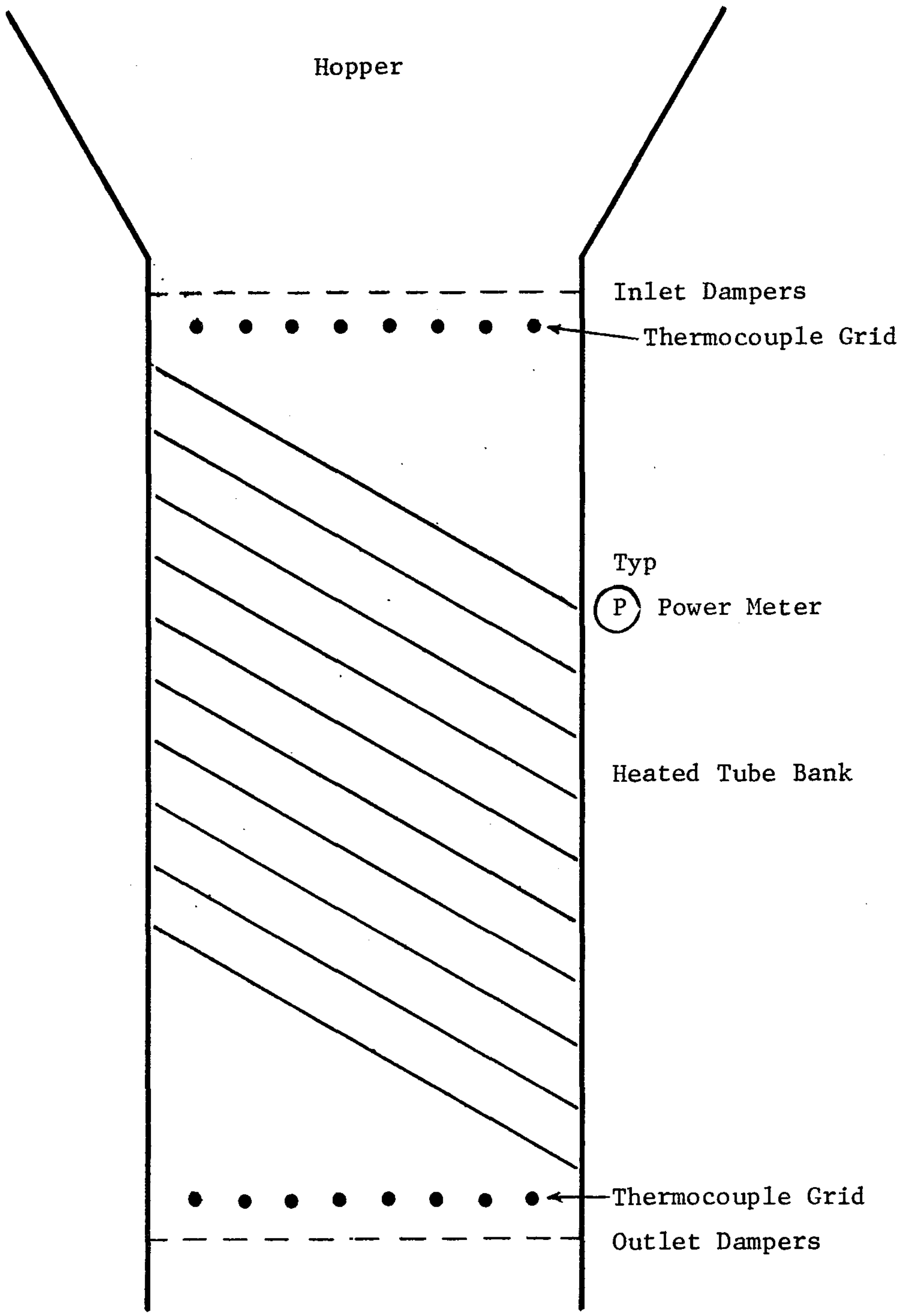
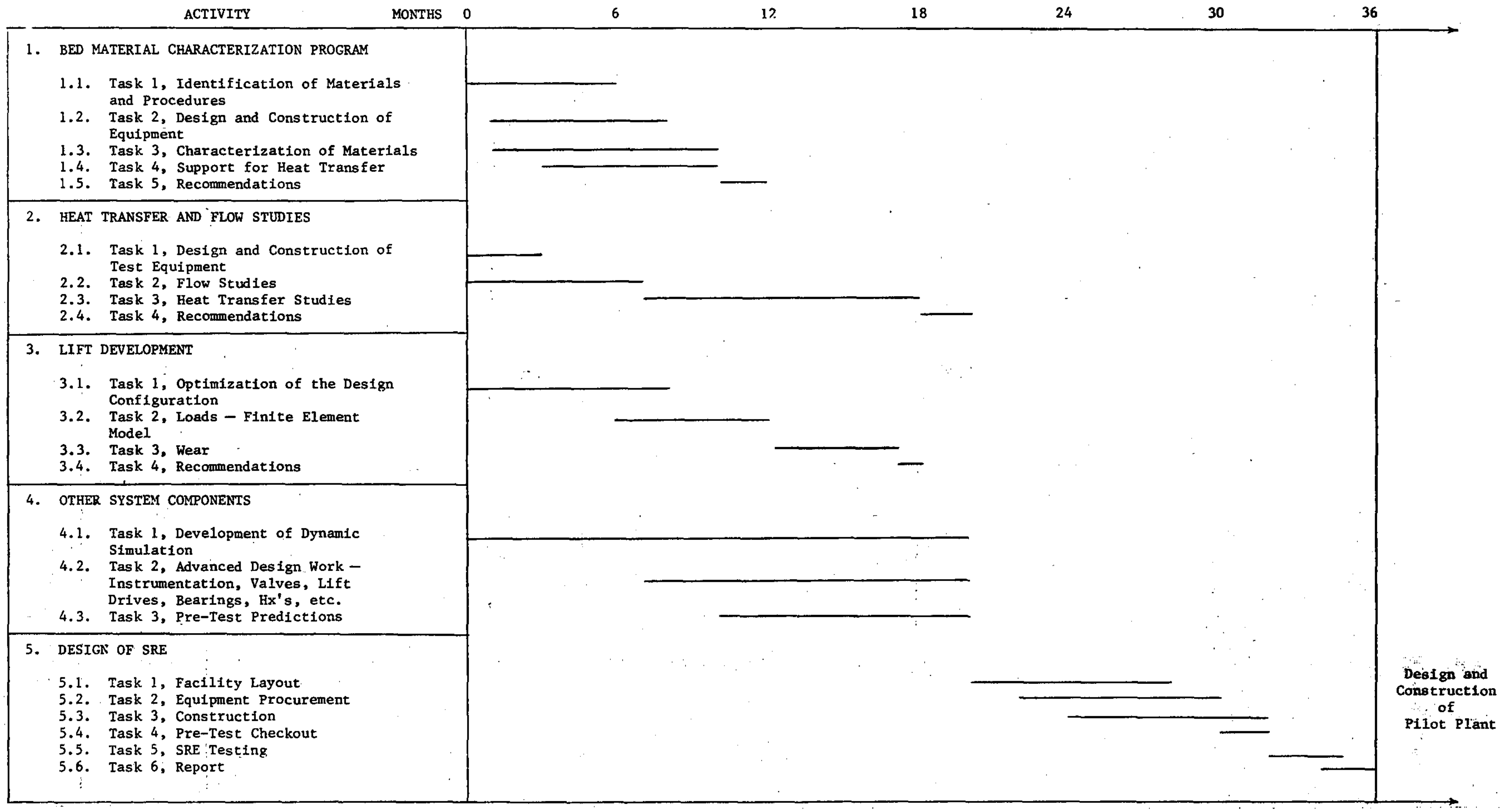


Figure 8-3. Diagram of Development Activities



9. CONCLUSIONS AND RECOMMENDATIONS

This section reviews and discusses the results of the study and includes an analysis of project results in relation to the goals of the thermal storage program. The objective of the study was to produce a commercial-scale thermal energy storage conceptual design offering cost/performance advantages over the oil/rock thermocline concept selected for the Barstow pilot plant. A further goal was to assess this storage concept for operation at temperatures that would permit generation of steam pressures and temperatures at or close to the conditions typical of modern-day power plants.

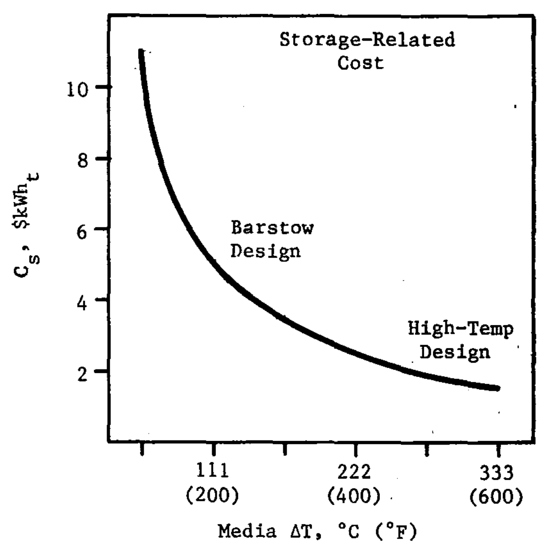
The following paragraphs review and analyze the characteristics of the moving sand bed system as applied to water/steam receivers using Barstow working fluid conditions and also as applied to higher working fluid conditions which would permit storage temperatures up to 538C (1000F).

At Barstow working fluid conditions, the moving bed TESS offers the advantages of chemically inert, low-cost storage media that is not subject to phase change (freezeup) temperature limitations. However, the system would be superior to an oil/rock system only if the thermal decomposition rate of the storage oil should prove to be unacceptable or if the oil costs should escalate beyond the projected range.

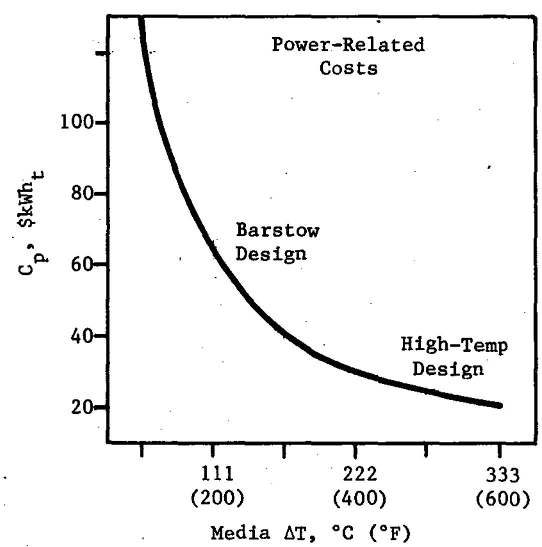
The moving bed TESS at Barstow working fluid conditions has an installed capital cost approximately 24% higher than the first generation goal for the Barstow oil/rock system shown in Table 9-1. The moving bed medium costs $1.8 million, or 6.9% of the total cost. The estimated media cost for the oil/rock system is $4.8 million or 23% of total costs, largely because of the high cost of oil. The oil/rock system utilizes relatively low cost equipment and components, while the low cost of the moving bed media is offset by the high cost of equipment and components necessary to handle it. Evaluation of round trip efficiencies led to the conclusion that no indirect storage system is likely to achieve an efficiency very much greater than that projected for the oil/rock/system. At the specified receiver working fluid conditions and specified steam cycle conditions, the oil/rock system is projected to have a round trip efficiency of 73.2%.

An ideal system with no heat losses and no parasitic losses would have a round trip efficiency of 77%. At the Barstow working fluid conditions the moving bed system appears to have a round trip efficiency about 3% less than that of the oil/rock system, or about 70.0%. The conclusion is that at Barstow working fluid conditions the moving bed concept could provide a backup to the oil/rock system, but it is not a clearly superior concept in this application.

At higher working fluid conditions beyond the temperature limitations of an oil/rock system, the moving sand bed concept provides much improved cost and performance characteristics. The concept developed in this study appears to be applicable with only minor modifications up to storage temperatures of 538C (1000F). Substantially less storage material and tankage are required at these storage temperatures due to the greater stored energy density. Savings in costs of power-related equipment also result from the lower material flow rates of the storage system. The sketches shown below illustrate the trend of cost with storage temperature for both power-related and storage-related equipment. Costs could tend to increase above 538C (1000F) because of the need to use structural materials capable of high-temperature operation. Development requirements for the system are considered independent of temperature up to approximately 538C (1000F). Beyond this temperature, requirements would tend to increase because of the need for higher-temperature materials.



$C_s = 1434(\Delta T)^{-1.06}$



$C_p = 14{,}960(\Delta T)^{-1.03}$

In conclusion, the moving bed concept appears to offer greatly improved economics at high storage temperatures and because of the lack of phase change of the storage material. The concept is also applicable to systems requiring storage temperatures as low as 111C (200F).

Table 9-1 compares project results to the thermal energy storage performance and cost goals. Because the charge and discharge steam conditions were fixed at the Barstow Commercial conditions, the first generation oil/rock storage goals should be used for comparison. At the Barstow working fluid conditions, the moving bed system appears to provide round trip efficiencies comparable to the oil/rock system and capital costs that are slightly greater than the oil/rock system costs. This relationship is expected to hold unless the costs of replacement oil should escalate beyond the projected range.

For high-temperature applications the moving sand bed concept shows very attractive economics, significantly below the second-generation storage cost goals for liquid metal and air receivers and with round trip efficiencies of approximately 90%.

Other factors considered in the study are discussed below. The specifications for development of the thermal storage concept stated that the concept be designed to be operable following an earthquake. This capability has been included in the design and results in some increase in equipment costs. However, the absence of hazard to the public from failure of storage equipment components would indicate that the elimination of seismic design requirements could be considered, particularly for locations with low seismic activity.

The operating and maintenance costs established for the moving sand bed concept are based on estimated rates of component wear and bed material attrition. This represents an uncertainty in the evaluation, which could only be resolved through a test program. Likewise, heat exchanger performance is based on laboratory measurements reported in the literature, limited analytical work, and observation of the behavior of plastic flow models built by B&W. The projected heat exchanger performance and sand flow behavior are uncertainties in the design that could affect equipment costs. A third issue that must be considered is the selection of sand bed material, which must also be established by a test program. A range of material costs have been considered in the evaluation, but the cost of the bed material remains uncertain.

The overall conclusion is that a thermal energy storage concept has been developed which is capable of operation over a wide temperature range and is compatible with all major receiver working fluids. The system has single-stage storage capability, a capability to operate at both high and low temperatures — 538 and 93C (1000 and 200F) — providing a high energy-per-unit volume and the absence of considerations related to media phase change.

The concept offers much improved economics for high-temperature applications, the system can provide a backup to the oil/rock concept. Uncertainties in the design which have been discussed previously should logically be resolved through a test program beginning with laboratory experiments to establish the behavior of bed material in system components, the most important of which are considered to be the heat exchangers.

Table 9-1. Thermal Energy Storage Performance and Cost Goal Summary (1980 dollars)[a]

		Round trip efficiency, %			Capital cost, $/kWh		
Application[b]	Solar interface	First generation	MBTESS	Second generation	First generation	MBTESS	Second generation
Barstow	Water/steam collector/receiver	70	70	80	50	62	38
Repowering	Molten salt collector/receiver	98	Est 90	98	30	25	15
IEA	Liquid metal collector/receiver	98	Est 90	98	109	25	47
EPRI/DOE Hybrid	Gas collector/receiver	80	Est 90	80	96	25	66

[a] Conversion from 1979 to 1980 dollars based on 1.087 factor from Business Conditions Digest, U. S. Department of Commerce, January 1981.

[b] Applications shown are all electrical power generating systems.

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[6] "Cost and Performance of Thermal Storage Concepts in Solar Thermal Systems Conceptual Design," Revised, Stearns-Roger Services, Inc., September 1980.

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The following manufacturers were consulted regarding drive motors, controls, and drive trains:

[23]	Philadelphia Gear Corp. King of Prussia, Pennsylvania 19406	re: Hydroviscous variable-speed drive
[24]	Robicon 100 Sagamore Hill Road Plum Industrial Park Pittsburgh, Pennsylvania 15239	re: a-c inverter drives
[25]	U.S. Motors Milford, Connecticut	re: Single speed a-c motors
[26]	Louis Allis Milwaukee, Wisconsin	re: d-c motors and controls
[27]	Rexnord Engineered Components Group Philadelphia, Pennsylvania 19154	re: Chain drives
[28]	Lufkin Gear Company Lufkin, Texas	re: Speed reducer

[29] P. Delaquil, III, C. T. Schafer, S. E. Faas, Cost-Performance Comparison of Water-Steam Receivers for Solar Central Electric Power Plants, SAND 80-8245, December 1980.

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APPENDIX A

Design Characteristics — Four Candidate ATESS Concepts

Table A-1. Operating Parameters

	Units	Oil/Rock	Air/Rock	Moving Bed	Molten Salt
Charge Rate (max)	10^6 Btu/h MW	870 255	887 260	same	same
Thermal Energy Stored 285 MWt x 6 HRS x 1.05 Heat Loss In 24 hrs.	10^9 Btu MWh	6.15 1802	6.13 1796	6.13 1796	6.33 1856
Discharge Rate (max)	10^6 Btu/h MW	973 285	973 285	973 285	same
Thermal Energy Output	10^9 Btu MWh	5.84 1710	same same	same same	same same
Charge Pumping Energy	10^6 Btu MWh	46.1 13.5	363 106	201 59	25.6 7.5
Discharge Pumping Energy	10^6 Btu MWh	55.6 16.3	432 127	273 80	40.6 11.9
Turnaround Efficiency	%	73.2	56.8	68.9	71.1
Storage Material Charge Flow Rates	10^6 lb/h 10^6 kg/h	8.74 3.96	(AIR) 16.6 7.54	17.5 7.94	11.5 5.21
Storage Material Discharge Flow Rates	10^6 lb/h 10^6 kg/h	10.1 4.58	(AIR) 18.2 8.26	19.3 8.76	12.6 5.72
Storage Material Temperatures (max/min)	^oF ^oC	600/450 316/232	635/425 335/218	630/425 332/218	633/425 334/218
Steam Charge Flow Rates	10^3 lbm/h 10^3 kg/h	883 401	890 404	890 404	891 404
Charging Steam Conditions	^oF @ psia ^oC @ MPa	950@ 1465 510@ 10.1	same	same	same
Steam Discharge Flow Rates	10^3 lbm/h 10^3 kg/h	906 411	908 412	908 412	912 414
Discharging Steam Conditions	^oF @ psia ^oC @ MPa	570@ 395 299@ 2.72	same	same	same

Table A-2. Storage Materials

Storage Materials	Units	Barstow Oil/Rock: Oil	Barstow Oil/Rock: Sand	Barstow Oil/Rock: Rock	Air/Rock: Rock	Air/Rock: Air	Moving Bed: Sand	Molten Salt: Molten Salt
Properties at Operating Temperatures (x/y = 400/600°F)								
a. Density	lb/ft^3	45.6/40.3	161.6/161.2	165	165	-	95	121.5/115.3
	kg/m^3	730/646	2589/2582	2643	2643	-	1522	1946/1847
b. Specific Heat	Btu/lb-°F	0.60/0.70	0.23/0.25	0.23/0.25	0.24	0.249@532°F	0.246	0.373
	J/kg- °K	2659/3102	1019/1108	1019/1108	1005	1043	1030	1653
c. Particle Size	in	-	0.0625	1.0	1.5	-	$17\text{-}29 \times 10^{-4}$	-
	mm	-	1.5	25	38	-	$44\text{-}74 \times 10^{-3}$	-
d. Bed Void Fraction		-	(0.25 ---------	Total)	0.39	-	0.40	-
Quantities	$10^3 lb$	16450	59500	119066	409200	-	127300	76600
	$10^3 kg$	7462	27000	54000	185610	-	57792	34750
	$10^3 ft^3$	311	370	722	2480	-	1340	630@425°F
	$10^3 m^3$	8.8	10.4	20.4	70.2	-	37.9	17.8

Table A-3. Heat Exchanger Characteristics, Air/Rock and Moving Bed, Chargers and Dischargers

Parameter	Units	CHARGER Air/Rock	CHARGER Moving Bed	DISCHARGER Air/Rock	DISCHARGER Moving Bed
Number of Heat Exchangers		9	9	9	9
Heat Exchanged @ (Max)	% Total 10^6 Btu/hr MWt	11 98.6 28.9	11 98.6 28.9	11 108 32	11 108 32
LMTD	^oF ^oC	*104.9/58.9/90.2 58/33/50	108.2/62/91.4 60/34/51	**71.5/71.5/107.4 39.7/39.7/59.7	71.5/69.5/105 39.7/38.6/58.3
Overall Heat Transfer Coefficient	Btu/hr-ft^2-^oF W/m^2-^oK	145/150/142 823/852/806	164/227/204.7 931/1288/1162	146/167/109 829/948/619	230.4/264/141.5 1308/1498/803
Heat Transfer Area/HX	ft^2 m^2	1885/6348/1080 175/590/100	1610/3992/737 150/371/68	1958/6605/743 182/614/69	1244/4300/582 115/399/54
Number of Passes	(shell/tube)	1:5/1:16/1:3	1:5	1:5/1:17/1:2	1:4/1:4/1:1
Number of Tubes	(tubes/pass)	114	130	114	160
Tube Dimensions (ID/OD)	in mm	0.62/0.75 15.8/19	0.59/0.75 15/19	0.62/0.75 15.8/19	0.62/0.75 15.8/19
Tube Lengths	ft m	20 16.1	68.9 20.9	20 6.1	89.6 27.3
Tube Pitch	type in mm	ǂ Triangular 2.10 Finned 53.3	Triangular 1.25x1.0x1.0 32x25x25	ǂ Triangular 2.10 Finned 53.3	Triangular 1.25x1.0x1.0 32x25x25
Storage Fluid Pressure Drop (Max)	psia kPa	0.340/1.140/0.221 2.344/7.860/1.524	Free Flowing	0.433/1.406/0.158 2.985/9.694/1.089	Free Flowing
Steam/Water Pressure Drop (Max)	psia kPa	15 103	same	**3.0/1.0/6.0 20.7/6.89/41.4	same
Steam/Water Temperature (Inlet/Outlet)	^oF@ psia ^oC@ MPa	950@1465/480@1460 510@10.1/249@10.07	same	250@400/570@395 121@2.76/299@2.72	same
Storage Fluid Temperature (Inlet/Outlet)	^oF ^oC	425/635 218/335	same	630/425 332/218	same

*Desuperheater/Condenser/Subcooler Sections

** Economizer/Boiler/Superheater Sections

ǂ 7 fins per inch, 0.049 in. thick, 0.36 in. high, annular

Table A-4. Heat Exchanger Characteristics, Oil/Rock and Molten Salt, Dischargers

Parameter	Units	ECONOMIZER Oil/Rock	ECONOMIZER Molten Salt	BOILER Oil/Rock	BOILER Molten Salt	SUPERHEATER Oil Rock	SUPERHEATER Molten Salt
Number of Heat Exchangers		5	1	5	1	5	8
Heat Exchanged @ (Max.)	% Total 10^6 Btu/hr MWt	3.8 37.34 10.94	19 186 54.5	14.6 141.6 41.5	73 710 208	1.6 15.56 4.56	1 9.65 2.83
LMTD	^oF ^oC	93 51.5	22.4 12.4	74 41	70 39	59 32.8	108 60
Overall Heat Transfer Coefficient	Btu/hr-ft^2 ^oF W/m^2-^oC	82 465	400 2268	146 829	440 2498	54 306	340 1927
Heat Transfer Area Hx	ft^2 m^2	4680 435	20800 1933	12950 1204	27800 2585	6,390 594	300 223
Number of Passes	(shell/tube)	1	1	1	1	1	1
Number of Tubes @	(tubes/pass)	800	1796	1232	3624	720	296
Tube Dimensions (ID/OD)	in mm	0.834/1.0 21.2/25.4	0.62/0.75 15.7/19	.584/0.75 14.8/19	0.527/0.625 13.4/15.9	0.584/0.75 14.8/19	.495/.625 12.6/15.9
Tube Lengths	ft m	24.5 7.47	60. 18.	60.5 18.4	46.8 14.2	51 15.5	6.25 1.9
Tube Pitch	type in mm	Staggered 1.25 31.8	Triangular 1.0 25.4	Inline 0.9375 23.8	Triangular 0.875 22.2	Staggered 0.9375 23.8	Triangular 0.875 22.2
Storage Fluid Pressure Drop (Max.)	psia kPa	7.0 48	5 34	5.0 34	5 34	7.0 48	5 34
Steam/Water Pressure Drop (Max.)	psia kPa	3.0 20.7	5 34	1.0 6.89	5 34	6.0 41.4	5 34
Steam/Water Temperature (Inlet/Outlet)	^oF ^oC	250/447 121/231	400/445 204/229	447/447 231/231	445/445 229/229	447/570 231/299	445/570 229/299
Storage Fluid Temperature (Inlet/Outlet)	^oF ^oC	480/450 249/232	465/425 240/218	587/480 308/249	617/465 325/241	595/587 313/308	633/617 334/325

Table A-5. Heat Exchanger Characteristics, Oil/Rock and Molten Salt, Chargers

Parameter	Units	Oil/Rock	Molten Salt
Number of Heat Exchangers		5	1
Heat Exchanged @ (Max)	% Total	20	*100% (29/57/14)
	10^6 Btu/hr	174	887.3
	MWt	51	260
LMTD	^oF	51.0	*107/61/90
	^oC	28.3	59/34/50
Overall Heat Transfer Coefficient	Btu/hr-ft^2-^oF	160	*320/520/280
	W/m^2 -^oC	909	1818/2954/1591
Heat Transfer Area @	ft^2	18000	*7520/15950/4930
	m^2	1670	698/1480/457
Number of Passes	(shell/tube)	1	1/1
Number of Tubes @	(tubes/pass)	1200	1680
Tube Dimensions (ID/OD)	in	0.620/0.75	.62/.75
	mm	21.2/25.4	15.7/19
Tube Lengths	ft	72	22.8/48/15
	m	21.8	6.95/14.6/4.57
Tube Pitch	type	Rotated Square	Triangular
	in	0.9375	1.077
	mm	23.8	27.4
Storage Fluid Pressure Drop (max)	psia	25	10
	kPa	170	68
Steam/Water Pressure Drop (max)	psia	15	9
	kPa	103	62
Steam/Water Temperature (Inlet/Outlet)	^oF	680 /480	950/480
	^oC	360 /249	510/249
Storage Fluid Temperature (Inlet/Outlet)	^oF	450/600	425/633
	^oC	232/316	218/334

* Desuperheater/Condenser/Subcooler Sections

APPENDIX B

System Analyses for Candidate Selection

Table B-1. Thermal Energy Storage System Cost Estimates

Oil/Rock

Tank

230,400 lb steel ≥ 1.5 in. wt [× $0.70/lb + $0.10/lb (stress relieving)]		=	0.184[a]
781,800 lb steel < 1.5 in. wt (× $0.65/lb)		=	0.508[a]
Two layers of 4-in. insulation (47,480 ft^2 × $0.88/$ft^2$)		=	0.042[b]
1/16-in. Al cover (23,740 ft^2 × $0.40/ft)		=	0.009[b]
305.6 tons insulation sand × $7.70/ton		=	0.002[b]
Excavation, 3040 yd^3 × $0.75/$yd^3$		=	0.002[c]
Foundation, 84.7 yd^3 × $114.4/$yd^3$		=	0.010[c]
Total		=	0.755
	× 4	=	3.020
Heat Exchangers			
2.101 × 10^5 ft^2 × $25.00/$ft^2$		=	5.253[b]
Pumps			
260 hp single-speed, $18,000 × 5		=	0.090[d]
250 hp three-speed, $20,000 × 5		=	0.100[d]
Storage Media			
59,533 tons of river gravel × $7.60/ton		=	0.452[b]
29,767 tons of sand × $7.70/ton		=	0.229[b]
8.218 tons of Caloria ($439/ton + $66.9/ton shipping)		=	4.158[b]
Total		=	4.839
TOTAL			13.302

[a]Richmond Engineering Co.

[b]Stearns-Roger.

[c]Process Plant Estimating Standards.

[d]Estimate based on Ingersoll-Rand data and Process Plant Estimating Standards.

Table B-1. (Cont'd)

Moving Sand Bed

Sand Storage Tanks

Structural steel, 754,700 lb × \$0.70/lb	=	0.528[a]
Roof insulation (two layers, 4-in. fiberglass)		
2×2535 ft^2 × \$0.88/ft^2	=	0.004[b]
2535 ft^2 × \$0.40/ft^2 (1/16-in. Al cover)	=	0.001[b]
Excavation, 2867 yd^3 × \$0.75/yd^3	=	0.002[c]
Foundation, 238 yd^3 × \$114.4/yd^3	=	0.027
Total	=	0.562
× 18	=	10.116

Heat Exchangers

Charging, 6333 ft^2 × \$20/ft^2 × 9	=	1.140[b]
Discharging, 6126 ft^2 × \$20/ft^2 × 9	=	1.103[b]
Total	=	2.243

Lifts and Motors

4.5-ft diam. screw, \$188,500 + 40% for installation	=	0.264[d]
4.31-ft diam. screw, \$163,500 + 40% for installation	=	0.229[d]
300 hp, 1800 rpm motor, \$8,843 × 0.97	=	0.009[c]
350 hp, 1800 rpm motor, \$9,570 × 0.73	=	0.010[c]
Reduction gear, 317 hp rating	=	0.007[c]
Reduction gear, 387 hp rating	=	0.009[c]
Total	=	0.528
× 18	=	9.504

Storage Media

60,280 tons silica flour (\$25/T + \$22/T shipping + 5% filling)	=	2.975[e/f]
30,362 tons insulating sand (\$7.7/T + 9% filling)	=	0.255[b]
TOTAL	=	25.09

[a] Richmond Engineering Co.

[b] Stearns-Roger.

[c] Process Plant Construction Estimating Standards.

[d] Hoover Universal, Inc.

[e] Pennsylvania Glass Sand Corp.

[f] Norfolk & Western Railroad.

Table B-1. (Cont'd)

Air/Rock

Tank

Steel structure, 1.8395×10^6 lb × \$0.65/lb		=	1.196[a]
Insulation, 35,426 ft^2 × \$0.88/ft^2		=	0.031[b]
1/16-in. Al cover, 17,713 ft^3 × \$0.40/ft^2		=	0.007[b]
Sand insulation, 411.3 tons × \$7.7/ton		=	0.003[b]
Excavation, 2045 yd^3 × \$0.75/yd^3		=	0.002[c]
Foundation, 97.7 yd^3 × \$114.4/yd^3		=	0.011[c]
Total		=	1.250
	× 9	=	11.250

Heat Exchangers

$9 \times 1.86 \times 10^4$ ft^2 × \$22.5/ft^2		=	3.767[c]

Fans and Motors

2414 hp induction motor and fan, \$145,000 + \$200,000		=	0.345[c]
3142 hp induction motor and fan, \$183,500 + \$200,000		=	0.384[c]
Total	× 9	=	6.561

Storage Media

1.250×10^5 tons crushed granite × \$8.50/ton		=	1.063[b]
TOTAL		=	22.64

[a]Richmond Engineering Co.

[b]Stearns-Roger.

[c]Process Plant Construction Estimating Standards.

Table B-1. (Cont'd)

Molten Salt

Storage Tanks

Structural steel		
Plate ≥ 1.5 in. (686,400 lb × \$0.70/lb + 0.10 stress relieving)	=	0.549[a]
Plate < 1.5 in. (895,000 lb × \$0.65/lb)	=	0.582[a]
Insulation (two layers of 4 in.) 23,229 ft^3/0.333 ft × \$0.88	=	0.061[b]
1/16-in. Al cover (1/2 × 23,229 × \$0.40/ft^2)	=	0.011[b]
537.2 tons of sand (\$7.70/ton)	=	0.004[b]
Excavation, 4187 yd^3 × \$0.75 yd^3	=	0.003[c]
Foundation, 168 yd^3 × \$114.4/yd^3, 150 lb/yd rebar)	=	0.019[c]
Total	=	1.229
× 2	=	2.458

Heat Exchangers

Steam drum	=	0.264[g]
Steel: 69,446 ft^2 carbon steel × \$25/ft^2 (U-tube, shell)	=	1.736[b]
Steel: 7,554 ft^2 low-chrome steel × \$27/ft^2 (U-tube, shell)	=	0.204[b]
Steel: 2,400 ft^2 carbon steel × \$22.5/ft^2 (tube, shell)	=	0.054[b]
Total	=	2.258

Pumps

783 hp charging pump	=	0.075[d]
856 hp discharging pump	=	0.075[d]

Storage Media

2.10 × 10^4 tons potassium nitrate × \$390/ton	=	8.190[e]
1.59 × 10^4 tons sodium nitrite × \$710/ton	=	11.289[e]
2.78 × 10^3 tons sodium nitrate × \$290/ton	=	0.806[e]
7.94 × 10^4 lb total × \$0.025/lb for unloading, mixing, etc.	=	1.985[f]
Total	=	22.870
TOTAL		27.14

[a]Richmond Engineering Co.

[b]Stearns-Roger.

[c]Process Plant Construction Estimating Standards.

[d]Ingersoll-Rand.

[e]VanWater & Rodgers Co.

[f]Sandia Laboratories.

[g]B&W Power Generation Division.

Table B-2. Round Trip Efficiency

		Oil/Rock	Air/Rock	Molten Salt	Moving Bed
1.	E_{OUT} (MWH_T)	1710	1710	1710	1710
2.	E_{HT}/E_L (MWH_T)	--	--	64.6	4 (Lift Loss)
3.	E_{PD} (MWH_T)	16.3	351	11.9	80
4.	E_{PC} (MWH_T)	13.5	212	7.5	59
5.	$0.036\ E_{OUT}$	61.6	61.6	61.6	61.6
6.	E_{IN} (MWH_T)	1802	2335	1856	1915
	η_{RT} (%)	73.2	56.5	71.1	68.9

$$\eta_{RT} = \frac{E_{OUT} \cdot \eta_{TESS}}{E_{IN} \cdot \eta_{SOLAR}}$$

where $\eta_{TESS} = 0.27$ $\eta_{SOLAR} = 0.35$

$$E_{OUT} = 285\ MW_t \times 6\ Hrs. = 1710\ MWH_t$$

$$E_{IN} = E_{OUT} + 0.036\ E_{OUT(1)} + E_{PC} + E_{PD} + E_{HT} + E_L$$

(1) - Steady state heat loss from storage

E_{PC} - Parasitic loss to pumps/lifts during charge

E_{PD} - Parasitic loss to pumps/lifts during discharge

E_{HT} - Parasitic loss to heat tracing (molten salt only)

E_L - Heat loss from lifts (moving bed only)

Table B-3. Availability, Reliability, Maintenance, Inspection and Service Life

	Category	O/R	MB	A/R	MS
		Rankings			
1.	Media decomposition/attrition	2	3	4	3
2.	Media compatibility with structural materials - corrosion/wear	4	3	4	3
3.	Media compatibility with water/steam	2	4	4	2
4.	Freezeup/Thaw difficulty	4	4	4	1
5.	Media Spill Consequence	1	3	4	1
6.	Heat Exchangers	3	4	4	3
7.	Valves	3	2	4	2
8.	Pumps	4	3	4	3
9.	Tanks	3	4	3	2
10.	Component Accessibility (after cooldown)	2	3	4	1
	TOTALS	28	33	39	21

1. O/R - Decomposition temperature is close to the operating conditions, no attrition. MB - No decomposition, some attrition. A/R - No decomposition, very little attrition of rock. MS - Minor decomposition at the Barstow operating conditions.
2. O/R - Nonexistent. MB - Some uncertainty exists as to the amount of wear that would take place. However, based on fluid cat. data - no significant wear is expected. A/R - Nonexistent. MS - Corrosion problems are controllable.
3. O/R - Explosion possible, highly flammable, extensive repairs. Note: All media are capable of causing fires.
4. O/R - No consequence. MB - No consequence. A/R - No consequence. MS - Media undergoes a phase change resulting in thaw or freezing problems.
5. O/R - Contamination of environment, explosive hazard. MB - Media easily cleaned up after cooling. A/R - No consequence. MS - Contamination of environment, explosion hazard.
6. O/R - Fouling of the tubes. MB - Utilizes low flow velocities to reduce wear, therefore essentially no consequence. A/R - No consequence. MS - Subject to tube - sheet crevice corrosion associated with minute tube-to-tubesheet leaks.
7. O/R - No consequence. MS - Some problems, will require some maintenance. A/R - No consequence. MS - Some problems, will require some maintenance.
8. O/R - No consequence. MB - Lifts should be reliable because of their low operating speeds and the performance record of rotating tube kilns which operate at much higher temperatures. A/R - Experience with induced draft fans for large power boilers indicates that the fans will be extremely reliable, especially since the gas is clean. MS - Some difficulties are expected with seals because of impurities in the salt and freezeups.
9. O/R - Some inspection and maintenance. MB - No consequence. A/R - Some inspection and maintenance. MS - Some difficulty in inspection and maintenance.
10. O/R - Drainage and cleaning required. MB - Drainage required. A/R - No consequence. MS - Drainage, cleaning,and reheat of component.

Table B-4. Development Requirements and Risks

	Equipment Item	A/R	MB	MS	OR
		Concepts/Scores			
1.	Heat Exchangers	4	1	2	3
2.	Valves	4	3	3	4
3.	Pumps	3	3	3	4
4.	Tanks	3	4	3	3
	TOTALS	14	11	11	14

1. A/R - No consequence. MB - New technology - development. MS - Characteristic design problems - limited technology. O/R - Possibility of fouling, decomposition, price rise.
2. A/R - No consequence. MB - Development required. MS - Development required. O/R - No consequence.
3. A/R - Possible development. MB - New technology - development. MS - Needs development. O/R - No consequence.
4. A/R - Ratcheting problem. MB - No consequence - FOAK engineering. MS - Development needed to reduce cost. O/R - Ratcheting problem.

Table B-5. Environmental and Safety Aspects

Environmental Event	A/R	MB	MS	O/R
	Concepts/Score			
1. Earthquakes	4	3	2	1
2. Missiles	4	4	1	1
3. Storms	4	4	1	1
4. Sabotage	4	4	1	1
Subtotals	16	15	5	4
Safety				
Consideration				
1. Storage media flammability	4	4	2	1
2. Storage media reaction with water/steam	4	4	2	1
3. Toxicity of airborne media particles or vapor	4	3	3	3
4. Media working pressure	4	4	4	4
5. Effectiveness of personnel protective equipment	3	3	2	2
Subtotals	19	18	13	11
TOTAL Environmental and Safety	35	33	18	17

E 1. A/R - No consequence. MB - No consequence, easily cleaned up. MS - Contamination of environment, explosive hazard. O/R - Contamination of environment, explosive hazard.

E 2. A/R - No consequence. MB - No consequence. MS - Explosive condition. O/R - Explosive condition.

E 3. A/R - No consequence. MB - No consequence. MS - Possible spread of contaminants, fire, etc. O/R - Possible spread of contaminants, fire, etc.

E 4. A/R - No consequence. MB - No consequence. MS - Possible spread of contaminants, fire, etc. O/R - Possible spread of contaminants, fire, etc.

S 1. A/R - No consequence. MB - No consequence. MS - Supplies its own oxygen. O/R - Highly flammable - fuel.

S 2. A/R - No consequence. MB - No consequence. MS - Steam explosion possible. O/R - Steam explosion possible.

S 3. A/R - No consequence. MB - Silica dust would require control. MS - Reaction with contaminants will produce vapors. O/R - Oil Vapors must be controlled.

S 4. All are being used essentially at atmospheric pressure.

S 5. A/R - Burn hazard. MB - Burn hazard. MS - Greater burn hazard. O/R - Greater burn hazard.

Table B-6. Applicability to Higher Range of Working Fluid Conditions

Consideration	A/R	MB	MS	O/R
1. Thermal decomposition	4	4	4	0
2. Compatibility with structural materials	4	4	3	0
3. Insulation	4	4	2	0
4. Ullage	4	4	2	0
5. Safety	4	4	2	0
6. Reliability/Availability	2	4	3	0
7. Service Life	4	4	4	0
TOTAL	27	28	20	0

1. A/R - No consequence. MB - No consequence. MS - No consequence. O/R - Unacceptable - decomposes at these temperatures.
2. A.R - No consequence. MB - No consequence. MS - Material change required that will increase cost - i.e. stainless steel.
3. A/R - No consequence. MB - No consequence. MS - Must avoid contact with insulation.
4. A/R - No consequence. MB - No consequence. MS - Phase change requires additional ullage space, plus heat traced ullage system components.
5. A/R - No consequence. MB - No consequence. MS - Some hazards.
6. A/R - Ratcheting problem. MS - No consequence. MS - No consequence.
7. All except O/R considered acceptable.

APPENDIX C

Source Documents for Literature Search

1. Active Heat Exchange System Development for Latent Heat Thermal Energy Storage, Joseph Alario and Robert Haslett, Grumman Aerospace Corp.

2. Active Heat Exchange System Development for Latent Heat Thermal Energy Storage, R. T. LeFrois, Honeywell, Inc., Technology Strategy Center.

3. SPRI EM-264, Project 225 (July 1976), Volume II, An Assessment of Energy Storage Systems Suitable for Use by Electric Utilities, Final Report, Public Service Electric & Gas Co., Newark, New Jersey.

4. SAN/1110-88-2, Volume 5, April 1977, Center Receiver Solar Thermal Power System, Phase I: Preliminary Design Report, Volume 5, "Thermal Storage Subsystem," Contract No. EY-77-C-03-1110, Department of Energy, Martin-Marietta Corp.

5. SAN-1108-8/2, November 1977, Central Receiver Solar Thermal Power System, Phase I, CRDL Item 2, Pilot Plan — Preliminary Design Report, Volume 3, Book I, "Collector Subsystem," Raymon W. Hallet, et al.

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7. SAND 79-8015, August 1979, A Description and Assessment of Large Solar Power Systems Technology, L. N. Tallerico, Sandia Laboratories, Albuquerque, New Mexico.

8. DOE/ET-0108, May 1979, Energy Storage Systems Program, Overview FY-1979, U.S. Department of Energy, Asst. Secretary for Energy Technology, Division of Energy Storage Systems.

9. SAND 80-8175, MCR-79-1369, December 1979, Internally Insulated Thermal Storage System Development Program, Final Report, Martin-Marietta Corp., Owen L. Scott.

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17. Solar Thermal Power Storage Applications Lead Laboratory Overview, Summary, Lee G. Radosevich, Sandia Laboratories.

18. SAN/1108-8/5, November 1977, Central Receiver Solar Thermal Power System, Phase I, CDRL Item 2, "Pilot Plant Preliminary Design Report," Volume 5, Thermal Storage Subsystem, Raymon W. Hallet, et al.

19. 10-MWe Solar Thermal Central Receiver Pilot Plant, R. N. Schweinberg, Department of Energy, and J. N. Reeves, Southern California Edison.

20. SAND 78-8221, August 1978, Thermal Energy Storage for Advansed Solar Central Receiver Power Systems, L. G. Radosevich, Sandia Laboratories.

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22. Thermal Energy Storage Systems Using Fluidized Bed Heat Exchangers, V. Ramanathan, T. E. Weast, K. P. Anath, Midwest Research Institute, Kansas City.

23. Heat Transfer Agents for High-Temperature Systems, Joel R. Fried, General Electric Company, Chemical Engineering, May 29, 1973.

24. SAND 78-1315, June 1979, User's Manual for Computer Code SOLTES-1, Simulator of Large Thermal Energy Systems), Fewell, Grandjean Sandia Laboratories, Albuquerque, New Mexico.

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26. Thermal Storage Experience at the MSSTF and Plans for the Future, Harrison and Randall, Sandia Laboratories, Albuquerque.

27. Preliminary Requirements for Thermal Storage Subsystems in Solar Thermal Applications, R. J. Copeland, SERI, SERI-RR-731-364, April 1980.

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31. Survey of Technology for Storage of Thermal Energy in Heat Transfer Salt ORNL-TM-5682, M. D. Silverman, et al., Oak Ridge National Laboratory, Oak Ridge, Tennessee (1977).

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APPENDIX D

TANK Evaluation for Candidate Design

Evaluation of Tank Designs for Four TES Systems

The conceptual design of the Thermal Storage Units (TSU) or tanks employed in four (4) Thermal Energy Systems were received, investigated and evaluated in the following areas:

(1) Applicable Codes and Standards

(2) Material Supports and Foundations

(3) Seismic Criteria and Aseismic Design

(4) Thermal Analysis and Thermal Stress Evaluation

(5) Thermal Bed Ratcheting Evaluation

(6) Cost Evaluation

Many conclusions or judgements could only be made in a general manner and on the basis of experience gained in high temperature, high pressure nuclear reactor vessel design and fabrication.

Review results and/or conclusions follow by area headings:

1. <u>Applicable Codes and Standards</u>

   Though the API Std. 650 Code is a proven oil storage tank standard, the only temperature limitation involved is to preclude a brittle fracture failure mechanism which by itself is inadequate for high temperature design and thermal cycling.

   Considering the thermal storage media and temperature levels (450 - 600 F), this code is considered inadequate and should only be used as a guide for TSU material selection, design, and fabrication to minimize costs when compatible with a more stringent code.

   This code does not degrade material physical properties with increasing temperature (including the coefficient of thermal expansion) nor does it impose a corrosion allowance (shell thickness loss as a function of time).

Most illustrated details in the API Code are inadequate and unacceptable for high temperature service. The bottom shell welded joints are partial penetration welds and too rigid for the anticipated thermal differentials; thus, weld cracks would result. The roof to shell joints also appear to be of a similar nature with similar problems.

The proper code for the intended service is the ASME boiler and Pressure Vessel (Unfired) Code, Section VIII, Division 2. This code allows ample alternatives to design and fabricate TSU's at reasonable costs consistent with contemporary safety requirements for this service. This code is recommended as the minimum acceptable code for this purpose.

The ASME B&PV Code Section III, Division 1, NC (Class 2) could also be used and though it contains all of the features of Section VIII it has more stingent requirements and is compatible with NRC Regulatory Guide 1.60 Seismic requirements. Because this code allows a very detailed structural analysis, including a fatigue analysis, with reduced stress margins, thinner design wall thickness could result with attendant cost reductions in both material and fabrication.

The significant results of the review are summarized on the attached Code Evaluation tabulation, Table 1.

2. Materials, Supports and Foundations

The ASTM A537 (or ASME SA 537) Class 2 material specified for the "oil/rock" TES, though acceptable for this service, may not be the best choice because in thicknesses over 1½" a 200 to 250 F preheat and an 1100 F (or alternative) stress relief are required. These requirements can be very burdensome and costly for field fabrication and are to be avoided if possible.

Because of differential thermal growth between steel, earth, and concrete, an earthen bottom in combination with a reinforced concrete ring, as proposed in the "oil/rock" system ("Barstow Proposal") is inadequate and would result in broken concrete or crippled steel because of an $\sim$4.547 inch difference in thermal growth. High temperature tank supports, which carry the tank deadload to the foundation must provide either free vessel thermal dilation motion (both directions) or support flexibility to prevent undue restraint stresses in the tank shell.

Conventional reinforced concrete foundations can be used consistent with the foregoing when the thermal conductance into the concrete does not heat the concrete above 150 F on a full time basis.

3. Seismic Criteria and A Seismic Design

The dynamic seismic design, as promulgated in the US-NRC Regulatory Guide 1.60 is generally recognized as the most rational basis for the design of structures to resist earthquakes. In the interest of public safety and welfare, it is recommended that a dynamic seismic analysis be performed per NRC Reg. 1.60 and 1.61 (Damping Values).

The seismic response of structures to a seismic occurrence can be compared and evaluated on an approximate basis by determining their individual natural frequencies for comparison. The greater (or larger) the natural frequency value the less responsive the structure will be to a seismic input. Other factors, such as total mass, support location (elevation), stiffness, mass center relative to the support, etc., also affect the structure response but will not be developed at this time.

The natural frequencies of the four types of TSU's were determined and the results are provided on the attached tabulation titled, "Seismic Evaluation", Table 2.

4. Thermal Analysis and Thermal Stress Evaluation

Regardless of the TES system used or the tank preliminary design selected, extensive thermal analysis (gradients and heat conduction) will be required in order to verify the adequacy of the tank design. Major areas of necessary investigation follow:

(a) Tank support to concrete interface to limit concrete to 150 F (precludes loss of strength from dehydration).

(b) Tank shell to shell weld joints of varying size plates (discontinuities).

(c) Tank shell to support (or reinforcement) surface thermal gradients and temperatures.

(d) Tank bottom to shell connection or joint.

(e) Tank top (or head) to shell connection or joint.

(f) Tank shell or bottom/top plate joints to internal structural supports.

(g) Thermal stresses, as related to each of the above (a) through (f), for acceptable levels.

The "oil/rock" system structural analysis[1], as provided on pages 4-48 through 4-59, was reviewed and found to be inadequate and incorrect with regard to coefficients of thermal expansion and thermal stress. Page 4-49, second sentence from the bottom, states that .."the coefficient of expansion of steel is higher than the rock-sand mixture, a gap will occur....", and the values by material given on page 4-54 are consistent with this statement but the solids values listed are incorrect. The International Critical Tables, Vol. II, Page 54, provide the following thermal expansion data for granite rock:

Type Location	°C	/ °F	$\propto \times 10^{-6}$ per °C	$\propto \times 10^{-6}$ per °F
Quartz manzonite,	20 to 100	68 to 212	9	5.03
Westerly, R.I.	100 to 200	212 to 392	14	7.82
	200 to 300	392 to 572	20	11.18
Biotite,	0 to 100	32 to 212	7.6	4.24
Milford, Mass.	100 to 200	212 to 392	13	7.27
Gneissoid	0 to 100	32 to 212	7.2	4.02
Bronford, Conn.	100 to 200	212 to 392	17	9.50
Muco te-biotite	0 to 100	32 to 212	6.1	3.41
Troy, N.H.	100 to 200	212 to 392	12	6.71

The low temperature range (20 to 100) value of 7.92 x 10^{-6} $^{o}C^{-1}$ is an acceptable value but it should not be used in place of the higher temperature (392 to 572) value of 20 x 10^{-6} $^{o}C^{-1}$ at the high temperature steady state condition. The difference between these values is a factor of 3 and the granite will outgrow the carbon steel to an interference in temperature instead of the claimed gap. All subsequent thermal stress calculations based on these expansion values will also be incorrect. For example, when the value of 20 x 10^{-6} $^{o}C^{-1}$ is used in the F_t equation on page 4-58, the thermal stress becomes 37700 psi instead of the indicated 20860 psi for a discrepancy of 1.81 to 1. When the hoop stress (F_a) equation, is revised for the larger 90.5' O.Dia tank, F_a becomes 15718 psi. Accordingly, F_{tot} then becomes 53418 psi, which is 0.89 of the yield stress (F_y) and 1.78 times the selected material allowable (F_o). This would be a most unsatisfactory situation which would be worsened by material temperature degredation and thickness corrosion loss that have not been considered.

An evaluation of tank shell thermal stress consistent with thermal insulation placement was performed and the results are provided in the attached Table 3, "Thermal Stress Evaluation".

5. Thermal Bed Ratcheting Evaluation

The "thermal bed ratcheting" problem, as described and calculated in the "oil-rock" (Barstow) report[1] pages 4-49 through 4-68, exists and must be accommodated in all tank designs using a solid storage media. Unfortunately, as described in the foregoing Section 4, inadequate granite rock coefficients of linear thermal expansion ($\propto$) were used and the calculated results are not valid for a range of 450 to 600 F. Using the higher temperature coefficient $\propto$, the granite rock bed will thermally outgrow the vessel diameter by 1.325 inch interference which overstresses the vessel shell. This problem can be solved by a separate segmented spring loaded ("girdle") type internal shell which can accommodate the differential thermal motions. This arrangement has been incorporated in the proposed "air/rock" TES system.

An evaluation of "thermal bed ratcheting" for the four TES systems was performed and is summarized in Table 4.

6. Cost Evaluation

A relative comparison of costs between the tanks of the four (4) TES systems was performed on the basis of tank weight and quantities. Each tank weight included only structural steel and piping. The thermal bed weight, motors and similar mechanical/electrical equipment were omitted.

Using a dollar per pound cost for steel material, fabrication cost, delivery, and installation/erection, a total tank cost can be approximated by the product of this cost and the weight.

The cost evaluation is summarized and presented in the attached Table 5.

Conclusions

If the following set of values are assigned to the ratings given in Tables 2 through 5, the four (4) TES systems can be evaluated on an overall basis:

Best - 4
Good - 3
Bad - 2
Worst- 1

The overall evaluation matrix results in the following value totals by TES system:

Air/rock -11
Moving bed -11
Oil/rock - 9
Molten salt- 9

Note: These results are on the basis of tank hardware design and weight (cost) only.

All of the problems described in areas (1) through (5) above can be solved at a cost which could affect area (6) to an unknown extent. Also, because the thermal bed and machinery costs are not included in the Cost Evaluation (Area 6), an overall cost estimate for each of the four (4) TES systems must be performed and evaluated.

Ultimately, two factors: (a) total TES system cost, and (b) thermal bed storage capability (i.e., capacity in BTU's per cubic foot and effectiveness) will determine the TES system selection. These two factors are not within the scope of this tank study.

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[1] Central Receiver Solar Thermal Power System, Phase I CRDL Item 2, Pilot Plant Preliminary Report, Vol 5, Thermal Storage Subsystem, SAN/1108-8/5, Sandia Laboratories.

TABLE D-1

Code Evaluation

Code	Adequacy	Recommendations
API Std. 650 April 15, 1977	• Inadequate for design at temperatures to 650°F. • UBC Seismic considered inadequate by contemporary safety standards. • No fatigue analysis required.	• Use as a guide for material, selection, design and fabricated details consistent with ASME Unifired Vessel Codes, Sections VIII or III.
ASME B&PV Code Section VIII, Division 2	• Acceptable for design temperatures of 400 to 700 F. • Includes a 1.2 factor on allowables for seismic as a requirement. • Fatigue analysis for sizing wall thickness at temperature.	• Minimum acceptable Code for materials, design, fabrication and test. • Conservative safety factors. • Allows various analytical alternatives.
ASME B&PV Code Section III, Division I, NC (Class 2)	• All of above plus more stringent requirements. • Compatible with NRC Reg. Guide 1.60.	• More comprehensive analytical methods <u>may</u> <u>permit</u> lesser wall thicknesses with material/fabrication cost savings for production units.

TABLE D-2

Seismic Evaluation

System Type	Weight Per Tank (lbs)	Mom. of Inertia $I\ (in^4)$	Nat. Freq. fn, (cps)	Rating
Air/Rock	31,495,350	7.84×10^8	2.871	Best
Oil/Rock	57,620,000	5.02×10^8	2.616	Good
Molten Salt	89,385,300	6.77×10^8	1.446	Bad
Moving Bed	15,746,620	0.84×10^8	1.030	Worst

TABLE D-3

Thermal Stress Evaluation

System Type	Thermal Insul. Location	Thermal Stress (psi.)	Rating
Air/Rock	Internal	9177	Best
Oil/Rock	External	19665	Better
Moving Bed	External	26876	Good
Molten Salt	External	27269	Worst

TABLE D-4

Thermal Bed Ratcheting Evaluation

(70 to 600 F)

System Type	Tank Dia. (ins.)	Δd Tank/Bed (ins.)	Diametral Gap (ins.)	Rating
Moving Bed	600	0 *	0 *	Best
Molten Salt	1200	0 *	0 *	Best
Air/Rock	1260	0 **	0 **	Good Includes internal "girdle" tank to accomodate "thermal racheting" w/o slump
Oil/Rock	1086	∝ = 0.0000066 3.799 6.449 ∝ = 0.000011.19	-1.325 Interference	Barstow report used inadequate coeff. of thermal expansion ∝ for granite at temp. Thus; the interference

* Thermal storage media accomodates to shell movements.

Bed is isolated and "girdle" internal tank accom tes bed expansion.

TABLE D-5

Cost Evaluation
(Steel Only)

System Type	Weight Per Tank (lbs)	No. Tanks	Total Weight (lbs)	Rating
Moving Bed	882,620	18	15,887,160	Lowest Cost
Molten Salt	13,515,840	2	27,031,680	Second Lowest Cost
Oil/Rock	8,291,540	4	33,166,160	Second Highest Cost
Air/Rock	6,682,420	9	60,141,780	Highest Cost

APPENDIX E

Theory of Hot Wire Method

The hot wire or line source method physically consists of a small diameter heater wire placed in an "infinite" medium, for which the thermal conductivity and diffusivity are to be measured. The term "infinite" is defined from a thermal transport sense and normally results in reasonable sample sizes in practice.

The solution of the cylindrical Fourier transient heat conduction equation for the temperature field in an infinite homogeneous medium surrounding an infinitely long line heat source is (Carslaw and Jaeger, 1959)[1]:

$$T(r,t) = \frac{q}{4\pi k}\left[-Ei\left(\frac{-r^2}{4\alpha t}\right)\right]$$

where

q = heat generation per unit length,

t = time,

k = thermal conductivity of the surrounding medium,

r = radius,

α = thermal conductivity of the surrounding medium,

ρ = density,

c = specific heat,

T = temperature.

-Ei(x) is the exponential intetral $\int_x^\infty \frac{1}{x}\exp(-x)dx$, where $x = r^2/4\alpha t$.

In real systems the finite size and heat capacity of the wire and the interface coefficient introduce additional terms to the series solution which vanish for sufficiently large values of time. Therefore, the thermal conductivity of a fluid surrounding an infinite line heat source, such as a hot wire, can be found from the slope of a plot of temperature versus the log of time when the heat generation per unit length is known.

$$k = \frac{q}{4\pi}\,\frac{\ell n(t_2/t_1)}{T_2 - T_1}$$

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[1]H. S. Carslaw and J. C. Jaeger, Conduction of Heat Solids, Oxford University Press, Second Edition, pp 344-345.

Figure E-1 is an example of the data that would be executed from this technique. The thermal conductivity is proportional to the linear portion as discussed above. Thermal diffusivity can be found from the following equation (ASTM STP 660)[2]:

$$t_o = \frac{r_o^2}{4\alpha}$$

The constant time, t_o, is found by plotting the inverse of the derivative of the wire temperature against time, t. The linear portion of the curve is extrapolated back to the time axis where the intercept is t_o. This constant time can also be used in the thermal conductivity evaluation to eliminate the error in neglecting the series expansion terms. Figure E-2 shows an example of this procedure.

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[2]ASTM Special Technical Publication 660, "Thermal Transmission Measurements of Insulation," 1978, pp 157-158.

Figure E-1. Method for Determining Thermal Conductivity Using Hot Wire Technique

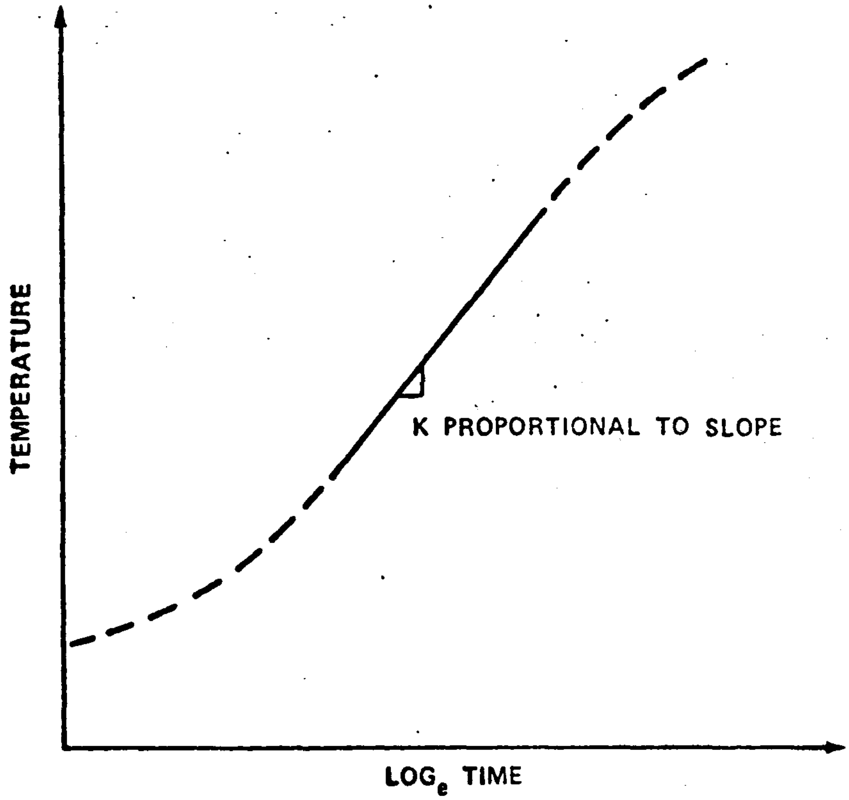
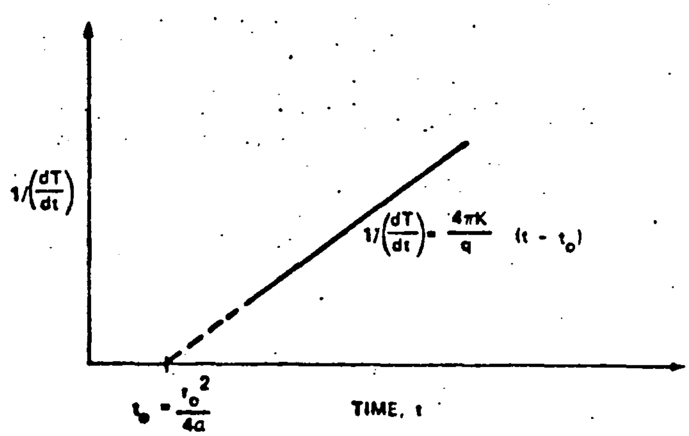


Figure E-2. Method for Determining Thermal Diffusivity Using Hot Wire Technique



APPENDIX F

Cost Estimate Calculations for Moving Bed Thermal Energy Storage System

Excavation

Volume of Excavation

$\frac{1}{2}[(270 \times 92) + (220 \times 64)] \times 22 = 4.281 \times 10^5$ ft^3 (15,860 yd^3)

$2 \times \frac{1}{2}[(23 \times 20) + (23 \times 59)] \times 30 = 2.726 \times 10^5$ ft^3 (1010 yd^3)

$\frac{1}{2}[(23 \times 20) + (23 \times 48)] \times 30 = 2.346 \times 10^5$ ft^3 (870 yd^3)

$3[1/3(30 \times 41) \times 20] = 2.460 \times 10^4$ ft^3 (910 yd^3)

$4 \times 1/3\left(\frac{14 \times 41}{2}\right) \times 30 = 1.148 \times 10^4$ ft^3 (430 yd^3)

$2 \times 1/3[\frac{1}{2}(25 \times 41) \times 30] = 1.032 \times 10^4$ ft^3 (380 yd^3)

$3 \times \frac{1}{2}(22 \times 31.6) \times 20 = 2.086 \times 10^4$ ft^3 (772 yd^3)

$12 \times 1/3[\frac{1}{2}(12.3 \times 5.74) \times 17.6 = 2.49 \times 10^3$ ft^3 (90 yd^3)

$3 \times \frac{1}{2}\{\frac{1}{2}(71 + 20) \times 51 + 1(48 + 20) \times 30\} \times 14 = 7.015 \times 10^4$ ft^3 (2600 yd^3)

$3 \times \{\frac{1}{2}[(71 \times 25) + (20 \times 25)] \times 52\} = 1.774 \times 10^5$ ft^3 (6570 yd^3)

$3 \times \frac{1}{2}[25.5 \times 52] \times 20 = 3.978 \times 10^4$ ft^3 (1470 yd^3)

$6 \times 1/3[\frac{1}{2}(25.5 \times 52) \times 25.5] = 3.381 \times 10^4$ ft^3 (1250 yd^3)

Total of items above	=	32,210 yd^3
10% for access cuts	=	3,220 yd^3
		35,430 yd^3
× 4	=	141,720 yd^3

Using eight Caterpillar D-8K dozers and four 988-B front loaders, 2100 yd^3/h can be removed. Costs/hour:

8 dozers	$ 600.00	
4 loaders	285.00	1.417×10^5 yd^3/2100 yd^3/h + 4 = 72 h
4 foremen	66.13	
12 operators	238.00	(1152 mh)
Overhead*	137.00	(*45% of labor cost)
Total	$ 1326.00/hour	

Equipment moving charges: 12 × 2 × $65 = $1600

TOTAL = 1326 × 72 + 1600 = $97,000

(Ref 1, Sec 2-18)

Excavation

Volume of Excavation

$\frac{1}{2}(106.3 + 62.0) \times 493 \times 51 = 2.116 \times 10^6$ ft^3 (7.84×10^4 yd^3)

$2[1/3(25.5)^2 \times 51] = 2.211 \times 10^4$ ft^3 (819 yd^3)

$2[1/3(25.5 \times 42.8) \times 51] = 6.312 \times 10^4$ ft^3 (2340 yd^3)

$2[\frac{1}{2}(25.5 \times 51) \times 62] = 8.063 \times 10^4$ ft^3 (2990 yd^3)

$2[\frac{1}{2}(51 \times 110 \times 15) - \frac{1}{2}(39 \times 15 \times 25.5)] = 6.923 \times 10^4$ ft^3 (2560 yd^3)

$4[1/3(\frac{1}{2} \times 56.27 \times 27.39) \times 25.5] = 2.620 \times 10^4$ ft^3 (970 yd^3)

$4[1/3(\frac{1}{2} \times 27.39 \times 39.11) \times 84.5] = 6.035 \times 10^4$ ft^3 (2240 yd^3)

Excavation for two buildings = 2.438×10^6 ft^3 (9.03×10^4 yd^3)

2 = 4.876×10^6 ft^3 (1.81×10^5 yd^3)

Using eight D-8K dozers and four 988-B loaders at 2100 yd^3/hour,

Costs per hour:	Eight dozers		\$ 600.00
	Four loaders		\$ 285.00
	Field crew		
	Four foremen	(16.54 × 4)	\$ 66.13
	12 operators	(48.09 × 12)	\$ 577.08
	Overhead*		\$ 289.44
	Total		\$ 1817.65/hour

1.81×10^5 yd^3/2100 yd^3/hour + 4 hours = 90 hours

\$1817.65/hour = \$163,600

Moving equipment on and offsite: 12 × 2 × \$65 = \$1600

10% profit = \$16,500

TOTAL = \$181,700

(Rejected because of increased price.)

*Overhead = 45% of labor cost.

Backfill and Compaction

Volume of structure below grade (per structure):

$V_{tank} = 1.691 \times 10^5 \text{ ft}^3$

$V_{tube\&exit} = 3.413 \times 10^4 \text{ ft}^3$

$V_{equip\,cavern} = \underline{5.115 \times 10^4 \text{ ft}^3}$

Total $= 2.544 \times 10^5$ f6^3 $\times 4 = 1.018 \times 10^6 \text{ ft}^3$ (37,690 yd^3)

Volume of initial backfill:

141,720 yd^3 - 37,690 yd^3 = 104,030 yd^3

Volume of final backfill = 37,690 yd^3

Backfill and Compaction Costs:

20 Caterpillar D-4D dozers	$ 375
20 Caterpillar 930 front loaders	500
20 towed static 4' × 4' rollers	70
20 pan-type compactors	35
20 2400-gal. water trucks	220
Total	$ 1220/hour
10 foremen (20.48)	$ 205
40 operators (16.03)	641
20 truckdrivers (13.40)	268
60 laborers (14.00)	840
Total	$ 1954/hour
Overhead (45% of labor)	879

Output = 900 yd^3/hour

Time required =
141,720/900 + 4 = 162 hours

21,060 manhours

Charge per hour: $4053

$4053/hour × 162 hours =

$4053/hour × 162 hours = $656,600
(Ref 1, Sec 2-21)

Backfill and Compaction

Initial Backfill and Compaction

Volume of structure below grade level (per tank), $\underline{ft^3}$:

$V_{tank} = ½(72.1 \times 20.75) \times 178 + 2 \times 1/3(72.1 \times 36.04 \times 20.75) = 1.691 \times 10^5$

$V_{tube\&exit} = 3(8.5 \times 8.5 \times 19.75) + 3(6.5^2 \pi \times 74.97) = 3.413 \times 10^4$

$V_{equip\,cavern} = 3[(15 \times 10 \times 51) + (40 \times 10 \times 15) + (20 \times 10 \times 5) = 5.295 \times 10^4$

$V_{total} = 2.563 \times 10^5\ ft^3 (9{,}490\ yd^3)$
$\times 4 = 1.025 \times 10^6\ ft^3\ (38{,}000\ yd^3)$

Volume of initial backfill:

$4.876 \times 10^6\ ft^3 - 1.025 \times 10^6\ ft^3 = 3.851 \times 10^6\ ft^3\ (143{,}000\ yd^3)$

Volume of remaining backfill: $1.025 \times 10^6\ ft^3\ (38{,}000\ yd^3)$
(Ref 1, Sec 2-21, p. 3)

Equipment

20 Caterpillar D-4D bulldozers	$ 19.75/hour
20 Caterpillar 930 loaders	25.00
20 towed static 4' by 4' rollers	3.50
20 pan-type compactors 21" by 24"	1.75
20 2400-gallon water trucks	11.00
Total	$ 61.00 × 20 = $1220.00/hour

Field Crew

10 foremen	10 × 20.48	
40 operators	40 × 16.03	Output = 900 yd^3/hour
20 truckdrivers	20 × 13.40	
60 laborers	60 × 14.00	Time required = 205 hours
Total	$1954.00/hour	

$ 1220	equipment
2954	labor
879	overhead*
405	profit

$ 4458/hour × 205 hours = $913,890

*Overhead = 45% of labor cost.

Equipment Cavern

Item	Material	Labor	Subcontract	References
Continuous footings	10" 20" 137.5' linear			
Forms	$ 86	303 (18.3 mh)		1, Sec 3-5
Concrete	$317	160 (11.5 mh)		1, Sec 3-5
Reinf. steel	(600' #4 straight, 100' #6 bent)		223	1, Sec 3-5
Total	$403	$463 (29.8 mh)	223	$1089
Floor (59.7 × 6.7 × 6") (393 ft^2)				
Sand fill and vapor barrier (4")		(6.3 mh)	137	1, Sec 3-5
Reinforcing steel	(700' #4 straight)		189	1, Sec 3-5
Concrete	$330	77 (5.6 mh)		1, Sec 3-5
Screed	$ 28	53 (3.2 mh)		1, Sec 3-5
Total	$358	$130 (15.1 mh)	$326	$814
Walls 2880 ft (double layer concrete block)				
Concrete blocks, mortar	$4700 (6360 blocks)	$7180 (491.6 mh)		1, Sec 4-1
Rinforcing truss	$ 626 (4310')	$ 412 (25.9 mh)		1, Sec 4-1
Reinforcing steel #6	$3820	$1058 (74.6 mh)*		1, Sec 4-1
Moisture barrier	$ 478	$ 800 (57.6 mh)		1, 7-0, 7-1
Total	$9624	$9450 (649.7 mh)		$19,074
Ceiling 47' × 10' × 3" (steel-supported concrete)				
10 WF 10x33 beams	$ 825 (25¢/lb)	$147 (10 mh)		2
Steel deck 16 ga.		(4.3 mh)	$1025	1, Sec 5-4
Concrete	$ 237	$ 96 (6.4 mh)		1, Sec 3-6
Screeds	$ 10	$ 19 (1.1 mh)		1, Sec 3-20
Moisture barrier	$ 79	$143 (9.4 mh)		1, Sec 3-20
Total	$1151	$405 (312 mh)	$1025	$2581
Shaft 10' × 15' × 50.7' (2550 ft^2) Single-layer concrete block				
Block & mortar	$2119 (2869 blocks)	$3530 (221.7 mh)		1, Sec 4-1
Reinforcing steel	(7600' #5 straight, 1800' #3 bent)	(88.6 mh)	$2900	1, Sec 4-1
Moisture barrier	$ 423	$ 708 (51 mh)		1, Sec 4-0
Total	$2542	$4238 (361.3 mh) (1087)	$2900	$9680
OVERALL TOTAL			$33,238 × 12	= $398,900

Foundations and Footings

Wall Footings

Total wall weight (not including insulation) = 420,200 lb
No. fottings per structure = 58
Supported weight per footing = 7245 lb
If soil bearing load of compacted soil = 2000 psf, A(of footing) = 5.4 ft^2

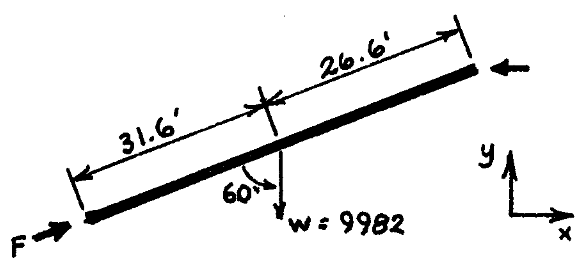
Wall Footing — 2'6" × 2'6" × 10½

Concrete	\$ 9.00	(\$45/yd^3)
Concrete labor	\$14.12	(1.0 mh)
Reinf. steel	\$ 7.93	(16' #4, ½" straight, 10.7 lb; 8#5-5/8", 8.3 lb)
	\$31.05 × 58 = \$1800	(Ref 1, Sec 3-3)

Roof Footings

Total roof wt 433,860 lb structure, 115,120 lb insulation = 549,000 lb total

No. of beams = 58
Supported wt per beam = 9982 lb
F_y = 9982 lb, F_x = 12,300 lb
A_x = 9982/2000 × 1.5 = 7.5 ft^2
A_y = 12,300/1000 × 1.5 = 18.45 ft^2



Continuous Footing 280' × 102' Overall 20" wide, 24" deep

Concrete	\$ 4249	(\$45/yd^3)
Concrete labor	1283	(90.9 mh)
Reinf. steel	5097	(6200' #5-5/8", 5800' #5 bent)
Forms	1222	
Form labor	2797	(168.8 mh)
	\$14,650	(Ref 1, Sec 3-1)

Aggregate base under floors 4" thick (ρ = 148 lb/ft)
28,000 ft surface per tank
710 tons aggregate base (\$6.00/ton) = \$4262.40)
12 hours fine grinding equipment and cres

Caterpillar 313vating scraper	\$ 38.00 (equipment)	\$ 16.03
Caterpillar motor grader	33.00	16.03
Rooter	27.00	16.03
Water truck	11.00	13.40
Foreman	--	17.84
Grade checker	--	17.30
Laborer	--	14.00
Total	\$109.00/hour	\$110.63/hour
+45% of labor for overhead		\$49.78/hour

Total cost for one structure \$7,495.00
(Ref 1, Sec 1-43)

Total foundation costs for four structures $95,780

Pillar Foundations

If a bearing load of 20,000 lb is available (at excavation depth of 52 ft),

Load = 264,000 lb $A = 13.2\ ft^2$

1 pad 4' × 4' × 10.5"

Concrete $23.40 m $16.94 L   (1.2 mh)

Reinforcing steel (30' #6, 10' #6 bent) $24.

Total $64 × 24 = $1544

Storage Structure

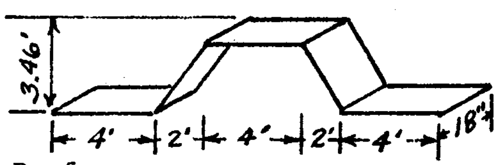
Walls

2 walls ¼ in. plate 270 ft × 16.75 ft (10.2 lb/ft^2)	92,260 lb
2 walls ¼ in. plate 267 ft × 16.75 ft (10.2 lb/ft^2)	91,230 lb
2 walls ¼ in. plate 92 ft × 16.75 ft (10.2 lb/ft^2)	31,440 lb
2 walls ¼ in. plate 89 ft × 16.75 ft (10.2 lb/ft^2)	30,410 lb
58 vertical support beams W = 14 × 22 (22 ft long)	28,070 lb
186 horizontal bracing beams 8I × 18.4 (12 ft long)	41,070 lb
186 horizontal bracing beams 7I × 20 (12 ft long)	44,640 lb
310 horizontal bracing beams 7I × 15.3 (12 ft long)	56,920 lb

wt of plate	245,340 lb
wt of structural shapes	170,700 lb
wt of bolts & ties (1% of total)	4,160 lb

Floor

2 floor sections 270 ft 181 ft × 33 ft trapazoid (+10% waste) 3/16 in. plate	201,150 lb
2 floor sections 92 ft base × 53 ft triangle (10% waste) 3/16 in. plate	41,030 lb
350 interlocking sand retaining sections as shown below	



16 gage steel (2.55 lb/ft^2)	26,780 lb
wt of plate	242,188 lb
wt of 16 gage	546,980 lb

Roof

18 gage × 26 in. wide corrugated steel sheet (2.32 lb/ft^2)

4 270 ft × 181 ft × 53 ft trapozoid (+10% waste)	122,000 lb
4 92 ft base 53 ft triangle (+10% waste)	24,890 lb
6 roof columns W = 36 × 230 (90 ft long)	124,200 lb
2840 ft roof support beams W = 18 × 35 (various lengths + 10%	99,370 lb
7130 ft roof cross supports 4I × 7.7 (various lengths + 10%)	54,890 lb

wt of corrugated sheet	176,890 lb
wt of structural shapes	278,460 lb
wt of bolts & ties (2% of total)	8,510 lb

Cost Estimates MSB TESS

Bottom feeder

8.5 ft diam. × 19.75 ft ½ in. plate + base
11,130 lb of plate
15% allowance for structural supports = 1790 lb structural shapes

Total

Plates A-36 521,000 lb		
Material (0.212 $/lb)	$110,452	(ref. 2)
Transportation (0.038 $/lb from Geneva, Utah)	19,798	(ref. 1, section 100-700)
Erection costs (labor, supplies, equipment, $0.30/lb)	156,300	(ref. 3)
	$287,000	
Structural shapes 467,200 lb		
Material (0.213 $/lb)	$ 99,514	(ref. 2)
Transportation (0.038 $/lb from Geneva, Utah)	17,754	(ref. 1, section 100-700)
Erection costs (0.30 $/lb)	140,160	(ref. 3)
	$257,000	
Insulation		
Floor and walls		
18 in. thick layer of ρ = 95 lb/ft^3 sand		
2905 tons required		
Material ($7.70 ton)	= 22,370	(ref. 6)
Fill and compaction		
Equipment $0.252/ton	= 730	(ref. 1, sections 2-18 and 2-21)
Labor (0.04 mh/ton) (1 foreman, 1 operator, 4 laborers) $0.544/ton	= 1,590	
	25,000	
Roof		
27,500 ft^2 thermal insulating wool type II 4 in. thick × 0.88 $/ft^2	=$ 24,200	(ref. 6, ref. 7)
28,800 ft^2 thermal insulating wool type II ½ in. thick × 0.11 $/ft^2	= 3,168	(ref. 6, ref. 7)
Installation 115% of material cost $31,473 (2251.3 mh)		(ref. 6, ref. 7, section 15-80)
Total	$ 59,000	

Corrugated Roof Sheets

Corrugated roof sheets 18 gage (4 tanks)

587,600 lb × 70.7¢/lb = \$415,400 (Ref 1, Sec 5-7)

16-Gage Floor Sections

16-gage floor sections (4 tanks)

107,100 × 70¢/lb = \$74,970

MZ32 Sheet Piling

MZ32 sheet piling (26,770 ft^2)

856,640 lb × \$0.35/lb	(material & shipping)		\$299,800
× \$0.20/lb	(erection)		\$171,300
		Total	\$471,100

Roof Center Columns

24 WF 36 × 230 95' long

Material cost (\$0.251/lb) = \$131,600

Erection cost (\$0.30/lb) = \$157,300

Insulation Calculations

$\Delta T = 630°F - 65°F = 565°F$

$K = 0.044$ Btu/h-ft^2-°F @ 350°F   Thermal insulating wool type II*

$K = 0.022$ Btu/h-ft^2-°F @ 320°F   Dry sand

$k = 0.60$ Btu/h-ft^2-°F @ 68°F   Moist sand

*1 inch thickness

$$q_k \text{ (walk \& floor)} = \frac{630°F - 65°F}{[(1.5/0.22) + (5/0.6)] \text{ ft/(Btu/h-ft -°F}} = 37.29 \text{ Btu/h-ft}^2$$

where

Δx dry sand = 18 in.

Δx moist sand = 60 in.

$A = 40{,}906$ ft^2 Heat loss per tank = 1.53×10^6 Btu/h (floor & walls)

Max heat loss allowed = 3.96×10^6 Btu/h (total)

Max heat loss allowed from roof = 2.43×10^6 Btu/h or 84.6 Btu/h-ft^2

$$q_k \text{ (roof)} = \frac{0.44 \text{ Btu/h-ft}^2\text{-°F/in.}}{\Delta x \text{ in.}} (565°F) = 84.6 \text{ Btu/h-ft}^2$$

where $\Delta x = 2.9$ in.

Allowing for 4 in. I beams,

$$Q_k \text{ (roof)} = \left(\frac{0.44 \times 1404}{1 \text{ in.}} + \frac{0.44 \times 27{,}375}{5 \text{ in.}}\right) \times 565°F = 1.71 \times 10^6 \text{ Btu/h}$$

$$= \left(\frac{0.44 \times 1404}{1/4 \text{ in.}} + \frac{0.44 \times 17{,}375}{4\text{-}1/4 \text{ in.}}\right) \times 565°F = 3.00 \times 10^6 \text{ Btu/h}$$

Sand Moving Equipment

Screws

48 sections 46.5 ft long, \$65,000 apiece (Ref. 4)
(weight apiece = 18,600 lb)

Shipping from St. Joseph, Missouri - \$2,220 (Ref. 1, section 100-700)

Capital cost - 67,200 × 48 = 3,227,000

Assembly and erection -
35% of purchase cost = \$1,092,000 (Ref. 5)

Total screw cost = \$4,319,000

Screw casing

(12) 186 ft long, 7.5 ft diameter casings of ¼ in. plate

54,290 ft² plate	→ 5.537×10^5 lb × 0.37 \$/lb	= \$204,900	(Ref. 3)
15% of above for stiffness	→ 8.306×10^4 lb × 0.26 \$/lb	= \$ 21,600	(Ref. 2)
Fabrication & erection, \$0.30/lb		= \$191,000	(Ref. 3)
Insulation 1.5 in. calcium silicate \$6.80/ft²		= \$358,000	(Ref. 1, section 15-82)
	TOTAL	\$775,500	

Screw casing support

Vertical load = 7.38×10^4, horizontal load 5.17×10^4 lb

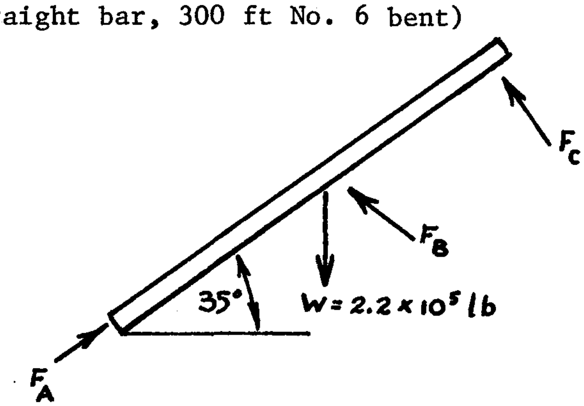
Soil bearing load = 2,000 lb/ft² vertical, 1,000 lb/ft² horizontal

1 pad for each casing - 8 ft × 6 ft × 6 ft (10.7 yd³)

Concrete - \$482 (material), 93 (6.6 mh) (Ref. 1, sec 3-3)

Reinforcing steel - (1300 ft No. 6 straight bar, 300 ft No. 6 bent)
\$946

Total = 1521 × 12 = 18,300



$F_A = 1.26 \times 10^5$ lb
$F_B = F_C = 9.01 \times 10^4$ lb

2 steel columns per pad 81 × 24, 4 ft long with base plates

192 lb (columns), 190 lb (base plates)

Material cost (0.25¢/lb) = $ 96

Erection cost (1.25¢/lb) = 478

$574 × 12 = $6,882

Total - 18,3600 + 6882 = 25,200

Thrust bearing, 27 in. bore

Material = $14,000 (Ref. 9)

Shipping = 59 (Ref. 1, section 100-700)

Installation = 1,911 (100 mh) (Ref. 1, Construction Cost Trend Report)

TOTAL $15,970 × 12 = $192,000

Roller bearings

(2) pillow blocks ($2,018 × 2)	= $4,036	
Shipping	= 50	(Ref. 1, section 100-700)
Shaft - 7-1/8 in. diameter steel shaft	= 112	(33.02 $/100 lb + 25%)
Maching & installation (100 mh)	= $1,965	
TOTAL	$6,163 × 60 = $369,800	

Bearing Casing & supports

Casing - 205 ft^2 (+50% extra), ¼ in. plate	= $ 775	(Ref. 3)
Supports - 50% of above, various shapes	= 242	(Ref. 2, 3)
Labor ($1.25/lb)	3,921	
TOTAL	$4,968 × 24 = $119,000	

Total of above = $5,801,000

Overhead (45% of labor = $686,000

Complete cost = $6,487,000

Pipe Sizes and Costs

Total mass flow rate 8.92×10^5 lb/h for charging heat exchangers

Total mass flow rate 9.06×10^5 lb/h for discharging heat exchangers

Max water flow 5 fps $\rho = 50$ lb/ft^3

$A_{charging} = 143$ in.^{2} D = 13.5 in. P = 1450 psia 16 in. Sch 100

$A_{discharging} = 145.0$ in.^{2} D = 13.6 in. P = 405 psia 14 in. Std

420 ft 16 in. Sch 120	(90.00 \$/ft)	= \$37,800	installation (168 mh)	= \$3600
420 ft 14 in. Std	(54.57 \$/ft)	= \$22,900	installation (122 mh)	= \$2600
370 ft 12 in. Sch 120	(63.34 \$/ft)	= \$23,400	installation (130 mh)	= \$2800
370 ft 10 in. Std	(40.48 \$/ft)	= \$15,000	installation (87 mh)	= \$1900

3 Tees 16 in. Sch 120	3 × \$600 + 3 × 639 (30 mh)	= 3700
3 Tees 14 in. Std	3 × \$285 + 3 × 396 (18.6 mh)	= 2000
3 Tees 12 in. Sch 120	3 × \$260 + 3 × 469 (22 mh)	= 2200
3 Tees 10 in. Std	3 × \$116 + 3 × 281 (13.2 mh)	= 1200

3 gate valves 16 in.	3 × 5995 + 3 × 115 (5.4 mh)	= \$18,300
3 gate valves 14 in.	3 × 4389 + 3 × 102 (4.8 mh)	= \$13,500
3 gate valves 12 in.	3 × 2586 + 3 × 90 (4.2 mh)	= \$ 8,000
3 gate valves 10 in.	3 × 1941 + 3 × 77 (3.6 mh)	= \$ 6,100

Total of above = \$165,000 (reference 1, section 15.43)

×2 = 330,000

Discharging Heat Exchanger

All tubes 0.75 in. OD, 0.065 in. ID carbon steel

Economizer

15,000 ft tube	$ 9,300
12 in. ID header (2), wt = 0.5 in.	
Carbon steel 52 in. along straight	500
Fabricator cost	48,200
	$58,000

Boiler

46,000 ft tube	$28,800
36 in. ID header (2), wt = 1.5 in.	
Carbon steel 52 in. along straight	6,700
Fabrication costs	$67,500
	$103,000

Installation costs (ref. 5)

15% of component cost - $38,000
½ of this is direct labor @ $17.48/mh
mh = 1087

Superheater

9,000 ft tube	$ 5,600
12 in. ID header (2), wt = 0.5 in.	
Carbon steel 52 in. along straight	500
Fabrication cost	31,700
	$37,800

Shell plating

1200 ft^2 of ¼ in. steel plating
12,240 lb × $0.75/lb = $9,200

Total of above	$208,000
+ 15% misc	$239,000

Insulation

1,455 ft^2 4 in. calcium silicate $16,300

Shipping weight = 158,000 lb
transported in four sections $19,400 for 1 complete heat exchanger

Heat Exchangers

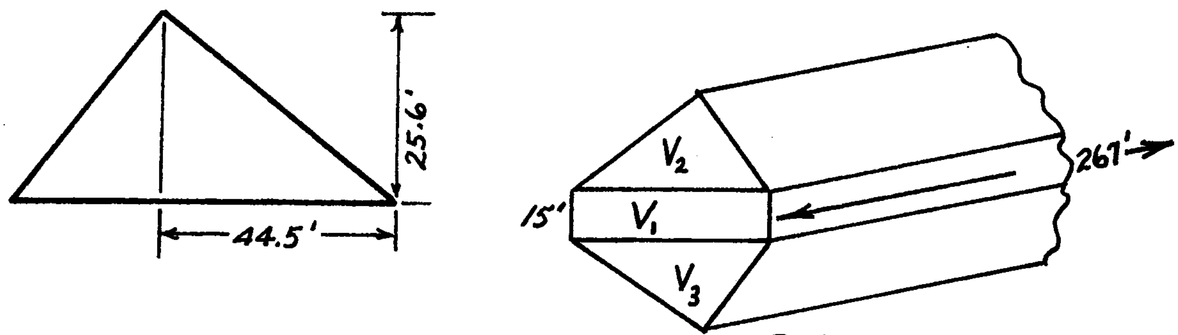
Charging heat exchanger

All tubes 0.75 in. OD, 0.08 in. wt

Subcooler			
5400 ft carbon steel tubes	$ 3,940	(Ref. 10)	
Construction costs	17,000	(Ref. 11)	
(2) 12 in. ID headers 1.4 in. wt			
Carbon steel 46 in. along straight	1,500	(Ref. 11)	
	$ 22,400		39,400
Condenser			
44,000 ft carbon moly tubes	$134,600	(Ref. 10)	
(2) 36 in. ID headers 4.125 wt			
Carbon steel 46 in. along straight	19,100	(Ref. 11)	
Fabrication costs	40,000	(Ref. 11)	
	$193,700		250,000
Desuperheater			
26,000 ft Croloy-1 tube	$ 95,400	(Ref. 10)	
(2) 12 in. ID headers 1.4 in. wt			
Carbon steel 46 in. along straight	1,500	(Ref. 11)	
Fabrication costs	65,000	(Ref. 11)	
	$161,900		149,000
Shell plating			
1160 ft^2 of ¼ in. steel plating			
11,860 lb ×0.75	$ 9,000		
Total of above	$387,000		
+15% misc	$445,000		
Insulation			
1,415 ft^2 4 in. calcium silicate	$ 15,800		
Shipping weight = 150,000 lb transported in four sections	$ 18,375 for 1 complete heat exchanger		

Controls & Instrumentation

108 thermocouples	59,400
12 pressure transducers	10,800
72 sand level detectors	28,100
12 sand flowmeters	5,000
12 sand control valves	12,000
12 steam flowmeters	14,400
12 steam control valves	12,000
1 charging microprocessor	50,000
1 discharging microprocessor	50,000
Heat exchanger instrumentation	
12 level detectors	19,000
Control panel & recorders	17,000
TOTAL	278,700



PER DWG. SK112480F-3
FIGURE 5-5

$$V_1 = 15 \times 89 \times 267$$
$$= 356{,}445 \text{ ft}^3$$

$$V_2 + V_3 = \frac{44.5 \times 25.6 \times 2}{2} \times 2 \times 267 - \left[\left(\frac{1}{3}\,\frac{44.5 \times 25.6 \times 2}{2}\right) 44.5 \times 4\right]$$

$$= 608{,}333 - 67{,}593$$
$$= 540{,}740 \text{ ft}^3$$

$$V_T = V_1 + V_2 + V_3 = 897{,}185 \text{ ft}^3 \text{ per tank}$$

$$4 \text{ tanks } V = 3{,}588{,}740 \text{ ft}^3$$

Volume of dividing walls (internally) $= 89 \times 40.6 \times 1.5 \times 2$ div
$= 10{,}840 \text{ ft}^3\text{/tank}$

$$V_{net}\text{/tank} = 897{,}185 - 10{,}840 \text{ ft}^3$$
$$= 886{,}346 \text{ ft}^3$$
$$4 \text{ tanks} = 3{,}545{,}380 \text{ ft}^3$$

Total mass of storage media (sand) needed to meet storage requirements = 1.34×10^6 ft^3 from Task 3.0

$$V_A = 1.34 \times 10^6 \text{ ft}^3.$$

Mass of sand for system is equal to required amount for storage plus "dead" sand

$$V_{TO} = V_A + V_D, \text{ where } V_D = \text{dead sand}$$

V_D = dead sand left in all tanks when emptied.

The dead sand is left in the tanks in the shape of a cone according to the angle of repose.

For estimating purposes, the maximum dead sand per tank is obtained by subtracting the active sand from the net free volume of a tank

$$V_D/\text{tank} = V_{net}/\text{tank} - \frac{V_A}{2}$$

$$V_D/\text{tank} = 886,346 - \frac{1,340,000}{2}$$

$$V_D/\text{tank} = 216,346\ \text{ft}^3/\text{active tank}$$

The active tank (hot or cold) does not fill completely and the incomplete fill space should about equal the dead fill space of an empty tank.

$$\therefore V_D/\text{total} = V_D/\text{tank} \times 2 = 432,692\ \text{ft}^3$$

Total amount of sand needed to operate is equal to active sand plus dead sand

$$V_{TO} = V_A + V_D = 1.34 \times 10^6 + 0.433 \times 10^6\ \text{ft}^3$$

$$V_{TO} = \underline{\underline{1.77 \times 10^6\ \text{ft}^3}}$$

$$V_{TO} = \underline{\underline{1.77 \times 10^6\ \text{ft}^3}}$$

$$= \underline{1.68 \times 10^8\ \text{lb}}$$

$$= \underline{\underline{84,202\ \text{tons}}}$$

For estimating cost of sand we assume a 20% factor for processing losses since this is Barstow site sand.

$$\therefore V_{CE} = V_{TO} \times 1.2$$

$$= 84,202 \times 1.2 = \underline{\underline{101,042\ \text{tons}}}$$

$$= \underline{2.02 \times 10^8\ \text{lb}}$$

$$= \underline{\underline{2.13 \times 10^6\ \text{ft}^3}}$$

Use 101,000 tons construction grade sand delivered
\$13.55/ton = \$1,355,000 (Ref. 6)

4 500-hp reversible impactors,	41,500 × 4	\$166,000	(Ref. 1, section 100)
4 500-hp motors & drives	20,000 × 4	80,000	(Ref. 1, section 100)
Installation (64 mh × 4)(16.95 \$/mh)		4,300	(Ref. 1, section 100)
4 troughed conveyors	44,700 × 3	134,100	(Ref. 1, section 100)
Operation (4,224 mh)(14.87 \$/mh)		62,800	(Ref. 1, section 100, TR)
		\$447,000	

Total sand cost = \$1,802,000

Most expensive - 101,000 tons of silica flour shipped by rail from Tulsa, Oklahoma =

\$25/ton for sand
+ 22/ton = \$4,747,000

Wage Differences

Los Angeles, California and Houston, Texas (from Richardson Construction Cost Trend Report)

Carpenter	1.05	0.94	0.90
Concrete finisher	0.87	0.92	1.06
Iron worker	0.94	0.82	0.87
Laborer	1.10	0.79	0.87
Operating engineer	1.05	0.87	0.83
Pipefitter	1.11	0.85	0.77
Sheet metal worker	1.34	1.06	0.79
Truck driver (const)	1.01	0.64	0.63

Excavation & backfill	72%
Equipment caverns	89%
Foundations & footings	89%
Structures	85%
Insulation	72%
Screws	87%
Piping	.77%
Heat exchangers	.87%
Aux	.87%

Modification to Increase Operating Temperature to 1000F

Media

Cp @ 425F = 0.218 Btu/lbm-°F	Cp @ 500F = 0.235 Btu/lbm-°F
Cp @ 630F = 0.259 Btu/lbm-°F	Cp @ 1000F = 0.268 Btu/lbm-°F
Cp (avg) = 0.239 Btu/lbm-°F	Cp (avg) = 0.252 Btu/lbm-°F
ΔT = 205F	ΔT = 500F
HC = 49.0 Btu/lb	HC = 126 Btu/lb

Media mass and volume are reduced by 61%

Site preparation costs	reduced by 61%
Equipment caverns	reduced by 66.7%
Foundations & structures	reduced by 61%
Insulation	reduced by 61%
Auxiliary equipment	same price
Control & instrumentation	reduced by 39%

Heat exchangers - all tubes and headers of 304 CRES

Charging heat exchanger		Discharging heat exchanger
$439,000	drum & headers	$367,000
102,000	plating, supports & misc	84,000
20,000	5 in. insulation	20,000
18,000	shipping	19,000
$579,000		$490,000

Screw conveyors

3.5 ft diameter $1,283/ft $\quad 1{,}398 = 1{,}283\left(\frac{6.25}{3.5}\right)^x \quad x = 0.148$

6.25 ft diameter $1,398/ft

6.5 ft diameter C $\quad C = 1{,}398\left(\frac{6.5}{6.25}\right)^{0.148} = 1{,}406$

All costs decrease by 66%

Pipe cost - quite variable, material costs the same, reduce labor by 2/3

Media cost - reduced by 61%

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9. Private communication with Torrington Bearing Co.
10. Private communication with B&W Tubular Products Division.
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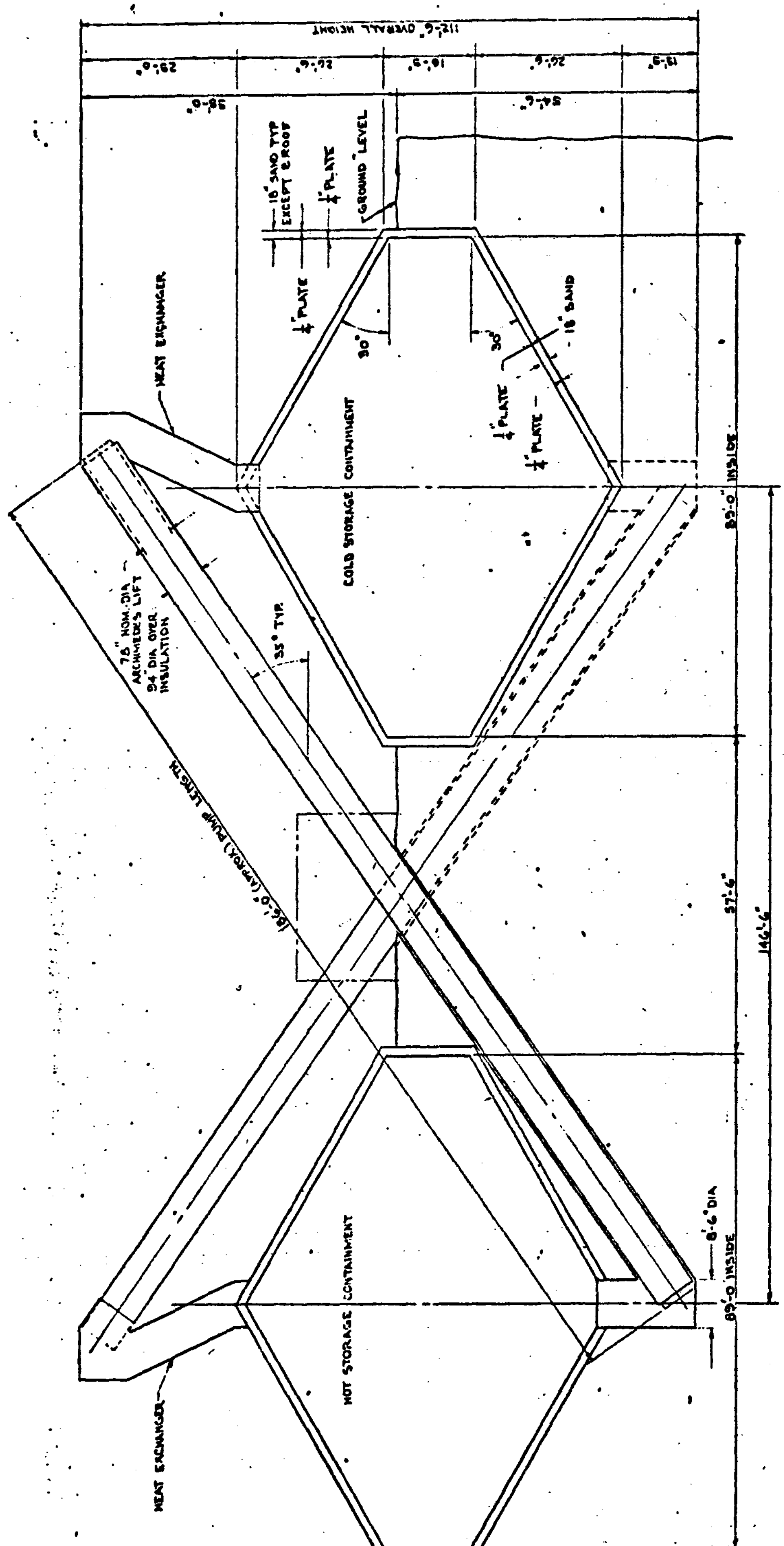

112'-6" OVERALL HEIGHT
29'-0"
22'-6"
16'-9"
24'-6"
13'-9"
58'-6"
54'-6"
18" SAND TYP EXCEPT @ ROOF
1/4" PLATE
GROUND LEVEL
HEAT EXCHANGER
1/4" PLATE
30°
30°
18" SAND
1/4" PLATE
1/4" PLATE
COLD STORAGE CONTAINMENT
82'-0" INSIDE
78" NOM. DIA ARCHIMEDE'S LIFT 94" DIA OVER INSULATION
35° TYP.
136'-0" (APPROX.) PUMP LENGTH
37'-6"
146'-6"
HOT STORAGE CONTAINMENT
8'-6" DIA
89'-0 INSIDE
HEAT EXCHANGER

SECTION A-A

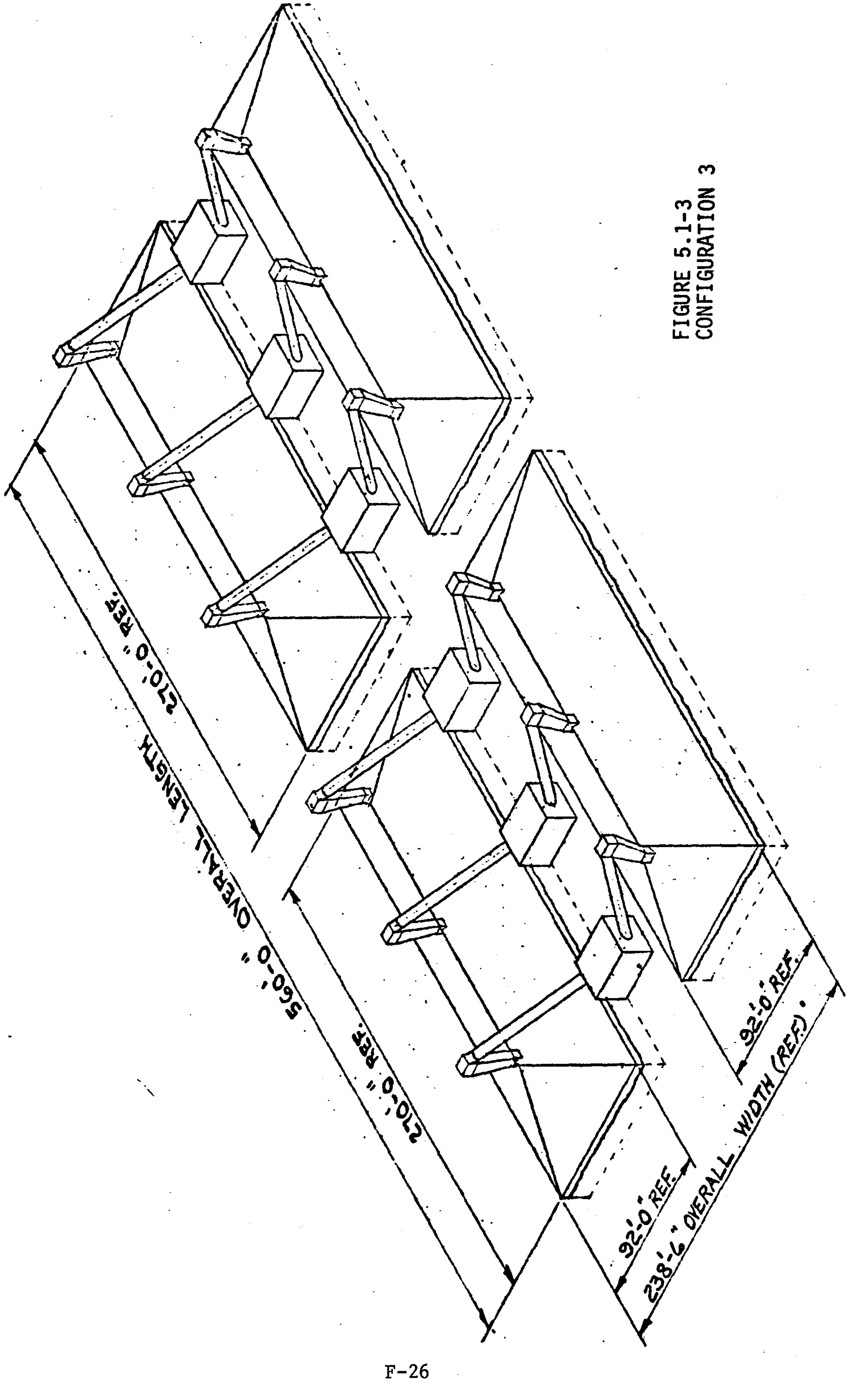
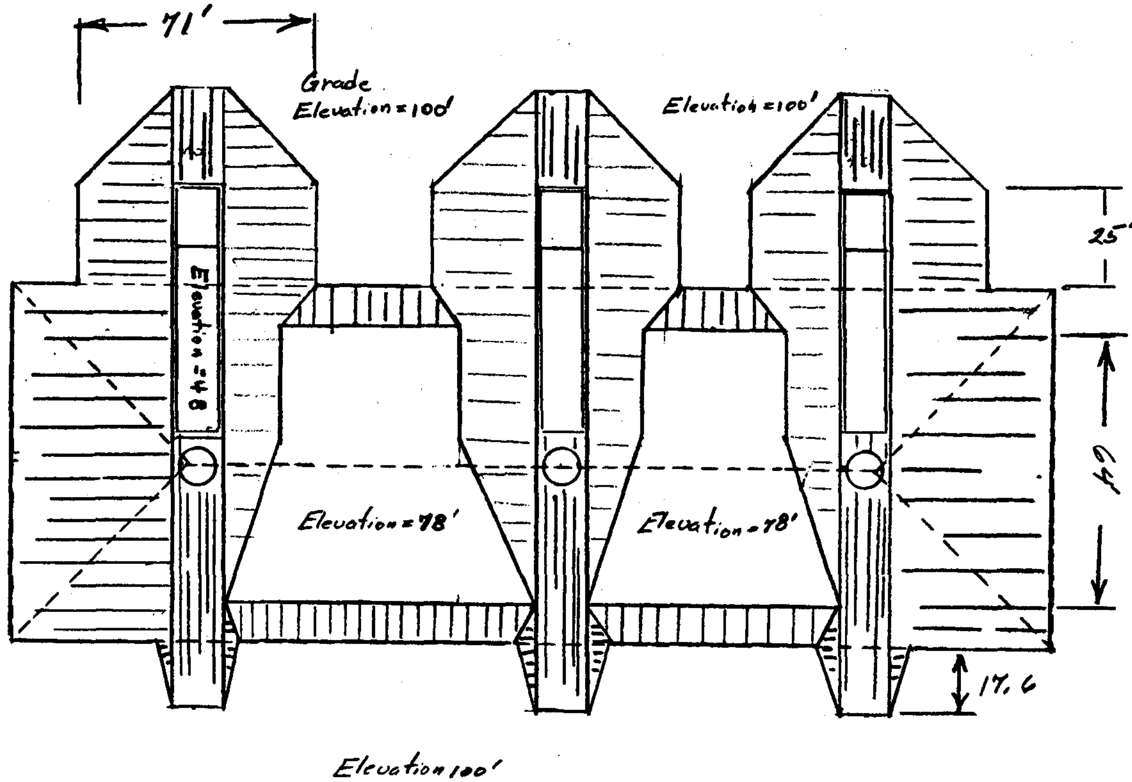


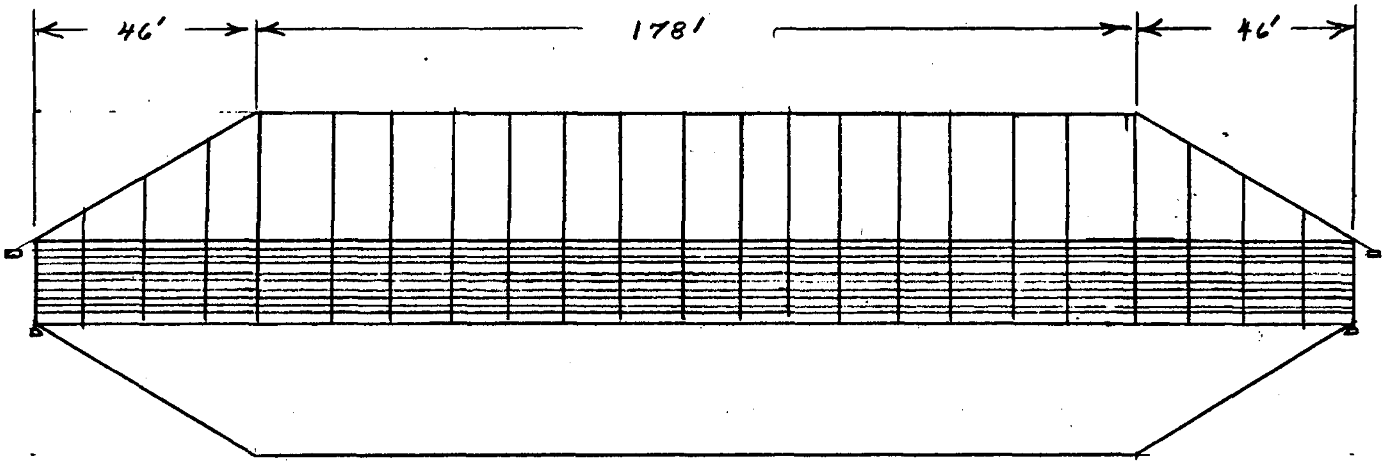
FIGURE 5.1-3
CONFIGURATION 3

EXCAVATION PLAN
SAND STORAGE TANK
MOVING SAND BED TESS
1/16/81

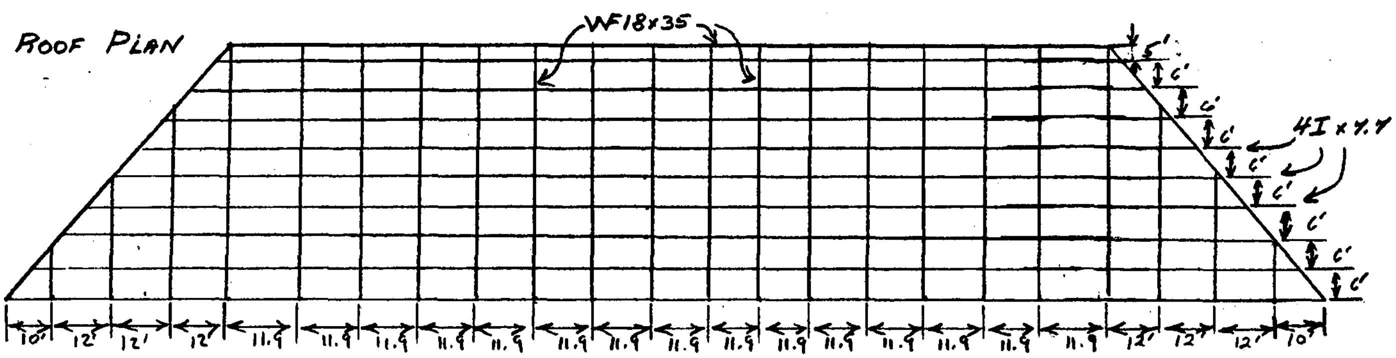


FRONT WALL & ROOF
SAND STORAGE TANK
MOVING SAND BED TESS
1/12/89

FRONT VIEW



ROOF PLAN



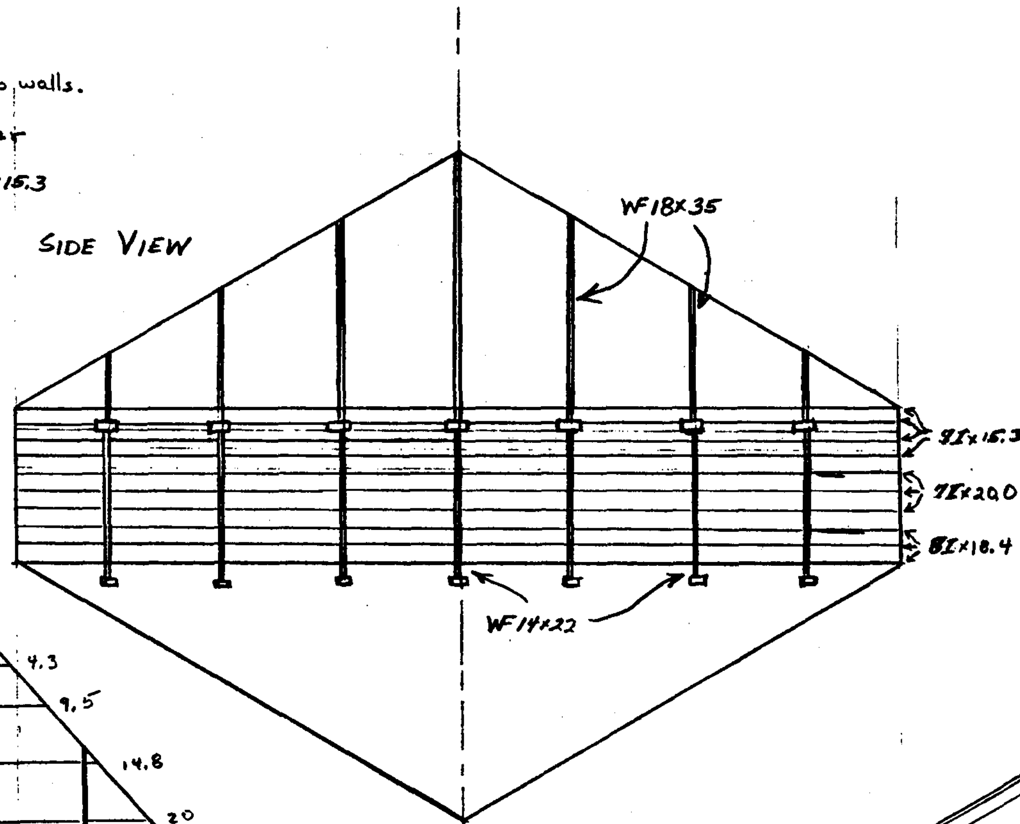
Notes

1. WF 18x35 Beam (Carbon Steel) are external with roof supported below beams. Horizontal bracing, 4I x 7.7, are attached as shown below on upper and lower roof plates

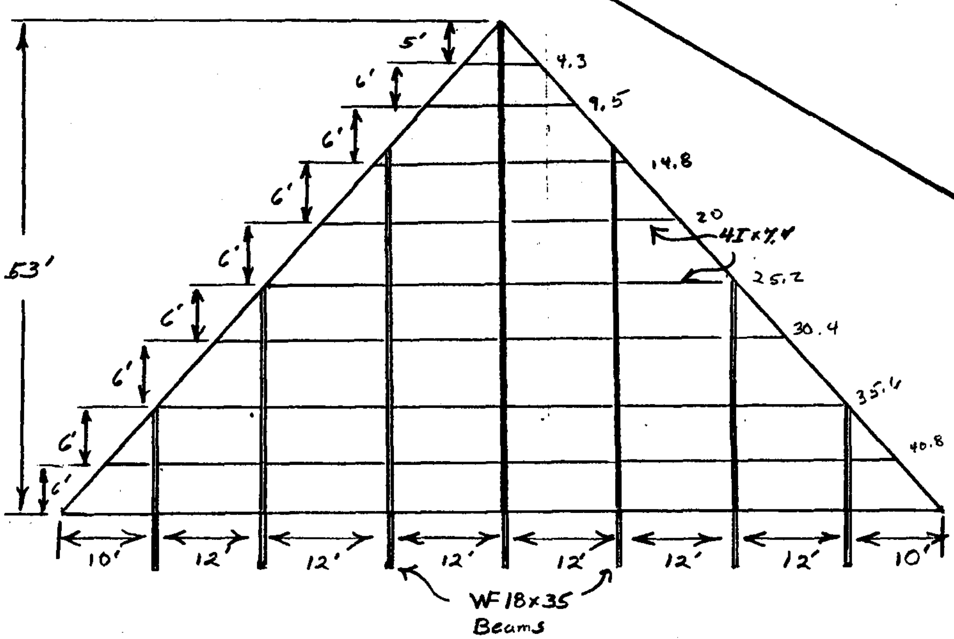
2. WF 14x22 Support beams are external to walls. Horizontal bracing are attached on inner wall as shown. One level of 7I x 15.3 on outer wall.

SIDE WALL & ROOF
SAND STORAGE TANK
MOVING SAND BED TESS
1/12/81

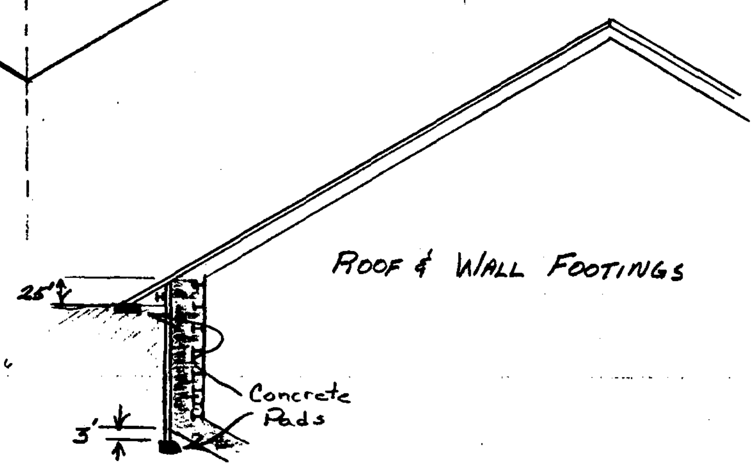
SIDE VIEW



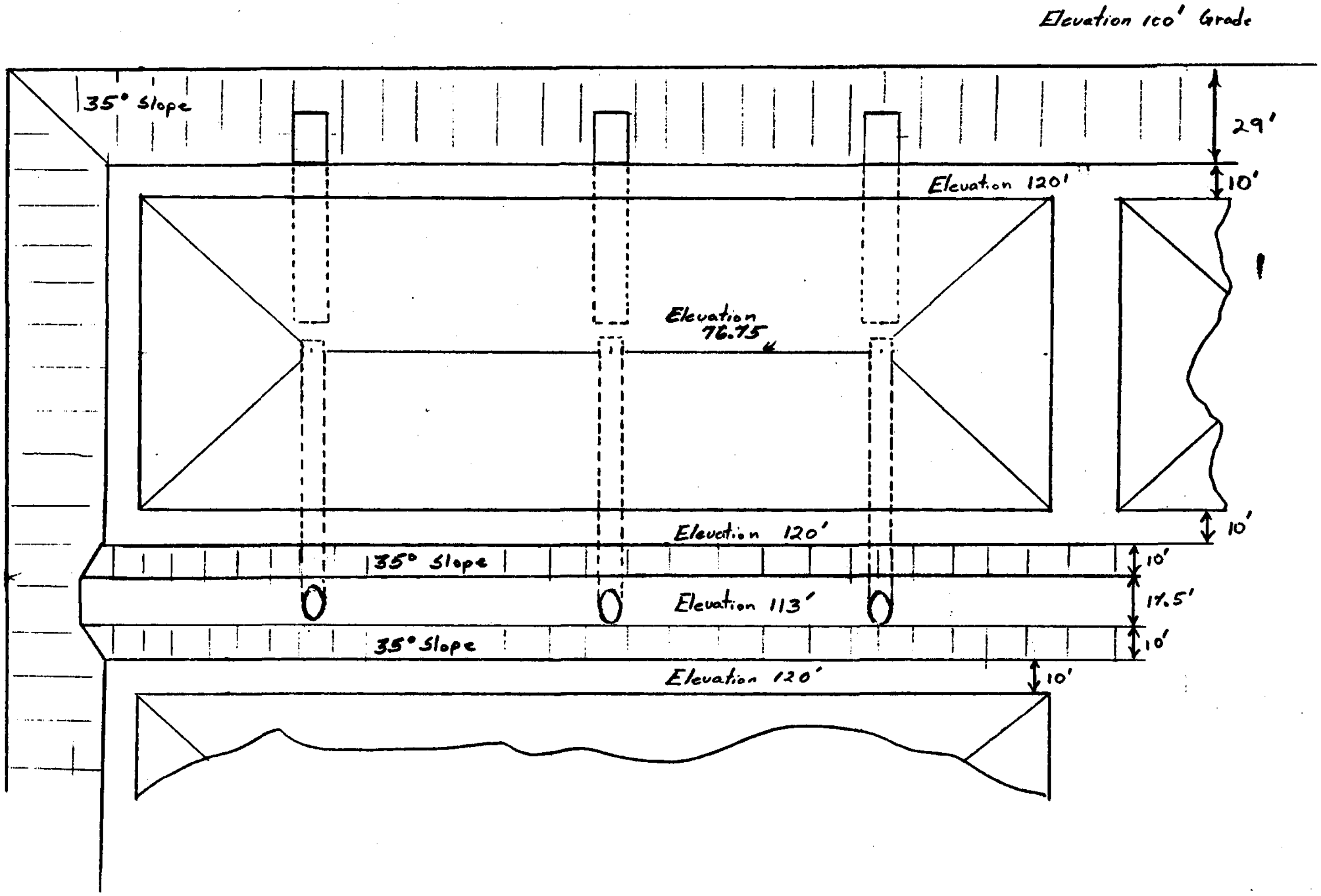
ROOF PLAN



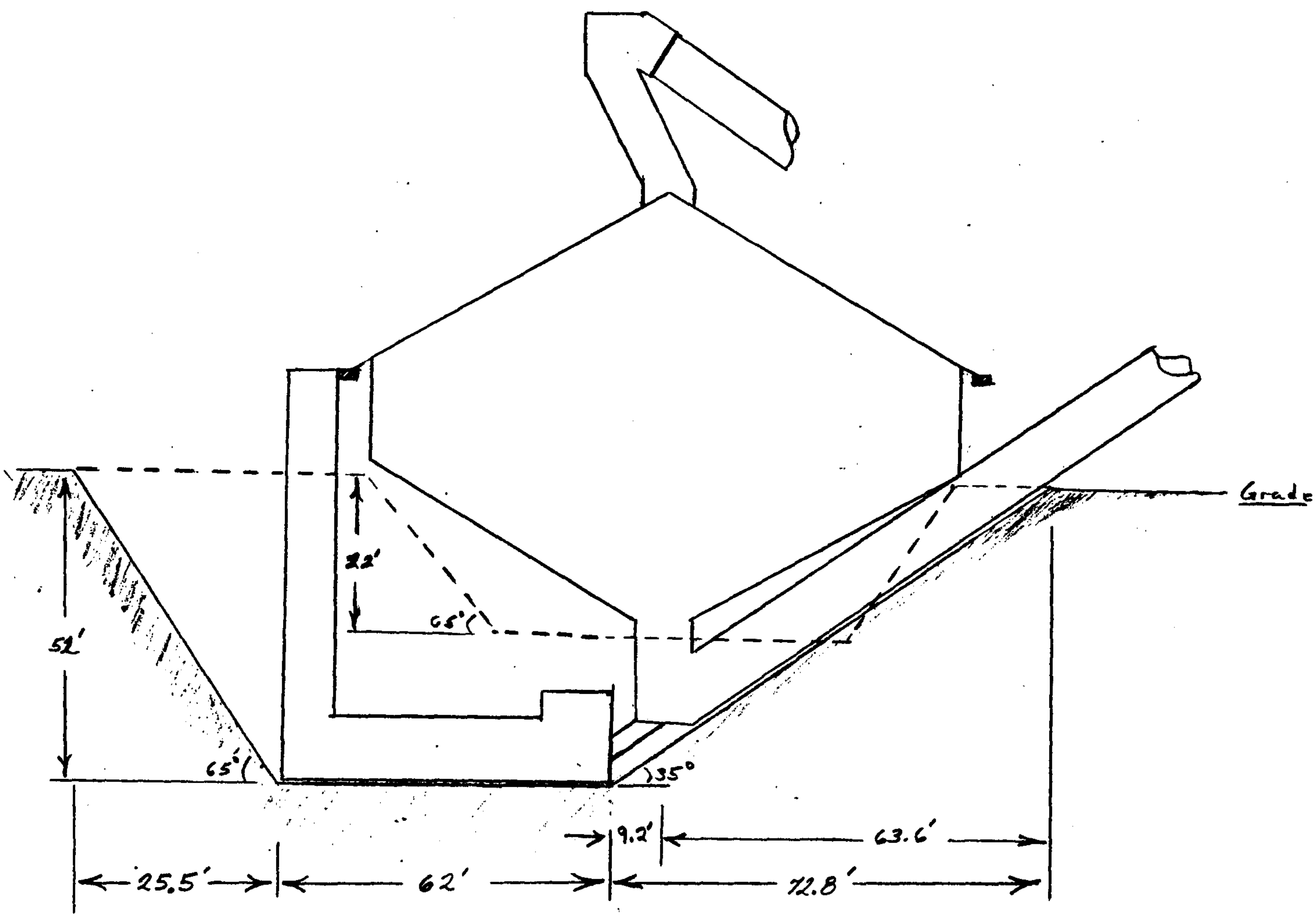
ROOF & WALL FOOTINGS



BACKFILL & COMPACTION PLAN
SAND STORAGE TANKS
MOVING SAND BED TESS
1/14/81



EXCAVATION, SIDE VIEW
SAND STORAGE TANK
MOVING SAND BED TESS
1/12/81



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